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Cavitating vortices in the guide vanes region related to the pump-turbine pumping mode rotating stall

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Abstract. Reversible pump-turbines used in Pumped Storage Power Plants are among the most cost-efficient solutions for storing and recovering large amount of energy in short time. Presented paper is focused on the pump-turbine pumping mode part-load instabilities, among them the rotating stall and the cavitating vortex in the distributor region. Rotating stall can be described as a periodic occurrence and decay of the recirculation zones in the distributor with its own rotational characteristics frequency. Unstable behaviour can result in high radial forces, high pressure fluctuations and local velocity fluctuations that can in some cases lead into the occurrence of the cavitating vortex in the distributor region, even though the distributor is located in the high pressure region. Computationally demanding calculations have been performed using commercial CFD code. Analysed results have been compared to the experimental data obtained in the ČKD Blansko Engineering hydraulic laboratory.

1. Introduction

To maintain the stability of the electrical grid, the pump-turbines need to switch between the generating and the pumping mode several times per day what expose them to extended operating under off-design conditions. Main instabilities in the part load pumping mode operating are the rotating stall in the distributor and the appearance of the cavitation in different pump-turbine regions. Rotating stall has been initially investigated in the axial and the centrifugal compressors [1, 2], much later also for the case of the pump-turbines [3, 4, 5]. Sometimes, the rotating stall is related to the positive slope of the performance curve also called hump zone [6] which is unstable and potentially dangerous operating region. If presented, it can lead into uncontrollable changing of the discharge through the machine and consequently strong vibrations. Operating under described conditions should be completely avoided. However, rotating stall can be present even if the slope on the performance curve is negative.

The intensity of the rotating stall in pump-turbines can vary. As shown several times experimentally and numerically [3, 6, 7] changing discharge and guide vane opening angle can lead into different number of the stalled cells and different rotating stall frequency. Various shapes of rotating stall influence pressure fluctuations, radial forces acting on the impeller as well as guide vanes vibrations and can have negative effects on the machine. Sometimes, when the rotating stall is very intensive, appearance of the cavitating vortex is possible in the distributor region as it was mentioned by [3]. However, the phenomenon have not been so far phenomenologically described in the literature.

2. Experimental data

Experimental measurements took place in ČKD Blansko Engineering hydraulic laboratory [7]. Investigations have been performed on the pump-turbine model. Additional to the standard performance measurement instrumentations, 8 pressure sensors have been distributed around vaneless space between the impeller and the guide vanes. One constant guide vane opening a_0 = 20 mm have been analysed. Rotational speed of the machine have been set to n = 1400 min^{-1}. Guide vane channel and vaneless space have been visually observed during the measurements.

The goal of the experimental setup was to measure low frequency pressure pulsations in pumping and generating mode. This paper will focus on the part load pumping mode only, where the phenomenon called rotating stall occurs together with related flow instabilities. Measurements have been done for whole part load regime, however, for the analysis, operating points at best efficiency (BEP) and at Q = 0,65 Q_{BEP} have been chosen.

Figure 1 presents comparison of the pulsations for the operating point at BEP and investigated point at Q = 0.65Q_{BEP}. It can be clearly observed that at the BEP, the flow is stable, therefore without pressure pulsations and instabilities. On the contrary, one can see periodic pressure pulsations at Q = 0.65 Q_{BEP}. Governing frequency of the low frequency pressure pulsations is f = 0.59 Hz, which corresponds to around 2.5% of the pump-turbine rotation frequency. Pressure measurements at mentioned operating point confirm the presence of the rotating stall.

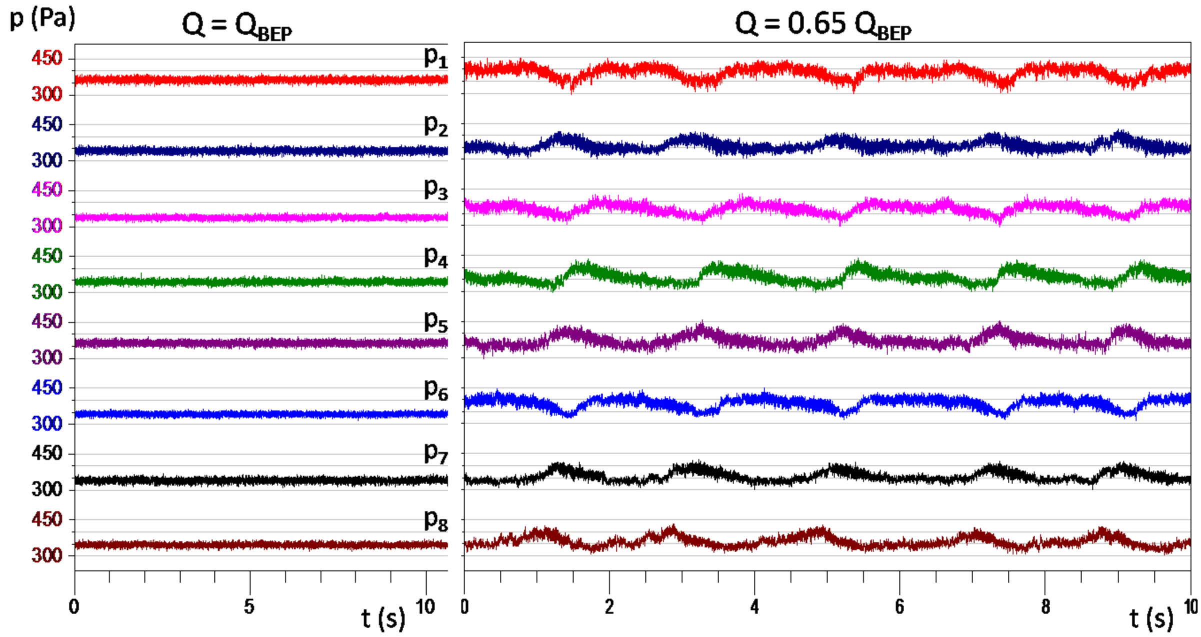

Figure 1. Pressure fluctuations at Q = Q_{BEP} (left) and Q = 0.65 Q_{BEP} (right)

Occasionally, during the pressure measurements at Q = 0.65 Q_{BEP}, the cavitating vortex have been observed in the distributor between the guide vanes, as seen on figure 2. Sometimes, there was one vortex, attached to the suction side of the guide vane (figure 2, left). At some other instants, the phenomenon have been observed as several separate, smaller cavitating vortices, as seen on figure 2, right. In both cases, the vortices occur only for a short time. It should be pointed out that cavitation in the distributor region is highly unusual due to very high pressure in the surrounding. Analysis of the cavitating vortex is the main subject of the presented paper and have been additionally analysed numerically.

Figure 2. Cavitating vortices in the distributor

3. Numerical setup

For the flow analysis, the numerical flow simulation (Computational Fluid Dynamics) software is nowadays the most common tool. It uses a set of Navier-Stokes equations to compute transport of mass and momentum in all parts of the computational domain. Commercial software has been used for the simulation. Transient simulation were performed using URANS equations and turbulence model based on k-ε model. Chosen time step corresponds to 2° of the impeller revolution.

Meshing of the domains has been done by commercial software, using structured and unstructured mesh. Total mesh contains of around 10 million cells, special attention has been put into meshing the distributor region, since this would be the place where the rotating stall occurs. Rotational speed of the impeller has been set to $n = 1400$ min^{-1}, guide vane opening has been set to $a_0 = 20$ mm. Around 20 revolutions of the impeller have been simulated, each fifth time step has been saved for the analysis. Steady results have been used as an initial solution for transient simulations.

4. Numerical results

Unsteady CFD analysis have been focused on the rotating stall parameters and related phenomena. Operating point at $Q = 0.65\ Q_{BEP}$ have been chosen for the comparison to the experiment. Three regions with high velocity have been found (figure 3) what corresponds to the experimental findings. Numerical rotating stall frequencies have been estimated to 0.5 Hz. Since the frequencies of the rotating stall are very low, more impeller revolutions should be simulated for more accurate frequency prediction. However, we can say that the phenomenon has been accurately described by using CFD and simple k-ε based turbulence model.

Cavitating vortex in the distributor channel have been occasionally observed during operating under rotating stall. The vortex is attached to the guide vanes and reaches the next upstream channel, as seen on figure 4. Constant Q-criterion has been used for the vortices representation. Estimated time of cavitating vortex appearance is around 0.011s what corresponds to around one quarter of the impeller revolution. The vortex appears and disappears several times on different positions around the distributor during the impeller revolution. The appearance time is very short, which means that the phenomenon is visually demanding to observe. The vortex appears when the discharge through the guide vane channel is partially blocked. Firstly, the blockage appears at the hub side and later also at the shroud side. The moment when the flow is still strong at the shroud side and blocked at the hub side is favourable for the occurrence of the cavitating vortex in the guide vane channel. Comparing experimental and numerical cavitating vortex (figure 2, figure 4) show very good agreement.

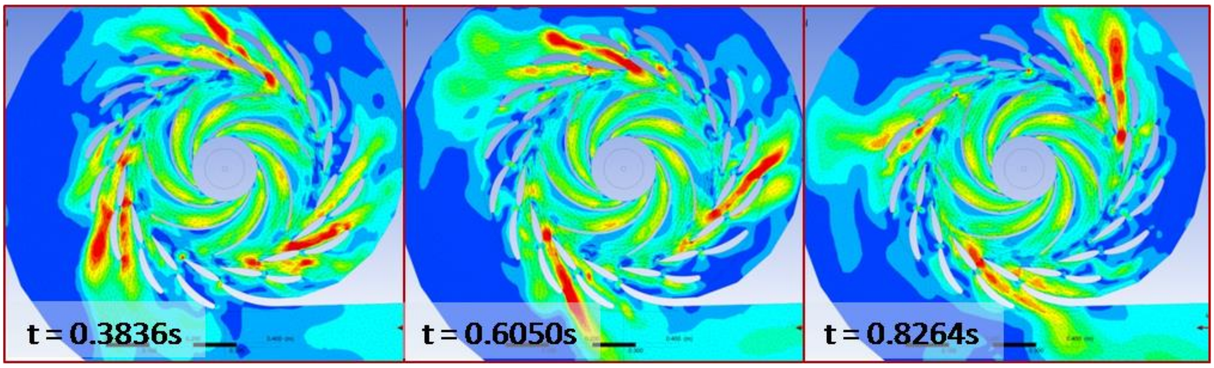

Figure 3. Meridional velocity time sequence on the mid-span

5. Conclusions

Rotating stall conditions have been simulated using CFD at the operating point Q = 0.65 Q_{BEP}. Results have been compared to the experiment. In both cases the same number of rotating stall cells (three) have been found. Similar rotating stall frequencies have been found experimentally and numerically. Cavitating vortices in the distributor region that have been observed during the experiment, have been successfully simulated by the CFD. The analysis explains the governing mechanisms of the cavitating vortex. Methodology can be used for pump-turbine impeller and guide vanes geometry optimisation. The findings will be used for a development of a new modern pump-turbine designed for wide range of operation.

6. References

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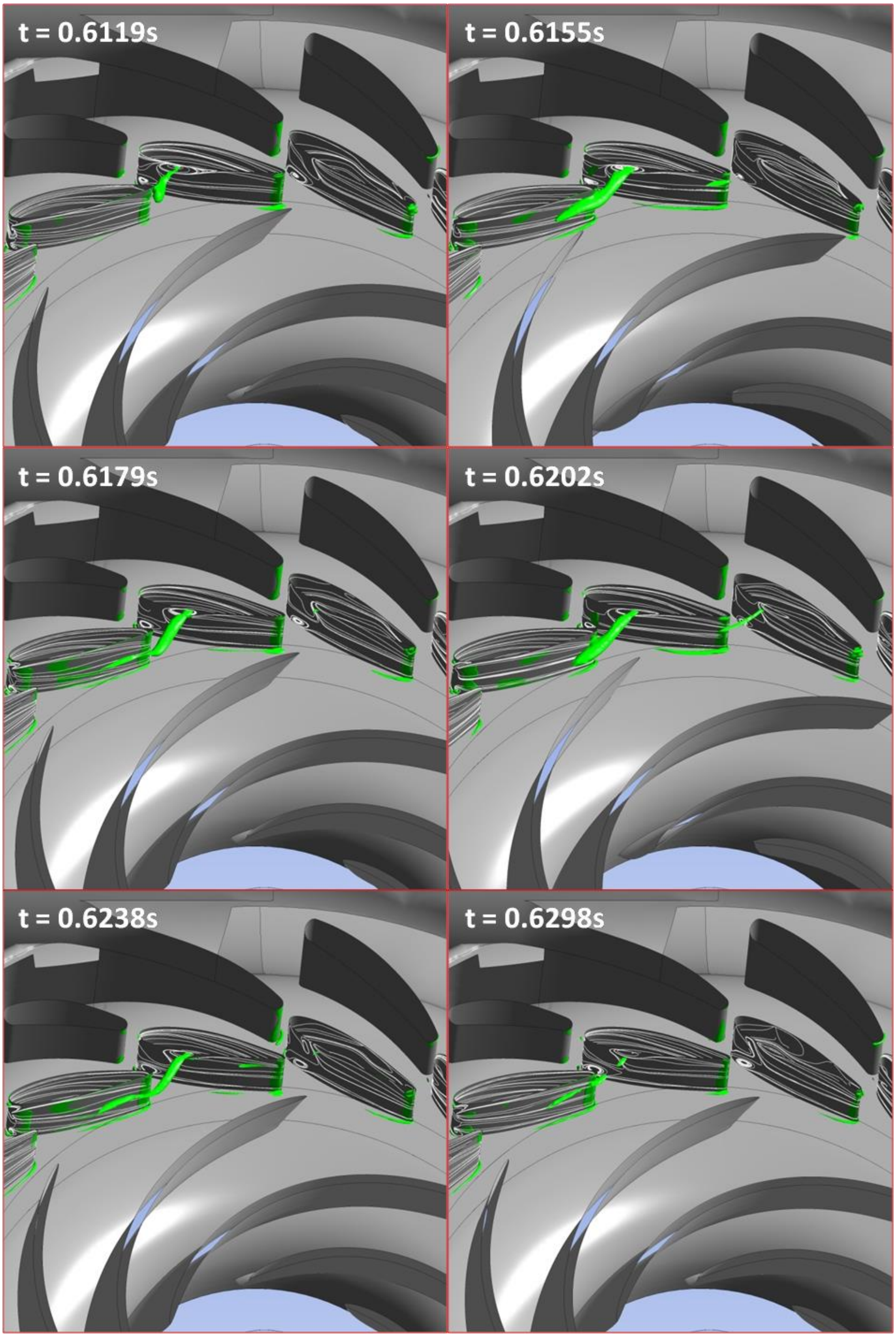

Figure 4. Cavitating vortex development - CFD analysis