3D Numerical Simulation versus Experimental Assessment of Pressure Pulsations Using a Passive Method for Swirling Flow Control in Conical Diffusers of Hydraulic Turbines

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Abstract. The hydraulic turbines operated at partial discharge (especially hydraulic turbines with fixed blades, i.e. Francis turbine), developing a swirling flow in the conical diffuser of draft tube. As a result, the helical vortex breakdown, also known in the literature as "precessing vortex rope" is developed. A passive method to mitigate the pressure pulsations associated to the vortex rope in the draft tube cone of hydraulic turbines is presented in this paper. The method involves the development of a progressive and controlled throttling (shutter), of the flow cross section at the bottom of the conical diffuser. The adjustable cross section is made on the basis of the shutter-opening of circular diaphragms, while maintaining in all positions the circular cross-sectional shape, centred on the axis of the turbine. The stagnant region and the pressure pulsations associated to the vortex rope are mitigated when it is controlled with the turbine operating regime. Consequently, the severe flow deceleration and corresponding central stagnant are diminished with an efficient mitigation of the precessing helical vortex. Four cases (one without diaphragm and three with diaphragm), are numerically and experimentally investigated, respectively. The present paper focuses on a 3D turbulent swirling flow simulation in order to evaluate the control method. Numerical results are compared against measured pressure recovery coefficient and Fourier spectra. The results prove the vortex rope mitigation and its associated pressure pulsations when employing the diaphragm.

1. Introduction

New trends in the energy market require turbines to operate more and more frequently in transient and unsteady regimes in order to regulate the grid. Hydraulic turbines (i.e. Francis turbine), were designed to operate at, or in the neighbourhood of the best efficiency point (BEP). Far from such optimal regime, hydraulic turbine operation is hindered by unwanted flow instabilities, with associated low-frequency phenomena developed in swirling flows [1]. An important component of the turbines with low and medium head is the hydraulic draft tube cone, because this is responsible in a large proportion of the hydraulic losses in the system. Consequently, the efficiency of the turbine is significantly affected by the performance of the draft tube [2].

Particularly, more than 50% of the kinetic energy is recovered within the draft tube cone [3]. At partial discharge the flow downstream to the Francis turbine runner evolves in a precessing helical vortex (also known as vortex rope). The precessing vortex rope is formed between the main flow (close to the cone wall), and the stagnant region (close to the cone axis). According with Nishi et al. [4], the vortex rope is explained as a rolling up shape of a vortex sheet which is surrounding the stagnant region at the centre. In the draft tube cone appear pressure pulsations which are responsible for vibrations in all hydraulic system due to the shape of the vortex rope. As a consequence the vortex rope may produce runner blade breaks and even cracks, or ogive removal [5].

Jacob [6], performed the experimental investigations into a Francis turbine model in order to identify the operating regimes associated to the decelerated swirling flow and its instabilities. It was concluded that at 70% from the nominal flow, the pressure pulsations associated to the vortex rope are the highest. Also, depending by the operating regimes, two or three vortices are formed increasing the frequency and decreasing the pressure pulsations of the swirling flow from the draft tube cone.

Different techniques have been proposed in order to eliminate or to mitigate the instabilities developed in the draft tube cone at partial load operation. Given by the energy injected in the draft tube cone these methods can be divided into active [7] or passive [8]. These methods lead to some improvements in reducing the pressure pulsations for a narrow regime but they are not effective or even increase the unwanted effects. Resiga et al. [9, 10] have proposed a novel and robust method to mitigate the vortex rope: a water jet is injected along the discharge cone axis. This technique was investigated on a test rig developed at the "Politehnica" University of Timisoara. Also, Susan-Resiga et al. [11] demonstrated that a 2D axisymmetric simulation is able to capture the formation and development of swirling flow phenomena at levels similar to a 3D numerical simulation. The only observation is that for a 2D axisymmetric simulation the pressure pulsations cannot be investigated, consequently the maximum amplitude peaks cannot be identified.

A passive method to mitigate stagnant region associated to the vortex rope in the draft tube cone of hydraulic turbine is presented in this paper. The method involves the development of a progressive and controlled throttling (shutter), of the flow cross section at the bottom of the conical diffuser (Fig. 1-down). The adjustable cross section is made on the basis of the shutter-opening of circular diaphragms (Fig. 1), while maintaining in all positions the circular cross-sectional shape, centred on the axis of the turbine. The vortex rope occurs when the turbine is operated at part load. The stagnant region and the pressure pulsations associated to the vortex rope are mitigated when it is controlled the diaphragm [12]. The opening of the diaphragm can be automatically correlated with the turbine operating regime.

The present paper focuses on 3D turbulent swirling flow simulation in order to evaluate the new control method. Numerical results obtained are compared against measured pressure data. The second and the third sections of the paper present the experimental and numerical setup. Section 4 presents the numerical results and validation against experimental data. In last section are summarized the conclusions.

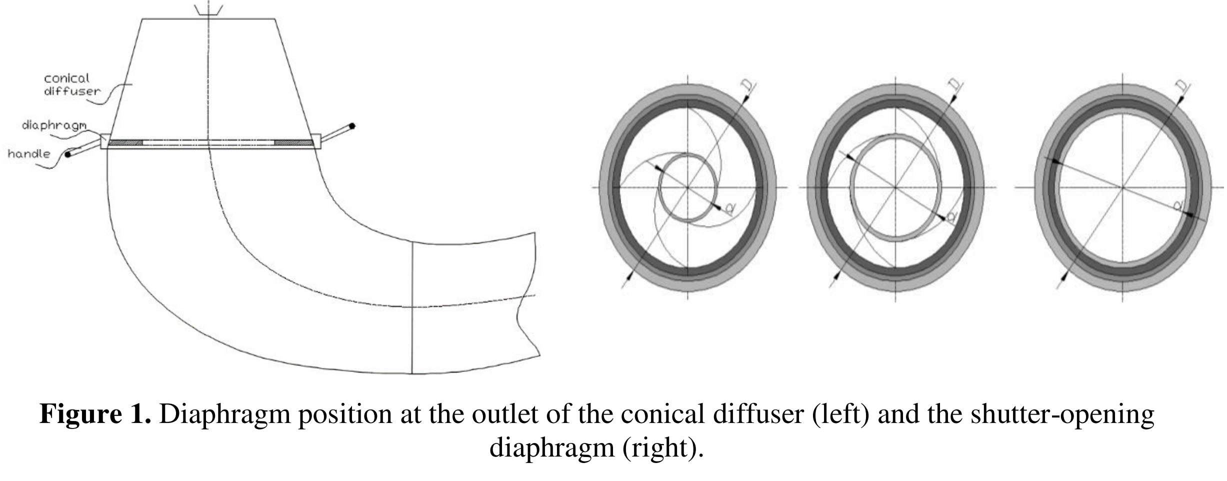

Figure 1. Diaphragm position at the outlet of the conical diffuser (left) and the shutter-opening diaphragm (right).

2. Experimental setup

In order to investigate experimentally the diaphragm passive method, we are using the test rig with a closed loop hydraulic circuit (Fig. 2a) described in [13]. Instead of testing the diaphragm on a model hydraulic turbine, we have designed and built a special swirl apparatus, Fig. 2b. The swirling flow apparatus, included in the main hydraulic circuit, contains two main parts: the swirl generator and the convergent-divergent test section. The swirl generator has an upstream annular section with stationary and rotating blades for generating a swirling flow. It has three components: the ogive, the guide vanes and the free runner, see the detail in Fig 3b. The ogive with four leaned struts sustains the swirl generator and supplies the jet nozzle. In the cylindrical section with $D_S = 150\,mm$ are installed the guide vanes and the free runner. Nozzle outlet with $D_n = 30\,mm$ is located close to the throat section with $D_t = 100\ mm$.

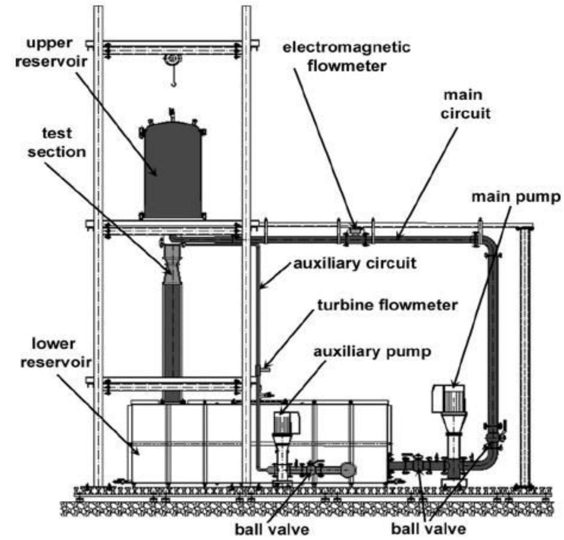

Figure 2a. Experimental closed loop test rig. Sketch of the test rig with the main elements.

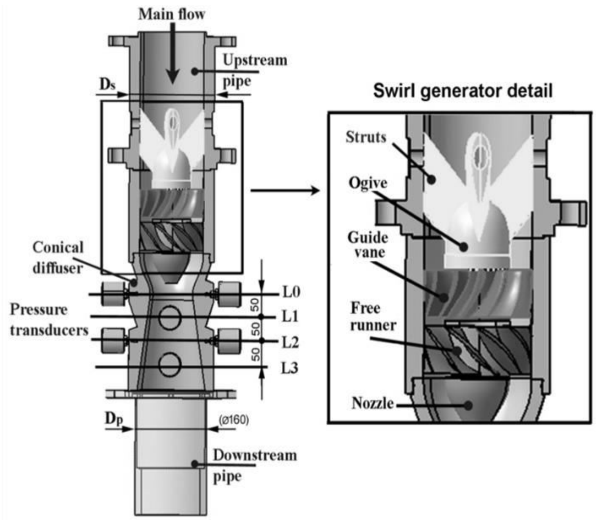

Figure 2b. Cross-section through the swirling flow apparatus and detail of the swirl generator.

This swirl generator provides a swirling flow configuration at the inlet of the conical diffuser quite similar to the corresponding flow downstream a Francis runner operated at partial discharge. As a result, the decelerated swirling flow in the cone develops a precessing vortex rope with the same Strouhal number as the one corresponding to the Francis turbine model investigated in [14]. The cone half-angle is 8.6 degrees, similar to the compact discharge cones used in the modern draft tubes for hydraulic turbines. However, in our case the ratio between the cone length ($L = 200\,mm$) and the throat diameter ($D_t = 100\,mm$) is quite large ($L/D_t = 2$) in order to capture the entire vortex rope in the conical diffuser. The results have been obtained for a test rig discharge of Q=0.03 m^3/sec. In addition, all experimental investigations has done under overpressure conditions. The hydraulic circuit was, fully filled with water. As a result, only non-cavitating vortex rope, has considered, in our investigations, meaning no air volume trapped inside.

Table 1. The parameters corresponding to three cases with different interior diameters of diaphragm.

Diaphragm interior diameter *d [m]*	Diaphragm interior area $A_d\,[m^2]$	Test section outlet area $A_o\,[m^2]$	Shutter area $A_r\,[m^2]$	Areas ratio $A_a\,[\%]$
0.113	0.01	0.02	0.01	50
0.100	0.0078	0.02	0.012	60
0.088	0.006	0.02	0.014	70

The diaphragm control method implemented on our swirl apparatus is sketch in Fig. 3. The main component of this new method for mitigating the vortex rope is the diaphragm (Fig. 1 - right). Three values of the diaphragm interior diameter of d = 0.113, 0.100, 0.88 m are considered in the experimental and numerical investigations. The diaphragm is located at the cone outlet (Fig. 3). Table 1, shows the ratio between diaphragm interior area and the outlet test section area with D = 0.16 m.

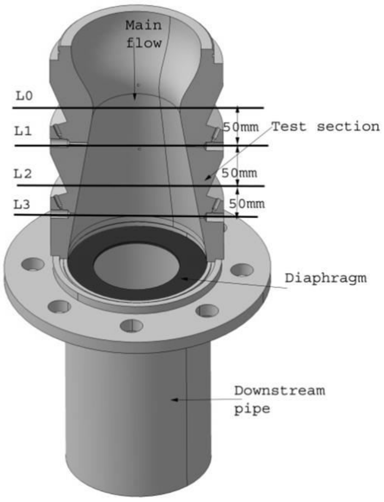

Figure 3. Representation of the passive method with diaphragm implemented on the swirl apparatus.

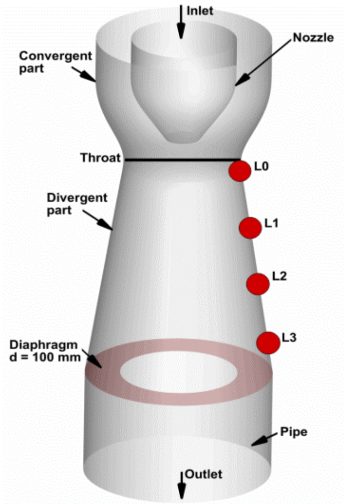

Figure 4. 3D computational domain.

3. Numerical setup

The computational domain with diaphragm is presented in Fig. 4. A structured mesh with 2.7M cells is generated on each computational domain (with and without diaphragm). Boundary conditions imposed for each case are the following: radial velocity profiles and turbulent quantities on the inlet surface and average pressure on the outlet of the section, respectively. The inflow boundary conditions are obtained computing the 3D turbulent flow in the swirl generator apparatus [15]. As a result, the inlet radial profiles of the velocity components (axial, circumferential are plotted in Fig. 5 and negligible radial component) as well as the turbulent quantities (kinetic energy and turbulence dissipation rate) corresponding to a runner speed of 920 rpm are imposed on annular inlet section. 3D unsteady numerical simulations with and without diaphragm were performed using the FLUENT code with RSM turbulence model in order to assess the new approach.

The time step for the numerical simulations in both cases was t = 0.1 ms. All numerical solutions were converged down to residuals as low as 10^{-4}. Pressure monitors denoted L0…L3 have been achieved on 4 levels. The axial distance between two consecutive pressure taps located on the cone wall is 50 mm.

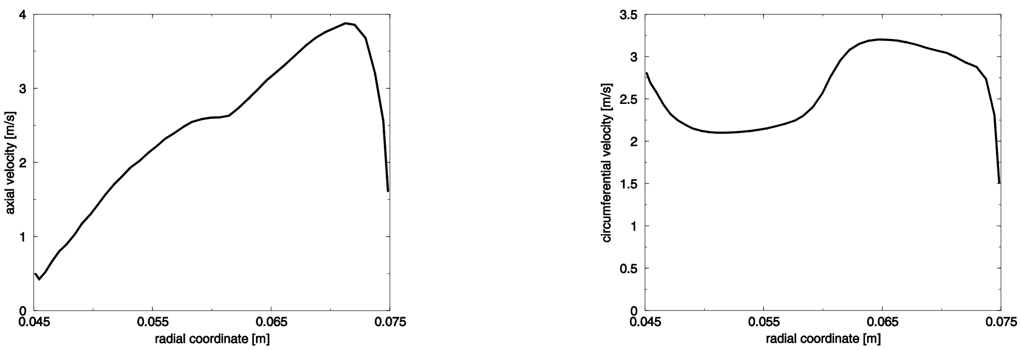

Figure 5. Velocity profiles imposed as inflow conditions: axial (left) and circumferential (right) components.

4. Numerical results validation against experimental data

4.1. Mean pressure analysis

The main purpose of the discharge cone is, to convert the excess of kinetic energy at the runner outlet into static pressure by decelerating the swirling flow downstream the hydraulic turbine runner. In practice, this dynamic-to-static pressure conversion is quantified by the so-called pressure recovery coefficient, usually evaluated with the wall pressure. The wall pressure evolution is expressed in dimensionless form with the pressure coefficient,

$$c_p = \frac{\overline{p} - \overline{p}_{L0}}{\rho V_t^2 / 2} \text{ where } V_t = \frac{Q}{\pi D_t^2 / 4} \tag{1}$$

where $\overline{p}_{L0}$ is the mean pressure measured by the transducers at the L0 level (Fig. 3), $\overline{p}$ is the mean pressure measured further downstream on the cone wall, ρ is the water density, and V_t is the throat average discharge velocity with $D_t = 0.1\ m$ the throat diameter of the swirl apparatus at level L0, respectively. The pressure coefficient defined in Eq. (1) is plotted in Fig. 6 for the levels L1, L2 and L3, where the axial coordinate is made dimensionless by the throat radius. The measurements were repeated N > 10 times for each operating regime, and the standard deviation for the measurement set,

$$\sigma = \sqrt{\frac{\Sigma\left(p_i - \overline{p}\right)^2}{N-1}} \tag{2}$$

One can easily observe a significant increase in the pressure recovery when the diaphragm is switch on. Also, can be observed a good correlation between experimental wall pressure coefficient vs. 3D numerical simulation. For example, in the middle of the conical diffuser, level L2, the wall pressure recovery coefficient is practically doubled. It is clear that by mitigating the vortex rope, the dynamic-to-static pressure conversion is more efficient. For real turbines, this improved pressure recovery in the discharge cone is reflected in an increase of the overall turbine efficiency far from the best efficiency point, especially for low-head hydraulic turbines, since the main fraction of the hydraulic losses at such operating points are associated with the swirl in the draft tube cone. At the L1 level, the pressure recovery has a small increase and an overestimated increase is recorded at the L3 level, because of recirculation zones occurring near the diaphragm wall. However, from a dynamic point of view, there is a significant improvement. Figure 7 shows the loss coefficient and kinetic-to-potential conversion ratio versus shutter area, respectively. For this, we introduce the following integral quantities in order to analyses the kinetic-to-potential energy transformation process, as well as its efficiency:

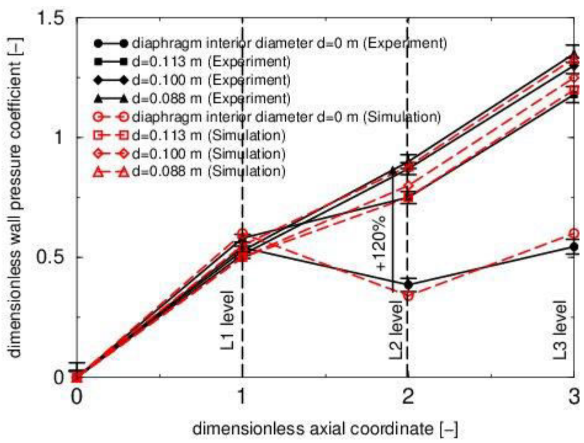

Figure 6. Pressure coefficient on the cone wall, experiment vs. 3D numerical simulation.

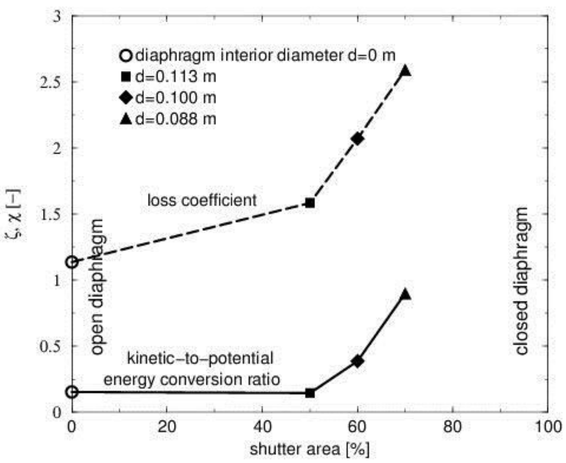

Figure 7. Energy loss coefficient and kinetic-to-potential energy conversion ratio vs. shutter area, respectively.

Flux of potential energy

$$\Pi(x) \equiv \int_{S(x)} p(x,y)V \cdot ndS, [W] \tag{7}$$

Flux of kinetic energy

$$K(x) \equiv \int_{s(x)} \frac{\rho V^2(x,y)}{2} V \cdot ndS, [W] \tag{8}$$

Flux of mechanical energy

$$E(x) \equiv \Pi(x) + K(x), [W] \tag{9}$$

A dimensionless loss coefficient ζ is usually defined as:

$$\zeta(x) \equiv \frac{E_0 - E(x)}{K_0} \tag{10}$$

The kinetic-to-potential energy conversion ratio can be quantified as:

$$\chi(x) \equiv \frac{\Pi(x) - \Pi_0}{K_0 - k(x)} < 1 \tag{11}$$

In our case we compute the dimensionless coefficients (loss coefficient ζ and kinetic-to-potential energy conversion ratio χ), between the throat of the cone and outlet of the domain. We examine their variation with respect to the diaphragm shutter area. One can see from Fig. 7 that the hydraulic loss coefficient and kinetic-to-potential energy conversion ratio reach the maximum values for the highest shutter area of diaphragm (70%). The evolution of the loss coefficient ζ in the cone, emphasizes the rapid increase in the hydraulic losses at partial discharge. This is also associated with an increase in the overall performance of the cone as shown in the variation of the kinetic-to-potential energy conversion ratio. It is obviously when throttling the flow at the outlet of the cone, using the diaphragm, the hydraulic losses increases but also the pressure recovery increase.

4.2. Unsteady pressure field analysis

The self-induced flow instability of the decelerated swirl in a conical diffuser develops a precessing helical vortex with an associated pressure fluctuation that hinders the hydraulic turbine operation. This is the reason, why we focus in the present section on the unsteady part of the pressure measurements, as well as on the effect of the diaphragm on the pressure fluctuations level.

The numerical results are validated against experimental data, in order to assess the numerical setup. Since the unsteady part of the pressure signal is periodic, we characterize it using the vortex rope precessing frequency and the dominant amplitude. In dimensionless form, the precessing frequency is expressed using the Strouhal number,

$$Sh = f\frac{D_t}{V_t} \tag{3}$$

and the pressure fluctuation is dimensionless with the kinetic term, $\rho V_t/2$.

Let us now examine the cause of the significant pressure pulsation reduction displayed in Fig. 9, in comparison with Fig. 8, which in turn is lower than the corresponding level measured without diaphragm. The decrease of the amplitude of the pressure fluctuations at the L0 to L3 levels in the cone, occurs with the increase of the shutter area from the outlet test section. One can see that when the area of the outlet test section is shutter, the precession frequency remains unchanged and the corresponding amplitude of the pressure fluctuations is significantly decreasing. We conclude, that the diaphragm approach has the potential to effectively mitigate the pressure fluctuations in decelerated swirling flows with precessing vortex rope, while improving the pressure recovery as shown in Figs. 6, 7.

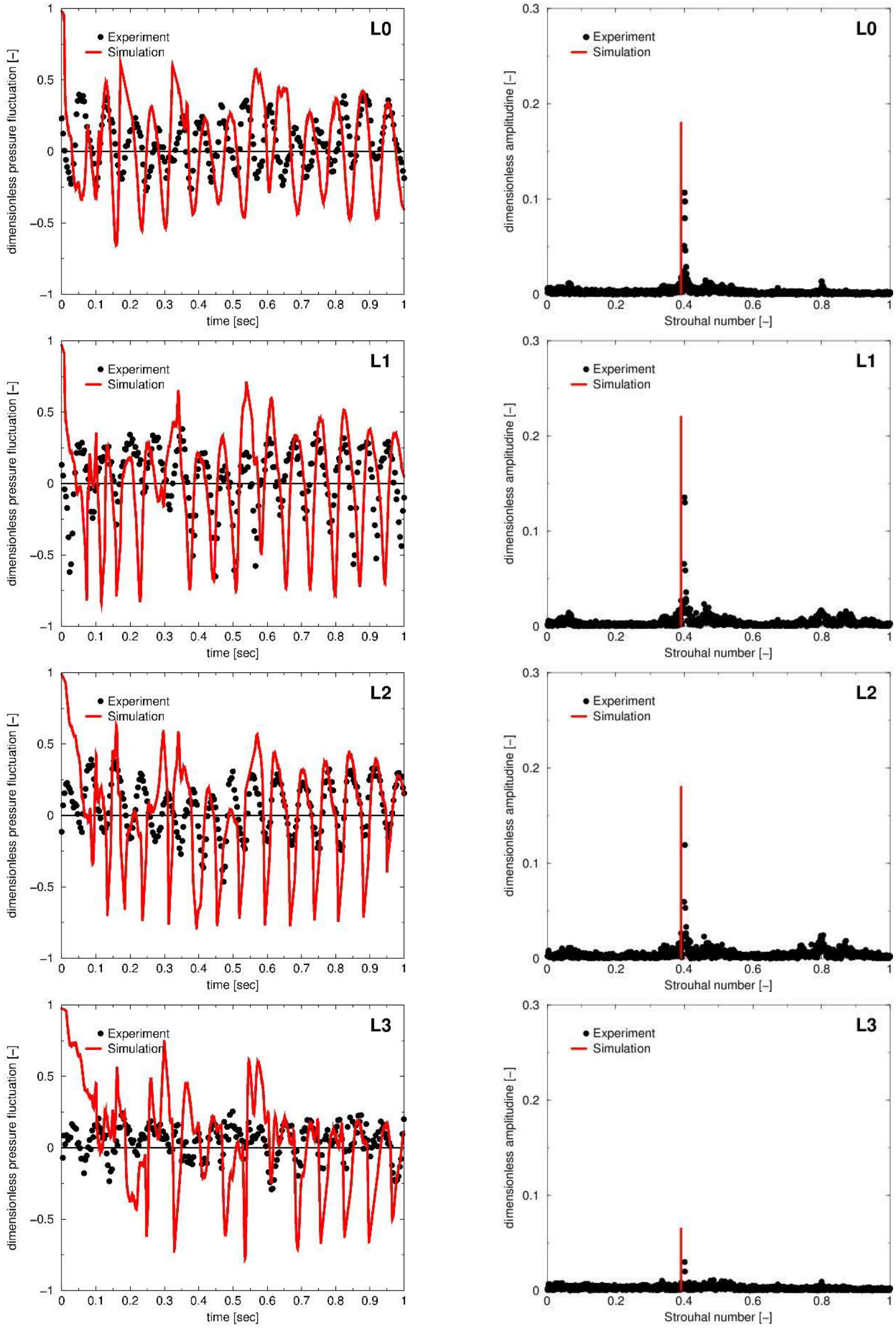

Figure 8. 3D numerical simulation vs. experimental data of pressure fluctuations (left) on the cone wall and associated Fourier spectra (right) without diaphragm, respectively.

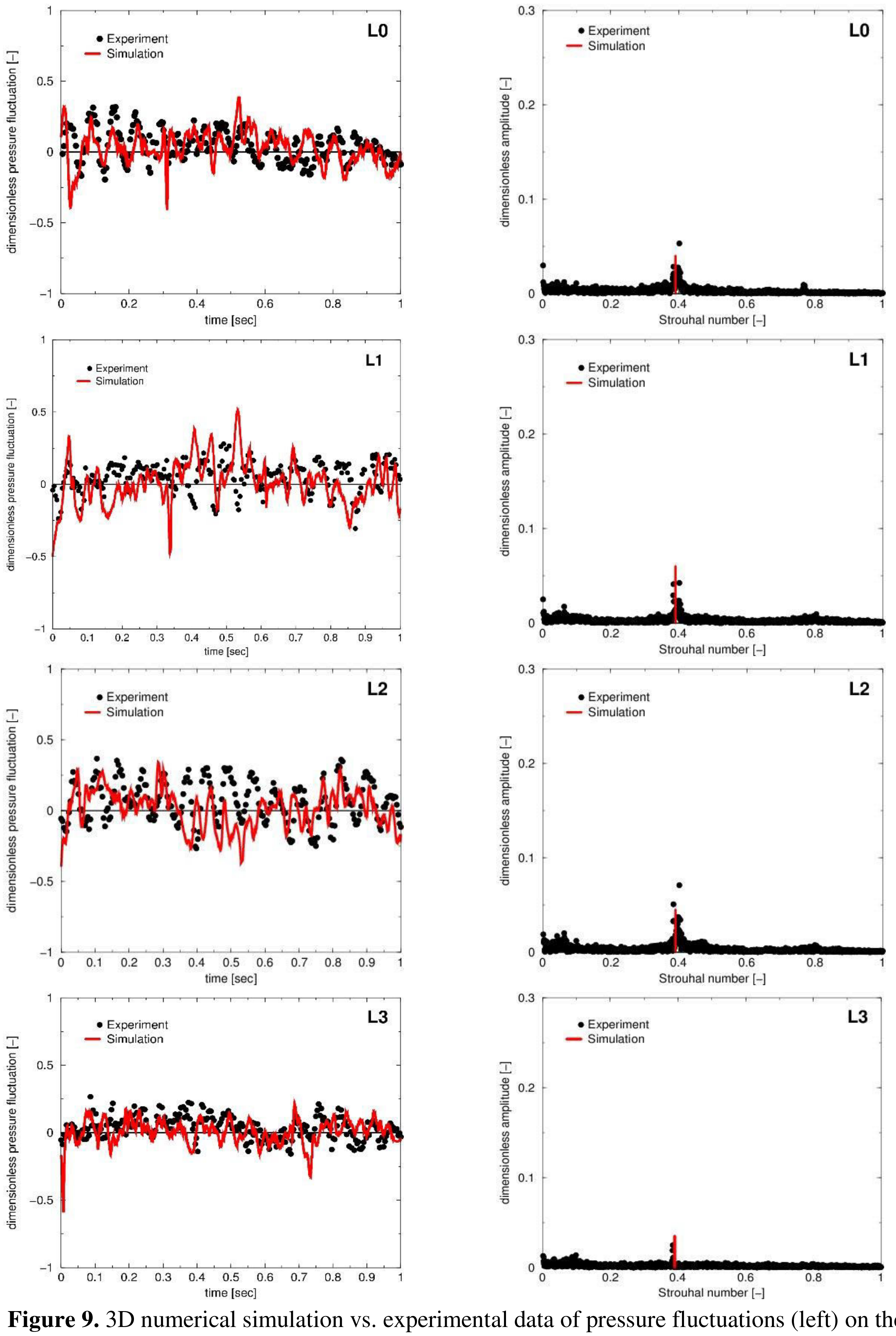

Figure 9. 3D numerical simulation vs. experimental data of pressure fluctuations (left) on the cone wall and associated Fourier spectra (right), with diaphragm d = 0.113 m, respectively.

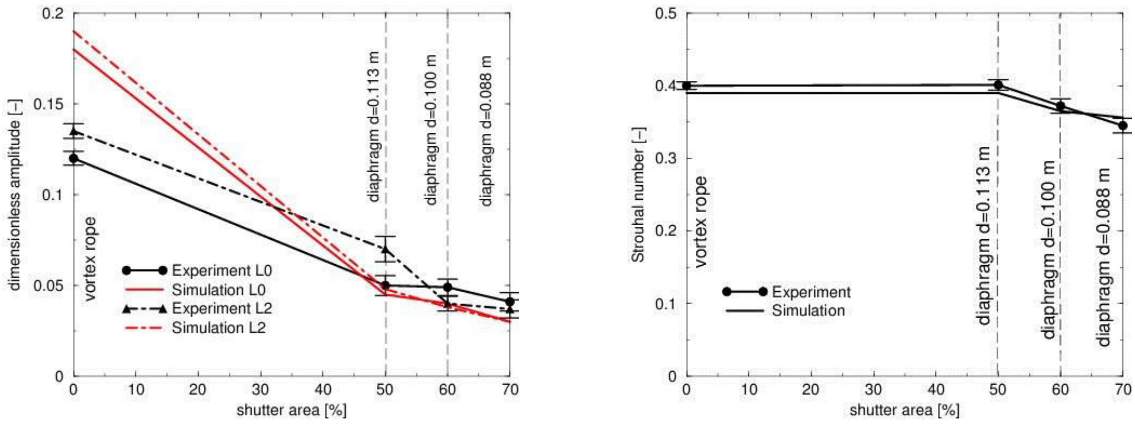

Figure 10. Dimensionless amplitude (left) and Strouhal number (right) versus shutter area from the cone outlet.

Figure 10 shows the global values of the amplitude and Strouhal number, respectively for L0, L2 levels for all studied cases. First off all can be see an overestimated amplitude value in the case without diaphragm (vortex rope), between experiment and numerical simulation. The comparison for the cases with diaphragm shows a better validation. It is obviously the decreasing of amplitude when the diaphragm is employed. The Strouhal number have almost a constant value for all cases and also can be observed a good validation between experiment and simulation.

5. Conclusions

The paper introduces a new method for mitigating the swirling flow with helical vortex from conical diffuser of hydraulic turbine. The method involves the development of a progressive and controlled throttling (shutter), of the flow cross section at the bottom of the conical diffuser. The adjustable cross section is made on the basis of the shutter-opening of circular diaphragms, while maintaining in four positions (d = 0, 0.113, 0.100, 0.88 m), the circular cross-sectional shape, centered in the axis of the cone. The pressure coefficient along to the cone wall is determined based on experimental data. Also, the pressure fluctuation on the wall at four levels is obtained. Next, full 3D unsteady numerical simulations with and without diaphragm were performed for all four cases in order to assess the performances of the new passive control method. The numerical results are compared against experimental data. As a result, a good agreement between numerical results and experimental data is obtained. Consequently, unsteady pressure pulsations and kinetic-to-potential energy recovery were found, in order to evaluate the draft tube cone efficiency using the passive method with diaphragm. The pressure recovery at level L2 of the cone wall is practically doubled and the amplitude of the unsteady pressure signals associated to the self-induced instability are mitigated up to 65% while the frequency remains unchanged. Also, the improvement in kinetic-to-potential energy recovery is important, while the hydraulic losses increases. As a solution, the operating of a real turbine which use this method needs to be performed with a compromise between mitigation of pressure pulsations and kinetic-to potential recovery energy with minimum hydraulic losses. According to Fig. 7 are two operating fields, one between zero diaphragm opening up to 50% opening, respectively, where is no kinetic-to- potential energy recovery in the cone, with minimum hydraulic losses. The 2nd field is between 50% to 70% diaphragm opening, where the kinetic-to-potential recovery energy increase up to 80% and the hydraulic losses increases up to 60%. Also, in the 2nd field the decreases in amplitude of pressure pulsations is the highest. We recommend the operation in the area of 40% - 50% opening of diaphragm, where is a good compromise between energetically and dynamically components.

Consequently, in our opinion, the above conclusions recommend the diaphragm method to be considered for either new or refurbished hydraulic turbines to improve both efficiency and safety of the operation far from the best efficiency point.

Acknowledgments

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