

Resonance investigation of pump-turbine during startup process

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Abstract. The causes of resonance of a certain model pump-turbine unit during startup process were investigated in this article. A three-dimensional full flow path analysis model which contains spiral case, stay vanes, guide vanes, runner, gaps outside the runner crown and band, and draft tube was constructed. The transient hydraulic excitation force of full flow path was analyzed under five conditions near the resonance region. Based on one-way fluid-structure interaction (FSI) analysis model, the dynamic stress characteristics of the pump-turbine runner was investigated. The results of pressure pulsation, vibration mode and dynamic stress obtained from simulation were consistent with the test results. The study indicated that the hydraulic excitation frequency ($Z_g * f_n$) Hz due to rotor-stator interference corresponding to the natural frequency of 2ND+4ND runner mode is the main cause of resonance. The relationship among pressure pulsation, vibration mode and dynamic stress was discussed in this paper. The results revealed the underlying causes of the resonance phenomenon.

1. Introduction

With the improvement of technology of hydraulic turbine, there is a trend to increase the power concentration of the units. As a consequence, heads and fluid velocities are higher and the hydraulic excitation forces on the turbine runners increase. Furthermore, the operation range of hydraulic turbines is widened in order to satisfy the end-users' demand of larger regulation capacity. For the pump-turbines, pressure fluctuation induced by the rotor-stator interaction are the mainly excitation forces [1,2]. High excitation forces may lead vibration and high stress levels which can produce damage by fatigue [3,4]. Therefore, how to predict the natural frequencies and mode shapes of the runner as well as the dynamic stress induced by hydraulic excitation force during the design stage is of paramount importance. Based on these characteristics, the investigation of runner resonance has an important significance to the safety and stable operation for pump-turbine.



The natural frequencies and mode shapes in water are important parameters for runner design. Accurately predicting the frequencies and mode shapes of runner might be very useful. By taking effective measures, it is possible to make the runner avoid resonance, forced vibration or fatigue failure. In the practical operation, because the runner is surrounded by water, the vibration of runner structure will make the surrounding fluid vibrate together. Thus it will produce a certain added mass by water, which will change the modal characteristics of runner.

At present, a lot of research achievements of modal behavior of runner in water have been obtained both from test and simulation. It is show that the natural frequencies and mode shapes of runner obtained by experiment which submerged the runner in an open water reservoir agree quite well with the finite element simulation results [5,6,7]. However, considering the runner is stalled in real flow passage in the practical operation, there is a big difference between the channel shape and open reservoir. Furthermore, the solid boundary conditions around runner like bottom ring and head cover will change the added mass effect. Especially when the gaps between the solid boundary and runner are very small and the runner has enough deformation, the effect is very significant [1]. And when it comes to pump turbine with high head, the gaps between runner and the solid boundary is very small, so it is necessary to study the dynamic characteristics of the runner in real flow path.

At present, a lot of researchers using harmonic response analysis based on the fluid structure interaction to analyze the runner dynamic stress caused by rotor-stator interaction. Considering computing time and storages, the analysis methods and objects are simplified, and so far only some important excitation frequencies have been chosen to analyze instead of all pressure pulsation frequencies [3,4,8,9].

In this study, resonance phenomenon of the pump turbine was observed during the startup process. In Toshiba Hydraulic Research Laboratory, extensive studies on this phenomenon have been carried out by model test. The results show that the amplitude of dynamic stress significantly increased at about 88% of rated speed. This was apparently caused by some resonance of the runner at this speed.

The causes of resonance of the model pump-turbine unit during startup process were investigated by simulation in this article. The modal characteristics of the runner in the working flow passage were analyzed in detail. Besides, based on one-way fluid-structure interaction, transient dynamic analysis method was applied in the investigation of vibration stress caused by hydraulic excitation force due to interference between runner blades and guide vanes.

2. Fluid-structure interaction theory

It is well known that the discretized structural dynamics equation can be formulated as follows:

$$\mathbf{M}_s \ddot{\mathbf{U}} + \mathbf{C}_s \dot{\mathbf{U}} + \mathbf{K}_s \mathbf{U} = \mathbf{F}_s \quad (1)$$

Where, \mathbf{M}_s is the structural mass matrix; \mathbf{C}_s is the structural damping matrix; \mathbf{K}_s is the structural stiffness matrix; \mathbf{U} , the nodal displacement vector; \mathbf{F}_s , the applied load vector.

Theoretically, to take into account the fluid effect on the submerged vibrating structure, the system has to be treated as a fluid structure interaction problem. In other words, the structural dynamic equation has to be coupled with the fluid equations. The equations for the fluid structure interaction problem by using the traditional acoustic method [10], can be written in form of this [11].

$$\begin{bmatrix} \mathbf{M}_s & \mathbf{0} \\ \mathbf{M}_{fs} & \mathbf{M}_f \end{bmatrix} \begin{Bmatrix} \ddot{\mathbf{U}} \\ \ddot{\mathbf{P}} \end{Bmatrix} + \begin{bmatrix} \mathbf{C}_s & \mathbf{0} \\ \mathbf{0} & \mathbf{C}_f \end{bmatrix} \begin{Bmatrix} \dot{\mathbf{U}} \\ \dot{\mathbf{P}} \end{Bmatrix} + \begin{bmatrix} \mathbf{K}_s & \mathbf{K}_{fs} \\ \mathbf{0} & \mathbf{K}_f \end{bmatrix} \begin{Bmatrix} \mathbf{U} \\ \mathbf{P} \end{Bmatrix} = \begin{Bmatrix} \mathbf{F}_s \\ \mathbf{0} \end{Bmatrix} \quad (2)$$

In the modal analysis, the damping effect doesn't take into account, and the structure vibrates freely. In this way, the fluid structure coupled equation can be simplified as follows:

$$\begin{bmatrix} \mathbf{M}_s & \mathbf{0} \\ \mathbf{M}_{fs} & \mathbf{M}_f \end{bmatrix} \begin{Bmatrix} \ddot{\mathbf{U}} \\ \ddot{\mathbf{P}} \end{Bmatrix} + \begin{bmatrix} \mathbf{K}_s & \mathbf{K}_{fs} \\ \mathbf{0} & \mathbf{K}_f \end{bmatrix} \begin{Bmatrix} \mathbf{U} \\ \mathbf{P} \end{Bmatrix} = \mathbf{0} \quad (3)$$

For the M_f , it's just considering the fluid density and geometry dimensions.

However, in the dynamic stress calculation, the fluid added mass effect is calculated. The applied loads vector contains the pressure, gravity and centrifugal force. The transient pressure loads are obtained from three-dimensional full flow path CFD simulation, in which, the flow fluid not only considers the viscous, but also irrotational.

3. Simulation model

The model pump turbine runner has 6 blades (Z_r) and a diameter of 404mm. The number of guide vanes Z_g is 20. The rated head is 492m, and rated speed n_r is 4929 rpm. Material density of runner is 7.75g/cm^3 , Elastic modulus is 207GPa, with Poisson's ratio 0.3, and the damping ratio of 0.01.

Considering the operating conditions of high head pump turbine are complex and changeable, in order to obtain more accurate hydraulic excitation force characteristics for runner, it is necessary to consider the clearance flow of the gap in the CFD simulation. Accurate analysis of the hydraulic excitation force plays an important role to the inner flow characteristics as well as the structural dynamic response. Therefore, a three-dimensional full flow path model was constructed in this study. Except for the main flow channel which includes spiral case, stay vane, guide vane, runner, and draft tube, the clearance flow outside the crown, band and seals was also taken into account. The calculation model was shown in Figure 1.

The runner part was selected for modal and dynamic stress analysis, which contains runner structure and water domain that inside and outside it, as it is showed in Figure 2. However, the hydraulic excitation force applied on runner structure surfaces for dynamic stress calculation was obtained by CFD simulation with full flow passage.

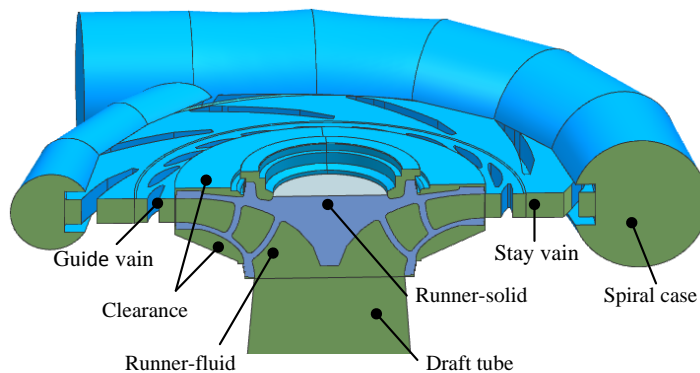


Figure 1. Calculation model

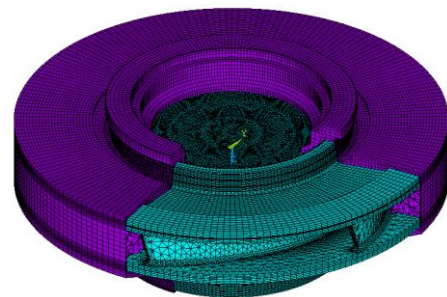


Figure 2. FEM model for dynamic characteristics analysis

The resonance point frequency of this model pump-turbine unit at laboratory is 1446 Hz, in other words, the resonance speed is about 4400 rpm. In order to investigate the inner flow fluid of the pump-turbine and the dynamic stress characteristics of the runner, and make a comparison to the measured results to verify the viability of numerical methods, 5 typical working conditions were selected for simulation in this study, as listed in Table 1.

Table 1. Simulation conditions

$n(\text{rpm})$	4200	4400	4500	4700	4929
$H(\text{m})$	357.3	392.1	410.1	447.4	492
$Q(\text{m}^3/\text{s})$	0.2079	0.2178	0.2228	0.2327	0.244

4. Modal analysis

In this paper, the modal behavior such as natural frequencies and mode shapes of this model pump-turbine runner was investigated. The modal analysis of runner in different conditions has been carried out, like in air, in open water pool and in actual working flow passage.

The typical mode shapes of runner both obtained from test and simulation were compared in Figure 3. It can be clearly noticed that the two make a good agreement. Furthermore, it is observed that the resonant mode shape of the runner was a combination of 2ND+4ND mode[1].

The natural frequencies of runner in air, open water and work condition were listed in Table 2. By comparing these results with measured values, it is found that the resonance frequency of runner in actual working flow passage was 1380Hz, which was very close to the test value of 1446Hz. Therefore, the modal analysis of runner in actual working condition might obtain more accurate results.

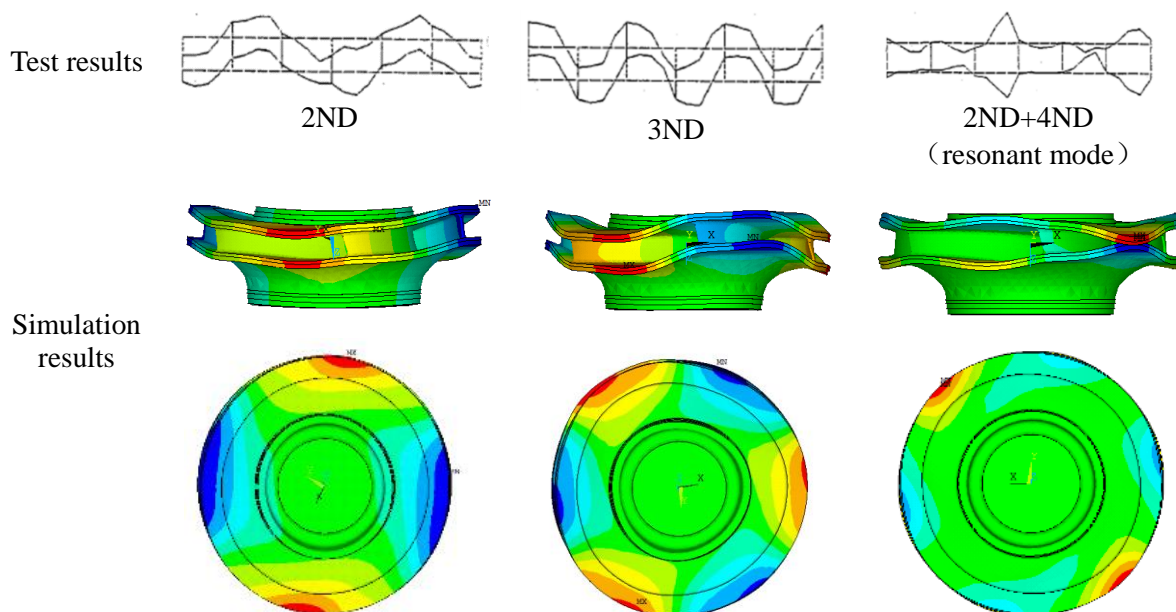


Figure 3. Comparison of runner mode shapes between test and simulation

Table 2. Comparison of natural frequencies of runner between test and simulation.

In air (Hz)		In water (Hz)	
Test	Simulation	Open water	Working flow passage
1787.5	1768.74	1442.53	916
2525	2618.25	2163.64	1555
2793.4	2798.34	1969.13	1367
3003.1	3009.21	2535.98	1380
3362.5	3425.19	2852.83	2259
3896	3845.01	3239.44	2780

5. Calculation of hydraulic excitation force

Unsteady simulation results of different rotating speed conditions were shown in Table 3

Considering the flow field stability and the calculation time, in this study, the total time for calculation of each condition is 20 times of rotation period time. Pressure pulsation results data of several rotation periods that after convergence were selected for analysis. Figure 4 presented the pressure pulsation results of different monitor points at the rated speed n_r (4929 rpm) condition. The above was the time process of pressure and the below was the corresponding frequency spectra. Here, Monitor P2 is the record point in the outlet of guide vane and near the head cover; Monitor P3 is located in the external gap of the runner crown; And Monitor P4 is the record point in the straight cone section of draft tube.

Table 3. Unsteady CFD simulation results of full flow path of model pump-turbine

$n(\text{rpm})$	4200	4400	4500	4700	4929
$\Delta H(\%)$	-0.214	-0.218	-0.267	-0.255	-0.115
$\Delta Q(\%)$	-4.14	-4.32	-4.17	-4.17	-3.98

The results show that, the dominant frequency of monitor P2 is $Z_r * f_n$ (f_n represent the rotating frequency), that is the blade passing frequency, mainly affected by the interference of runner blades. While the dominant frequency of monitor P4 is $0.9 f_n$.

A comparison of pressure pulsation results between simulation and test was made here. The result of P2 at rated speed condition was selected. As shown in Figure 5, the red dash line represents the test results, and blue solid line represents the calculation results. By comparing the time process and frequency spectra, it is clearly that both have the same change trend, and the dominant frequency also highly consistent.

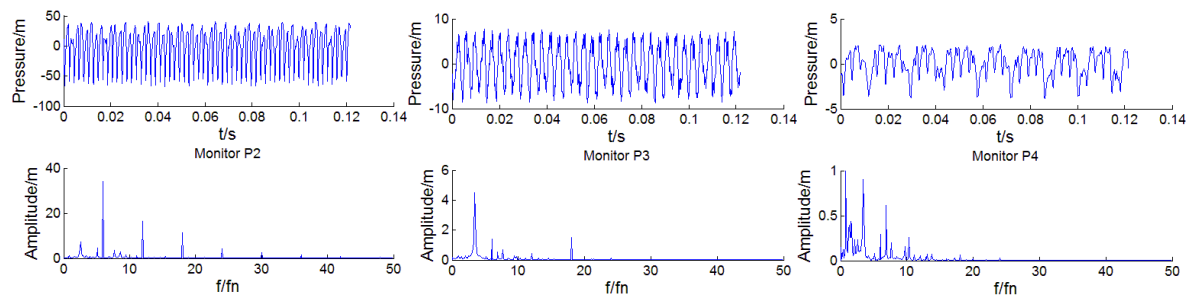


Figure 4. Time process and frequency spectra of pressure pulsation of each monitor point

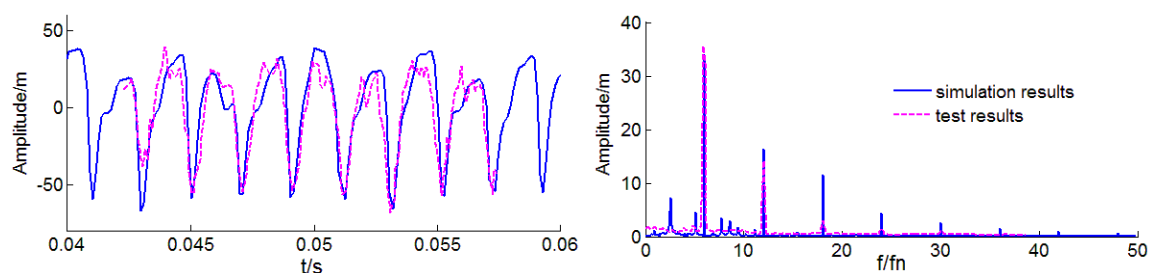


Figure 5. Comparison of pressure pulsation between simulation and test (P2, $n = n_r$)

6. Dynamic stress calculation

One-way FSI coupling method was applied in this paper. The influence of water around the runner was considered in dynamic stress analysis, thus acoustic-structural coupling calculation method was introduced to calculate the fluid added mass effect directly. In this way, it overcomes the shortcomings of wasting time and resources when using two-way FSI method, and also avoids the inaccuracy of without considering the fluid added mass effect on the structural response in the previous researches.

The distribution contours of stress and displacement of runner were presented in Figure 6. It is shown that, the maximum deformation was located at the inlet of runner crown and band between two blades, and the largest stress concentration area appeared at the fillets at the blade roots around the inlet edges.

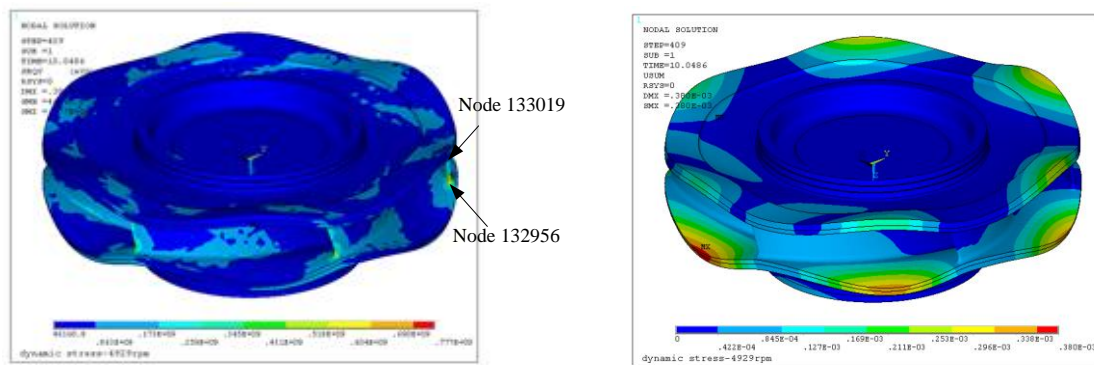


Figure 6. Stress and displacement distribution contours of runner

The calculation dynamic stress results of the corresponding test location nodes of runner crown and band at the resonance speed condition were shown in Figure 7. It is obviously that the dominant frequency of dynamic stress is $Z_g * f_n$, which suggested that the stator-rotor interaction effect between the guide vane and runner blades was the main cause of resonance for pump turbine runner.

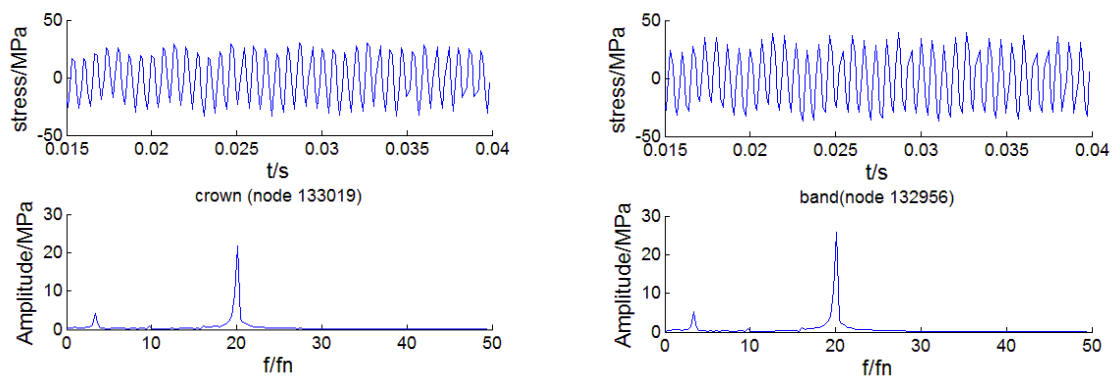
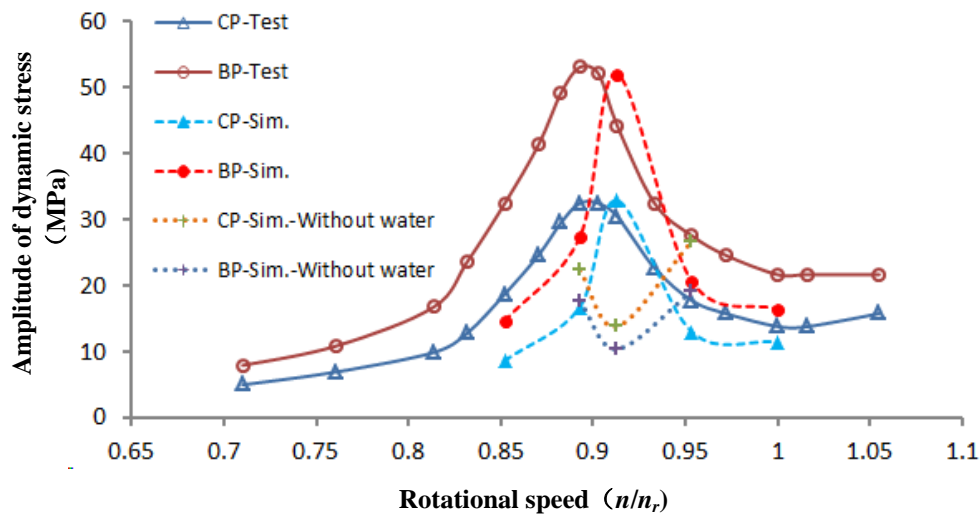


Figure 7. Time process and frequency spectra of dynamic stress results ($n=0.91n_r$, resonance condition)

The amplitude of dynamic stress results that using different calculation model were presented in Figure 8. By comparing to the test results, it is confirmed that the results of the FEM model which considering the fluid added mass effect was basically identical with the test results. The amplitude of dynamic stress increases significantly at about $0.91n_r$, it means that, resonance of the runner occurs at this speed. However, the calculation results without considering the fluid added mass effect are very different from test results.



CP - Crown Point, BP - Band Point, Sim. - Simulation

Figure 8. The comparison of dynamic stress results between calculation and test

7. Conclusion

The causes of resonance of a certain model pump-turbine unit during startup process were investigated in this article. The following conclusions were obtained.

1) The modal analysis method of runner in real working flow passage was investigated. The results of natural frequencies obtained by this method were more agreeable with the test results. The resonance characteristic of the model pump-turbine runner was accurately analyzed. The actual resonant mode observed in runner vibration was a combination of 2ND+4ND mode.

2) The hydraulic excitation force obtained from unsteady CFD simulation of full flow path was more reliable when taken into account the clearance flow outside the runner crown, band and seals. This might have a greater influence to the pressure fluctuation and runner dynamic stress calculation.

3) Based on one-way FSI coupling, and considering the fluid added mass effect, the transient dynamic stress calculation method can be more accurately to predict the resonance characteristics of runner. The results of dynamic stress obtained from simulation and test were quite consistent. The calculated and measured results both show that, the amplitude of dynamic stress was increased significantly near the resonant speed.

Acknowledgments

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