Pump-turbine Rotor-Stator Interaction Induced Vibration: Problem Resolution and Experience

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Abstract. Pressure pulsations induced by rotor-stator interaction (RSI) phenomenon are one of the major sources of excitation resulting in severe vibrations of pump-turbines and powerhouses. By now, many articles focus mainly on the RSI mechanism by methods of theoretical analysis, fluid-structure calculation and model test verification. There are few reports published on equipment and powerhouse vibrations induced by RSI. Therefore, in this paper we took a real RSI induced problem at Zhanghewan (ZHW) pumped storage power plant (PSPP). We employed frequency spectrum technology on signals of powerhouse vibrations and pressure pulsations in stable and transient processes, combined eigen-frequency measurement and numerical calculation of powerhouse localities, and then confirmed that partial powerhouse resonated with RSI induced pressure pulsation in the vaneless area (gap between guide vanes and runner blades). We statistically evaluated the relationship between the ratio of guide vane pitch diameter D₀ to the high-pressure runner diameter D₁ and vaneless area pressure pulsation from many PSPPs in China, and proposed solutions that new runner should use. The runner alteration work was successful. Model tests and field measurements show that vaneless pressure pulsation decreases and the powerhouse shows acceptable vibration levels. Many experiences were accumulated during this work, including that from an engineering perspective by enlarging the vaneless area it is possible to lower RSI induced pressure vibrations. Also, much attention should be paid on local anti-vibration calculations in the powerhouse designing period. Our research and review provide a new perspective for designing, refurbishing, rehabilitating of pump-turbines.

1. Introduction

In recent decades, sustainable development philosophy of green energy is deeply rooted among the population worldwide, and many renewable sources such as wind power and solar energy are being rapidly developed. But their unpredictable and intermittent characteristics can have a significant impact on the security, stability, reliability and efficiency of the power grid, which restrains the development of renewable energy. Pumped storage units with attributes of flexibility, reliability and safety are widely used in the power grid for its rapid response for power regulation [1]. Many PSPPs have been built all over the world, especially in China [2]. Several factors relate to the safe and stable operation of the unit and powerhouse, including structural design, equipment manufacturing and installation. Different from normal Francis turbine, some specific hydraulic induced instabilities can occur in a reversible pump-turbine. Firstly, the S-shape characteristic induced instabilities occur in turbine mode operation, such as water hammer in load shedding [3,4] and speed disturbance in parallel for low water head [5]. Secondly,
hump induced instability occurs in pump mode operation [6,7]. Because of bi-direction operation and complicated hydraulic design, many units have serious pressure pulsations in their operating zone [8,9]. Therefore, the powerhouses can have concerns of severe vibration [10,11].

For the turbine, rotor-stator interaction (RSI) is a hydraulic phenomenon occurring in the vaneless area (gap between guide vanes and runner blades) under specified speed and dependent on the number of guide vanes and runner blades [12]. The frequency components of RSI induced pressure pulsations are multiples of the blade passing frequency. When this pressure pulsation frequency coincides with those of the powerhouse or unit natural frequencies, a fluid-excited resonance results impacts by two aspects. The first is structural damage in hydraulic passages [10, 13]. The second is local or integral severe vibrations which bring potential risks for the power station [14]. Tanaka’s research indicates that RSI intensity depends on the geometric parameters of runner, such as thickness of guide vanes and runner blades and operational parameters, such as working head, unit rotational speed and unit output. He developed a model to determine dihedral vibration modes in high-head pump-turbine [13].

Recently, with the rapid development of pump-turbine technology, many PSPPs with high-head large-capacity units have been constructed in China, such as Changlongshan PSPP, with a unit rated head 710m and rated power 350MW, Yangjiang PSPP, with a unit rated head 653m and rated power 400MW. Since the water passage including the vaneless area in pump-turbine is relatively long and narrow compared with a typical Francis turbine, RSI induced problems become much more challenging.

Therefore, civil engineers should also pay much attention on the powerhouse and anti-vibration design of units.

For RSI induced failures, Hideo Ohashi described a pump failure in literature [25], but no similar problems of pump-turbines have been reported in the released literature. Much research from scaled models enrich our knowledge of the mechanism and guide people to mitigate the problems, but it cannot completely eradicate the risk of RSI induced failure occurrence on the prototype. In this paper a real power plant case, Zhanghedian (ZHW) PSPP, was studied and which exhibited RSI induced powerhouse vibrations and we present the solutions. These solutions provide a new perspective for other PSPPs around the world which have similar problems.

2. Problem statement

ZHWW PSPP is located within the metropolitan area of Shijiazhuang City in Hebei province of China. Total installed capacity of the power station is 1000MW with 4 reversible pump-turbines each having a rated capacity of 250MW. It is linked to the South power system of Hebei province for peak and load regulation (operating range from 50% to 100% of rated output in generating mode), frequency regulation, synchronous condenser operation and emergency reverse functions. The rated working head of the pump-turbine is 305m, and rated rotational speed is 333.3r/min. The pump-turbine has 9 blades, 20 guide vanes, 20 stay vanes. Each of the two penstocks is bifurcated into two branches. In December 2007 the first unit was connected to the grid and put into commercial operation, and the commercial operation for the last unit was in February 2009.

After the unit commissioning, significant vibrations were noticeable in the powerhouse floors during generating mode at higher load. High frequency noise was observed. The noise amplitude measured in motor-generator floor was around 84~85dB(A), busbars floor 90~91dB(A), turbine floor 91~96dB(A), and at the turbine pit nearly 102dB(A). The intensities of noise and vibration became larger as the
generating load increased until a maximum at rated output. Because the units are operated in generation mode with rated output most of time, vibrations were affecting the structure of powerhouse, and resulted in failures of automation devices which caused abnormal stops of the unit and thus negative impact for the power grid. Meanwhile, the severe vibration and noise could lead to nausea and discomfort of operators and maintainers in the powerhouse and cause occupational health problems for the staff. In order to solve these problems, dedicated workshops have been held since 2011 between the owner, manufacturer, consultant and designer, and many tests have been performed by independent third parties on the vibrations of powerhouse and unit, pressure pulsation in the water passage, noise in each floor of powerhouse, etc. Modal characterizations have been performed on the partial and whole powerhouse structure. Vibration source and dynamic behavior of the powerhouse were found by these activities, which was the groundwork for the final solutions.

2.1. Vibration at Stable Condition

Vibration of the unit and powerhouse, pressure pulsation and noise were measured under stable operating conditions. In figure 1, floor vibrations in time-domain are presented for pumping operation and generating operation at rated power (G.F, B.F, T.F, and S.C.F are respectively the abbreviations for Generator Floor, Busbar Floor, Turbine Floor and Spiral Case Floor). Corresponding RMS vibration velocity values are indicated in figure 2.

![Figure 1. Time-domain vibration waveforms of each powerhouse floor](image1)

From figure 1, we can see that under rated output operation each powerhouse floor vibrates in a simple harmonic oscillation and there are different time delays between different floors. In generation mode each floor vibrates more than that in pumping mode. For different floors, RMS values of each floor in descending series are busbar floor, turbine floor, generator floor and spiral case floor (figure 2). This result is consistent with the intuitive feeling on-site.

![Figure 2. Powerhouse vibration RMS value comparison in pump mode and generating mode](image2)

Frequency cascades with the load varying of vaneless area pressures and some measuring point of powerhouse floor vibrations are presented in figure 4. We obtain that the dominant frequency in each floor is 100Hz, and the amplitude of 50Hz is also prominent. The amplitudes of 100Hz has a similar trend with the RMS values shown in figure 2, which indicates that the 100Hz component plays significant role in the powerhouse vibration.

The components of 50Hz and 100Hz may come from two sources. One is polar vibrations of the generator. Considering the guide vanes number, turbine blades number and rotational speed, the other one is the first-order harmonic of blade passing frequency and RSI induced vibration frequency from vaneless area (second order harmonic of blades passing frequency). For pump-turbine of blades number $Z_s$ and guide vanes number $Z_r$, the generation equation of RSI is given by [13]:
\[ m \times Z_s + v = k \times Z_r \]  

where, \( k \) is the order of excitation harmonics in stationary frame; \( m \) is the order of excitation harmonics in rotating frame; \( v \) is the order of diametrical mode; \( k, m \) are positive integers, and \( v \) is integer.

For the pump-turbine of ZHW, the RSI induced frequency component in the formula (1) is expressed by \( m=1, v=-2 \) and \( k=2 \). As indicated in figures 1 and 2, the powerhouse floor vibration is depending on the operation mode. In order to determine whether its intensity depends on the active power, a load variation test in generating mode was performed. Pressure fluctuations in the vaneless area were measured simultaneously with the vibrations. The transducers were installed directly on the turbine head cover without using a long water pipe to avoid pressure signal distortion [26]. The trends between the powerhouse floor vibration as well as the amplitudes of vaneless area pressure fluctuations versus the active power are shown in figure 3. We observe that the powerhouse floor vibrations remain relatively constant in the active power range from 120 to 200MW, whereas the amplitudes of RSI pressure pulsations in vaneless area increases gradually with the increasing active power.

![Figure 3. Vaneless Pressure (VP) RSI amplitude and powerhouse floor vibration RMS values](image)

![Figure 4. Frequency cascades of vaneless area pressures and powerhouse vibration measuring points](image)

For the vaneless area, the main frequency is the blades passing frequency at 50Hz as it is expected in the proximity of the leading edge (figure 4a). In the powerhouse floors the main frequency is RSI frequency at twice the blades passing frequency (figures 4b and 4c). Theoretically, frequencies of powerhouse floors are determined by two factors. One is the resonance of natural frequencies caused by excitation sources of limited energy injected in the system. The other one is forced frequency excitation caused by sources of sufficiently large energy. As for the powerhouse of PSPPs, each floor is connected to the surrounding solid rock providing large mass and a damping effect. The powerhouse can likened to an infinitive mass which cannot be forced to vibrate by limited energy input. Therefore, the reason to cause the powerhouse vibration is that the natural frequency coincides with the excitation source frequency. In figure 4, we can obtain that in generating mode at rated output, the main frequency in vaneless area is one single blade passing frequency of 50Hz, which has the biggest energy in all frequencies. But, no obvious 50Hz has been excited in the powerhouse floor. One main reason is that the pressure field corresponding to the fundamental blade passing frequency at 50Hz is limited to the vicinity of the leading edges of the runner but is less prone to propagate over large distance. For RSI frequency, although in the vaneless area it is the second component and the amplitude is significantly smaller than the amplitude of 50Hz, certain orders of natural frequency of the powerhouse coincide with or are close to 100Hz, and the successive injecting energy is sufficient to excite the powerhouse floor. When the unit runs in generation mode, 100Hz amplitudes of vaneless area pressure and powerhouse floor vibrations rise as the load increases, which can be seen from figures 3 and 4.
2.2. Transient Process

Vibration of the powerhouse floor and the units reflect the effects of hydraulic, mechanical and electrical sources of excitation. By observing the transient unit stop before and after the deactivation of the field current, we can discriminate the source of the 100Hz component from electrical source coming from the rotor poles excitation and hydraulic excitation as RSI induced excitation.

In Oct. 2015, main inlet valve (MIV) closing test under water discharge was carried out. The data acquired in this test have been analyzed to confirm the source the 100Hz component. Before closing of the MIV, the unit was running at rated output. After getting the order from the electricity dispatching agency, the closing sequence was triggered locally from the control cubicle of the MIV. During the closing, the output decreases continuously to a given value, then the generator circuit breaker disconnects the machine from the grid and the guide vane closes following the predefined sequence. Refer to figure 5a for the operation parameters and figure 5b for the typical time records of vibration measuring points during the whole process.

![Figure 5. Waveform records during MIV closing under water discharge](image)

The signals are unsteady as it can be expected during a fast variation of the parameters during the transient process. A short-time-Fourier-transform (STFT) applied on the time series of the vibrations is clearly demonstrating that the RSI component dominating in generation mode remains after opening of the generator circuit breaker and during the closing of the guide vanes. The frequency decreases logically with the unit rotation speed. Since there is no field current and the generator circuit breaker is disconnected in this whole period, there is no electrical excitation imposed on the whole system. Therefore, the vibrations of the unit and powerhouse is likely excited by hydraulics. During the whole transient process, a clear linear dependence between main frequency and rotational speed is observed. For ZHW PSPP with 9 blades and 20 guide vanes, the main frequency is 18 times rotational frequency, which is characteristic of the RSI frequency.

2.3. Structural Calculation and Modal Test Verification

Structural dynamic calculation of the powerhouse has been performed with commercial Finite Elements Method software and indicated that the first vibration mode of powerhouse was Y-axis vibration with frequency 13.3Hz, the second vibration mode was X-axis vibration with frequency 15.9Hz, the third mode was a torsional oscillation with frequency 17.4Hz, and the residual order vibration modes differed slightly. These frequencies of main vibration modes were far away from RSI frequency, i.e. there is sufficient safety margin to avoid the resonance of the powerhouse with pressure pulsation induced by RSI. It is hard to identify local resonance of higher orders from integral structural calculation of powerhouse. The experimental modal analysis carried out on-site with shock hammer excitation showed that busbar floor exhibits a local resonance, while the modal simulation of powerhouse as a single entity cannot explain this fact. So, local natural vibration simulation and modal measurements have been carried out. Some results are provided in figure 6.

The calculation results indicate that the first two frequencies of vertical busbar pillar vibration mode are 85.0Hz and 98.5Hz (figure 6a) respectively. Powerhouse vibration measurement in generation mode shows that generator floor connecting with pillars of busbar floor reach up to 1.5g with main frequency
of 100Hz, which has been previously identified as RSI induced frequency. The local vibration modal frequencies are close to RSI frequency (figure 6c). According to Chinese standard frequencies between vibration sources and modal frequencies shall have a frequency margin not less than 20%. For ZHW PSPP, the overall structure respects this criterion, but it is evident that some local components of the structure do not satisfy this requirement. The measurements in operation also confirm that the powerhouse is amplifying locally the excitation source.

From above analysis, we obtain that for operation points at high power output, the pressure pulsation induced by RSI is the main excitation source which causes severe powerhouse vibration of ZHW PSPP. The powerhouse as a single entity shows no resonance. However, some local components of the powerhouse resonate with the pressure pulsations coming from the vaneless area. This phenomenon plays a significant role in the overall feeling of noise and vibration observed in the plant.

3. Solutions to the problem

3.1. Determination of the improvement program

There are two methods to attenuate the powerhouse vibrations. One is to optimize the structural response by increasing, for example, the rigidity of the powerhouse to change the natural frequency of local structures. The other one is to optimize the pump-turbine to reduce the intensity of vaneless area pressure pulsations induced by RSI at their source. Generally, PSPPs are constructed in underground powerhouses. For PSPPs in commercial operation, changing the powerhouse structure will result in out-of-service of units, huge economical losses, and the correction result is difficult to control. Optimization of the pump-turbine design can reduce pressure pulsations in the vaneless area, and therefore reduce powerhouse vibration. The investment of this program is low cost and feasible. Therefore, without any change in civil works or embedded components, the program to optimize the pump-turbine was selected.

During the hydraulic design period of the pump-turbine, different ways to reduce RSI effects on unit and powerhouse should be carefully considered. Factors influencing vaneless area pressure pulsations include the ratio \( \frac{D_0}{D_1} \) of guide vane pitch circle diameter \( D_0 \) to the high pressure runner diameter \( D_1 \) (i.e. the relative width of vaneless area), the cavitation conditions, the height of guide vanes, the application of desynchronized guide vanes, and some other factors as indicated in [24]. Among these factors, the most applicable methods are to change the \( \frac{D_0}{D_1} \) ratio and to optimize geometrical design parameters of the runner. The increasing of \( \frac{D_0}{D_1} \) can enlarge the width of the vaneless area, and then attenuate the flow interferences between runner blades and guide vanes to reduce pressure pulsations in the vaneless area [28,29]. It is important to point out that runner blades number has been discarded from the scope of geometric parameter changes in the case of ZHW. Since the number of guide vanes and the rotation speed were not modifiable, changing the number of runner blades would have changed the frequencies of hydraulic excitation source, and the anti-vibration calculation should be re-implemented, resulting in unforeseeable risks. For ZHW PSPP, vibration of the unit and powerhouse in pumping mode is acceptable and the optimization of the pump-turbine runner to attenuate amplitude of RSI was finally mainly focused for turbine operation.
3.2. Program implementation and evaluation on model test

Based on the assessments done, the targets of the optimization include the reduction of the vaneless pressure pulsation induced by RSI, the pump cavitation behavior in the lowest head operating area as well as the S-shape in the turbine coupling area. Pump-turbine upgrading was fulfilled by optimizing the high-pressure runner diameter. The hydraulic profile of runner blades leading edge was also modified and as well as the guide vanes hydraulic profile. Reduced scale model test is used to evaluate the new runner and to predict prototype stability within limits of transposability. The testing protocol was including the characterization of the efficiency, the limits of cavitation and the pressure pulsations in entire operating range, the stability in pump mode and in coupling zone, and the transient characteristics. For prototype rated net head, pressure pulsations scaled up from the model test are indicated in figure 7 for the vaneless area of the original runner and the new runner.

Figure 7. Vaneless area pressure pulsation transposed from model test at rated water head

We obtain that for the new runner the peak-to-peak amplitude of vaneless area pressure pulsations with 97% confidence level under the rated head of 305m decreases significantly. Specifically, between 50% and 100% of load, the overall pressure pulsation reduces by 26%~75% and the amplitude of RSI induced pressure pulsation between 50%~80%. S-shape zone characteristic curves of original runner and new runner are provided in figure 8. For large guide vane openings in the normal operation zone, the runaway curve of the original runner forms an S-shape turbine coupling area with positive slope, which will result in large rotational speed variation at low head (start-up condition with two units parallel-in). While the runaway curve of the new runner in the S-shape turbine coupling area exhibits a negative slope, which means that the speed can be stabilized in the low head, and even in the lowest water head the turbine still has a margin of 25m. The original pump-turbine employed desynchronized guide vane (MGV) to achieve parallel-in coupling condition in generating mode. The application of MGVs caused severe problem of pressure pulsation and aggravated the powerhouse vibration, which was confirmed by corresponding research in [30]. Moreover, the application of MGVs makes the control logic of guide vanes more complex and increases the risk of out-of-synchronization of guide vanes. Therefore, MGVs were removed for the rehabilitated pump-turbines. The new design greatly improved the stability of the unit.
3.3. Final qualification of the new runner

The effectiveness of the corrective program is evaluated by field tests. After rehabilitation works, series of tests, including unit stability and powerhouse vibration have been performed to investigate new runner characteristics. Key parameters are listed in Tab.1 for reference. Because several parameters were frozen during the refurbishment and excluded from the scope of the redesign, a compromise of a small efficiency loss for the pump turbine runner was expected. Despite of the enlargement of vaneless space and the modification of the guide vane profile, the power generation and absorption meets the requirement of operation.

| Table 1. Key parameters comparison of original and new runners. |
|---------------------------------------------------------------|
|                  | Original runner | New runner |
|------------------|----------------|-----------|
| Gross head (min/max) | 291/346        | 291/346   |
| Rated net head   | 305            | 305       |
| Rated speed      | 333.3          | 333.3     |
| Rated output     | 255            | 255       |
| Maximum power absorption | 263.2   | 269       |
| Runaway speed    | 451            | 445       |
| Rated flow rate  | 73.6           | 74.0      |
| Minimum flow rate of pump | 61.5     | 60.1      |

Under rated head, peak-to-peak values and frequency spectrum of vaneless area pressure pulsations from 125MW to 250MW are shown in figure 9. The vaneless area pressure pulsation measured decrease significantly at rated head, reaching 60%~70% reduction at rated load. Comparing with figure 4a, RSI induced pressure amplitude in vaneless area also decreases to a very small value. For the busbar floor where the vibration of the powerhouse was previously the most serious, a portable vibrometer was used to map the vibration RMS value distribution as indicated in figure 10 before and after rehabilitation. We note an average decrease of 70%, above the expected value of 40%. It is also interesting to point out that after rehabilitation head cover vibration decreases remarkably by an average value of 50%, which indicates a stable and safe operation of the pump-turbine. Concerns of excessive powerhouse vibration have been resolved successfully by runner replacement and guide vane profile modification.

Figure 9. New runner characteristics of vaneless pressure pulsation (amplitude and frequency)

Figure 10. Busbar vibration comparisons before (left) and after (right) rehabilitation. Red areas indicate the highest vibration level
4. Conclusions

Reversible pump-turbines are widely employed in PSPPs all over the world and several are under construction in China. Some of them have a high head and large capacity. For the hydraulic design, the RSI induced problem has become more challenging for engineers. In this paper, we introduced the refurbishment case of ZHW PSPP. The excitation source of the powerhouse has been identified by a series of tests, including stable performance tests such as vibration and pressure pulsation measurements. Modal calculation and experimental tests of the powerhouse structure have been performed to verify the source of elevated vibrations of the structure and have shown that the local natural frequency of the civil-work structure coincides with the RSI induced frequency. The assessment of the hydraulic source of pressure pulsations has also shown that the vaneless area pressure pulsations, although being within the acceptance criteria during the original design, could possibly be reduced with a more modern design. The solutions to the vibration concerns were thus focused on the replacement of the original runner and the modification of the guide vanes profile. The strategy applied in this case is to enlarge the vaneless space to reduce the RSI induced pressure pulsation. The model and prototype tests indicate that although the efficiency of the new runner decreases slightly, pressure pulsations in the vaneless area decrease significantly. The new design also provided additional stability in S-shape coupling area and enough head margin allowing to remove the desynchronized guide vanes originally used for parallel-in synchronization at low head. After this rehabilitation, ZHW PSPP increased its reliability, lowered the noise and vibration amplitudes, and finally improved the environmental condition for the staff. This experience enhances our awareness of hydraulic characteristics of pump-turbines. The lessons learned provide great value for other operating PSPPs which may have similar concerns, and guidelines on hydraulic design of new pump-turbines. Generally, we obtain the following notes from this typical case:

1) During the pump-turbine design period, attention should be paid to the vaneless area pressure pulsations, especially the phenomenon induced by RSI,

2) The RSI induced component of the vaneless area pressure pulsations can be effectively decreased by enlarging the vaneless area. The consequences of resulting pulsations, such as powerhouse severe vibration and high frequency noise can be significantly mitigated,

3) During the powerhouse structural design period, much emphasis should be focused on checking local anti-vibration calculations, especially on avoiding hydraulic, mechanical and electrical excitation sources. Sufficient margins should be guaranteed between powerhouse structure natural frequencies and excitation source frequencies.

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