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Connection between inlet velocity field and diffuser flow instability

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Abstract

Simulations of the flow in a straight diffuser were carried out for different inlet velocity profiles equivalent to the real turbine operating points. Frequencies and magnitudes of pressure pulsations of the rotating vortex rope were recorded. Two types of vortices were detected. The first one is a thin vortex with higher frequency and relatively low pressure amplitudes corresponding to higher part load turbine operation. For lower loads, further from best efficiency point, a huge vortex rotating on higher diameter appears, which is accompanied by massive pressure pulsations, but with lower frequency. Connections of these two regimes with inlet circumferential velocity component are discussed in the paper.

Keywords: instability, diffuser, vortex rope, hydraulic turbine

1. Introduction

Hydraulic turbines, especially Francis turbines, exhibit instability of the flow in the xit diffuser (i.e. in draft tube in hydraulic machine terminology) downstream of the runner for part load operation, it means for lower flow rates. This instability has a form of precessing helical vortex filament. Cavitation occurs within the vortex core if the pressure drops below the level of vapor pressure. This phenomenon is usually referred in literature as a *cavitating vortex rope*, see Fig. 1. It is a significant source of pressure pulsations, vibrations and noise. The physical mechanism for its occurence is associated with vortex breakdown of the swirling flow,

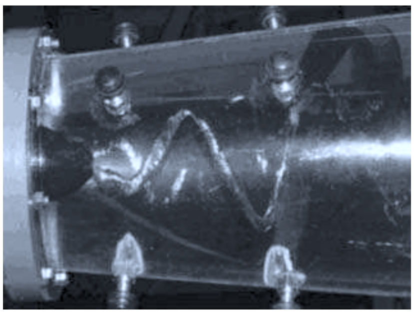

Fig. 1. Vortex rope in a diffuser [21]

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which was studied first in tip vortex flows over wings [18]. Further studies, mainly confined to swirling flow in pipes [4, 7, 10], were seeking deeper insight into physics of this phenomenon. Theories explaining vortex breakdown as an instability of the flow or upstream propagation of internal waves similar to transition from supercritical to subcritical regime in open channel flows were developed. Unfortunately none of them provided full clarification of this complex flow. Overview of the past research over a more than thirty year period can be found in a paper [14]. Focus on vortex breakdown in hydraulic machine diffusers was evoked by papers [16, 17, 8, 9, 25]. Qualitatively large step was made by development of CFD techniques and improved measurement methods (PIV) in the nineties. Computational simulations further expanded the possibilities to investigate this problem, especially using parametric modeling [26, 23, 24, 5]. Prelimery numerical experiments also appeared to find some link between inlet velocity profile and vortex rope origin [27], but this study was empirically driven. Detailed measurements of the internal flowfield of a diffuser flow with vortex breakdown recently appeared [12], but the lack of measured inlet boundary conditions disqualifies this study from using in analytical or computational studies of the problem.

Quite extensive theoretical, experimental and computational research has been carried out since the first observing and description of vortex breakdown in 1957, but we still lack deeper understanding of all influences on origin of this phenomena and their relations to vortex rope properties (frequency of the precessing motion, helix shape, pressure amplitudes, etc.). Equally we miss analytical description of the mechanism, which triggers the instability. Attempts for analytical explanation of vortex rope behavior are reported in [1, 6, 13, 19, 20].

2. Inlet velocity field

2.1. Description of the inlet velocity field

Flow leaving the hydraulic turbine runner has significant circumferential velocity component especially for operating points off the optimum. Circumferential velocity induces, by centrifugal force, low pressure zone along the draft tube axis. This pressure drop is further amplified by decreasing velocity in the downstream direction, which is caused by the diffuser flow. All these effects result in severe adverse pressure gradient accompanied by axial flow reversal. Strong coupling between axial and circumferential velocity components was noted first by Batchelor [3].

Although velocity distributions are different for different turbines, they always posses common feature: vorticity concentration along the draft tube axis. Vortex filament is formed, which is quite sensitive to disturbances. Eventually it looses its stability and transforms into a helical vortex.

2.2. Decomposition of the inlet velocity field

An extensive research has been performed on FLINDT turbine (Francis turbine with specific speed $n_q = 92$, see Fig. 2) and draft tube at EPFL Lausanne [2, 5]. Research included detail measurements of the velocity field on draft tube inlet using Laser Doppler Anemometry (LDA). Susan-Resiga et al. [28] carried out analysis of the flow downstream the FLINDT runner for operating points near the best efficiency point. Principle idea they took was decomposition of the velocity field into three distinct flow types, each characterized by axial and circumferential velocity component. The first flow type (index “0”) is purely axial flow and solid body rotation, the second one (index “1”) is a counter-rotating Batchelor vortex with co-flowing axial stream

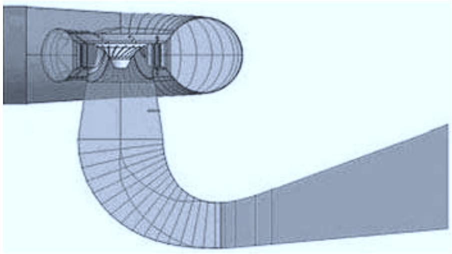


Fig. 2. FLINDT Francis turbine with elbow draft tube [5]

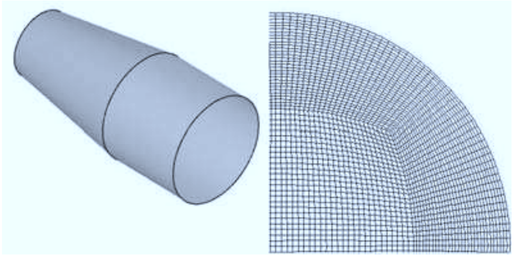


Fig. 3. Simplified straight draft tube (the outlet pipe is shortened in the picture) and cross-section of the computational mesh (only a quarter is displayed)

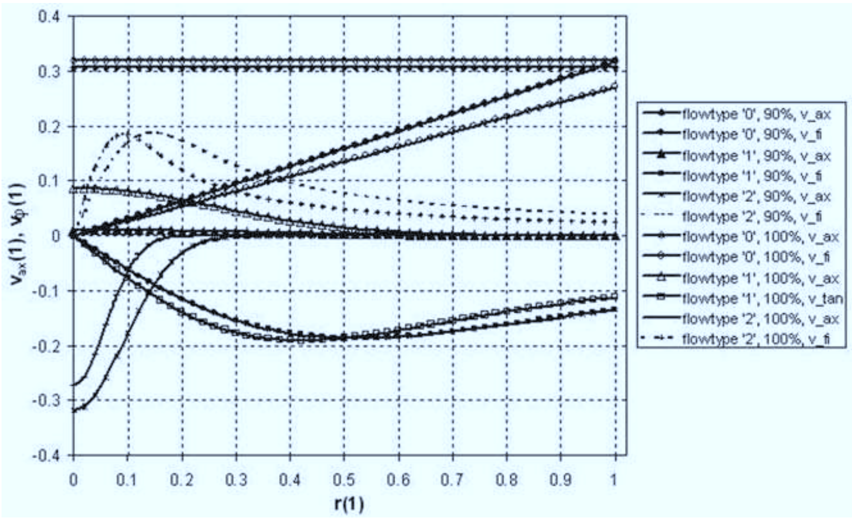


Fig. 4. Decomposition of the inlet velocity field into three basic flow types

and the third one (index "2") is a co-rotating Batchelor vortex with counter-flowing axial stream. Complete velocity field is a summation of all these flow types, see eqs. (1), (2) and Figs. 4, 5.

$$v_{ax} = U_0 + U_1 \exp\left(-\frac{r^2}{R_1^2}\right) + U_2 \exp\left(-\frac{r^2}{R_2^2}\right), \tag{1}$$

$$v_\varphi = \Omega_0 r + \Omega_1 \frac{R_1^2}{r}\left[1 - \exp\left(\frac{r^2}{R_1^2}\right)\right] + \Omega_2 \frac{R_2^2}{r}\left[1 - \exp\left(\frac{r^2}{R_2^2}\right)\right]. \tag{2}$$

Parameters U_0, U_1, U_2, Ω_0, Ω_1, Ω_2, R_1, R_2 were obtained by fitting the experimental data for respective velocity profiles.

Only the best efficiency point (BEP or 100 % discharge) and two operating points with lower discharges (90 % and 70 % of BEP discharge) are taken into account in this paper. Unfortunately the point furthest from BEP (70 % of the optimum discharge) cannot be described by equations (1), (2), but only by the raw measured data. Velocity profiles in non-dimensional form are plotted in Fig. 4.

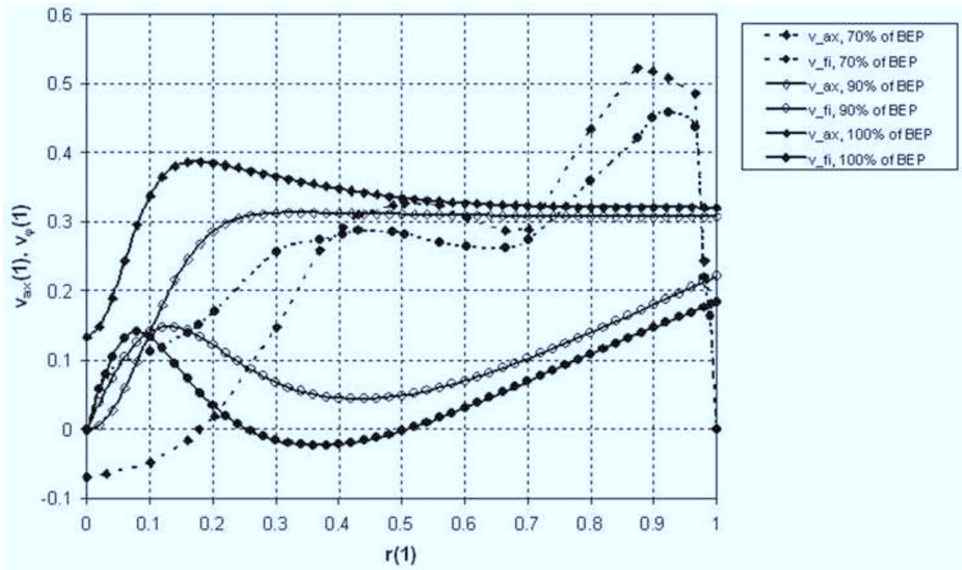


Fig. 5. Inlet velocity profiles for three operating points

3. Computational modeling

3.1. Geometry

The real geometry of elbow draft tube was simplified and only a straight cone was modeled (Fig. 5), but the change of the diffuser cross-section in the downstream direction was respected. Half of the diffuser opening angle is 8.5 degrees and its length to diameter ratio is two. The diffuser continues as a straight pipe another seven diameters downstream to prevent ill posed outlet boundary condition. Whole domain consists of about two milion finite volumes.

3.2. Flow model

Flow was modeled using Fluent 6.3 commercial CFD code. Reynolds stress turbulence model was employed to capture the highly swirling flow. Its suitability for this complex flow was confirmed by previous study [24]. Unsteady simulations were performed as one-phase, i.e. cavitation was not included in the simulation. Velocity profiles from Fig. 5 were prescribed on the inlet whereas constant pressure (atmospheric pressure) was set at the outlet boundary condition. Turbulence quantities on the inlet were based on LDA measurements [15].

4. Results

4.1. Phenomenological description

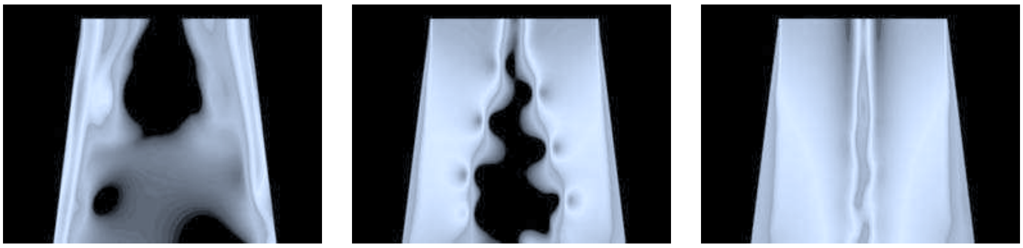
Going down with the discharge means increase of non-dimensional swirl number, which is a ratio of angular momentum flux and axial momentum flux

$$S_r = \frac{1}{R}\frac{\int r v_{ax} v_\varphi \,\mathrm{d}S}{\int v_{ax}^2 \,\mathrm{d}S}. \tag{3}$$

Table 1. Swirl numbers for inlet velocity profiles

70 % of BEP	90 % of BEP	100 % of BEP
0.22	0.15	0.11

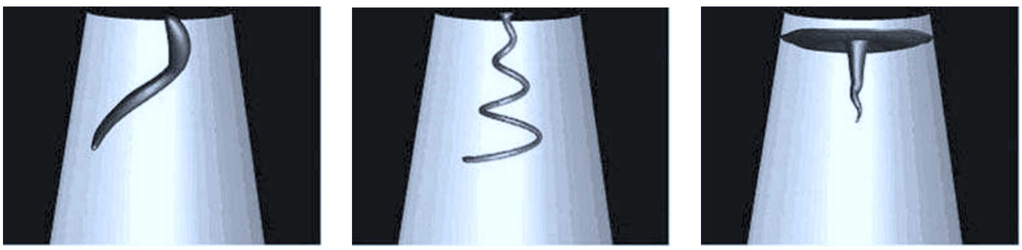
Higher swirl indicates more likely appearance of the vortex rope. Fig. 6 depicts axial velocity fields within the diffuser. The black spots inside denote zones of axial flow reversal. If compared with Fig. 5, it is observed that the reversals occur just in the regions of vortex rope. It can be stated that more massive swirl also means more massive backflow. There is no flow reversal for BEP. For 90 % of BEP discharge there is a region of backflow along the diffuser axis, through the vortex rope and the "corkscrew" domain. For even lower discharge (and higher swirl) the zone of flow reversal is only located along the vortex rope. This would suggest that there are two types of vortices with possibly different mechanisms of origin.



a) 70 % of BEP b) 90 % of BEP c) 100 % of BEP

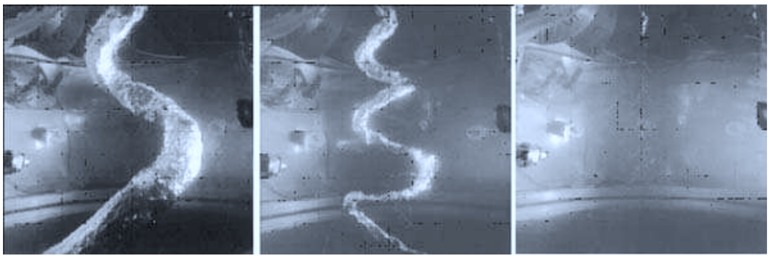
Fig. 6. Axial velocity reversal

One-phase computational simulations only allow identification of the vortex rope with contour of constant very low pressure, Fig. 7. Anyway the agreement with experimental visualizations is very good. Fig. 8 shows photos from observations of vortex rope on very similar turbine ($n_q = 82$) by Jacob [11]. Present computations confirm shapes of the vortex filament for off optimum operating points and moreover even slight instability for BEP. Relatively thin vortex rope of almost constant diameter exists for 90 % of BEP discharge, which is stretching three turns in the downstream direction. On the other hand both computation and experiment show huge vortex rope rotating near draft tube wall with high ascend of the helix for 70 % of BEP discharge. Unfortunately the computational simulation is not able to track the vortex rope further downstream because of excessive numerical dissipation.



a) 70 % of BEP discharge b) 90 % of BEP discharge c) 100 % of BEP discharge

Fig. 7. Vortex rope visualized by contour of constant pressure (computation)



a) 70 % of BEP discharge b) 90 % of BEP discharge c) 100 % of BEP discharge

Fig. 8. Vortex ropes experimentally visualized by cavitation [11]

4.2. Pulsations

It was noted earlier that vortex rope is a source of significant pressure pulsations, which might threaten to turbine operation. The dominant frequencies of the vortex rope rotation obtained from FFT of the diffuser wall pressure recorded during the unsteady simulation are depicted in Fig. 9. Experimentally measured value for non-cavitating flow was only accessible for the lowest discharge (70 % of BEP discharge) and is in excellent agreement with computation. Fig. 10 suggests increase of frequency of the vortex rope motion with increasing discharge, possibly with sudden jump around the BEP. On the other hand pressure pulsations demonstrate growth with decreasing discharge. The growth is more pronounced when going from 90 % to 70 % of BEP discharge.

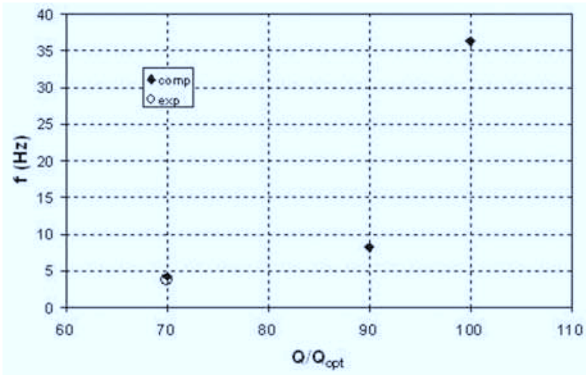


Fig. 9. Dominant frequencies of the pressure pulsations

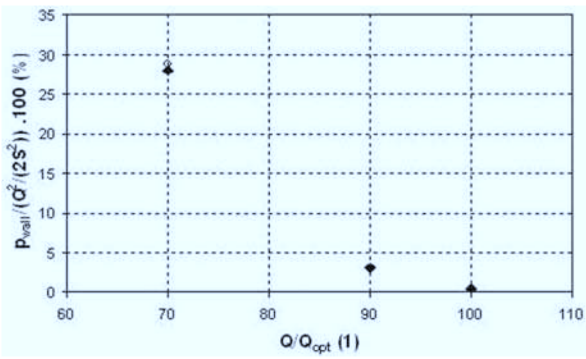


Fig. 10. Pressure pulsation amplitudes

5. Conclusion

Vortex rope occuring in the draft tube of Francis turbine was studied. The simplified straight diffuser simulations captured the unsteady behavior of the swirling flow and appearance of the vortex rope. Both qualitative and quantitative agreement between computations and experiment was achieved (11 % overestimation of the vortex rope rotating frequency, 2.5 % underestimation of the pressure amplitude). It is worth to stress that frequency and pressure amplitude prediction is of the same error as simulations of complete moving runner and draft tube in [5]. Present study confirms that only the draft tube alone and even straightened can only be taken into account when simulating the vortex rope motion. This underlines the importance of the inlet velocity field and the inlet diffuser opening angle on the flow stability. On the other hand rest of the draft tube (i.e. elbow and the outlet section) only have minor effect and their exclusion from computations does not influence overall behavior. Current paper also reports finding of two different types of the vortex ropes:

1. Thin vortex rope with higher rotating frequency, which is formed by Kelvin-Helmholtz instability on the interface between 2 counterflowing streams. This vortex rope forms for operating points close to best efficiency point, where significant minimum in circumferential velocity profile in the mid between axis and diffuser wall is present.
2. Thick vortex rope, which develops from the former type for lower discharges and high swirl rate. This type is characterized by massive pressure amplitudes.

It should also be emphasized that vortex rope instability causes significant drop of hydraulic efficiency, see [22]. It is obvious then that such runners that prevent unstable flow field should be designed. Therefore the future research will be focused on effect of the individual terms (i.e. the flow types) in eqs. (1), (2) on swirling flow instability and vortex rope behavior.

Acknowledgements

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