Vibration Reduction Design of Torpedo Engine Based on Modal Contribution Method

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Abstract. Aiming at improving the structure of torpedo engine, the damping technology based on modal contribution method was proposed. Based on the results of the linear combination analysis of the modal participation factors, the vibration control method of the constrained damping layer was designed to control the dominant modes with large contribution. Firstly, the dynamic model of torpedo engine support structure is built, and the modal frequency, modal displacement and normalized modal stiffness are obtained. The dominant modes in the target frequency band are calculated, and the first 24 modes are analyzed. According to the mode shape of dominant mode, the constrained damping layer is designed for the engine. The optimized structure was verified by numerical simulation and the corresponding experimental results. The results showed that the damping technology based on modal participation factor can significantly reduce the vibration response of the engine support structure. This method creatively combines the dynamic design with the damping design, and obtains an effective damping method for engineering, especially for the military industry field with narrow space and strict weight requirements.

1. Introduction

The engine is one of the main vibration sources of the thermal power torpedo. In order to reduce the radiation noise of the torpedo, it is necessary to eliminate the vibration from the source, that is, to reduce the vibration generated by the engine itself. At present, torpedo weapon requires effective control of structural vibration in wide frequency band. In engineering, damping layer technology is generally used to control the vibration response. The damping layer control measures have obvious effect in the position with large vibration deformation, and even have a negative effect on the position with small deformation.

In this paper, through the modal contribution analysis of the target point in the broadband, the main modes to be controlled are accurately located. According to the characteristics of the mode shape changes of the main modes, the specific location of the damping vibration control is determined. To achieve targeted damping vibration control technology, improve the pertinence of control measures, and avoid damping waste.

2. Basic Principle of Modal Contribution Analysis

For the linear time invariant system, using the modal coordinate method, its base vector is the modal mode of the system, and the modal mode is the natural balance state of the deformation energy when the structure is in undamped vibration. In this state, the modes are uncoupled, and the vibration response can be expressed as the sum of the contributions of the modes.
Where \( \{x\} \) is the displacement response of the structure; \( \{\varphi_r\} \) is the displacement mode of the structure; \( q_r \) is the modal coordinate of each mode, which actually represents the component of \( \{\varphi_r\} \) in the whole response \( \{x\} \), that is, the contribution of each mode. For real modal systems, the \( r \)-th modal coordinates is

\[
q_r = \frac{\{\varphi_r\}^T \{F_r\}}{k_r - \omega^2 m_r + j\omega c_r}
\]

Where \( \{F_r\} \) is the \( r \)-th modal force; \( k_r \) is the \( r \)-th modal stiffness; \( m_r \) is the \( r \)-th modal mass; \( c_r \) is the \( r \)-th modal damping. In this method, the physical coordinates are reduced. A complete model with \( n \) degrees of freedom. Suppose that we are only interested in some physical coordinates of the system, regardless of other coordinates, then the reduced coordinates, \( \{x\}_{1\times n} \), are obtained.

\[
\{x\}_{1\times n} = \sum_{r=1}^{N} \frac{\{\varphi_r\}_{n\times l}^T \{\varphi_r\}_{l\times n} \{F_r\}_{n\times l}}{k_r - \omega^2 m_r + j\omega c_r}
\]

In this case, the modal contributions of each order under reduced degrees of freedom can be obtained as follows:

\[
q_{r} = \frac{\{\varphi_{r}\}_{l\times n}^T \{F_{r}\}_{n\times l}}{k_{r} - \omega^2 m_{r} + j\omega c_{r}}
\]

After calculating the modal parameters of each order, the contribution of each order can be obtained by the above formula. The total modal contribution of each order in the excitation frequency band is obtained by integrating the modal contribution of each order in the excitation frequency band to be considered.

### 3. Vibration Reduction Design of Engine Body

#### 3.1 Finite Element Model of Engine Body Structure

The engine body structure includes two parts: cylinder block and wobble plate box. The two parts are connected by bolts, and the bolts are evenly distributed. During the finite element modeling, the following assumptions are made: 1) there is no relative movement between the cylinder block and the swing plate box; 2) there is no friction damping at the connection position of the cylinder block and the swing plate box, and the structural damping comes from the material damping.

The engine body structure is divided into 10 nodes and 4 plane elements. The discrete finite element model is shown in Figure 1. Under the Cartesian coordinate system, the radial stiffness of each spring is \( 2.4 \times 10^6 \) N/m, the axial stiffness is \( 4.57 \times 10^5 \) N/m, and the torsional stiffness is \( 1.1 \times 10^5 \) N/m.
3.2 Calculation of Modal Contribution

The first 25 modal parameters of the engine body structure are calculated by NASTRAN software, and modal parameters such as modal displacement and normalized modal stiffness are calculated. As shown in Table 1: the modes with the largest contribution to the vibration response of engine body structure in the frequency band of 200Hz ~ 3000hz are the 8th, 14th and 24th modes. It can be seen from table 2 that the vibration mode of the above three modes is mainly the breathing mode of the swing plate box, and the vibration mode is obvious at the installation guide slot position; the bending mode of the end cover of the swing plate box changes obviously at the connection position of the swing plate box and the vibration isolation ring.

| Order | x -contribution (1e-3) | y -contribution (1e-3) | z-contribution (1e-3) |
|-------|------------------------|------------------------|----------------------|
| 8     | 0.1077                 | 0.0094                 | 0.0062               |
| 14    | 0.1024                 | 0.0154                 | 0.011                |
| 24    | 0.1482                 | 0.0188                 | 0.0175               |

3.3 Damping Layer Design of Engine Body Structure

Due to the high stiffness of the engine body structure, small vibration displacement, high vibration reduction requirements; at the same time, the size of the outer shell is limited, the type of damping layer here is selected as the constrained damping layer. The damping material layer is laid after the trench is excavated at the place where the damping material is to be laid, and then the aluminum alloy plate is used to constrain the damping material above the damping material. When the damping layer is compressed or extended by bending vibration, the deformation of the constraint layer is much smaller than that of the damping layer, resulting in shear stress and strain in the damping layer, thus dissipating the vibration energy of the structure.

According to the mode characteristics of each mode with large modal contribution, the position of damping vibration control of engine body structure is determined, and the structure after the damping layer is arranged is shown in Figure 2.
4. Numerical Simulation and Test Verification

4.1 Numerical Simulation Verification

Considering the effect of the actual stress state and load directionality of the above engine body structure on the response, respectively, y-harmonic load is applied to the rear bearing, X-harmonic load is applied to the thrust bearing. Among them, the X direction is the axial direction, the normal direction of the roller guide groove working face is Z direction, and the Y direction follows the right-hand rule. The direct response method is used to calculate the harmonic response of the two structures, and the force amplitude is 1N. The excitation frequency range is from 200Hz to 6150Hz, and the frequency step is 50Hz.

The calculation results are shown in Fig. 3, 4. Compared with the response results under three working conditions, after the damping layer is laid, the x-direction and y-direction obviously restrain the vibration. In the target frequency range of 20Hz to 3000Hz, the vibration reduction effect is very obvious, and the vibration peak value is obviously reduced.

![Figure 3](image3.png)

**Figure 3.** Comparison of dynamic responses between before and after optimization in x-direction unit load

![Figure 2](image2.png)

**Figure 2.** Layout of damping layer of main structure
4.2 Test Verification

The power test was carried out on the structure before and after the constrained damping layer was laid. And use LMS.Test.Lab software to pick up the time-domain acceleration vibration data at the target point. The target point frequency domain comparison chart is shown in Figure 9.

Table 2. Comparison diagram of the total level on the target point

| Measure point | Status          | Total level |
|---------------|-----------------|-------------|
| Target point 1| Before optimization | 105.8       |
|               | After optimization | 102.4       |
| Target point 2| Before optimization | 103.2       |
|               | After optimization | 99.1        |
5. Conclusion

According to the main structure of the torpedo engine, the 26-order modal parameters below 3000 Hz are obtained through modal analysis, and the vibration contribution of each mode in the 20Hz~3000Hz frequency band of the target point is calculated. The structure adopts damping layer vibration reduction measures, and carries out frequency response simulation analysis and test analysis on the original engine structure and the improved structure respectively. The following conclusions are obtained:

1) The comparison of the response simulation analysis under three load conditions shows that the target frequency band is 200Hz~3000Hz, the vibration reduction effect is very obvious, and the vibration peak is significantly reduced.

2) Through experiments, the response amplitude of the structure before and after the improvement is compared. The response amplitude of the improved structure at the dominant modal frequency is greatly attenuated. Compared with the target point vibration energy level in the 20Hz~3000Hz frequency band, it is reduced by about 3dB.

3) The damping layer vibration reduction design method based on the modal parameter contribution method creatively proposed in this paper is effective and feasible, and has achieved obvious vibration reduction effects in engineering.

6. References

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