Development trends in the field of reversible pump-turbines – Study of pumping and generating mode off-design conditions

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Abstract. The role of pumped storage power plants (PSP) in the electrical grid systems is changing in last years. Demands for switching from pumping to generating mode are becoming more and more frequent. Moreover, the operating ranges of the reversible pump-turbines used in PSP systems are getting wider in order to use PSP as a regulator and a stabilizer of the electrical grid. Main challenges in the pump-turbines development are hydraulic instabilities that occur in both pumping and generating modes. The paper focuses on the cavitation and the rotating stall that occurs at pumping mode partial load and can cause uncontrollable switching of the operating point, strong pressure oscillations and can damage the pump-turbine. Moreover, at the generating mode, the characteristics form an S-shaped curve in the turbine, turbine break and reverse pump quadrants. This S-shape leads to unstable behavior of the turbine when coupling to the grid or to surge transient phenomena in case of an emergency shutdown. The paper presents results obtained by Computational Fluid Dynamics (CFD) tools for both types of instabilities. Additionally, pumping mode rotating stall has been compared to the experiment performed in the hydraulic laboratory. Three rotating stall cells with rotational frequency 2.5% of nominal pump-turbine frequency have been identified at the operating point 65% of nominal discharge. Cavitating vortices related to the rotating stall have been found in the guide vanes region. S-shaped curve has been identified in the generating mode and reproduced by CFD. Unstable flow patterns have been analyzed at various unstable operating regimes and compared to the stable ones. Understanding the causes of the instabilities leads to the improved pump-turbine design that enable safer, more flexible and more reliable operating with less unwanted instabilities. Moreover, together with mechanical and manufacturing optimization, the acquired knowledge helped to develop pump-turbine with significantly wider operating ranges in pumping and generating modes.

1. Introduction
The demand for pumped storage power plants (PSP), using reversible Francis turbine, is increasing every year. One of the reasons is the growing number of weather-conditioned sources of energy such as wind and solar power plants. To provide a reliable electrical grid, the grid needs to include power plants that can balance the differences between demand and supply of the electricity. A PSP with reversible Francis runner that has a wide operating range and enables a fast transition from the generating to the pumping mode is very suitable for this task. Besides new PSP projects, refurbishments of the pump-turbine runners represent an important part of the market.
The development process of a new pump-turbine runner is related with several major challenges. The customer demands and final goals of the development process are the operation of the pump turbine from very low to maximum output in the generating mode and non-restricted operation in the pumping mode. To achieve that, the whole operating range should be free of hydraulic instabilities. An additional reason for the refurbishment is very often also the improvement of the total efficiency of the cycle. Development of the new runner with the expected reliability and performance must be supported by an effective cooperation between hydraulic and mechanical designers and by application of precise manufacturing technology.

Both, the generating and the pumping mode instabilities have been analyzed during the study in order to prepare the new runner design for a 2x325MW pump storage power plant Dlouhe Strane in Czech Republic, which will be able to operate from 0 - 100% output power [1] in pumping and generating mode. This paper presents different internal studies of the same problematics that was collected lately in order to combine knowledge about pump-turbine instabilities, analyze it and as a consequence, to develop the pump-turbine with eliminated or at least reduced instabilities and non-restricted operating range demanded by clients.

2. Hydraulic characteristics and instabilities
Switching between the pumping and the generating mode several times a day and a flexible operation demands a wide operating range for reversible pump-turbine, including operating under highly off-design regimes. Typical four quadrants characteristics (figure 1) represents discharge factor \( Q_{11} \) (\( Q_{11} = Q / D^2 H^{0.5} \cdot m^3/s \)) dependence on speed factor \( n_{11} \) (\( n_{11} = n D / H^{0.5} \cdot rpm \)) for different guide vanes openings. The efficiency, as well as the reliance on pump-turbines, depends on both, the pumping and the generating mode. Moreover, off-design operating areas, like start-ups and partial load operation, play a key role in the designing and the optimizing of the reversible machine.

![Figure 1. Typical 4-quadrant pump-turbine characteristics with marked unstable operating regions.](image)

The problematic S-shaped area close to the runaway curve can be observed in the generating mode for all guide vane openings on figure 1. The phenomenon has been intensively analyzed numerically and experimentally by [2], [3], [4], [5], [6]. Meanwhile, instability in pumping mode can be observed as a hump zone on the performance curve and has been lately studied by [7], [8], [9], [10]. Another possibility to present and describe the system instabilities is a net head characteristic curve. The instabilities are especially related to the slopes of the curves (figure 2). As explained by [6] and [11], the slope gives a necessary but not sufficient condition for the instability of the system. Additional
conditions are elasticity, inertia and the amount of energy dissipated in the system. Once the transferred energy is larger than the dissipated one, the system becomes self-excited.

- At pumping off-design operating mode, the criterion for the instability is defined by: \( \frac{dH}{dQ} > 0 \)
- For generating mode, the instability definition can be: \( \frac{dQ_{11}}{dn_{11}} > 0 \)

Negative slope in the generating mode means that the energy is transferred to the fluid and can cause instabilities in the system. On the other side, the positive slope in the pumping mode means that the energy, transferred to the fluid does not contribute to the head rise, but dissipates and potentially causes the rise of unsteady flow patterns such as flow separation, recirculation zones and vortices. If unstable formation rotates around the pump-turbine with its own characteristic frequency, the phenomenon is called rotating stall and it can appear in the pumping and in the generating mode as well.

![Figure 2](image)

**Figure 2.** Unstable operating region for pumping regime (left) and generating regime (right).

3. **Pumping mode rotating stall**

Cavitation and rotating stall are considered main hydraulic instabilities in the pumping mode operation. Cavitation in the pumping mode regime mostly occurs at the impeller leading edge, where local pressure drops below vaporization pressure. However, in combination with phenomena called rotating stall, it is possible that the cavitation occurs also in the high pressure distributor region.

Rotating stall is a phenomenon presented at partial load operation and has been investigated several times for the case of the pump-turbines [7], [8], [10]. If present, it can lead to uncontrollable changing of the discharge through the machine and consequently strong vibrations. The intensity of the rotating stall in pump-turbines can vary. As shown several times experimentally [12] and numerically [9] changing discharge and guide vane opening angle can lead into a 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. If the rotating stall is very intensive, the appearance of the cavitating vortex is possible in the distributor region. Operating under described conditions should be completely avoided. However, rotating stall can be present even if the slope on the performance curve is negative. Rotating stall has been investigated experimentally and numerically in order to propose a hydraulic design that would be free of hydraulic instabilities and would satisfy very demanding criteria of non-restricting operation.

3.1. **Experiment**

Experimental measurements took place in ČKD Blansko Engineering hydraulic laboratory in 2003 [12]. Additional to the standard performance measurement instrumentations, 8 pressure sensors have been distributed around the vaneless space between the impeller and the guide vanes. 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.
Measurements have been done for the whole part load regime, however, operating points at best efficiency (BEP) and at \( Q = 0.65 \, Q_{\text{BEP}} \) have been chosen for the analysis. Figure 3 (left) shows pressure fields around the distributor at \( Q = 0.65 \, Q_{\text{BEP}} \). Three pressure cells were formed and rotated around the distributor with the governing frequency of \( f = 0.59 \, \text{Hz} \), which corresponds to around 2.5\% of the pump-turbine rotation frequency. Measurements confirmed the presence of the rotating stall.

![Figure 3](image)

**Figure 3.** Pressure fields around distributor \( Q = 0.65 \, Q_{\text{BEP}} \) (left), cavitating vortices attached to guide vanes at different instants (right).

Occasionally, during the pressure measurements at \( Q = 0.65 \, Q_{\text{BEP}} \), the cavitating vortex was observed in the distributor between the guide vanes, as seen on figure 3 (right). Sometimes, there was one vortex, attached to the suction side of the guide vane (figure 3, up-right). At some other instants, the phenomenon was observed as several separate, smaller cavitating vortices (figure 3, down-right). In both cases, the vortices occurred only for a short time. It should be pointed out that cavitation in the distributor region is highly unusual due to high pressure in the surrounding.

3.2. **Numerical analysis**

Commercial CFD software was used for the numerical flow analysis. Navier-Stokes equations are basis to compute the transport of mass and momentum in all parts of the computational domain. Transient, single phase simulations were performed using URANS equations and turbulence model based on k-\( \varepsilon \) model. Turbulence model is essential for the successful reproduction of the complex phenomena such as rotating stall. The model should be robust enough to enable the convergence with wall functions that enable the exact prediction of the first flow separation on the guide vanes. Numerical domain contained around 10 million cells, special attention has been put to distributor region. Time step corresponds to 2\(^\circ\) of the impeller revolution, which seems to be a good compromise between the computational time and the quality of the obtained results. Around 20 revolutions of the impeller were simulated. Mass flow rate was imposed at the inlet of the domain and static pressure at the outlet. No-slip condition was applied on the solid walls. Unsteady CFD analysis was focused on the rotating stall parameters and related phenomena. Operating point at \( Q = 0.65 \, Q_{\text{BEP}} \) has been chosen for the comparison to the experiment. Three regions with high velocity have been found (figure 4) in between three cells of blocked discharge which corresponds to the experimental findings. Separation zone regions periodically occurred and disappeared at the guide vanes surfaces and caused backflow from stay vanes and even spiral case region. The detailed description of the rotating stall origins is given in [10]. 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.

Figure 4. Meridional velocity distribution during rotating stall occurrence (Q = 0.65 Q_{BEP}).

Cavitating vortex in the distributor channel was occasionally observed during operating under rotating stall. The vortex was attached to the guide vanes and has reached the next upstream channel, as seen on figure 5. Constant Q-criterion was used for the vortices representation. Estimated time of the cavitating vortex appearance was around 0.011s what corresponds to around one quarter of the impeller revolution. The vortex appeared and disappeared several times in different locations 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 on 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 favorable for the occurrence of the cavitating vortex in the guide vane channel. The comparison of the experimental and the numerical cavitating vortex (figure 3 - right, figure 5) shows very good agreement.

Figure 5. Cavitating vortex development (Q = 0.65 Q_{BEP}).
4. Generating mode rotating stall

Rotating stall in generating mode occurs in the S-shaped area in the characteristic close to runaway curve and causes the uncontrollable changing of the discharge through the machine and its speed. Since the coupling of the machine to the grid is done at runaway condition (T = 0), the rotating stall can make the coupling difficult. The demand for fast switching between pumping and generating mode of the pump-turbine is then not satisfied.

Not only the discharge and guide vane opening affect the character of the rotating stall: obviously, the mode of the turbine also plays its role. Characteristic frequency and number of stalled cells of generating mode differ from those of pumping mode. Also, the generating mode rotating stall affects the runner rather than the distributor region: the vortices occur primarily in runner channels.

During the development of a new runner, the available data were considered: first the experimental measurement from the hydraulic laboratory; second the numerical analysis of a different pump-turbine with similar specific speed. The numerical analysis was performed on geometry where there were difficulties with coupling to the grid.

4.1. Experiment

Generating mode instabilities investigation was part of the measurements performed in 2003 mentioned above [12]. On Figure 6 there is a record of pressure pulsations in the vaneless gap (left) and a low frequency pressure field animation based on these pulsations (right). The record is for model guide vane opening $a_0 = 26$ mm, because the low frequency pulsations were most noticeable for this opening.

![Figure 6](image)

**Figure 6.** Pressure pulsations in vaneless gap (left), low frequency pressure field (right).

The high frequency oscillations represent impeller rotation frequency multiplied by the number of guide vanes; the low frequency pulsations are caused by rotating stall and its governing frequency corresponds to around 68 % of the impeller rotation frequency. The pressure field has one separation zone and rotates in the same direction as the impeller. The amplitude of low pressure pulsation is approximately the same on all the pressure sensors and its value corresponds to approximately 25 % of measured net head value.
4.2. Numerical analysis
The transient, single phase numerical analysis using URANS equations and turbulence k-ε model was performed for a model dimension turbine. The time step corresponded to 2° of the impeller revolution. Computational domain consisted of around 10 million cells, special attention was paid to the distributor region and the runner. Mass Flow Rate boundary condition was prescribed at the inlet of the domain (spiral case inlet) and Opening boundary condition (Relative Pressure) at the draft tube outlet. No-slip condition was applied on the solid walls. Both rotor-stator interfaces were set to Transient Rotor Stator option. Operating point at \( Q = 75 \text{ l/s} \) was chosen for the analysis and the model guide vane opening was set to \( a_0 = 12 \text{ mm} \). Small guide vane opening and the discharge close to runaway speed was chosen with respect to the conditions common for machine-to-grid coupling. The instabilities during the start of pump-turbines were recently studied in [6]. The presented numerical simulation confirms the findings of [14]: the generating mode rotating stall occurred only after about 12 impeller revolutions. Therefore at least 25 revolutions were necessary to get reasonable results. Rotating stall with one separation zone was found. From figure 7 it is obvious that the pressure field rotates in the same direction as the impeller. The governing frequency of the rotating stall determined by means of fast Fourier transform is 21 Hz which corresponds to 84 % of impeller rotational frequency. The high frequency pressure pulsations represent impeller rotation frequency multiplied by the number of guide vanes. The amplitude of low frequency pressure pulsation is approximately same on all the pressure sensors and its value corresponds to around 35 % of computed net head value.

![Figure 7. Pressure pulsations in vaneless gap (left), low frequency pressure field (right); \( Q = 75 \text{ l/s} \).](image)

The flow patterns in the runner have been visualized by streamlines. Three different time steps are presented for hub-to-shroud span 0,15; 0,5 and 0,85 on figures 8, 9 and 10 respectively. It is clear that the rotating stall intensity differs with the span. The closer to the hub the more profound rotating stall appears. There are flow separations and related vortices in basically all the runner channels for span 0,15 (figure 8) whereas close to the shroud the flow looks much steadier (figure 10). From this point of view, the mid-span appears to be most useful for the observation of rotating stall and the movement of stalled
cells. The position of rotating stall vortices in the runner channel is discussed in [2] and [13], the influence of runner geometry has been discussed by [15].

Figure 8. Streamlines in the runner; span 0.15.

Figure 9. Streamlines in the runner; span 0.5.

Figure 10. Streamlines in the runner; span 0.85.

Besides the pressure field and flow patterns, the guide vane torque was monitored. On figure 11 there is a comparison of guide vane torque progress within 3 impeller revolutions for two operating points: 75 l/s, unstable point with the presence of rotating stall and 220 l/s, point within normal operating range. For the stable operating point, the torque fluctuates around the fixed value (the mean value is much bigger than the dynamic component). In case of unstable point, the torque changes from positive to negative values and the amplitude is much higher than the torque mean value. The fluctuation of the guide vane torque might be a good indicator of rotating stall. Since the torque measurement on the guide vanes is relatively simple measurement, this method can be used experimentally for detecting unstable operating points governed by rotating stall mechanisms. The flow past guide vanes under unstable conditions is extensively studied in [3].
Figure 1. Guide vane torque (blue - \(Q = 75\) l/s; orange – \(Q = 220\) l/s).

5. Conclusions
The main goal of the study has been to improve the understanding of the instabilities that occur at the off-design operation of the reversible pump-turbine in order to enable wide operating range or at the pumping and generating mode. Current paper has been focused on the partial load pumping mode instabilities such as rotating stall and cavitation. Moreover, generating mode instabilities, such as S-shaped characteristic and related generating mode rotating stall has been analyzed by CFD. In pumping mode, CFD analysis has been used to successfully reproduce the rotating stall and related cavitating vortex that has been observed during an experiment in the hydraulic laboratory. The numerical governing frequency of the rotating stall is \(f = 0.5\) Hz and a good approximation of experimental value \(f = 0.59\)Hz. Cavitating vortex has been visualized and compared to the experimental one. Generating mode rotating stall has been identified at the operating point 75 l/s, which is close to the runaway conditions. Rotating stall frequency 21 Hz has been identified, which corresponds to 84 % of impeller rotational frequency. Flow conditions have been compared to the stable operating point. Knowledge that has been obtained during research work has been used and will be used in the future to develop hydraulically stable pump-turbines. The study has been done in the scope of the development project of pump-turbines with reversible Francis runner and non-restricted operating ranges from 0-100% in pumping and generating mode. Based on the obtained knowledge, the hydraulic shape of the pump-turbine impeller, guide vanes and size of the vaneless gap have been optimized together with mechanical design during first rehabilitation project called PSP Dlouhe Strane in Czech Republic It already enables described operating range. After a successful replacement of the first runner, the company has been awarded a contract for replacement of the second runner, which is now under realization in the company.

6. References
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