Research of Damping and Dynamic Stress for Impeller of Reactor Coolant Pump

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Abstract

For the impeller of reactor coolant pump, the calculation of dynamic stress is very important for predicting its life and reliability. In this paper, the natural frequencies of impeller in the water environment and the hydrodynamic damping ratio are obtained by the numerical simulation using the fluid-solid coupling method. At the same time, the material damping ratio of impeller is calculated by the nonlinear material damping model. It is found that the hydrodynamic damping ratio is much bigger than the material damping ratio by one order of magnitude, so the influence of the damping ratio of the material can be neglected. At the same time, the frequency response function curve of the impeller is obtained by experiment. The modal parameters such as the damping ratio and frequency are obtained, and the experimental results and numerical simulation results are compared. The 2 nodal diameters (ND) modal frequency error is 1.4%, the damping ratio error is 7.8%. The frequency domain characteristics of pressure pulsation can be obtained by CFD analysis of impeller. According to the damping ratio, the dynamic magnification coefficient is obtained, and the dynamic stress of impeller is calculated by the pseudo-static method.

Keywords

Reactor coolant pump — Wet modal-Damping ratio — Pressure pulsation — Dynamic stress

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INTRODUCTION

Reactor Coolant Pump（RCP） is a device that provides power for coolant delivery and discharges heat and is a component of the pressure boundary of nuclear power plant. Impeller is the high-speed rotating part, and also is the core component of RCP. Impeller requires long-term, safe and reliable operation under high temperature, high pressure, which is related to the integrity of Reactor Coolant Pump operation and has a direct impact on the transport of coolant. The key factors of impeller stress analysis include damping analysis, pressure pulsation analysis, resonance check and vibration response analysis.

Impeller will be subject to pressure fluctuations due to uneven flow runners in the actual work[1-2]. Disturbance can result in a series of problems such as vibration, and then affect the security of overall structure. Chiang[3] studied the response of blade pressure fluctuations. Yao[4] discussed the time-domain characteristics of pressure fluctuations of double suction pump. The working medium of RCP is liquid, and modal analysis of impeller needs to consider the additional mass of liquid[5]. David[6] studied the influence of the axial nearby rigid distance and the fluid depth on the dynamic response of a sumerged disk. Presas[7] developed and validated a method to accurately determine the FRF of submerged and confined structures by using PZTs.Liang[8] discussed the effect of added mass. The impeller damping ratio in the water environment can be divided into material damping ratio and hydrodynamic damping ratio. Lazan[9] summarized the relationship between energy dissipation of material damping and the stress amplitude through a large number of experiments. Rao[10] discussed and analyzed blade damping ratio, and proposed a method to calculate the equivalent viscous damping ratio. Bidkar[11] studied nonlinear aerodynamic damping ratio. Due to the influence of water environment, the wet modal vibration characteristics are different from those of dry modal. Tanaka[12] analyzed the vibration characteristics and dynamic stress of pump with high head, and found that modal frequency in the water decrease accordingly because of the added mass. At the same time, Rao[13] studied the calculation of the dynamic stress of the last stage of steam turbine. Singh[14] discussed the SAFE interference diagram of turbine blade vibration.

In summary, the material damping ratio, the pressure pulsation characteristics of fluid and the impeller dynamic stress have some in-depth researches. However, the numerical study and the experimental study of the damping ratio in the water environment are rare. And the study of impeller dynamic stress is also less when considering hydrodynamic damping ratio. In this paper, the damping ratio of impeller was obtained by fluid-structure interaction (FSI) in the water environment, and results of damping ratio were verified by experiments. Variation characteristics of impeller damping ratio, including material damping ratio and hydrodynamic damping ratio, with amplitude changing in wet modal was analyzed, and the proportion of each damping ratio was shown, too. Impeller surface pressure fluctuations was obtained through CFD analysis. The resonance of impeller was analyzed which provided a basis for strength check of impeller in wet modal.

1. NUMERICAL CALCULATION AND EXPERIMENTAL TEST OF IMPELLER DAMPING RATIO

Damping ratio of impeller in the water includes material damping ratio and hydrodynamic damping ratio, and total damping ratio is the sum. Material damping is the vibration energy loss caused by friction among material internal lattice, when structure is vibrating. Material damping ratio of blades is not only related to the material and the mode of blades, but also depends on stress level. Impeller is working around fluid medium. The fluid forceon blades, shroud and hub is the main reason which leads to impeller vibration. If fluid does positive work on impeller, vibration will be intensified; or it will consume energy and have damping effect, which called hydrodynamic damping dissipation energy, and the damping ratio is called hydrodynamic damping ratio.

1.1 NUMERICAL CALCULATION OF HYDRODYNAMIC DAMPING RATIO

Hydrodynamic damping dissipation energy can be expressed as

$$W = \int_{t_0}^{t_0+T} \int_A p\, \vec{v} \cdot \hat{n} dAdt \qquad (1)$$

where, P is the pressure on boundary surface, v is velocity, and n is unit normal vector on surface.T is the period of mode vibration .

If W is positive, fluid does positive work on boundary. If W is negative, fluid does negative work on boundary and has damping effect. Energy loss made by fluid is not only on blades, but also on hub and both sides of shroud. To conform the computational accuracy of damping ratio, the work on blades, shroud and hub should be taken into consideration.

Hydrodynamic damping ratio is defined as

$$\xi = \frac{W}{4\pi U_0} \qquad (2)$$

where U_0 is the modal strain energy, which can be obtained by modal analysis.

$$U_0 = \int_V UdV = \int_V \frac{\sigma_a^2}{2E} dV \qquad (3)$$

where U is unit modal strain energy. σ_a is stress amplitude. E is Young's modulus. Through the modal analysis, modal parameters were obtained. The stress amplitude is calculated by assuming an amplitude value. And the through (3), U_0 can be obtained.

The impeller parameters is in table 1. The impeller is a 1: 2.5 scale model including 6 blades and 15 guide vanes.

To obtain W in different modes, the mode shapes of the wet modal should be set as vibration boundary condition. By calculating the displacement and the force on the blade surface, W can be obtained. The model parameters of impeller are shown in table 1. The result of hydrodynamic damping dissipation energy with 2-ND first mode is in table 2. It is shown that the energy dissipation varies with amplitude and is proportional to the second power of amplitude. Likewise, modal strain energy varies with amplitude and also is proportional to the second power of amplitude, too. According to formula (2), hydrodynamic damping ratio with different amplitude can be obtained. It is shown in Figure 1 that hydrodynamic damping ratio is almost invariant with different amplitude.

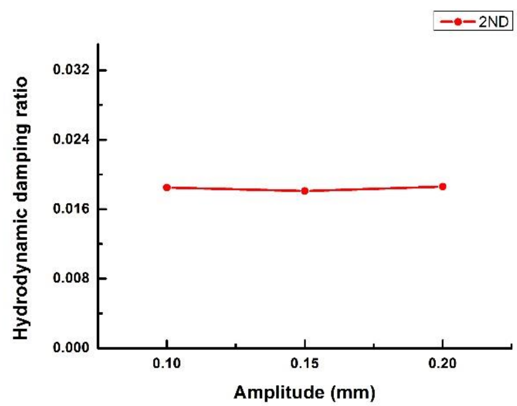


Figure 1. Hydrodynamic damping ratio with different amplitude

Hydrodynamic damping ratio is proportional to energy dissipation, and is inversely proportional to total strain energy. In the small amplitude, the energy dissipation of impeller is proportional to the pressure and velocity. And the pressure and velocity are proportional to the amplitude . Therefore the energy dissipation of impeller is proportional to the second power of amplitude. At the same time, the strain energy is proportional to the second power of amplitude. So hydrodynamic damping ratio is invariant with amplitude when amplitude is small. It is also shown that hydrodynamic damping ratio do not vary with amplitude in both

Hydrodynamic damping ratio is almost considered to be a constant in the same mode according Fig 1 and Table 2.

Table 1. Impeller parameters

Propertied	Inlet diameter (mm)	Outlet diameter (mm)	Axial length (mm)	Number of blades	Number of guide vanes
Value	286	350	205	6	15

Table 2. Hydrodynamic damping ratio in the 2-ND first mode

Maximum amplitude (mm)	0.100	0.150	0.200
Disspation energy of hydrodynamic damping (J)	-2.87	-6.40	-11.549
Modal strain energy (J)	12.4	28.1	49.4
Hydrodynamic damping ratio	0.0185	0.0181	0.0186

1.2 NUMERICAL CALCULATION OF MATERIAL DAMPING RATIO

Material damping ratio is defined as

$$\xi = \frac{D_0}{4\pi U_0} \tag{4}$$

where U_0 is modal strain energy, D_0 is material damping dissipation energy.

$$D_0 = \int_V D dV = \int_V (11732.04 \times \left(\frac{\sigma_a}{\sigma_f}\right)^{2.4}) dV \tag{5}$$

where σ_a is stress amplitude, σ_f is fatigue strength of the material.

According modal analysis, U_0 and D_0 can be obtained, then material damping ratio can be calculated.

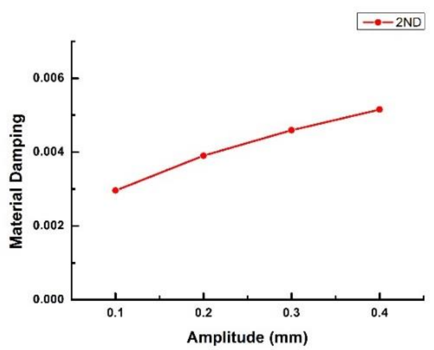


Figure 2. Material damping ratio with different amplitude

As shown in Figure 2, in the same mode, material damping ratio varies with amplitude, and the relationship is nonlinear. Comparing Figure 1 and Figure 2, in the same mode with the same amplitude, hydrodynamic damping ratio is about one order of magnitude higher than material damping ratio. Therefore, in a range of small amplitude, the effect of material damping ratio can be ignored, so that hydrodynamic damping ratio is approximately equal to total damping ratio. So, total damping ratio can be considered to be a constant for a specific mode, and it is invariant with amplitude. For this reason, the problem can be simplified.

Hydrodynamic damping ratio of impeller in wet modal is obtained by numerical simulation, and it is measured in the test at the same time. Comparing the two results, there is a difference of

7.8% in 2-ND first mode. What's more, research on material damping ratio shows that material damping ratio is far less than hydrodynamic damping ratio in water environment. Therefore, it can be considered that hydrodynamic damping ratio is almost equal to total damping ratio in water environment.

1.3 EXPERIMENTAL TEST OF IMPELLER DAMPING RATIO

During the test, impeller damping ratio is obtained via the single-input multi-output analysis method(SIMO). In the experiment, the model pump was placed in the bucket with diameter of 1 m and the depth of 1 m.

The measuring system software that is called DASP is used to obtain frequency-response function of model pump, including acquisition system and signal processing system. The sensor used in the test is acceleration sensor, with sensitivity of 50 mv/g, frequency range of 0.3-12000 Hz. Sensors are distributed on shroud and blades uniformly in circumferential direction as shown in Figure 3. Use theimpulse-response method, and the impeller was freely suspended in the middle of the bucket in the experiment as shown in Figure 4.

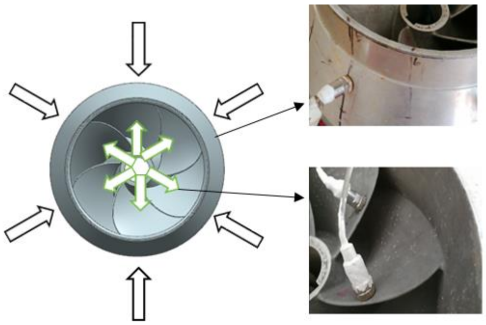


Figure 3. Distribution of sensors

Frequency-response function curve measured in the test is shown in Figure 5. The half-power bandwidth range of frequency response function of wet modal is wide, and modes are relatively dense, which caused by big damping. The coherence function graph is shown in Figure 6. The coherence coefficient of the first-order model and the second-order model are 0.94573 and 0.93676 that are meeting the requirements of coherence. According to frequency response function curve, modal frequency and modal damping ratio are obtained. Eigensystem Realization Alogrithm(ERA) and Infinite Impulse Response (IIR) algorithm can be used to analyze frequency response function curve. The ERA, as a model parameter identification method in time-domain, has received more and more attention in recent years. And when using IIR, the characteristic equation coefficients are deduced by IIR filter theory. The first peak of frequency response function corresponds to 2-ND first mode. The results of ERA algorithm and IIR algorithm were compared in Table 3. In 2-ND mode, the frequency is 1136 Hz by ERA algorithm, and frequency is 1132 Hz via IIR algorithm. The error is 0.35%. In 3-ND mode, the frequency calculated via ERA algorithm is 1329Hz, and that is 1330 Hz via IIR algorithm, the error is 0.075%. That means the results via the two algorithms are almost same, and IIR algorithm will be used.

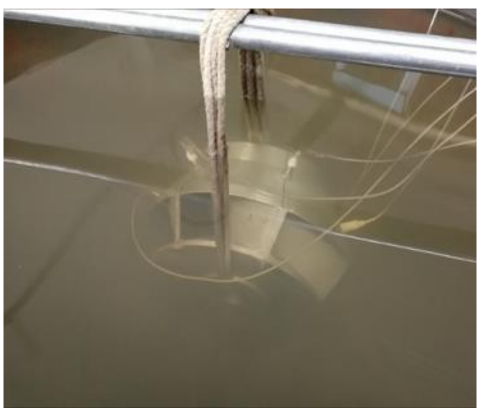


Figure 4. Hanging impeller in the bucket

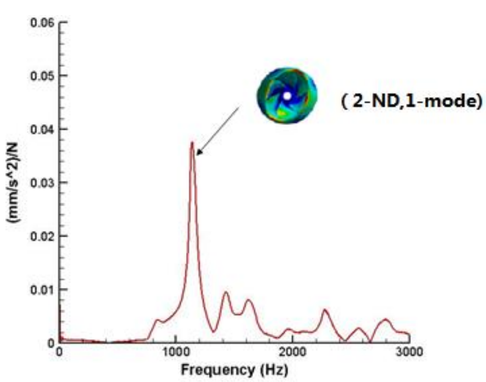


Figure 5. Measured frequency response function

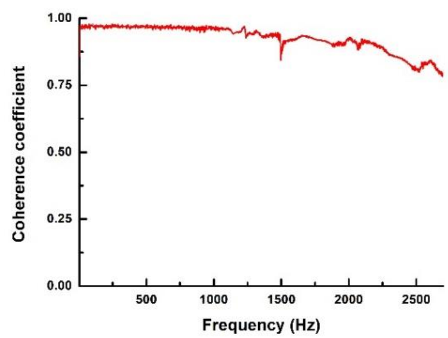


Figure 6. Coherence coefficient function

Table 3. Different Methods of Modal Parameter Identification

	2-ND	3-ND
ERA (Hz)	1136	1329
IIR (Hz)	1132	1330
Error (%)	0.350	0.0750

With the methods of numerical simulation and experiment, the result of impeller damping ratio with 2-ND first mode is shown in Table 4. There is a difference of 7.8% between the two methods. Both the accuracy of test and modal damping ratio detecting method are the reasons that cause the difference.

Table 4. Numerical Simulation and Experimental Comparison of Damping Ratio of Impeller 2-ND First Mode

	2-ND First Mode
Experiment	0.0197
Simulation	0.0182
Error (%)	7.80

2. RESONANCE VIBRATION CHECK AND DYNAMIC STRSS ANALYSIS OF IMPELLER

Due to the uneven distribution of runners, impeller is under pressure load in water environment, which caused corresponding dynamic stress and amplitude. To analysis dynamic stress on impeller, besides calculating damping ratio, it is also necessary to calculate pressure fluctuation.

Analysis of impeller resonant is an important part of strength analysis. Via Fast Fourier Transformation(FFT), pressure fluctuation signals in time domain can be transformed into signals in frequency domain. By modal analysis, interference diagram of impeller can be obtained. Then through pseudo-static analysis, dynamic response and amplitude can be analyzed, too.

2.1 NUMERICAL SIMULATION OF IMPELLER IN WET MODAL

Numerical calculation of impeller in wet modal is completed. On wet modal condition, the coupling of impeller surface and fluid, as well as the effect of added mass of water should be taken into consideration. Sound velocity, density and some other acoustic parameters should be defined. In table 6, there are some impeller material properties used in the calculation. Impeller is in the middle of the sink. Figure 7 shows FEM numerical modelof impeller and water for wet modal analysis.

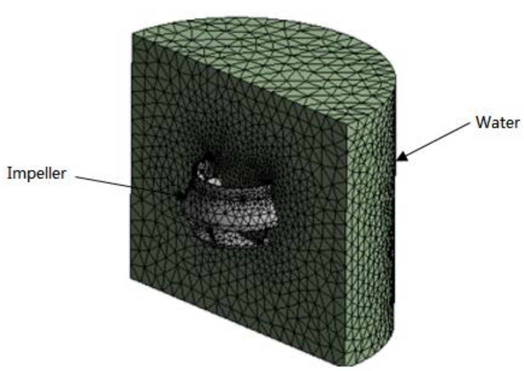


Figure 7. FEM model of impeller and water

As shown in Table 5, the results of impeller natural frequency in wet modal are almost consistent via experiment and numerical simulation. There is a difference of 1.2% in 2-ND first mode, and 1.1% in 3-ND first mode.

Table 5. Numerical simulation and experimental comparison of natural frequency of wet modal

	2-ND	3-ND
Experiment (Hz)	1132	1330
Simulation(Hz)	1149	1315
Error (%)	1.50	1.10

2.2 CALCULATINO OF IMPELLER PRESSURE FLUCTUATION

Due to the uneven distribution of runners and the rotor-stator interaction, there are pressure fluctuating loads on impeller surface. In order to do resonance vibration check and study dynamic stress, characteristics of pressure fluctuation should be analyzed first.

Steady and unsteady numerical simulations are done on impellerchannel. Impeller model includes 6 blades and 15 vanes. Computational runners includes impeller runners and vane runners, adding extended parts to inlet and outlet. Results will be recorded when impeller rotating 3° every time, and there are 6 periods recorded in total. After obtained pressure fluctuation time-domain characteristics of impeller, the information can be transformed into frequency-domain characteristics. Figure 8 shows impeller channel model for CFD calculation.

2.3 RESONANCE VIBRATION CHECK OF IMPELLER

As shown in figure 9, which is interference diagram of impeller, impeller frequency with 2-ND first mode is close to the exciting force which is 46 times the rotation frequency. The slope of the exciting force line is the rotation frequency. When the natural frequency and excitation frequency (rotational frequency and its multiples) is close, the resonance phenomenon is easy to occur. Therefore, impeller resonance with 2-ND first mode should be analyzed.

Dynamic response of resonance can be calculated by quasi-static method. Equation (6) and (7) explain the relationship between static and dynamic response.

$$\Delta_d = \Delta_s \cdot Q \tag{6}$$

$$\sigma_d = \sigma_s \cdot Q \tag{7}$$

Where σ is stress, Δ is amplitude, subscript d represents the result of dynamic response, subscript s represents the result of static response, Q is dynamic amplification factor, and it is defined as

$$Q = \frac{1}{2\xi} \tag{8}$$

Table 6. Properties of the impeller material

Propertied	Density	Young's modulus	Poisson's ratio
Value	2600kg/m^3	71GPa	0.330

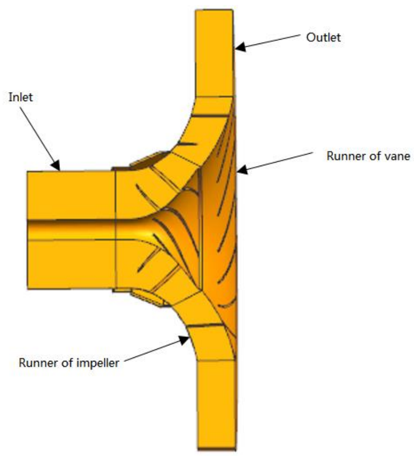


Figure 8. Impeller channel model for CFD calculation

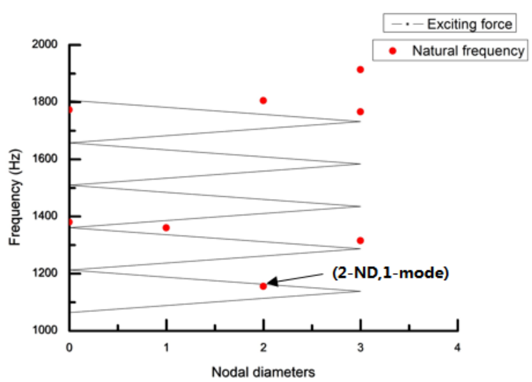
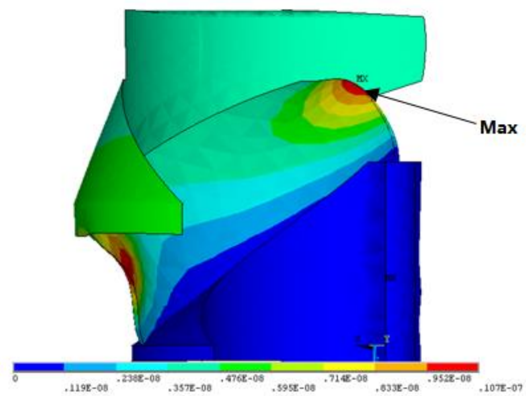


Figure 9. Interference diagram of impeller

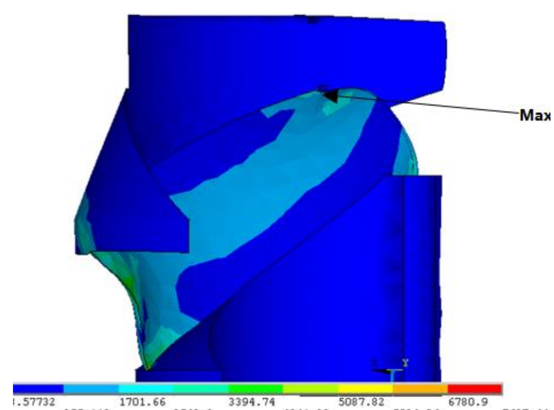
Amplification factor can be calculated using damping ratio obtained before, then impeller dynamic response can be calculated according to static stress and amplification. Equation (8) shows that for the given load, the dynamic stress is inversely proportional to the damping ratio.

3. RESULTS AND DISCUSSION

Through the interference diagram of the impeller, the resonance mode is found as 2-ND first mode. Fig. 10 shows the static displacement and equivalent stress of the impeller sector under 2-ND first mode. The maximum static stress was 0.007627 Mpa and the maximum displacement was 1.07 × 10-8 mm as can be seen in Fig 10. The maximum static stress is at the blade root, and the maximum displacement occurs at the medium diameter of blade. Table 7 and Table 8 shows the process of quasi-static analysis.



(a)



(b)

Figure 10. Static displacement (a) and equivalent stress (b) of impeller sector in 2-ND first mode resonance

Through the quasi-static method, the resonance of the 2-ND first mode of impeller is analyzed. As can be seen from Table 7 and Table 8, the modal damping ratio is 1.972%, the amplification factor is 25.35, the modal resonance amplitude of sector is 1.529 $\times 10^{-6}$mm, and the equivalent dynamic stress is 0.1933MPa. The impeller is a 1:2.5 scale model in this paper, so the natural frequency of the impeller is higher than the frequency of the prototype impeller. When the resonance is achieved, the frequency of exciting force is often high. And the pressure pulsation energy is basically concentrated in the low frequency. So when the scale model is calibrated, the stress and amplitude are small. This paper provides a calculation method of the dynamic stress of the impeller in the water environment. For a prototype impeller, the fluid exciting force of the low frequency may cause a high dynamic stress.

Table 7. Quasi - Static Analysis of Sector of Impeller

Rotation Frequency Multiplier of Impeller	Static Displacement /mm	Equivalent Static Stress / MPa
46	6.03×10-6	0.00763

Table 8. Quasi - Static Analysis of Sector of Impeller (dynamic analysis)

Damping Ratio	Dynamic Amplification Factor	Maximum Amplitude /mm	Equivalent Dynamic Stress / MPa
0.0197	25.4	1.53×10-4	0.193

4. CONCLUTION

In this paper, the damping ratio of impeller in the water environment was analyzed. It is found that the material damping ratio changes with the amplitude, and this change is a nonlinear relationship in the same mode. However, the hydrodynamic damping ratio does not change substantially with the amplitude change, and is a constant in the same mode. At the same time, hydrodynamic damping ratio is higher than the material damping ratio by an order of magnitude in the same mode and amplitude. Therefore, ignoring the material damping ratio, hydrodynamic damping ratio of impeller can be approximated as a total damping ratio in the water environment. And the experimental results and numerical simulation results of the wet modal of the impeller were compared. The natural frequency error is 1.5% in 2-ND first mode and the natural frequency error is 1.1% in 3-ND first mode. The results show that the numerical simulation results were in good agreement with the experimental results.

Through interference diagram of the impeller, the resonance zone of the impeller can be found in the water environment. When the exciting force frequency is 46 times greater than the rotation frequency of impeller, the impeller is close to the resonance region of 2-ND first mode. The damping ratio is 1.972%, and dynamic amplification factor is 25.35 in 2-ND first mode resonance. By means of quasi-static method, the amplitude and dynamic stress were obtained in the resonance of 2-ND first mode.

Based on the analysis of the damping ratio, wet modal and pressure pulsation of the impeller, the vibration response analysis process of impeller was established, which can provide a reference for the strength calculation of impeller.

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