Modal Analysis and Experimental Study of Marine Gear Box

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Abstract. In this paper, the finite element model of marine gearbox is established, using Lanczos method performed the modal analysis of the gearbox system mathematical, the modal frequencies and modal shapes of each order of the box were obtained, and the modal experiment of marine gearbox is carried out. The results show that by comparing the data obtained from the experimental modal analysis with the theoretical calculated values, it is verified that the theoretical analysis values and experimental test values are in better agreement, and the maximum frequency error is 4.04%, the fea model is reliable, thus theoretical model could better reflect the natural vibration characteristics of the box on the marine gear box.

1. Introduction
At present, scholars and experts at home and abroad have conducted extensive research on experimental modal analysis, and have long involved in the fields of machinery, construction, civil engineering, aerospace and other fields. In 2007, Wang Ji et al. [1] conducted a modal analysis of the gearbox for serious vibration and noise problems in marine gearboxes. In 2004, Yang Ping [2] and others made modal analysis and research on the structural characteristics of diesel engines and the characteristics of vibration noise and strength stiffness in their work. This provides a way to improve the research and development of old and new models. The study of mechanical structure provides a reference. In 2006, Yang Chuanqi [3], passed the test of the dynamic characteristics of the XK717 CNC milling machine and did experimental modal analysis and mechanism optimization analysis of the overall machine tool. In 2011, Zhu Zhuangrui [4] had established a body dynamics model with the shell unit as the main body, performed experimental modal analysis on the body and modified the model based on the results. In 2015, H. Nahvi et al. [5] used experimental modal analysis and finite element modal analysis techniques to diagnose the location and depth of cracks on cantilever beams. In 1999, A. D. Gupta used [6] calculation and experimental modal analysis to evaluate and analyze the performance of the armored vehicle's turret. In 2009, R. Farshidi et al. used [7] the method of air to generate excitation for the most cantilever beam for experimental modal analysis. In this paper, modal analysis and experimental research are carried out for marine gearbox.

2. Modal Analysis of Gear Box
The dynamic characteristics of the vibration system are described by means of modal analysis, which is also the premise of calculating the dynamic response of the vibration system under excitation.
2.1. Theory of modal analysis

On the analysis of complex gearbox system as well as the finite element method program to realize the modal analysis of system structure is not reality, this paper has established by using Abaqus software to gear box system modal analysis, the mathematical model and using Lanczos extraction gearbox system natural vibration properties of eigenvalue and eigenvector, modal analysis theory to realize this system.

2.1.1. Lanczos method. When Lanczos is used to solve the eigenvalues, for given $M$, $K$ and $u$, which is written as:

$$Au = uT$$

The type is $u = [u_1, u_2, \ldots, u_r]$.

The matrix is shown as follows:

$$T = \begin{bmatrix}
\alpha_1 & \beta_2 \\
\beta_2 & \alpha_2 & \beta_3 \\
\beta_3 & \alpha_3 & \ddots \\
\ddots & \ddots & \beta_r \\
\beta_r & \alpha_r & \end{bmatrix}$$

(2)

On this basis, the Lanczos vector representing the eigenvalue problem can be obtained by introducing $\phi_r = uZ$ ($Z$ is the order matrix $r \times r$), transforming the original eigenvector to generate the Lanczos vector, at the same time, and transform the generalized eigenvalue equation:

$$TZ = Z \lambda$$

(3)

By solving the standard eigenvalues of (3), the following Eigen solutions can be obtained:

$$Z = [z_1, z_2, \ldots, z_r], \ \lambda = \text{diag}(\lambda_1, \lambda_2, \ldots, \lambda_r)$$

(4)

Further eigenvalue solution of the original problem can be obtained as follows:

$$\phi_r = QZ$$

(5)

In this way, the eigenvalue problem of $A$ is transformed into the eigenvalue problem of $T$. From the above derivation process, it can be seen that Lanczos method is used to extract the eigenvalues of the matrix $T$ obtained by calculation and simplification, which are the approximate values of characteristic roots of the matrix $A$. Therefore, Lanczos method can effectively and quickly extract the eigenvalues of real modes in the subspace. This method is used in modal analysis of large Marine gearbox system.

2.1.2. Finite element model and modal analysis. The finite element modal analysis process of ABAQUS is mainly divided into three steps, namely pre-processing, loading solution and post-processing. The pre-processing is mainly to create the solid model and the finite element model, define the element attribute, divide the mesh and modify the finite element model. The loading solution
includes analysing the type and characteristics of the load, determining the action position and size of the load, selecting the calculation type and setting various parameters. Post-processing is to view the final analysis results, that is, natural frequency and relative stress and deformation, can be observed in the general post-processing modal analysis results.

2.1.3. Modal analysis results. In Abaqus, the model of the box on the gear box was calculated by modal analysis, and the modal natural frequency and modal mode of the box system on the gear box were obtained by finite element analysis in the suspended state. In the dynamic problem of the structure, the response of the structure is usually controlled by the first mode of vibration, but only the first mode is excited in the modal experiment. Therefore, only the first 12 main modes and mode of vibration are extracted in the finite element modal analysis of the box on the gear box. Table 1. Shows the natural frequency of vibration in the first 12 orders of a large marine gearbox system. Figure 1. - Figure 2. Shows the mode mode of vibration in the first 2 orders of the box on the gearbox.

By comparing the frequencies and modes of vibration of each order, it can be seen that the modes of vibration of the box on the gear box are complicated, and the mode displacement trend of the upper box in the box is obvious. The natural frequency of the first order mode of the upper box is 213.18HZ, and the upper box is twisted along the vertical axis.

| Modal order number | 1   | 2   | 3   | 4   | 5   | 6   |
|--------------------|-----|-----|-----|-----|-----|-----|
| Natural frequency  | 213.18 | 299.33 | 336.04 | 491.37 | 503.54 | 540.69 |
| Modal order number  | 7   | 8   | 9   | 10  | 11  | 12  |
| Natural frequency  | 565.91 | 627.43 | 726.78 | 779.24 | 799.37 | 844.39 |

Figure 1. The first modal vibration mode.    Figure 2. The second modal vibration mode.

Since the operating conditions of the gear box are 1000, 2000 and 3000 respectively, the rotation characteristic frequencies of the gear box system under each operating condition are 16.7HZ, 33.3HZ and 50HZ, respectively. By comparing with the modal frequencies in Table 1., it can be seen that the rotation frequencies of the input, output and transmission shaft are far away from the natural frequencies of the upper box.

3. Experimental Modal Analysis of Gear Box

3.1. The establishment of modal experiment test system.
In order to better correspond with the finite element simulation results, the X direction in the finite element model is defined as longitudinal, the Y direction as vertical, and the Z direction as transverse. Then, the driving point of force hammer is set on the upper box. Figure 3. Shows the point diagram of force hammer in area 1.
According to the results of theoretical modal analysis, experimental modal analysis is carried out. Removing the upper box from the gear box, hang the upper box in mid-air with the lifting ring, and obtain the test mode of the upper box by force hammer. In order to generate as many modes of vibration as possible, the excitation position should avoid the stagnation point of the structure's natural mode of vibration as far as possible. Through the analysis of the finite element model, it is found that the four load-acting nodes of the no. 1, no. 2, and no. 3 and no. 4 regions in the model are not the stagnation points of the structure's natural mode of vibration. Therefore, when the impact load is applied to the above region by force hammer, the response effect of the nodes is better. When the gear box works, the gear drive to the box body is through the connection of the bearing, so the connection is the excitation point. The inner part of the gearbox is the response unmeasurable point, and the outer part of the upper box is the response measurable point. In order to select the best number and combination of measurable points, a total of nine measuring points were set around the box on the gear box, named 1-9 measuring points respectively, and acceleration sensors were arranged on these measuring points, which were the same as those of the finite element model.

3.2. Experimental modal analysis of gear box

During the modal test, the force pulse signal applied to the box by the force hammer should have the following characteristics: the peak value is large and the waveform is smooth, so as to improve the signal to noise ratio and stimulate the high frequency response of the gear box system. Therefore, it is necessary to increase the impact velocity and avoid the phenomenon of double strike when hammer is used to hit the excitation point on the box body. Suppose the cross power spectrum of the impulse force signal and the response signal is $S_{xx}(\omega)$, then the expression of the coherence function is:

$$\gamma_{qq}^2(\omega) = \frac{|S_{qq}^2(\omega)|}{S_{xx}(\omega) \cdot S_{qq}(\omega)}$$

In the modal test, the acceleration response of the measuring point 1 in the corresponding direction of 0-0.06s and the amplitude-frequency function and coherence coefficient spectrum in the frequency range of 0-5000HZ at the three-direction load in the no.1 region are shown in Figure 5.-6. Respectively.

In the modal test, the acceleration response in the corresponding direction of the measuring point 4 in the three-direction load in the no.2 region at 0-0.06s, the amplitude-frequency function and the coherence coefficient spectrum in the frequency range of 0-5000HZ are shown in Figure 7.-8. Respectively.
3.3. Experimental modal analysis results of gear box

It can be seen from the spectral value of the coherence coefficient in the frequency domain of 0-5000Hz of each measuring point in section 3.3 that the coherence coefficient of each point fluctuates in the frequency domain but its mean value is relatively high, close to 1, indicating the reliability of the test data. Through ME ‘scopeves analysis software, the modal parameters of the test results were identified, and the modal frequencies and corresponding damping ratios of the gearbox system were obtained. The test values are compared with the theoretical modal analysis values mentioned above, as shown in Table 2.

| Modal order number | 1     | 2     | 3     | 4     | 5     | 6     |
|--------------------|-------|-------|-------|-------|-------|-------|
| Calculated natural frequency | 213.18 | 299.33 | 336.04 | 491.37 | 503.54 | 540.69 |
| Experimental natural frequency | 217   | 306   | -     | 493   | -     | 558   |
| Experimental damping ratio (%) | 0.18  | 0.24  | 0.75  | 0.43  | 0.46  | 0.57  |
| Frequency error (%) | 1.76  | 2.18  | 0.33  | 3.1   | 2.1   | 2.1   |
| Modal order number | 7     | 8     | 9     | 10    | 11    | 12    |
| Calculated natural frequency | 565.91 | 627.43 | 726.78 | 779.24 | 799.37 | 844.39 |
| Experimental natural frequency | 576   | 608   | -     | 749   | -     | 827   |
| Experimental damping ratio (%) | 0.93  | 0.29  | 0.46  | 0.57  | 0.57  | 0.57  |
| Frequency error (%) | 1.75  | 3.2   | 4.04  | 2.1   | 2.1   | 2.1   |
It can be seen from Table 2. that some calculated values have no corresponding test values, because the box body on the gear box is connected to the lifting ring, and the finite element model of theoretical modal analysis cannot include this factor. In addition, the force hammer cannot excite all the natural frequencies and modes of vibration of the equipment. Modal theory in this paper, selection, calculation and experiment, comparing the similar value of maximum order modal frequency error, 10 to 4.04%, the theoretical analysis with good value and test value, showing the case on the finite element model is reliable, the theoretical model can well reflect the inherent vibration characteristics of box system based on, so the model can be used for gear box on the recognition of impact load of the study.

4. Conclusion
In this chapter, based on the theory and method of modal real eigenvalue analysis, the solution process of modal parameters is derived. A three-dimensional finite element mathematical model for calculating the natural vibration characteristics of the gearbox is established. Lanczos method was used to conduct modal analysis on the mathematical model of gear box system, and the modal frequencies and mode shapes of each order of the box were obtained. It was found that:

(1) The mode of the system is relatively complex, and the box of the system has a variety of motion trends at each order of mode frequency, such as complex bending, torsion, sag, bulging, and oscillation. By comparing the mode frequency of each order with the rotation characteristic frequency of gear box, it is found that the gear rotation frequency is far away from the characteristic frequency of each order.

(2) Before the modal experiment, the frequency response of measuring points at different positions on the box to the force hammer impulse force was extracted through the simulation experiment, which can provide a reference for the selection of measuring points in the real test. The hammering method is adopted to the modal experiment, by using data obtained from experimental modal points and the theoretical calculation value contrast value found that theoretical analysis and experimental test values conform to good, maximum frequency error is 4.04%, which shows that the established finite element model is reliable, the theoretical model can be well reflected in the inherent vibration characteristics of gear box. At the same time, the damping ratio of the gear box system is obtained by analyzing the experimental data of the single mode identification method.

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