Investigations on the unstable flow in a prototype reversible pump turbine

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Abstract. In the present paper, influences of load variations on the pressure fluctuations especially in the vaneless space of a prototype reversible pump turbine are experimentally investigated in generating mode. Original pressure data obtained through on-site measurement are further analyzed in both time and frequency domains in order to show the propagation mechanism of the pressure fluctuations. Our work reveals that the dominant characteristic frequency is the blade passing frequency (BPF, \( f^* = 1 \)) between 52% and 90% partial loads and its harmonics (\( f^* = 2 \)) between 90% and 100% of rated power, respectively. Detailed results are analyzed and discussed with the aid of several typical demonstrating figures in the present paper.

1. Introduction

Nowadays, the pump turbines are operated with frequent load variations in order to adopt the massive renewable energies (e.g., solar and wind energy [1]) and enhance the stability of the power grid. However, load variations could generate huge pressure fluctuations in the pump turbines, leading to the negative long-term effects on the turbine operations. The present paper focuses on the influences of load variations on the propagation mechanism of pressure fluctuations in the flow channel of a prototype pump turbine.

The previous research on the pump turbines are mainly based on the model test and numerical simulation. Zhang et al. [2] comprehensively summarized the rotating stall (RS) in the pump turbines and pointed out that the RS is the physical origin of huge pressure fluctuations in the S-shaped region. Zuo et al. [3] summarized the pressure fluctuations in the vaneless space including the mechanism of rotor-stator interaction (RSI). Zhang et al. [4] summarized the methods for vortex identification based on five taxonomies (including "region/line", "invariant", "local/global", "2D/3D" and "Objective"). Their applications in hydroturbines are further summarized and compared in detail.

About the prototype turbine, Zhang et al. [5] studied the effects of load variations on the pressure fluctuations by steady test in a prototype pump turbine. For low partial loads, the pressure fluctuations are the highest due to the RSI in the vaneless space. For medium loads, the pressure fluctuations are medium and mainly induced be the swirling vortex in the draft tube. For high partial loads, the
pressure fluctuations are quite limited and still dominated by the RSI in the vaneless space but the dominant frequency is different with one for low partial loads. Zhang et al. [6] found that vibrations of the top cover are mainly attributed to the fluid inside the prototype pump turbine together with great influences of load variations and the vibrations of the brackets are mainly generated by the mechanical rotation of the rotor with limited influences of load variations. Li et al. [7] further investigated the effects of water head variations on the pressure fluctuations in a prototype pump turbine. Hu et al. [8] studied the characteristics of transient pressure fluctuations during load rejection of a 300MW prototype pump turbine and found that the intensity of RSI increases significantly at low partial load. Xia et al. [9] found that pressure fluctuations in the vaneless space are strongly affected by the distributed reverse flow vortex structures. Recently, a new vortex identification method (named as “Omega” method [10, 11]) was proposed with many advantages over the conventional methods (e.g. $Q$ criterion), which could be conveniently applied into a broad range of industrial-scale fluid flow (e.g. reversible pump turbine [12]).

The structure of the present paper will be introduced as follows. Firstly, the experimental setup will be briefly introduced including the basic parameters of the unit, the operational conditions and some dimensionless parameters. Then, the original pressure data are analyzed both in time and frequency domains in order to illustrate the physical origin and the propagation mechanism of the pressure fluctuations. At last, the primary findings of our analysis will be presented.

2. Experimental Setup

The experiment was performed based on a prototype reversible pump turbine in the pumped hydro storage power station. Some basic parameters of the pump turbine are listed in table 1. Three typical monitoring points along the flow channel were selected to collect the original pressure data (named as "P1", "P2" and "P3" for short in the following sections). Specifically, "P1" is located at the inlet of the spiral casing, while "P2" and "P3" are both in the vaneless space. The original pressure data were obtained through on-site measurement with the AK-4 type pressure transmitter (with its specific parameters shown in table 2) during the whole data acquisition process.

| Name                        | Value  | Unit |
|-----------------------------|--------|------|
| Number of impeller blades   | 9      | ---  |
| Number of active guide vanes| 20     | ---  |
| Number of stationary guide vanes | 20   | ---  |
| Rated rotational speed      | 333.3  | rpm  |
| Rated power                 | 250    | MW   |

Table 1. Basic parameters of the prototype pump turbine.

| Name                      | Value   | Unit |
|----------------------------|---------|------|
| Intrinsic error            | ± 0.21  | %    |
| Linearity error            | ± 0.06  | %    |
| Output signal              | 4–20 mA |      |
| Power supply               | 24 V    |      |
| Frequency response range   | 1000 Hz |      |
| Operational temperature    | -10–60 °C|     |

Table 2. Primary indexes for the performance of the AK-4 type pressure transmitter

Note: The pressure transmitter was sent to the Beijing Institute of Metrology and Testing Institute for calibration and verification before the experiment.

The average gross head was fixed during the present experiment. For facilitating further analysis, some dimensionless parameters are defined as follows with the nomenclature shown in table 3:
\[ t^* = \frac{t}{T}, \quad (1) \]
\[ p^* = \frac{p}{\rho g H}, \quad (2) \]
\[ \bar{p}^* = \frac{1}{t} \int_0^t p^* dt^*, \quad (3) \]
\[ f^* = \frac{f}{f_{BPF}}, \quad (4) \]
\[ f_{BPF} = nf_n, \quad (5) \]
\[ A^* = \frac{A}{\rho g H}, \quad (6) \]
\[ P^* = \frac{P_{output}}{P_{rated}}, \quad (7) \]
\[ \tilde{C}_p = \sqrt{\frac{1}{N} \sum_{i=1}^{N} (p_i^* - \bar{p}^*)^2}. \quad (8) \]

After the dimensionless calculation, \( P^* \) varies between 52% and 100% in the present experiment (including 52%, 60%, 70%, 80%, 90% and 100% load conditions).

| Symbol | Definition |
|--------|------------|
| Superscript "*" | Dimensionless value |
| Subscript "i" | \( i \)th data of the whole data set |
| \( p \) | Pressure |
| \( \bar{p}^* \) | Time averaged value of dimensionless pressure |
| \( \rho \) | Water density |
| \( g \) | Gravitational acceleration |
| \( t \) | Time |
| \( T \) | Impeller rotation cycle |
| \( f \) | Characteristic frequency |
| \( f_{BPF} \) | Blade passing frequency |
| \( n \) | Number of impeller blades |
| \( N \) | Total number of data sample |
| \( f_n \) | Rotational frequency of the impeller |
| \( A^* \) | Dimensionless amplitude in the frequency spectrograms |
| \( P_{output} \) | Output power |
| \( P_{rated} \) | Rated power |
| \( P_{peak-to-peak}^* \) | Dimensionless peak-to-peak value of the pressure fluctuation |
3. Influences of Load Variations on the Pressure Fluctuations

Figure 1 shows the variations of dimensionless pressure fluctuations ($p^*$) obtained at three monitoring points in the time domain during 52% partial load condition. Three red straight lines indicate the time averaged pressure ($\bar{p}^*$) of three monitoring points, respectively. It shows that the time averaged pressure decreases along the flow channel and the waveform of pressure signal in the time domains becomes more and more periodical from the inlet of spiral casing (“P1”) to the vaneless space (“P2” and “P3”).

Figure 2 shows the dimensionless peak-to-peak values of pressure fluctuations ($p_{\text{peak-to-peak}}^*$) versus load variations (from 52% to 100%) at three monitoring points. $P^*$=52%, 60%, 70%, 80%, 90% and 100%, respectively.

Specifically speaking, during 52% partial load, the values of $p_{\text{peak-to-peak}}^*$ are 8.47% and 9.82% at “P2” and “P3”.

Figure 3. Dimensionless standard deviation of pressure fluctuation ($\overline{\sigma_p}$) at three monitoring points. $P^*$=52%, 60%, 70%, 80%, 90% and 100%, respectively.
and "P3" respectively. In addition, for "P2" and "P3", the values of $p_{\text{peak-to-peak}}$ drop to the minimum at 80% partial load and then increase with the increase of the load. Figure 2 also shows that $p_{\text{peak-to-peak}}$ at "P3" are the largest one among three monitoring points in the whole load range, which indicates that pressure fluctuations are the most serious in the vaneless space. Different with "P2" and "P3", the values of $p_{\text{peak-to-peak}}$ at "P1" show little difference among six operational conditions.

Figure 3 shows the standard deviation of dimensionless pressure fluctuations versus load variations (from 52% to 100%) at three monitoring points. The $C_p$ reflects the degree that the actual pressure deviates from the average pressure. The trends of three curves are almost identical as those in figure 2. Similar to figure 2, the largest deviation of the pressure data appears at 52% low partial load condition especially at "P3" in the vaneless space. And for all the three monitoring points, the smallest deviation appears at 80% partial load condition. Among the three monitoring points, $C_p$ at "P1" at the inlet of the spiral casing is nearly not affected by the load variations.

**Figure 4.** Frequency spectrograms of dimensionless pressure fluctuations at three monitoring points. $P^*=52%$.

**Figure 5.** Frequency spectrograms of dimensionless pressure fluctuations at three monitoring points. $P^*=60%$.

Figures 4-9 show the frequency spectrograms of pressure fluctuations based on fast Fourier transformation (FFT) at three monitoring points at all six load conditions, respectively. The six figures indicate that the dominant frequency of pressure fluctuations at three monitoring points is the BPF ($f^*=1$) or its harmonics ($f^*=2$) at six load conditions. The amplitudes of above two characteristic frequencies are quite limited in the upstream components (e.g., "P1" at the inlet of spiral casing). In addition, the amplitudes of the harmonics of BPF ($f^*=3, 4, 5$ and $6$) can also be observed at "P2" and "P3" in the vaneless space and they almost disappear at "P1". Therefore, it can be inferred that physical origin of the pressure fluctuations with the characteristic frequencies being the BPF is located in the vaneless space, which are induced by the rotor-stator interaction (RSI) between the stationary guide vanes and the rotational impeller. Based on the dominant characteristic frequency, the whole load range (from 52% to 100% load conditions) can be divided into two parts as follows.
Figure 6. Frequency spectrograms of dimensionless pressure fluctuations at three monitoring points. $P^*=70\%$.

Figure 7. Frequency spectrograms of dimensionless pressure fluctuations at three monitoring points. $P^*=80\%$.

Figure 8. Frequency spectrograms of dimensionless pressure fluctuations at three monitoring points. $P^*=90\%$.

Figure 9. Frequency spectrograms of dimensionless pressure fluctuations at three monitoring points. $P^*=100\%$.

**Part I.** The load range of this part covers 52-90% partial load conditions (including 52%, 60%, 70%, 80% and 90% partial load conditions). Physical origin of pressure fluctuations is located at "P3" in the vaneless space with characteristic frequency being the BPF ($f^*=1$). With the increase of the load, the amplitudes of the dominant frequency ($f^*=1$) at "P3" are 0.03315, 0.02970, 0.02045, 0.01855, 0.01510 for 52%, 60%, 70%, 80% and 90% partial load conditions respectively.

**Part II.** The load range of this part only includes 100% rated power condition. Physical origin of pressure fluctuations is still in the vaneless space but with the dominant frequency being the harmonics of BPF ($f^*=2$). The amplitude of dominant frequency ($f^*=2$) at "P3" is the largest (0.01622). In this part, the pressure fluctuations with the characteristic frequency being the BPF ($f^*=1$) are much lower than in Part I because the unit is operated under the design condition.

As shown in figures 4-9, the pressure fluctuations of the characteristic frequencies being the BPF ($f^*=1$) and its harmonics ($f^*=2$) are more significant than other characteristic frequencies. Figures 10 and 11 summarize the amplitudes of above two frequencies at three monitoring points at all six load conditions. It shows that with the increase of the load, the amplitudes of the BPF ($f^*=1$) decrease while
the amplitudes of its harmonics ($f^* = 2$) decrease firstly and then increase at 70% partial load condition. The upstream monitoring points (e.g., "P1" at the inlet of spiral casing) are less affected by above two characteristic frequencies. According to the amplitudes in figure 10, it can be also inferred that physical origin of the BPF ($f^* = 1$) is located in the vaneless space.

**Figure 10.** Amplitude ($A^*$) of the BPF ($f^* = 1$) in the frequency spectrograms at three monitoring points. $P^* = 52\%, 60\%, 70\%, 80\%, 90\%$ and 100%, respectively.

**Figure 11.** Amplitude ($A^*$) of the harmonics of BPF ($f^* = 2$) in the frequency spectrograms at three monitoring points. $P^* = 52\%, 60\%, 70\%, 80\%, 90\%$ and 100%, respectively.

### 4. Conclusion

In the present paper, based on the experimental data in a prototype reversible pump turbine, both time and frequency domains are adopted to illustrate the influences of load variations on the pressure fluctuations especially in the vaneless space. The dominant frequencies of pressure fluctuations in different load range have been analysed in detail. Furthermore, two characteristic frequencies of the BPF ($f^* = 1$) and its harmonics ($f^* = 2$) are discussed with their propagation mechanism. The main conclusions of the present paper are summarized as follows.

1. The dominant frequency of the pressure fluctuations is the BPF ($f^* = 1$) between 52% and 90% partial loads and its harmonics ($f^* = 2$) at 100% rated power, respectively.
2. The physical origin of BPF ($f^* = 1$) is within the vaneless space and its corresponding amplitude gradually reduces with the increase of the load.
3. The pressure fluctuations at the inlet of spiral casing are quite limited and less affected by the load variations.

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