The Research on Vibration Mode of a Ship Pump Group

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Abstract. With the intention of reducing the vibration and noise of a ship pump group, rubber dampers were installed between the motor base and the foundation. The finite element analysis method was used to analysing the influence of the stiffness of the rubber damper on the damping effect. It was found that the rubber shock absorber significantly reduces the low-order modal frequency of the group. When the structure has good symmetry, the frequency of the translational mode is proportional to the square root of the stiffness of the rubber damper in the corresponding direction. And some impedance characteristics of the pump group were obtained by harmonic response analysis, which are beneficial to pick the right rubber damper to avoid the modal resonance frequency.

1. Introduction

Many of the auxiliary equipment in the ship are connected to the hull deck through the base, so the vibration of the auxiliary equipment is mainly transmitted to the hull through the base. Therefore, the impedance characteristics of the base are of great significance to the vibration and noise reduction of the whole group. In this paper, rubber dampers were installed between the motor cabinet and the foundation, which made the impedance of the system change abruptly. The vibration energy is absorbed and consumed by the rubber damper in the transmission path of vibration energy, thereby achieving the purpose of vibration and noise reduction. The research object of this paper is a ship water pump group, whose working speed is 300 RPM.

2. The Basic Theory

For the vibration of a multi-degree-of-freedom system, the complex-modal vibration differential equation [1] of a general viscous system can be described as

\[
M\ddot{x} + C\dot{x} + Kx = f(t)
\]  

(1)

\(M\) and \(K\) are the mass matrix and stiffness matrix of the system respectively, and \(C\) is the general viscous damping matrix. Let the exciting force as \(f(t)\) and the steady-state displacement response as \(x(t)\).

\[
f(t) = Fe^{j\omega t}
\]  

(2)

\[
x = Xe^{j\omega t}
\]  

(3)

The velocity is the derivative of the displacement

\[v = \dot{x} = i\omega Xe^{j\omega t}\]  

(4)

The acceleration is the derivative of the velocity
\[ a = \dot{v} = -\omega^2 X e^{j\omega t} \tag{5} \]

Substitute (2) - (5) to (1), and the original differential equation can be rewritten as:

\[ X = H(\omega) F, H(\omega) = (K + j\omega C - \omega^2 M)^{-1} \tag{6} \]

\[ Z_{D}(\omega) = H(\omega)^{-1} \tag{7} \]

\( H(\omega) \) represents the frequency response function matrix and \( Z_{D}(\omega) \) for Displacement impedance [2] matrix.

3. Finite Element Analysis for Models

3.1. The Finite Element Models

The foundation was modelled in ANSYS [3] with solid 187. In this model, the opening of the pump is aligned with the X direction, the opening of the motor cabinet is aligned with the Z direction, and the vertical direction is aligned with the Y direction. In the actual case, the foundation is connected to a test platform with high stiffness by 26 M20 bolts. The inner face and end face of the bolt hole were fixed to simulate the bolt connection between the base and the test platform, and the vertical displacement of the bottom surface of the base was restrained to set precise boundary conditions [4]. The foundation and the motor cabinet are made of special material high manganese steel, the other structures are made of common structural steel. The rubber dampers are installed in the four corners of the foundation.

3.2. Modal Analysis

Vibration modes are inherent and integral characteristics of the elastic structure. Modal analysis can be used to find out the characteristics of the main modes of a structure in a range of susceptible frequencies [5]. Then, the actual vibration response of the structure in this frequency band, generated by external or internal vibration sources, can be predicted.

3.2.1. Results of foundation

![Figure 1. The first mode](image1)
![Figure 2. The second mode](image2)
![Figure 3. The third mode](image3)

![Figure 4. The fourth mode](image4)
![Figure 5. The fifth mode](image5)
![Figure 6. The sixth mode](image6)
The frequency of first-order mode is 111.67Hz, which shows the translation of the whole structure along the x direction. The frequency of the second-order mode is 138.24Hz, and its mode shape is the translation of the structure along the z direction. The third-order mode has a frequency of 168.08Hz, and its mode shape is that the structure rotates around the vertical axis. The fourth-order mode has a frequency of 288.11Hz, and its mode shape is the stretching and compression of the opposite side. The fifth-order mode has a frequency of 315.01Hz, and its mode shape is diagonal stretching and compression. The frequency of the fifth-order mode is 323.01Hz, which shows the vertical translation

3.2.2. Results of motor cabinet

The motor cabinet was modelled with shell 181 with the boundary condition of completely free. The above are the mode shapes of the first six modes whose frequencies are not 0. It can be seen the four corners of the upper plate deformation are obvious, are the most vulnerable part of the cabinet.

3.2.3. Results of the assembly of the motor cabinet and foundation

Springs with three directional stiffness were used to simulate the rubber dampers between the base and the motor base. The mode of 33.317 Hz shows the motor cabinet moves along the X direction. The mode of 50.081 Hz shows the translation of motor cabinet in the Z direction. The frequency of the third-order mode is 58.738 Hz, and its mode shape is the motor seat is rotated around the vertical axis. The mode of 64.051 Hz describe vertical translation of motor base. Modes of 71.489 Hz and 81.779 Hz describe rotations around the Z and X direction correspondingly.
The foundation and motor cabinet have high modal frequency respectively, but after assembly with a rubber damper, the modal frequency of the assembly drops significantly. It is obvious that foundation has almost no movement, because the stiffness of the spring is very small compared with the foundation and motor cabinet, and the spring connection is the weakest part, and the stiffness of the spring connection has a significant influence on the mode. The foundation and motor cabinet were modelled according to the real models. Because the internal structure of the rotor is very complicated and its stiffness is large, the pump and the motor have little effect on the modal analysis results. To simplify the calculation, only the approximate shape of the pump and the motor is retained [6]. By adjusting its density, the position of the centre of gravity of the model is consistent with the position of the centre of gravity of the pump group in the real situation. At the same time, when the pump group is working, the pump inlet and outlet will be connected to the pipeline which also has a restraining effect on the pump group. The spring with three directional stiffness was used to simulate the pipeline. After the motor and pump body were assembled, the model mass increased and the frequency of first six modes decreased significantly, but the mode shape did not change much.

3.3. Summary and Simplified Hypothesis

Most vibration structures can be discretized into multi-degree-of-freedom systems with finite degrees of freedom. Within the linear range, the vibration response is the linear superposition of the main
vibration by degrees. If only the motion of a certain degree of freedom is considered, it can be simply treated as a single-degree-of-freedom system.

The differential equation of vibration in regular form for a single degree of freedom vibration system [8] is shown in the figure above can be written as:

$$\ddot{x} + 2\sigma \dot{x} + \omega_0^2 x = 0$$

(8)

Attenuation damping coefficient $\sigma = \frac{c}{2m}$, undamped natural frequency $\omega_0 = \sqrt{\frac{k}{m}}$.

### Table 1. The first six mode frequencies

|   | Foundation | Motor cabinet | Assembly of foundation and motor cabinet | Pump group |
|---|------------|---------------|-----------------------------------------|-----------|
| 1 | 111.67Hz   | 0 Hz          | 33.317 Hz                               | 10.694 Hz |
| 2 | 134.28 Hz  | 0 Hz          | 50.081 Hz                               | 14.383 Hz |
| 3 | 168.08 Hz  | 0 Hz          | 58.738 Hz                               | 15.087 Hz |
| 4 | 288.11 Hz  | 4.0559\times10^{-4} Hz | 64.051 Hz                               | 18.162 Hz |
| 5 | 315.01 Hz  | 1.5602\times10^{-4} Hz | 71.489 Hz                               | 21.208 Hz |
| 6 | 323.01 Hz  | 1.9086\times10^{-3} Hz | 81.779 Hz                               | 21.946 Hz |

It can be seen from equation (8) that the natural frequency is proportional to the square root of the stiffness and inversely proportional to the square root of the mass. Since the stiffness of the spring is smaller than that of the foundation and motor cabinet, the reduction of stiffness makes the modal frequency of the assembly decrease, which is in accordance with the law. After assembling the motor and pump, the increase of mass makes the modal frequency of the assembly further decrease, which also conforms to the law.

### 4. Verification and Analysis

With the attention to further study the effect of the spring stiffness on the modal frequency of the structure, three different sets of stiffness were set for the spring. The three sets of parameters are respectively from the rubber damper of the EA400, BE300 and E400. The translational motion in three directions were studied, and the motion in this direction was analogous to a single-degree-of-freedom spring-proton system. At the same time, assuming the motions in the three directions are orthogonal to each other, then the low-order natural frequency should be proportional to the square root of the stiffness of the spring.

### Table 2. Spring stiffness setting

|    | EA400(kg/cm) | BE300(kg/cm) | E400(kg/cm) | (BE300/EA400)^0.5 | (E400/EA400)^0.5 |
|----|--------------|--------------|-------------|-------------------|-------------------|
| KX | 1700         | 5500         | 2400        | 1.798692          | 1.188177          |
| KY | 6500         | 6000         | 13000       | 0.960769          | 1.414214          |
| KZ | 5000         | 17000        | 6200        | 1.843909          | 1.113553          |

The table below shows the 0-12th order modal frequencies. It can be seen the spring stiffness has a significant effect on the first six modal frequencies, but has little effect on the frequency of the seventh and higher modes. The energy of higher-order modes accounts for a lower proportion of the vibration energy of the structure and has little effect on the vibration of the entire structure [8]. Since the working speed of the pump is 300RPM that the exciting frequency of the structure is not high, it is enough to focus only on the first six modes.
Table 3. Frequencies of 0 to 12th mode

| Order | Foundation (Hz) | Motor cabinet (Hz) | Assembly of foundation and motor cabinet (Hz) | Pump group with EA400(Hz) | Pump group with BE300(Hz) | Pump group with E400(Hz) |
|-------|-----------------|--------------------|----------------------------------------------|---------------------------|---------------------------|--------------------------|
| 1     | 111.67          | 0                  | 33.317                                       | 10.694                    | 13.274                    | 12.634                   |
| 2     | 134.28          | 0                  | 50.081                                       | 14.383                    | 15.201                    | 17.804                   |
| 3     | 168.08          | 0                  | 58.738                                       | 15.087                    | 18.683                    | 19.418                   |
| 4     | 288.11          | 0.00041            | 64.051                                       | 18.162                    | 20.432                    | 22.745                   |
| 5     | 315.01          | 0.00156            | 71.489                                       | 21.208                    | 25.559                    | 24.758                   |
| 6     | 323.01          | 0.00191            | 81.779                                       | 21.946                    | 36.854                    | 28.901                   |
| 7     | 346.66          | 324.37             | 131.32                                       | 114.844                   | 114.8                     | 115.54                   |
| 8     | 379.58          | 376.15             | 139.42                                       | 119.653                   | 119.66                    | 119.68                   |
| 9     | 429.65          | 393.28             | 174.51                                       | 126.337                   | 148.387                   | 129.68                   |
| 10    | 480.38          | 407.53             | 289.77                                       | 138.905                   | 153.521                   | 140.72                   |
| 11    | 489.34          | 542.34             | 319.03                                       | 173.846                   | 177.736                   | 175.45                   |
| 12    | 489.9           | 598.58             | 325.57                                       | 175.332                   | 180.091                   | 175.86                   |

The lowest natural frequencies of translational motion in three directions were found, and the corresponding proportional relationship was calculated with the data of EA400 as the reference. It can be seen that except for the Z direction, that is, the direction of the opening on the motor cabinet, the simulation values of the other two directions are not much different from the expected values.

There are two reasons for the error of the results in the Z direction: one is that the foundation and the motor cabinet have openings on only one side, which destroys the symmetry of the structure in the Z direction; the other is that the eccentric installation of the motor also destroys the symmetry of the structure. This result confirms that the stiffness at the joint can significantly and clearly affect the modal frequency of the pump group once again. Since the translational mode frequencies in the vertical and X direction are proportional to the square root of the spring stiffness in the corresponding direction, it is convenient to modify the spring stiffness to change the frequency of the mode to reduce the coincidence with the working frequency and its multiple frequency, so as to avoid resonance.

Table 4. Results of simulation and prediction

| Direction | Simulation value | Predictive value | Error (%) |
|-----------|-----------------|-----------------|-----------|
| BE300     |                 |                 |           |
| X         | 1.181447        | 1.188177        | 0.566414  |
| Y         | 1.362755        | 1.414214        | 3.638700  |
| Z         | 0.980293        | 1.113553        | 11.96710  |
| E400      |                 |                 |           |
| X         | 1.747146        | 1.798692        | 2.865749  |
| Y         | 0.963434        | 0.960769        | 0.277382  |
| Z         | 1.407275        | 1.843909        | 23.67980  |

5. Harmonic Response Analysis
The harmonic response analysis is used to determine the steady-state response of a linear structure subjected to a sinusoidal load over time [9]. The purpose of harmonic response analysis is to calculate the curve of structure response with frequency. Thus, the designer can predict the continuous dynamic characteristics of the structure and verify whether the structure can overcome the harmful effects caused by resonance, fatigue and other forced vibration.
Calculating with the harmonic response module in ANSYS. A vertical downward force of 2000N was applied to the four corners of the upper panel of the motor base. According to the natural frequency range of the structure, the results of harmonic response in 0 Hz ~ 700 Hz are analyzed. The value calculated here is the displacement response of the centre of the upper plate of the motor cabinet.

| Table 5. Results of harmonic response analysis |
|-----------------------------------------------|
| Magnitude sort | 1  | 2  | 3  | 4  | 5  |
| EA400(Hz)      | 11 | 175| 14 | 21 | 221|
| BE300(Hz)      | 20 | 37 | 178| 15 | 222|
| E400(Hz)       | 29 | 24 | 176| 222| 13 |

It can be seen from the results of the harmonic response analysis that the frequencies with large amplification factor are still concentrated in the low-order frequency band. From the displacement response in figures 20-22, the resonance frequencies are keeping the same with the natural frequencies.
in mode analysis. The rationality of the results of the harmonic response analysis is verified. The working speed of the pump is 300 RPM, so the mode frequency should avoid 5Hz and its multiplier to prevent resonance. From the results of modal analysis and harmonic response analysis, the E400 can be used to avoid resonance.

6. Conclusions
The least rigid components in the assembly have a significant impact on the low-order mode and low-frequency vibration characteristics of the structure. Therefore, it is possible to use a rubber damper to adjust the modal frequency of the structure. After the rigidity of the base is raised to a certain extent, it has almost no effect on the improvement of the low-order mode frequency of the structure, and can only affect the frequency of the high-order mode. When the structure has good symmetry, the lowest order mode frequency of the corresponding direction is mainly proportional to the square root of the stiffness in the corresponding direction. From the results of modal analysis and harmonic response analysis, the E400 can be used to avoid resonance.

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