A study on the evolution of the instability in two model pump-turbine runners with large blade leans

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Abstract. Instabilities usually happen in the reversible pump-turbine in generation mode, especially in the S-shaped region. Flow separations, vortex formations and rotating stalls can exist in the runners and other components. The evolution of the instabilities in two model pump-turbine runners with large blade leans, runner A with a negative lean and runner B with a positive lean, is studied by means of numerical simulations in the present work. The guide vanes opening (GVO) is fixed as 15mm. Six operation conditions from the normal operation region to the turbine brake region are chosen to be simulated. In addition, processes with a decreasing discharge at the inlet of the volute are also calculated to trace the evolution of the instability. The pressure fluctuations in the vaneless gap are monitored in the whole annulus. Different flow structures can be detected from the power spectrum of the pressure fluctuations. As the discharge reduced, the amplitudes of the pressure fluctuations increase visibly from the normal operation region to the runaway conditions and slightly decrease if further reduce the flow rate. Rotor-stator interaction frequency and its harmonics dominate the flow field from normal operating region to turbine brake region with the frequency corresponding to \(7f_n, 14f_n\), where \(f_n\) is the runner rotating frequency, and etc. Low frequency components gradually develop near the runaway points and the wavelet method detects the inception of the rotating stalls in the vaneless gap in the turbine brake region. Different blade leans give arise to an obvious distinction of the flow distributions near the inlet of the runners and in the vaneless gap. The separation generates a swirl stripe extending to opposite direction dependent on the blade lean direction. In addition, due to the different location of the swirl stripes, the pressure fluctuation distributions are opposite for the runners. Higher fluctuations locate near the hub and near the shroud for runner A and runner B respectively.

1. Introduction
Among kinds of new energies, the hydropower is expected to play a major role in energy supply worldwide since it meets the demand for clean and secure energy supply. Especially, pumped storage power plants (PSPP), which employ reversible pump-turbine technology to produce and store electric energy efficiently, flexibly and securely, were developed quickly globally. PSPP can be competent at kinds of works including storing energy during valley-time and releasing during peak hour [1,2], electricity grid stabilization [3,4] and storing alternative energies like solar, wind and tidal [5]. As a result, this technology is wildly used all around the world.

Although it has satisfactory characteristics, it is also subject to some problems, such as instability. Due to its own property of frequently switching between pump mode and turbine mode, it usually operates at prolonged off-design conditions [6]. In addition, in the design process of reversible pump-turbines, pump mode is paid more attention since decelerated pump flow are more sensitive [7]. As a
result, some penalty has to be paid in turbine mode and more severe instability can appear in turbine mode.

Instability in generating mode usually rises in the so-called S-shaped region where the characteristic curves show a positive slope \( \left( \frac{dQ_{11}}{dn_{11}} > 0 \right) \). At the stationary point, the system cannot be stable but may switch between turbine mode and reverse pump mode. As a result, severe torque fluctuations as well as prominent head and flow rate fluctuation may happen [8, 9]. Zoom inside of the flow field, the pressure fluctuations at each part in the unit may become severe due to the interactions between the S-shaped characteristics of the runners and the hydraulic system [10, 11]. Obvious pressure fluctuations can give rise to mechanical vibration and even premature mechanical failure [12, 13], especially the impulse in the vaneless space between the runner and guide vanes. These fluctuations should be ascribed to the unstable flow structures such as vortex formations, separations, backflows, rotating stalls, vortex ropes and etc.

Rotor-stator interaction (RSI) is an inevitable source of pressure fluctuations due to the relative movement between runner blades and guide vanes [7, 14]. The pressure fluctuations generated by RSI is labelled with a frequency equal to rotational frequency of the runner times the number of runner blades. In the S-shaped region low frequency items could develop which stands for vortex formation or rotating stall [7]. With a relatively large guide vanes opening (GVO), rotating stalls can finally form and propagate in the vaneless space. As a result, a modulated impulse at 30%-70% rotational frequency can be detected in the passage of the runner and guide vanes as well as in the vaneless space [7, 15-17].

The present paper tries to display the evolution of the instability during the load rejection process with the discharge being reduced at a constant speed. Two runners designed by a 3D inverse method design system were investigated [18, 19]. The process begin from a condition out of the S-shaped region to a low discharge condition and the variations in process were analyzed.

2. Numerical model and method

![Figure 1](image1.png)

**Figure 1.** The numerical domain of the entire passage.

The commercial software ANSYS 16 were employed to simulate the flow in an entire passage model of a reversible pump-turbine as shown in figure 1 and the specifications of the runners are shown in table 1. Apart from the volute, stay vanes, guide vanes and the draft tube, two runners with large blade lean on the high pressure edges are studied. Runner A with a large negative blade lean \( \theta = -23^\circ \) and runner B with a large positive lean \( \theta = 30^\circ \) as well as a sketch of the blade lean are shown.

Model test of these runners were conducted on a standard hydraulic machinery test rig at the Harbin Institute of Large Electric Machinery, China as shown in Figure 2. Measurements were conducted following International Electrotechnical Commission (IEC) Standard 60193 [18]. The performance curves of the runners were measured.
Table 1. Specification of pump turbine and runners

| Number of runner blade/Z₁ | Number of guide vane/Z₉ | Number of stay vane/Z₈ | Height of guide vane/b₀ (mm) | Diameter of runner inlet/D₁ (mm) | Diameter of runner outlet/D₂ (mm) | Specific speed/Nₛₒ |
|---------------------------|-------------------------|------------------------|-----------------------------|---------------------------------|---------------------------------|------------------|
| 7                         | 20                      | 20                     | 51                          | 470                             | 210                             | 42.5             |

Figure 2. Sketch of test rig and test section.

Figure 3. Guide vanes position and the arrangement of pressure monitors.

A relatively large GVO was chosen as 15mm since at such GVO the characteristic curve presents an obvious S-shaped so that rotating stall were expected to develop. The radial gap between the trailing edge of the guide vanes and the runner’s leading edge was about 30.5mm. Twenty pressure monitors marked by red points were arranged in the vaneless space as shown in figure 3.

The computational domains of volute, draft tube and the torque were discretized by ICEM-CFD and the rotating domain including stay vanes, guide vanes and the runner were discretized by
TurboGrid. Four meshes (about 3,500,000, 4,400,000, 5,800,000 and 6,700,000 for the two runners) were tested at best efficiency point in the grid independent study and the third one was selected. Details of the mesh in each domain was reported in figure 4. At the BEP, average $y^+$ value in runner and guide vanes domain of the second mesh were less than 30.

RNG $k$-$\varepsilon$ model is a turbulence model derived by a rigorous statistical technique named renormalization group theory. It employs an analytically derived differential formula of effective viscosity to deal with the low Reynolds number cases. Therefore, it was employed to simulate the turbulence. Mass flow rate with a 5% free stream turbulent intensity was set at the inlet of the volute and static pressure was set at the outlet of the draft tube. The solid surfaces were set as smooth no slip condition. The interfaces were modelled with frozen stage and transient sliding method in steady and unsteady simulations. Scalable wall function was employed on the near wall surface. A time interval, $\Delta t=0.000333$ s, 1° of the runner revolutions, was employed in the unsteady simulations. A convergence criterion $\text{RMS}_{\text{max}}<10^{-4}$ with 10 internal coefficient loops were selected.

![Figure 4. Detail of the chosen mesh of each domain.](image)

Test conditions with constant flow rate were simulated first as shown in figure 5 with solid circles. Near the runaway conditions, $0.7f_n$, where $f_n$ is the runner rotation frequency, was detected for both runners standing for unsteady vortex formations. In addition, rotating stall frequency, $0.4f_n$ and $0.5f_n$ were identified in the turbine brake region for runner A and B respectively. Based on the result, a load rejection scenario was schemed to conquer the evaluation of the instability as shown in figure 5. The rotational speed remained at 500rpm during the simulation. Onset of the load rejection was set as 40kg/s which was higher than the discharge at runaway condition. A simple linear decline of the mass flow rate in about 40 runner revolutions was chosen according to the following equation:

$$Q=40\text{ (kg/s)}-50\text{ (kg/s)}\times\text{ runner revolutions}$$  \hspace{1cm} (1)

A normalized flow rate defined as $Q_n=Q/Q_{\text{ref}}\times100\%$. Here since the discharges of best efficiency point of runner A and B are almost the same (100kg/s) so $Q_{\text{ref}}$ is set to be 100kg/s for convenience. Unsteady simulation on operation point with a flow rate $Q_n=40$ was finished first whose result would
be the initial condition of the load rejection process. The process went through runaway condition and finally end on operation point 5 with a flow rate $Q_n=14$ in the turbine brake region.

3. Simulation result and analysis

3.1. Validation of the numerical method

The simulation results of the test conditions and the load rejection were compared with the model tests of the unit in terms of head and normalized discharge and normalized rotating speed defined as:

$$Q_{11} = \frac{Q}{D^2 N^2}, \quad n_{11} = \frac{n D^2}{N^2}$$

The comparison in Figure 6 indicates the numerical method can supply an approximate result of the experiments both in test conditions (Sim(T)) as well as in the load rejection process (Sim(L)). The points stand for the simulations of the test condition locate near the characteristics curve with relative error less than 6%. And transient head, $Q_{11}$ and $n_{11}$ of the load rejection simulation just vibrate along the characteristics curve matching the experimental data.
3.2. Analysis of the pressure fluctuations and flow structures

Figure 7. History of the pressure fluctuations in the vaneless space of the runners.

Figure 8. Wavelet of pressure fluctuations in the vaneless space, (a) runner A, (b) runner B.

Figure 7 shows the pressure fluctuations of GV5 (pressure monitor as shown in figure 3) in the vaneless space during the load rejection. It can be found that the amplitude of the fluctuation is higher from 40 to 30 and many kinds of frequencies are included in the signal. In the process of load rejection, several kinds flow structures are included in the signals but exist in different times. Therefore, Fourier Transform is not qualified for depict the evolution. Wavelet method is thus employed to analyze the pressure variation during the process. Figure 8 demonstrates the frequency variation of the pressure. Apart from the RSI component, low frequency components are the secondary element. The simulation
of load rejection clearly displays the flow structure evolution with showing the transformation from vortex formation to rotating stall.

Figure 9. (a) different operations during the process and (b) $C_q$ variation of different conditions

General feature of the wavelet result can be described as a process of three steps including duration of the $0.7f_n$, transition from $0.7f_n$ to $0.4/0.5f_n$ and development of $0.4/0.5f_n$. Based on the property, the process can be divided into three regions including vortex region, transition region and stall regions. Meanwhile, six conditions located in different regions are chosen to be deeply analyzed as shown in figure 9. It is found that the component corresponding to rotating stall in runner A appear earlier than runner B. The onsets of rotating stall frequency of runner A and B are about at OP4 and OP5 respectively.

The evolution of the flow structures accompanies with the blockage of the flow passages. Distribution of a parameter $C_q$ defined as:

$$C_q = \sqrt{\frac{\sum (7Q_i/Q_T-1)^2}{Z}}$$

where $Q_T$ is discharge, $Q_i$ is flow rate in passage-i, $Z$ is number of blades (7), is also demonstrated in the figure to represent the asymmetry of the flow in the runner. Higher $C_q$ can be taken as a proof of highly asymmetric flow distribution in the runner. The variation of $C_q$ also can be divided into three regions: increasing region, decreasing region and quick increasing region. These variations seem to be closely correlated with the regions shown in figure 9 (a). In the vortex region, the asymmetry develops, in the transition region, it is reduced and finally in the stalled region, an extremely asymmetric condition appears. It is interesting that for both two runners, the low frequency components as well as the asymmetry of the flow are suppressed. Actually, in such a region, a relative low pressure fluctuations were also founded in model tests and simulations [14].

Figure 10 demonstrates the flow characteristics in the middle span in the runners from OP3 to OP6 to explore the variation in the transition region. Radial velocity distribution contours indicate the complexity of the flow field near the inlet of the channels which is induced by separations and vortex formations. As the discharge reduced, the inflow (negative parts) becomes weaker and outflow (positive parts) becomes stronger. In addition, the outflow also arises and develops in the guide vanes channels. Three typical conditions marked with red circles in the figure 10. The channels of type one is normally working in turbine mode in which the water can normally flow into the downstream. Type two channels are almost blocked in which the flow is stopped by the vortex in the vaneless space and cannot flow downstream. Type three channels are labelled as partially working as a pump with obvious outflow areas locating near the guide vanes surface.
Figure 10. Contour of radial velocity in the middle span, (a) runner A, (b) runner B.

Figure 11. Three types of guide vanes channels, (a) turbine mode, (b) blocked, (c) pump mode

Flow conditions in such three types of channels are similar between the two runners and here the details of the flow in runner A are presented in figure 11. From OP3 to OP6, namely the transition
region, guide vanes channels are all of type one or type two with type three at the beginning. As the discharge reduced, more channels are blocked and finally some of them transform to the partial-pump mode. Rotating stalls are well-defined as some adjacent channels work in type three condition. In addition, from figure 10, it can be found that most channels are of the type two at OP4 and OP5. That means the flow rate of each channel is relatively low and the condition matches the analysis obtained from figure 9.

4. Conclusion

A load rejection process of a pump-turbine with two runners with large blade lean was studied in the CFD mode. In this process, the rotation speed of the runner keeps constant and the discharge is reduced by a slow speed. The process begins at a condition before runaway condition and end at the turbine brake region.

By wavelet method, the evolution of the instability was demonstrated with clearly identifying different unstable structures. Going through the unstable region, the characteristics of flow structures can be classified into three regions: vortex region, transition region and stalled region. The first region is labelled with unsteady vortex formation and it lasts in the turbine brake region. Then a transition region follows in which the rotating stall gradually replace the unsteady vortex formation to play the chief role. Finally, the rotating stalls develop as the discharge reduced furtherly. Apart from the transition region, both of the pressure fluctuations and asymmetry of the flow in the runner display an ascending fashion. While in the transition region, even the discharge is reducing, the pressure fluctuations and the \( C_p \) keep decreasing.

With a deep study of the transition region, three types of guide vane channels (turbine mode, blocked and partial pump mode) are identified. These types of conditions reflect the evolution of the unstable flow. In summary, during the transition region, the guide vane channels experience a variation that more channels transform into blocked type from turbine mode and finally some partial pumped channels appear and gather together to generate well-defined rotating stalls.

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