Special Pressure Fluctuation analyse of the pump turbine

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Abstract. In the 13th five-year planning, the pumped storage power stations have speeding up. A large number of pumped storage power stations will been installed. The current development of modern pump storage plants aims towards a higher flexibility and stability in operation, an extended operation range of the hydraulic machine. The stability of pump turbine has been paid more and more attention in the industry.

Hydraulic pressure pulsation is very important for the stability of pump turbine. For pump turbine, the pressure pulsation on vane-less area between guide vane and runner blade is very important. It is the fundamental source of unit stability and the direct embodiment of rotational and static interference.

This paper analyses the special pressure pulsation in the vane-less area of pump turbine, focusing on the pressure pulsation of one time rotating frequency and two times rotating frequency.

The amplitude and frequency of pressure pulsation and the source of pressure pulsation induced by this special frequency are analysed and detected from the perspective of numerical and model test. By optimizing the design of the runner, the pressure pulsation of that particular frequency is eliminated.

1. Introduction
The world's first pumped storage power station was born in 1882 in Zurich, Switzerland, and has a history of 127 years. By the end of June 2019, power plants with a capacity of 6,000 kilowatts or more had a capacity of 1.84 billion kilowatts. The total installed hydropower capacity is about 354 million kW (including 30.2 million kW pumped storage), the cumulative installed wind power capacity is 193 million kW, and the installed photovoltaic capacity is about 100 million kW.

As a large number of renewable energy sources are connected into the power grid system, the current hydropower units not only bear the base load, but also bear a large number of peak and frequency modulation tasks, so the stability and safety of the units are very high requirements.

For hydraulic turbine, pressure pulsation is the key index that affects unit stability. The core component of the pumped storage unit is the pump turbine, and the core heart is the runner, which is known as the "pearl in the crown" of the hydropower industry. The hydraulic design of the unit needs to take into account the performance indexes of the hydraulic turbine and pump under various operating conditions.

It is necessary to ensure stable operation in off-design conditions such as pump-turbine start-up, ultra-high head operation and no-load grid connection of water turbine, especially to ensure the safety of load rejection of multiple units. The number of design variables that affect the
performance of pump turbine is large and influence each other.

At present, the industry pays special attention to the pressure pulsation. In the power stations that have been put into operation, there are pumped-storage units whose pressure pulsation is too big, resulting in too much vibration of the units and the plant, too much noise, and even cracks, which reduce the reliability and economy of the unit operation.

How to control the pressure pulsation is an eternal topic in the development and design of hydraulic turbine. From the current main point of view, who controls the pressure pulsation, who has mastered the "method" of developing a high-quality turbine. To some extent, pressure pulsation is the decisive factor for winning the bid, but the implicit condition is that other indicators must be balanced, and one cannot lose the others. This brings great difficulty to hydraulic design.

Experimental tests on pressure fluctuations in pump turbines have been carried out since 1970. Grein [1] and Yamaguchi [2] both measured the stress and pressure fluctuations in a prototype and a model pump turbine. The similarities between data obtained in model test and prototype test were discussed in two papers, both focusing on the influencing factors on the amplitude and frequency of pressure fluctuations at different positions along the flow paths. Liu [3] did more detailed experiments to discuss different features of pressure fluctuations in different operating conditions. The results showed that in pump mode, the amplitude of pressure fluctuations was at minimum at the design operating point; the main frequency of pressure fluctuations in vane-less space was blade passing frequency; the amplitude of pressure fluctuations was 30% larger at critical cavitation condition than non-cavitation conditions. This article also discussed the pressure fluctuations at different openings of guide vanes.

Regarding the stress and vibrations caused by the pressure fluctuations in vane-less space, the method to reduce the amplitude of pressure fluctuations was discussed by Kawamoto [4]. Results in this article showed that by increasing the guide vane height by 40%, the amplitude of the pressure fluctuation at the runner periphery under turbine mode was reduced by 20~30%. This was an early attempt to reduce the stress fluctuations by reducing pressure fluctuations.

Generally speaking, early studies only provided limited sensible information on flow mechanism of pressure fluctuations in vane-less space.

Recent studies mainly involve the following topics: characteristics of pressure fluctuations in vane-less space, including the amplitude and main frequencies, by numerical and experimental methods; the mechanism study on the rotor-stator interaction, by means of theoretical and experimental analysis, to make clear the source of pressure fluctuations in vane-less space; the relationship between vibration and pressure fluctuation in engineering applications and experimental tests, to investigate the consequence of pressure fluctuations on the prototype and model running.

Unsteady CFD methods for simulating the amplitudes and frequencies of pressure fluctuations of pump turbines have been widely used, and RANS method is most popular among them. Unsteady simulations in normal operating condition were developed [5-7], and the results showed that for the monitoring point closest to the runner, maximum pressure amplitude is observed for the same component, indicating the strong influence of the potential effect in the interactions between the guide vanes and the rotating runner blades. The amplitude of this component decreases very fast backward to the stay vanes. Other studies simulated the off-design operating condition to investigate different feature of pressure fluctuations. Yan [8] simulated the hydrodynamics in a pump-turbine at off-design operating conditions in turbine mode and found that the low frequency components appear at runaway and low discharge conditions. Widmer [9] did similar simulation and found that during unsteady vortex formation, the vortices in the runner and the vane-less space fluctuate in time, and induce in-phase pressure fluctuations in the vane-less space. Yin [10] predicted pressure
fluctuations under low partial flow of pump mode and suggested that the low frequency pressure fluctuations are essential.

The methods of experimental studies have experienced rapid development, with the development of related techniques such as PIV and LDV technology. Comparing with early studies, more data with higher accuracy of pressure fluctuations in vane-less space can be obtained from the experiments, benefiting the study the characteristics of pressure fluctuations.

Wang [11] carried out experimental studies to investigate the pressure fluctuations in different regions on hump area on the pump turbine’s performance curve in pump mode. The experiment results showed that the pressure fluctuations in vane-less space increased with increasing openings of the guide vanes. The hydraulic performance was more unstable with larger guide vanes openings in the so-called ‘hump area’ on the performance curves. Thus the method of decreasing opening of guide vanes is often used to get through the hump area when the unit starts up in pump mode. Many other experiments have been carried out in EPFL. Hasmatuchi [12] investigated the flow hydrodynamics in a reduced pump-turbine model under off-design operating conditions in generating mode. Wall pressure measurements in the stator, synchronized with high-speed flow visualizations in the vane-less space between the runner and the guide vanes using air bubbles injection were performed. The results showed that at the best efficiency point, the pressure fluctuation was very low and it was mainly dominated by the blade passing frequency and the first harmonic. As the runner enters the “S-shape” domain on the performance curves, a substantial increase of the pressure fluctuation was observed.

Although some scholars have also conducted relevant studies on the pressure fluctuation, there is still special pressure fluctuation such as one time rotational frequency and two times rotational frequency on vane-less area. These frequency is hard to explain. From the perspective of PIV flow field test, this paper studies the internal flow state of special pressure fluctuation in detail, and provides some valuable references for the pressure fluctuation problem of pump turbine.

2. One time frequency pressure fluctuation
The tested object is one model turbine, the detailed specifications is listed in Table 1.

| Impeller diameter at inlet (mm) | 538 |
| Impeller diameter at outlet (mm) | 252 |
| Impeller blade number Z₁ | 5+5 |
| Guide vane number Z₀ | 16 |
| Maximum guide vane opening α (°) | 16 |
| Model Rotational speed (rpm) | 1100 |
| Height of guide vane (mm) | 39 |
| specific speed ns=np₀.5/H₁.25 | 82 |

Table 2. Main tested working points of the tested pump-turbine model.

| working points | n11 | Q11 |
|----------------|-----|-----|
| 42%P           | 38.7| 180 |
| 50%P           | 38.7| 239 |
| 100%P          | 38.7| 529 |

P-rated power
The tested pressure fluctuation points are arranged at casing, vane-less area, head-cover and draft tube in Figure 1. The measuring points are arranged along X and Y direction.
On the experiment, the one-time frequency pressure fluctuation is detected on vane-less area between guide vane and runner in Figure 2. From the Figure 2, the one-time frequency is evidence and the amplitude is larger than 10 times blade passing frequency and 5 times blade passing frequency.

Furthermore to analysis the one time frequency pressure fluctuation, the pressure fluctuation mainly take place on vane-less area. And the pressure fluctuation is also detected on spiral casing, head-cover and draft tube. However, the amplitude on casing is smaller than vane-less area and the amplitude on head-cover and draft tube is smallest in Figure 3.
According to the model test data, the tested data is converted to prototype data, the one-time frequency pressure fluctuation takes place on full power zone from zero load to full load. Especially the pressure fluctuation amplitude is increasing rapidly from 20% to 60% load. Its amplitude makes contribution about 50% to mixed frequency amplitude in Figure 4. And the taking place zone of one time frequency pressure fluctuation mainly focus on the smaller operation zone from 40% to 50% load.

Figure 3. The one time frequency pressure fluctuation of the pump-turbine model.

According to the working conditions occurring at 1 time rotational frequency, the working conditions at 42% load point were selected to carry out numerical calculation. The numerical results show that, as shown in Figure 5(a), there is a rotating stall vortex on the leading edge of the runner blade, and the stall vortex rotates along with the rotation of the runner, and the low-frequency pressure pulsation generated during the rotation is detected, so it is characterized by 1 time rotation frequency in the spectrum analysis.

Based on the analysis of the numerical results and experimental data, the source of the pressure pulsation in the vane-less area with one time rotational frequency. The direction of optimal design is to optimize the runner blade leading edge, but it is necessary to ensure that other performance indicators of the pump turbine are not affected.
By modifying the profile of the runner blade, it can be found that the stall vortex on the runner leading edge is well destroyed, as shown in Figure 5(b).

**Figure 5.** The flow phenomena of old runner on 42% load working point.
The section is Span=0.1, runner hub span=0, runner shroud span=1

The model test was conducted on the new runner to focus on the improving of pressure pulsation, and the pressure pulsation was monitored according to the range of 0-100% load, as shown in figure 6. It can be found that the pressure pulsation was improved in the whole range. The pressure pulsation was also monitored at 42% of the load points, and the time-domain diagram of the pressure pulsation was obtained and FFT transformation was carried out to obtain in Figures 7 and 8. It can be seen from Figure 8 that the pressure pulsation induced by one time frequency almost disappears with a small amplitude close to 0. The pressure pulsation at 5 times and 10 times rotating frequency is also improved, so that the frequency division amplitude of the first third main frequency is improved, thus improving the whole mixing amplitude.

**Figure 6.** The flow phenomena of optimized runner on full range load working point.
3. Two times frequency pressure fluctuation

The tested object is one model turbine, the detailed specifications is listed in Table 3. The object is tested for pressure pulsation, and the arrangement of measuring points is shown in Figure 1.

| Table 3. Main parameters of the tested pump-turbine model. |
|-----------------------------------------------------------|
| Parameter                                               | Value |
| Impeller diameter at inlet (mm)                         | 560   |
| Impeller diameter at outlet (mm)                        | 300   |
| Impeller blade number $Z_1$                             | 9     |
| Guide vane number $Z_0$                                 | 20    |
| Maximum guide vane opening $\alpha$ (°)                 | 24    |
| Model Rotational speed (rpm)                            | 1100  |
| Height of guide vane (mm)                               | 53    |
| $n_s=n_p^{0.5}/H^{1.25}$                               | 122   |

| Table 4. Main tested working points of the tested pump-turbine model. |
|-----------------------------------------------------------|
| Working points | $n_{11}$ | $Q_{11}$ |
|----------------|---------|--------|
| 49%P           | 47      | 383    |
| 55%P           | 47      | 427    |
| 100%P          | 47      | 776    |

P-rated power

In a head on the vane-less of pressure pulsation test and analysis, through the fast Fourier transform, the pressure pulsation can be found throughout the load range. The first main frequency of pressure
pulsation is 9 times frequency in most load ranges, but the first main frequency of pressure pulsation is 2 times frequency pressure fluctuation in range of from 49% to 55% load, as shown in figure 9.

Figure 9. The pressure fluctuation first main frequency in range of full load.

The pressure fluctuation frequency on different measuring points are shown in Figure 10. The two times frequency fluctuation propagates from the vane-less zone to the head-cover, casing and the draft tube.

Figure 10. The pressure fluctuation along different measuring points.

From the Figure 11, the two times frequency ($f_1/f_m=2$) is detected on the measuring points VS+Y and main frequency amplitude occupies cover 28.5% mixture frequency amplitude. Furthermore, the two times frequency propagates from the vane-less zone to the head-cover, casing and the draft tube. And the two times frequency is first main frequency on the vane-less and draft tube, it’s the second main frequency on casing.
According to the Figure 12, the shedding vortices are mainly located on the leading edge of runner. Two shedding vortices appear intermittently in the runner passage. So How to solve the two times rotational frequency pressure pulsation problem and it equals how to eliminate or attenuate the two shedding vortices.

The runner leading edge is optimized and two shedding vortices are eliminated. And the two times rotational frequency pressure fluctuation is reduce. And the first main frequency becomes 9 times rotational frequency on measuring point VS+Y, so the pressure fluctuation mixture amplitude becomes smaller in Figure 13. Furthermore, the first main frequency is 4 times rotational frequency on measuring points casing and DTHW, and the reason of 4times frequency is discussed on next paper.

4. conclusion
In this paper, the special pressure fluctuation is paid attention including one time rotational frequency and two times rotational frequency pressure fluctuation. By means of the fast Fourier transform, the root
cause of the pressure pulsation at one and two times the frequency was found, and the pressure pulsation at that particular frequency was eliminated by the optimized design of the runner.

Figure 13. The pressure fluctuation mixture and main frequency on 49%P working point of new runner.

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