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An experimental and numerical study of a three-lobe pump for pumped hydro storage applications

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Abstract. Pumped hydro storage (PHS) plays an important role as a matured technology that accounts for the vast majority of global energy storage capacity, and its expansion is therefore desirable. The expansion of PHS in mid- and high-water heads is limited to topographic features, but there is an untapped potential in low-head applications. For most of the PHS applications, a Francis reversible pump-turbine (RPT) is regarded as the most common and cost-effective machine, but it is not a suitable option for water heads of less than 30m. In its place, positive displacement machines like lobe pumps could potentially work as RPT machines and unleash new possibilities for low-head pumped hydroelectric storage. In addition, unlike bladed pump-turbines, lobe pumps-turbines present a fish friendliness design, an important attribute to preserve the aquatic wildlife. This work will therefore present a three-lobe pump that could potentially be used in low-head PHS. An experimental model for a lobe machine will be presented, and its results will be used to validate the computational fluid dynamic simulations. Numerical investigations will address the characteristic curves regarding water-head, rotation speed and flow rate.

Keywords: Lobe pump, pumped hydro storage, experiment, numerical validation.

1. Introduction

With a growing share of intermittent renewable energy sources, especially solar and wind power, and as a means to allow a higher penetration of these energy supplies, the implementation and expansion of energy storage systems will be necessary to maintain grid stability and enhance its flexibility. Energy storage technologies can detain energy during periods when demand or costs are low, or when electricity supply exceeds demand, and can return that same energy when demand or energy costs are high. Hoffstaedt et al. (2022) made an extensive review on technologies that could suit low-head pumped hydro storage (LH PHS), a promising field in terms of energy storage where systems typically operate at water heads lower than 30m, but that lacks further investigations and full scale experiments in real life scenarios. The same authors address different reversible pumps and turbines machines that could address the challenges in low-head applications, in lieu of Francis turbines which is the dominant technology for PHS in mid- and high-head applications but that are not suitable for low elevations. One of the technologies mentioned is the lobe pump, a positive displacement machine that can be reversible and also work in turbine mode. Regarding turbine mode, Sonawat et al. (2021) find positive displacement turbines in applications with low flowrate, high head and lower specific speeds, operating below the operational range of conventional turbines.

1.1 Positive displacement machines

Positive Displacement (PD) machines are a type of fluid handling equipment that work by trapping a fixed amount of fluid and then forcing it to move between two locations. Unlike centrifugal pumps, which rely on the kinetic energy of the fluid to move it, PD machines maintain a constant flow rate by changing the volume of the fluid trapped inside the chambers of the pump. This makes them ideal for applications where maintaining a consistent flow rate is important.

Hoffstaedt et al. (2022) explain that a lobe pump presents an efficiency that remains relatively stable despite changes in head. The lack of sharp blades contributes to a more fish-friendly design, particularly when operating at low rotation speeds. These qualities are desirable in a LH PHS application. However, lobe machines have yet to be proven as a viable alternative.

1.2 Design aspects of lobe pumps and turbines

Rotary positive displacement machine such as lobe pumps and turbines present clearances between rotors and casing, and also clearances between both rotors. These gaps are expected to prevent the rotors to touch each other or to touch the casing at any instant of the pump or turbine operation. If there is any contact point, changes in flow rate, pressure drop, leakages and surface wear may occur and affect the operation of the machine. Plus, depending on the clearance sizes, cavitation and unwanted vibrations can occur. Phommacchanh et. al. (2006) showed in their PD turbine that it achieved a higher efficiency than a conventional turbine and it could sustain high efficiencies under the wide range of operating conditions. They also found out that the efficiency of turbine mode is much improved when reducing the clearance between the rotating and stationary parts of the turbine because of the amount of water leakage through those gaps. Complementary, they noticed that the torque efficiency of small side clearances decreased noticeably in the range of low-pressure region. But the improvement of the volumetric efficiency of the turbine with small side clearances was much higher than the torque efficiency reduction. Also, PD turbine with a smaller side clearance obtained much higher output power and consequently higher efficiency than the case with a large clearance at the same head and flow rate. Kang et al. (2012) also made the same conclusions, and proved that smaller clearances between rotor and casing walls produced much higher efficiencies, while the gap between lobes should be verified individually for each case.

The rotor surface profile can affect performance considerably. Kang et al. (2012) investigated this topic and they found that less pressure drop was generated by the cycloidal lobe. They also stated that the combination of less vortex and lower speed helps epicycloidal lobe pump to take an advantage over circular lobe pump, since it would prevent backward flow from the discharge area to suction area. Another advantage for the cycloidal profile was that it generated a characteristic curve with higher slope ratio, which resulted in higher efficiency of the pump because of the smaller leakage level in pump mode. Plus, the average value of pressure head of epicycloidal lobes was nearly 10% higher than that of circular pump in all studied cases. Finally, multi-lobe pumps with up to 4 lobes were compared, and they found that both tri- and four-lobe pumps provided more stable output and higher capacity than the two-lobe design, with no important differences between the tri- and four-lobe pumps. At the same time, the use of multi-lobes did not improve pump performance.

Finally, in order to reduce pressure pulsation, Kurokawa et al. (2008) explained that one can install a small sized simple surge tank at the inlet and outlet of the turbine, and that could reduce the pulsation amplitude by 75%. However, it should be possible to reduce pulsation without installing supplemental (and expensive) structures. That is one of the reasons why the authors studied a pair of four-lobe type rotor twisted helically with a twist of 11º, and they compare it with a three-lobe rotor. The authors explain that pressure pulsation increases remarkably with an increase of rotational speed and differential pressure in a three-lobe straight turbine. In the other hand, the twisted rotors reduced to about 20% of that by straight three-lobe rotor. As the helical lobe presents a bigger surface area, leakages are expected to be higher through both tip clearances and centre clearances. Torque and leakage are almost dependent only on the differential pressure, same as the straight 3-lobe type, but torque efficiency becomes much higher.

Complementary, Sonawat et al. (2021) say that machines that suffer of large pulsations in the flow can cause the generation of noise, vibration and fatigue, leading to the structural damage or breakdown of the device. So, they suggest the use of twisted rotors in order to damp most of the flow fluctuations. Sonawat et al. (2020) had a special look at the occurrence of cavitation in PD turbines by doing multi-phase CFD simulations, and they showed that the utilization of a 45º twisted rotor could practically eliminate the occurrence of cavitation because the contact area of the fluid for both rotors was uniform irrespective of the orientation or rotation of the lobes. Hence there was no abrupt change in the fluid properties. In the other hand, the 45º twisted rotor produced 1.15% less theoretical hydraulic efficiency compared to the straight lobe design due to increase in the leakage losses associated with it. But a vibration free device ensures a better longevity of the machine, which also can reduce maintenance costs.

2. Experimental setup

A model scale experiment was designed and built by the Laboratory of Fluid and Thermal Science group at Chalmers University of Technology to verify how a lobe machine would behave in low rotation speeds and it was used to validate the numerical simulations in this work.

First, one set of experiments was executed by the Chalmers group and it did not contain a diffuser in the design, as shown in Figure 1. Later, a diffuser was introduced at the inlet in order to smooth the flow transition. The experiment with the modified geometry was carried out by the Norwegian group at Chalmers University. As shown in Figure 1, the experimental rig consisted of two pipes connecting to the main channel which contained the PD RPT. The geometry consists of a casing and two straight rotors with three lobes each that follow a cycloidal profile. Twisted rotors were not considered in this work because the low rotational speed is intended for this experiment, thus vibration and cavitation should not be present, and a straight surface shape should provide a better hydraulic efficiency. Driving and driven rotors are synchronized with a gearbox. Rotors were 3D-printed and manufactured in acrylic (PMMA), as well as the housing that constrains the water flow. The overall design and dimensions can be seen at Figure 1 and the rig in Figure 2.

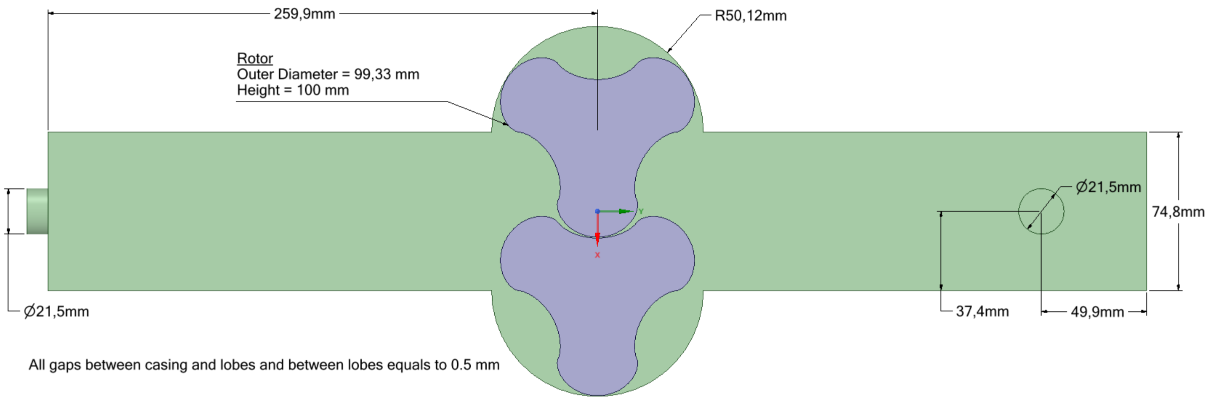
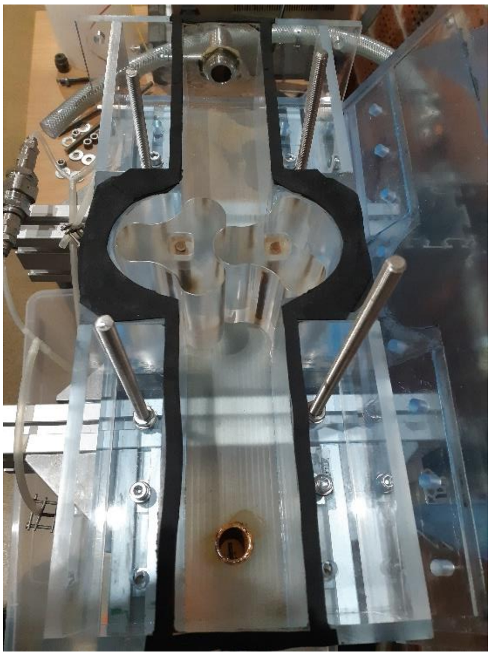
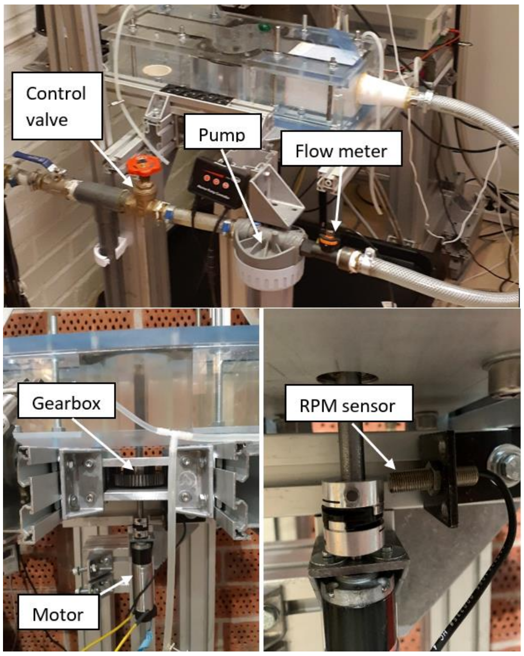


Figure 1. Lobe pump machine designed in model scale.





(a) (b)

Figure 2. Rig layout for the lobe machine. (a) Top view and (b) equipment for operating the rig.

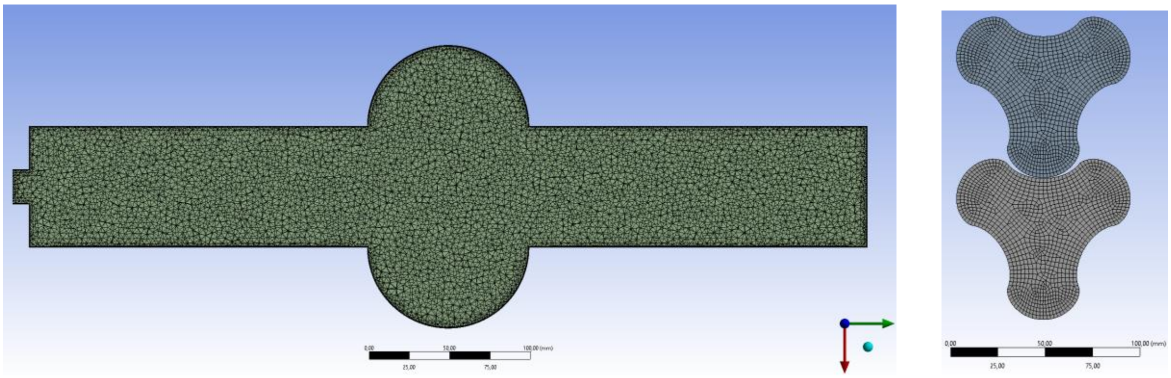
The test setup is a circular system, as depicted in Figure 2, where the water travels from the bottom outlet of the testing section, through a hose, through a flow meter, through a control valve, through an external pump, and back to the inlet of the testing section. The control valve is employed to adjust the necessary head during pump mode. The rotor lobes are rotated by a motor and a gearbox that connects the rotation of the two rotors, causing them to rotate at the exact same rotational speed but in opposite directions. An RPM sensor is used to monitor the rotational speed.

3. Numerical setup

Computational Fluid Dynamic (CFD) simulations were prepared in ANSYS 2020 R2 and CFX Solver. Transient simulations with single-phase water covering a single rotor revolution were sufficient to obtain stable pulsation values.

The immersed boundary method (IBM) was chosen to simulate the fluid-structure interaction in this problem. IBM treats the rotors as immersed bodies that indicates which numerical cells are inside the fluid or solid domain and interpolates values along the solid-fluid interface at the rotor surface (ANSYS CFX Refence Guide, 2020). IBM allows to gain computational time without the need to recalculate the numerical mesh for each time step. As consequence, the velocity in the fluid region that overlaps the immersed solid is enforced through a body force in the momentum equations. Therefore, a Momentum Source Scaling Factor (MSSF) is defined in CFX as a representation of the penalty method. In this method, fluid-structure interaction is modelled as a penalty force added to the fluid momentum equation that penalizes fluid velocity when it penetrates the structure. A higher MSSF value results in a closer match between fluid velocity and solid body velocity, leading to a lower error, but it also increases the likelihood of solver failure as the forcing term becomes larger, causing the numerical system to become stiff. So, an automatic solver control was set to control the MSSF. ANSYS CFX User Manual states that for a gear pump (a machine with similar behaviour to lobe pump), flow rate errors can be as high as 50% for MSSF equals to 10, and 9% for MSSF equals to 100.

Figure 3 shows the created mesh, and its statistics are described in Table 1. An inflation layer with 7 prisms was created to fill the space between the rotors and casing, so the leakages around the lobes could be properly modelled by the IBM method. The mesh refinement for the immersed body only needs to accurately depict the surface profile of the lobe.



(a) (b)

Figure 3. Mesh for (a) Fluid Domain (stationary) and (b) Immersed Solid domain (rotating).

Table 1. Mesh statistics

Mesh	Unstructured + Inflation
Number of nodes	916 507
Number of elements	2 277 848

Preliminary transient simulations showed that maximum Courant number achieved values under 1.0 over all the domain for all time steps, showing that a time step equivalent to 1º of rotation was sufficient to represent the CFD simulation in this work. Table 2 shows the calculated time-step and total simulation time expressions, which are only dependent on the rotational speed of the rotors.

Table 2. Analysis settings

Analysis Type	Transient
Total time for each simulation	1 complete turn, t = 1/(n[rpm]*60)
Time Step	1º of rotation, dt = 1/(n[rpm]*60)/360

Different boundary conditions for rotating displacement pump simulations were tested by Schiffer (2012), who also used IBM in ANSYS CFX. Thus, Table 3 indicates the boundary conditions that represented the experimental setup.

Table 3. Settings and Boundary conditions

Domain	CFX Options	Boundary conditions / Settings
Fluid (stationary)	Opening at low pressure side	Opening Pressure and Direction, Relative Pressure = 1 atm, Normal to boundary condition
	Opening at high pressure side	Opening Pressure and Direction, Relative Pressure = 1 atm + ‘measured pressure difference’, Normal to boundary condition
	Walls	No slip
	Turbulence Model	Shear Stress Transport, Automatic wall function
Immersed Solid (rotating)	Angular Velocity	30/35/40/45/50/55/60 rpm

4. Results and discussions

This study, following Schiffer (2012), verified the appropriate use of boundary conditions and allowed us to obtain and compare the average values for flow rate and pressure difference without the need to analyse the pulsation behaviour commonly found in positive displacement machines.

All the experimental tested points are indicated in Figure 4. It shows the characteristic pump curves for two configurations: one with and one without a diffuser at the inlet.

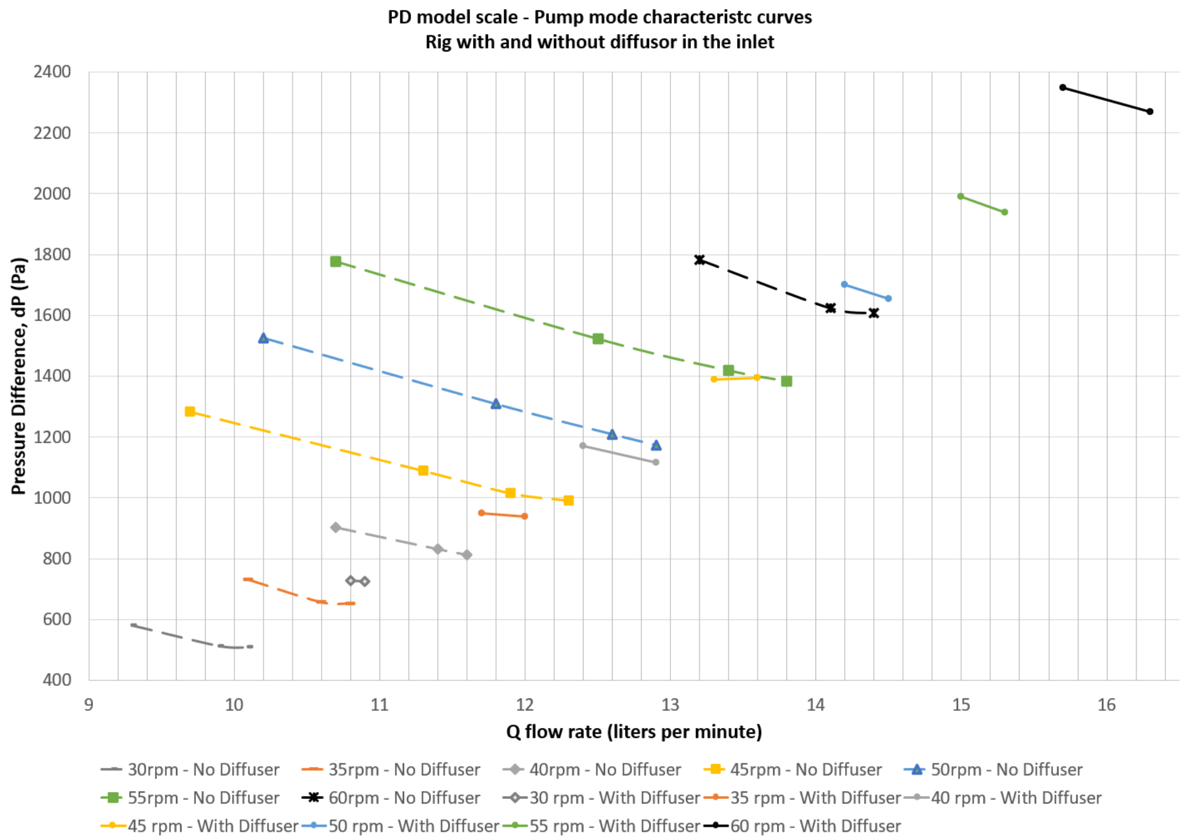


Figure 4. Experimental results for PD model scale

In Figure 4 the characteristic curves for both cases present a linear relation between flow rate and pressure difference, as expected in lobe pumps. Rotors operate between 30 and 60 rpm, and they produced flow rates between 9 and 16 litres per minute when pressure differences between 400 and 2400 Pa were imposed, which represents equivalent water-heads ranged from 4 to 23 cm. The higher the rotation speed, the higher the generated flow rate. We can also notice that, for the same rotation speed, the characteristic curves between the two geometries differ considerably, even though the only additional feature, the diffuser, was included in the inlet and did not interfere directly in the rotors zone. Thus, the diffusor was able to reduce drag by slowing down the fluid velocity and spreading it over a larger area, which reduced the turbulence and kinetic energy, and promoted a more efficient flow. So, it was observed that the small scale of the experimental model was very sensitive to variations in geometry or operation conditions, even small ones. Kang et al. (2012) addressed the same difficulties in measurements and measuring equipment can have significant errors.

Figure 5 shows the comparison between experimental and numerical data for the configuration without a diffuser in the inlet.

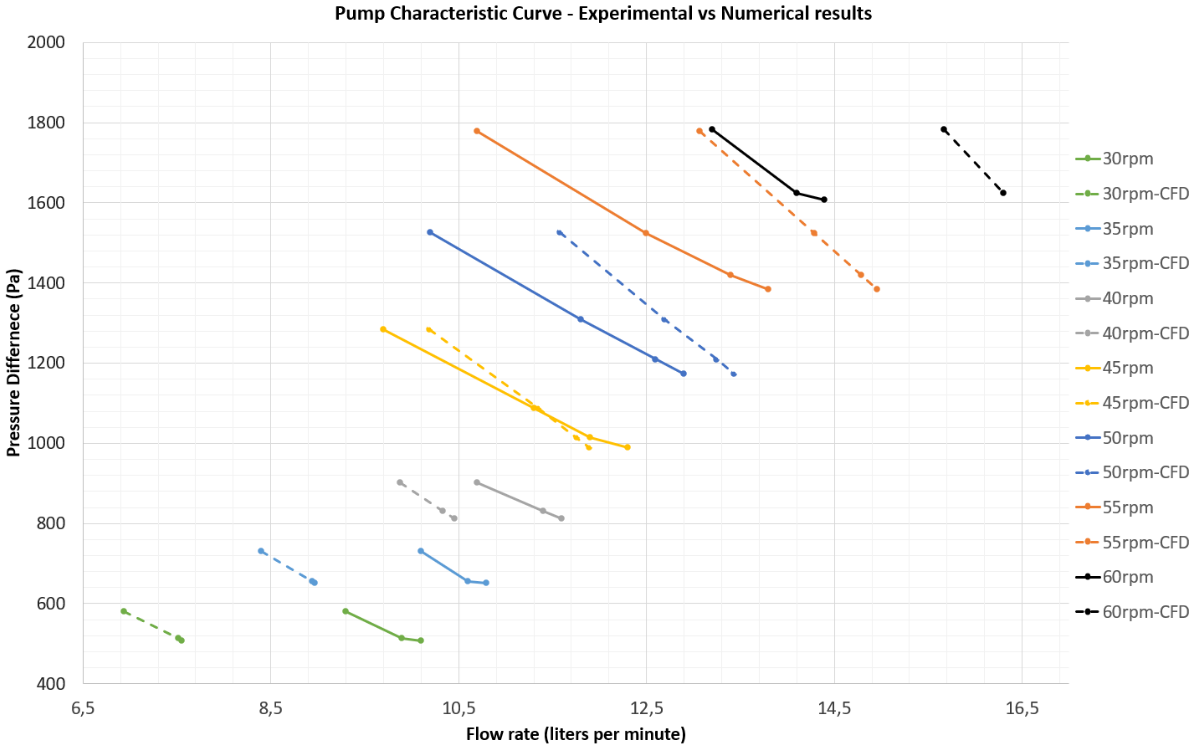


Figure 5. Comparison between experimental and numerical data for geometry without a diffuser.

In Figure 5, all points stayed below 5% error for rotation speed equals to 45 rpm. At 30 and 60 rpm, errors increased up to ±25%. We can also observe that the slope of the experimental and numerical curves remains the same inside the same set, but the numerical slope is a little higher than the experimental ones, which indicates that the numerical simulations underestimated the real leakages. Different reasons can explain those results. Initially, a higher MSSF could also decrease the error since IBM can induce unrealistic numerical leakage through the immersed solid bodies. Also, a more detailed mesh could be applied along the centreline of the model to accurately capture leakage between lobes. Moreover, the presence of bubbles during the experiment, although minimal, and the difficulty of maintaining the lobes with a constant rotor-rotor gap value could have generated abnormal behaviour in the system. Despite the differences explained above, errors were still under the estimated range provided by the ANSYS CFX Reference Guide.

Finally, due to the compact size of the test setup, the mechanical losses in the gearbox and seals prevented the runner from rotating in turbine mode, regardless of the head and flow rate. Hence, the motor had to be employed also in turbine mode to overcome those losses, so a net hydraulic torque could be estimated from the resulting pressure difference and flow rates. Despite the adjustments, significant uncertainties were still present and prevented this machine to provide reliable data in turbine mode.

5. Conclusions

A three-lobe pump was studied to investigate the potentials of positive displacement machines as reversible pump turbine units for low-head pumped hydro storage applications. Linear relationship between flow rate and water head was observed in both experiment and numerical analyses. In pump mode the model scale achieved around 16 litres per minute of flow rate when operating at 60 rpm, under water-heads close to 23cm, and in the presence of a diffuser. Because of the closed loop configuration of the experiment and scale of the model, it was not possible to test the machine in higher head conditions, and small variations led to big uncertainties. Numerical simulations followed the experimental trends, but despite the numerical errors being within the expect errors for the IBM technique, a mesh refinement located in the centreline should be implemented to improve the accuracy and reduce the discrepancy between CFD and experimental data. A higher MSSF value could also decrease unrealistic leakages through the solid bodies. Apart from the pump mode, the small-scale machine presented too high mechanical losses in the system that prevented it to operate in turbine mode. Therefore, the next step is to build another experiment with two open reservoirs in two different levels in order to provide the necessary setup to study the lobe machine in turbine mode, and as a full reversible pump-turbine machine intended for LH PHS. With a bigger model, experimental uncertainties are expected to decrease, and global efficiencies, torques and generated power can be estimated. With the new experiment, a numerical study could be validated for turbine mode and eventually provide an optimized design for the round trip of the PD RPT machine.

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