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REDESIGNED ADJUSTABLE DIAPHRAGM FOR CONTROLLING AND MITIGATING THE SWIRLING FLOW INSTABILITIES FROM THE CONICAL DIFFUSER OF HYDRAULIC TURBINES

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ABSTRACT

Currently, the energy market requires the use of renewable energy sources, especially photovoltaic and wind. During the year, these energy sources have large fluctuations. The machines that can compensate for the energy fluctuations given by fluctuating sources are hydraulic turbines. Thus, they end up operating at points other than those for which they were designed, with negative consequences for the operation of the machine. One of the consequences of operating hydraulic turbines at discharges other than the nominal one (for example, the Francis turbine) is the pressure pulsations that appear due to the self-induced instability generated by the swirling flow, in the conical diffuser. One of the modern techniques for mitigating the pressure pulsations that appear at partial discharges in hydraulic turbines (especially those with fixed blades) is the adjustable diaphragm introduced downstream of the conical diffuser. It has previously been shown that this technique can mitigate the swirling flow and the associated pressure fluctuations but can lead to an increase in hydraulic losses. Thus, the present work shows that by redesigning this diaphragm, a compromise can be found between the mitigation of pressure pulsations (dynamic component) and hydraulic losses, respectively the pressure recovery in the conical diffuser (energy component). The results obtained through 3D numerical simulation show that the pressure pulsations have a significant decrease, and the hydraulic losses are minimal when the turbine operates at 70% of the nominal flow.

Keyword: adjustable diaphragm; conical diffuser; hydraulic losses; hydraulic turbine; pressure pulsation; swirling flow

NOMENCLATURE

A_d	[m^2]	diaphragm interior area
A_o	[m^2]	test section outlet area
D_t	[m]	reference diameter from the throat of the test section,
V_t	[m/s]	reference velocity from the throat of convergent-divergent test section
d	[m]	diaphragm interior diameter
f	[Hz]	dominant frequency
ρ	[kg/m^3]	density
sh_r	[%]	diaphragm device shutter area
c_p	[-]	pressure recovery coefficient
$\overline{p}_{L0}$	[kPa]	mean pressure from the test section throat
$\overline{p}$	[kPa]	mean pressure
Q	[l/s]	nominal flow
h_p	[m]	hydraulic losses
p_{in}	[kPa]	pressure from the inlet test section
p_{out}	[kPa]	pressure from the outlet of the test section

1. INTRODUCTION

The acute problem faced by hydraulic turbines (especially those with fixed blades such as the Francis turbine), which are imposed by the energy market to operate at operating points far from the optimal one, is due to fluctuating energies such as photovoltaic and wind [1, 2]. This type of operation (at part load) leads to the appearance of the helical vortex or vortex rope phenomenon, in the conical diffuser of the turbine [3]. Vortex rope is the main phenomenon that produces severe pressure fluctuations in the conical diffuser leading over time to cracks or breaks of the runner blades, cracks in the bearings or shaft, tearing of the ogive, etc [4].

To eliminate or to mitigate the instabilities from the conical diffuser, different techniques have been implemented in hydraulic turbines [5-8]. These methods lead to reducing the pressure pulsations over a narrow regime, but they are not effective or even increase the unwanted effects [9-11].

The present paper is focusing on the modern adjustable diaphragm passive control technique, to mitigate the instabilities associated to the vortex rope from the conical diffuser of hydraulic turbines (e.g., Francis turbine), operated at part load conditions. It is mentioned that the proof of concept of the adjustable diaphragm control technique was presented previously [12, 13]. The results obtained previously clearly show that the mitigation of the swirling flow instabilities is directly proportional to the narrowing of the diaphragm surface towards the centre of the conical diffuser axis (the diaphragm is mounted at the outlet of the conical diffuser). Anyway, it is shown that the hydraulic losses have an increase since the shutter area of the diaphragm is close to the diffuser centre and exist a small area where the diaphragm can work without increasing the hydraulic losses but still mitigate the amplitudes associated to the instabilities of the vortex rope.

This paper presents a 3D numerical analysis of pressure filed from dynamic and energetic point of view using a redesigned adjustable diaphragm called IRiS device (Fig. 1). The second section presents the problem setup for numerical analysis, including the computational domain and boundary conditions. Section 3 analyzes the flow field quantifying the hydraulic losses and the unsteady pressure field with and without IRiS device. The conclusions are summarized in last section.

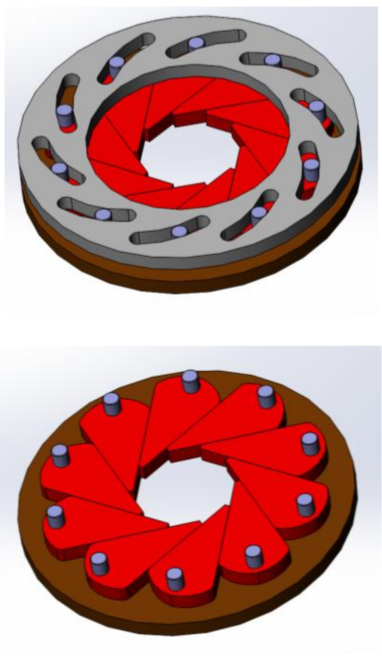

Figure 1. Sketch of the IRiS device.

2. COMPUTATIONAL DOMAIN AND BOUNDARY CONDITIONS

The computational domain corresponds to convergent-divergent part of the swirling flow apparatus developed at Politehnica University Timisoara (UPT) [14]. The convergent section is bordered by the annular inlet section and the throat (Fig. 2). The annular inlet section is considered just downstream to the runner blades. The divergent section includes a discharge cone with semi-angle of 8.5° similar to FLINDT project [15] and a pipe. Three values of the IRiS interior diameter of d = 0.143, 0.134, 0.124 m are considered in this numerical study. According to Figure 2 the IRiS device is located at the outlet of the conical diffuser. Table 1, shows the areas ratio between IRiS interior area and outlet test section area.

Table 1.

IRiS interior diameter *d* [m]	IRiS interior area A_d [m^2]	Test section outlet area A_o [m^2]	Shutter ratio sh_r [%]
0.143	0.016	0.02	20
0.134	0.014	0.02	30
0.124	0.012	0.02	40

The computational domain previously presented in [12] is presented in Fig. 2. A mixed mesh with approximately 2.8M cells is generated on each computational domain with and without IRiS, Fig. 3. The most distorted element has an Equisize skew value of 0.87.

Boundary conditions imposed for each case uses a velocity profile at the inlet, and the outflow condition on the outlet section. The inflow boundary conditions are obtained computing the flow upstream of the numerical domain in our previous research work. As a result, the inlet velocity profiles (axial, radial and circumferential velocity components) 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. Figure 4, shows the velocity profiles from the inlet test section [12]. 3D unsteady numerical simulations with and without IRiS were performed using the Ansys FLUENT 2023 R2 software in order to assess the new approach.

For the numerical setup it was used k-ω GEKO turbulence model. This turbulence model is relatively new introduced to Ansys FLUENT and captures more accurate the flow specifics to hydraulic machines, [17]. The advantage of this turbulence model is that it has enough flexibility to cover a wide range of applications. The model provides free parameters that you can adjust for specific types of applications, without negative impact on the basic calibration of the model.

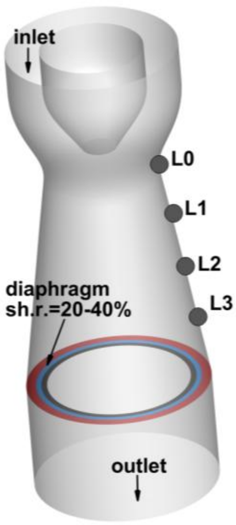

Figure 2. 3D computational domain of the case with diaphragm.

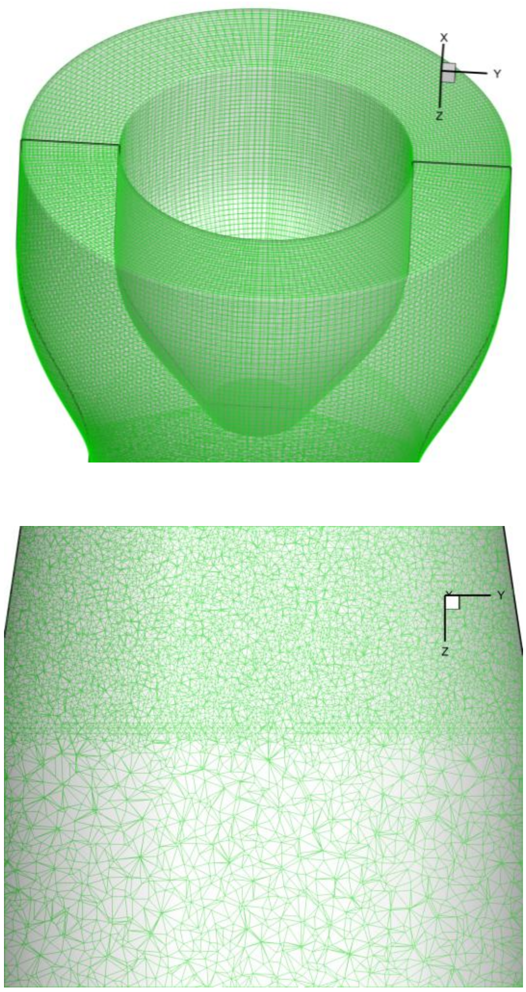

Figure 3. Detail of mesh domain. Inlet of the domain (up) and outlet (down).

This is a tool for model optimization but requires a proper understanding of the impact of these coefficients to avoid mistuning. It is important to emphasize that the model has strong defaults, so one can also apply the model without any fine-tuning, and one should make sure that any tuning is supported by high-quality experimental data.

For this study the default value of the turbulence parameters was applied. The time step applied for the unsteady simulation for all the investigated cases was t = 0.002 s. For our investigation, 5000 time steps were employed for each case, corresponding to a flow time of 10 seconds, for obtaining a stable flow structure. All numerical solutions were converged down to residuals as low as 10^{-4}. Pressure monitors denoted L0…L3 have been placed on 4 levels. The axial distance between two consecutive pressure taps located on the cone wall is 50 mm.

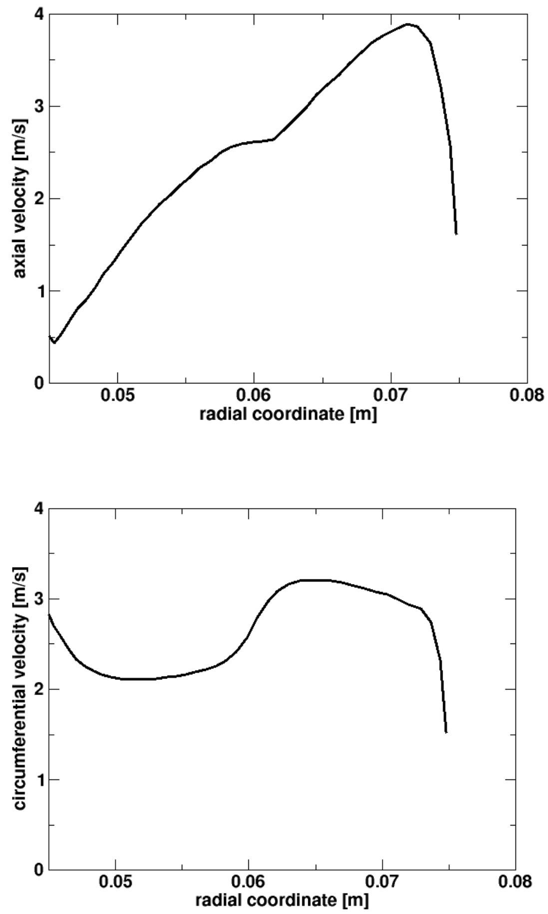

Figure 4. Velocity profiles from the inlet test section [12].

3. NUMERICAL RESULTS

3.1. Unsteady pressure analysis

Numerical simulations for turbulent swirling flow in the test section from UPT, have been performed for 3 cases of the IRiS device and for the case without IRiS. Figures 5 to 8, shows a first set of numerical results of pressure iso-surface for each case. One can clearly observe that the helical vortex structure still evolves during the IRiS shuttering for all cases, but the corresponding pressure pulsations are mitigated due to the eccentricity is significantly reduced. The above statement is supported by unsteady analysis of pressure signals from the pressure monitors of the domain (Fig. 9).

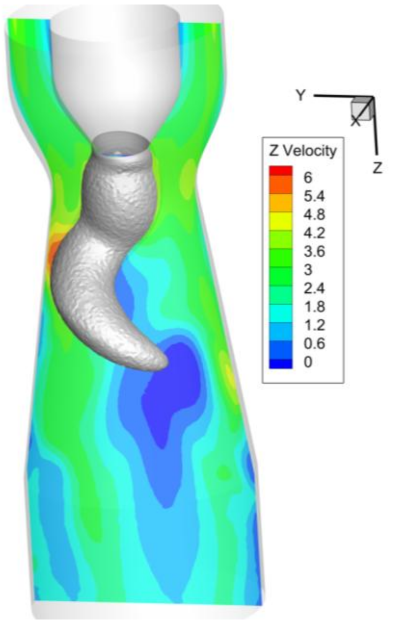

Figure 5. Pressure iso-surface for the case without diaphragm, t=10 s, p=37.700 Pa.

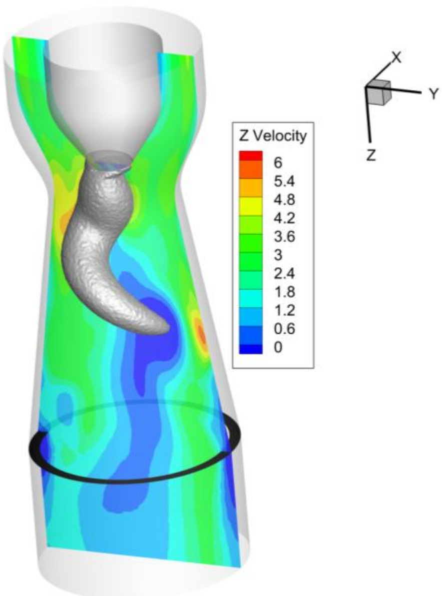

Figure 6. Pressure iso-surface for the case with diaphragm diameter of d = 0. 143 m, t=10 s, p=37.700 Pa.

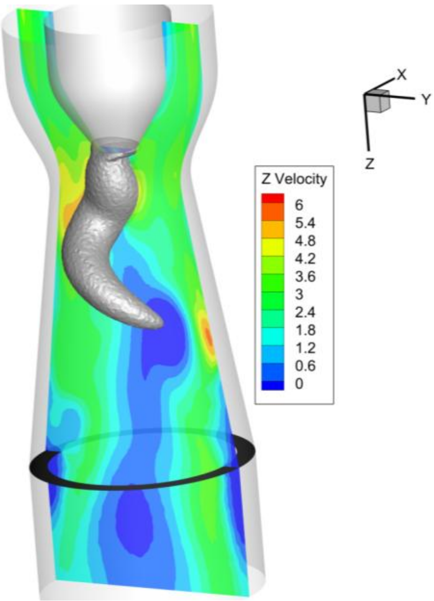

Figure 7. Pressure iso-surface for the case with diaphragm diameter of d = 0.134 m, t=10 s, p=37.700 Pa

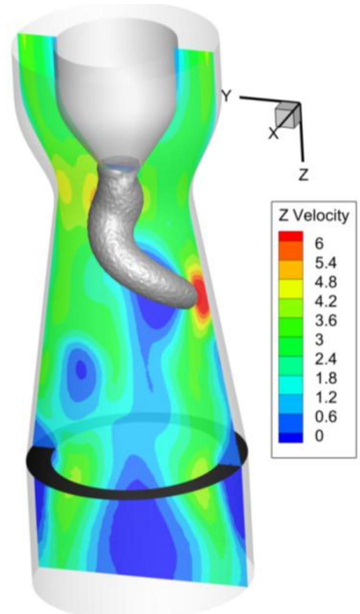

Figure 8. Pressure iso-surface for the case with diaphragm diameter of d = 0.124 m, t=10 s, p=37.700 Pa

For the investigated technique, in this paper, it is obviously that the IRiS device provides a significant drop in amplitude, while the frequency remains constant (f~14 Hz) for all cases. The shutter ratio sh_r = 20% and 40% provides the largest amplitude reduction (up to 60%), compared to the case without IRiS device. The sh_r = 30% has a smaller decrease (up to 32%), because at that configuration the vortex sheet occurs – the flow region formed between stagnant region and the main swirling flow [3].

We conclude that the passive method presented in this paper, has the potential to effectively mitigate the pressure fluctuations in decelerated swirling flow with precessing helical vortex.

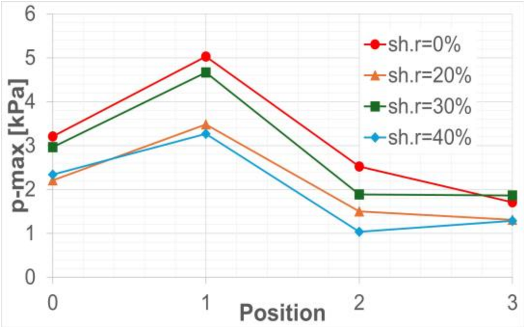

Figure 9. Pressure amplitudes corresponding to pressure taps from the test section domain.

3.2. Mean pressure analysis

The main purpose of the conical diffuser of hydraulic turbines is to convert as much as possible the kinetic energy at the runner outlet into pressure potential energy with minimum hydraulic losses.

This energy conversion is expressed by the wall pressure recovery coefficient c_p, which it is given in dimensionless form by Eq. (2),

$$c_p = \frac{\overline{p} - \overline{p}_{L0}}{\rho \frac{V_t^2}{2}} where V_t = \frac{Q}{\pi \frac{D_t^2}{4}} \qquad (1)$$

where $\overline{p}_{L0}$ is the mean pressure on the wall at L0 level,$\overline{p}$ is the mean pressure measured downstream on the cone wall, $\rho = 998$ kg/m^3 is the water density, V_t is the bulk velocity in the throat and $D_t = 2R_t = 0.1$ [m] is the throat diameter corresponding to L0 level, Q is the nominal flow. The distribution of the pressure coefficient c_p [-] along the cone wall is plotted in Figure 10. The wall pressure recovery in the first part of the cone up to L1 level is negligible if it uses the diaphragm. In contrast, a significant improvement of the wall pressure recovery is obtained downstream to L1 level delaying the separation flow on the wall. For instance, the pressure recovery coefficient (at the wall) is increased up to 30% in the middle of the conical diffuser (on L2 level). 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. It is expected when the pressure recovery on the cone wall is improved it will have diminished additional hydraulic losses associated to the vortex rope.

However, the hydraulic losses increase on the cone once the diaphragm obstructs the flow more and more. The hydraulic losses cannot be correctly quantified based on experimental data available.

Tănasă et al. [16], reveal that the hydraulic losses are less than half of the value got for the case with vortex rope up to the relative shutter area of 50% and no larger than twice up to the relative shutter area ratio of 70%, respectively.

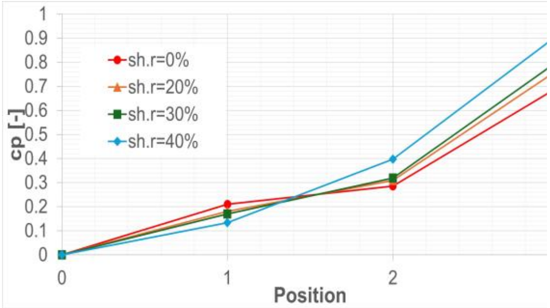

Figure 10. Pressure recovery coefficients vs. axial coordinate.

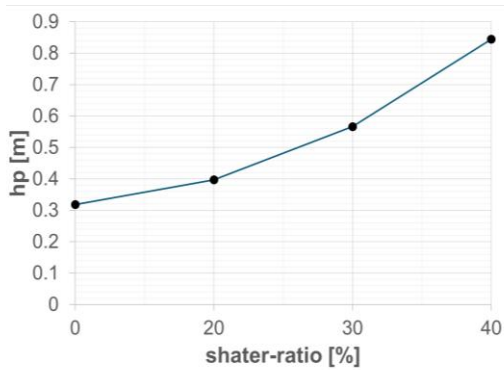

Figure 11. Hydraulic losses versus shutter ratio

Figure 11 shows the loss coefficient defined as:

$$hp = \frac{p_{in} - p_{out}}{\rho g}[m] \qquad (2)$$

We examine the *hydraulic losses* variation with respect to the diaphragm shutter ratio. One can see from Fig. 11 that the hydraulic losses reach the maximum values for the highest shutter ratio of 40%. The evolution of the losses 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 pressure recovery coefficient at the wall. It is obvious when throttling the flow at the outlet of the cone, using the IRiS device, the hydraulic losses increase, but also the pressure recovery increase.

4. CONCLUSIONS

The paper reintroduces the concept with adjustable diaphragm (as IRiS device), for mitigating the swirling flow instabilities from the conical diffuser of hydraulic turbines operated at part load conditions. Full 3D unsteady numerical simulations with and without IRiS were performed in order to assess the dynamic and energy recovery performances of the concept. The numerical results clearly show that the helical vortex evolves in a weaker structure when the IRiS is switched on. As a result, the amplitude of the unsteady pressure signals associated to the helical vortex are mitigated up to 60% in the amplitude with constant frequency in all cases. The evolution of the losses 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 pressure recovery coefficient at the wall. It is obvious when throttling the flow at the outlet of the cone, using the IRiS device, the hydraulic losses increase but also the pressure recovery increase. It is recommended to operate the IRiS device when the turbine operates over a wide range, far from the best efficiency point (BEP), so that hydraulic losses are minimal with high energy recovery and small pressure pulsations.

Consequently, in our opinion, the above conclusions recommend the IRiS device 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.

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