Influence of thrust bearing seating on acoustic radiation of submarine

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Abstract: [Objectives] In this paper, the influence of thrust bearing seating on the acoustic radiation of a submarine is studied. [Methods] By utilizing the Finite Element (FE) method and FE coupling fluid interface method, the structure vibration of the pressure hull is calculated with different distributions of longitudinal exciting forces. In this case, the symmetrical distribution of exciting forces and moments are beneficial for reducing vibration. According to the above conclusions, symmetrical thrust bearing seating is designed and the longitudinal stiffness of the system controlled through choosing the size of components. Two submarines are designed, one with conventional thrust bearing seating and the other one with symmetrical thrust bearing seating. The vibration response and radiation noise of the structures of the two submarines are forecasted. [Results] It can been seen that when the propeller longitudinal force is constant, symmetrical thrust bearing seating can significantly decrease the level of vibration and radiation noise. [Conclusions] The results can provide references for the acoustic optimization and design of the thrust bearing seating of submarines.

Key words: thrust bearing; structural vibration; radiation noise; peak frequency; vibration mode

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0 Introduction

Propellers are one of the main noise sources of submarines, which induce the vibration of hulls mainly through shafting and connecting seating, and radiate noise into water from the wet surfaces of submarines. As a result, the coupled vibration of propellers, shafting, seating, and hulls is brought about. Thus, in-depth research on the related problems is of great significance for reducing the propeller-induced structural vibration and radiation noise of hulls.

In general, longitudinal dampers and dynamic vibration absorbers [1-4] are installed in shafting to control the transmission of propeller exciting force to hulls through the shafting. Dylejko et al. [5] and Merz et al. [2-3] studied the coupled vibration of a shafting-hull system; the longitudinal exciting force of shafting, transferred to the hull through thrust bearings, was reduced by using dynamic vibration absorbers at the positions of these bearings; then, the transfer matrix method, the Finite Element Method (FEM), and the coupled method of structural finite element and fluid boundary element were used respectively to establish dynamic system models to optimize the structural parameters of dynamic vibration absorbers, so as to reduce the transmission of exciting force. In order to reduce the radiation noise of hull caused by the longitudinal exciting force of a propeller, Cao [5] and Yang [6] adopted transverse bulkheads as the thrust bearing seating to change the transmission paths of longitudinal exciting force. The results showed that this scheme not only reduced the vibration, but also lowered the longitudinal-transverse coupling degree of the systems. However, transverse bulkheads are mainly used to divide interior
space, forming watertight compartments and supporting submarine hull, and the use of transverse bulkheads as thrust bearing seating may affect the realization of their major functions. For example, poor water tightness of a compartment will influence the general arrangement in the compartment. In addition, submarines are usually transversely symmetric while vertically asymmetric, so using bulkheads as bearing seating will result in excessive paths for force transmission, making it difficult to effectively control force transmission in all directions. Li et al. (1) analyzed the natural characteristics of coupled vibration in a shaft-hull system and their variations with the stiffness of thrust bearings. According to the results, the stiffness of thrust bearings changed the longitudinal vibration frequency of shafting, exerting an obvious influence on the energy transfer of longitudinal vibration. The longitudinal vibration of shafting not only caused the longitudinal resonance of the hull, but also brought about the bending vibration of the hull, forming a longitudinally and transversely coupled mode of the shaft-hull system. The longitudinal vibration control of shafts could reduce the vibration of the coupled system. Pan et al. (8) studied the transmission characteristics of propeller exciting force from shafting to simply supported plates. Specifically, the exciting force was obtained experimentally; the relationship between thrust bearing stiffness and propeller speed was measured, considering the influence of oil films on thrust bearings; the longitudinal propeller exciting force caused by the wake flow field and relevant vibration response of elastic plates were studied in particular.

The above research focused on the law of propeller exciting force transferring to hulls through shafting, reduced vibration and noise of submarines by using transverse bulkheads as the thrust bearing seating, and abated the transmission of longitudinal propeller exciting force to hulls by installing dampers in shafting and dynamic vibration absorbers on thrust bearing seating. In order to reduce the structural vibration and radiation noise of a hull caused by longitudinal propeller exciting force, considering that the internal arrangement of a submarine and the longitudinal displacement of thrust bearing seating are required to satisfy relevant restrictive conditions, this paper plans to redesign the structure of thrust bearing seating from the perspective of acoustic design of structural-vibration transmission paths, in order to make longitudinal propeller exciting force act symmetrically on a submarine. In addition, it analyzes the vibration and radiation noise of the whole submarine in order to verify the validity of the design method.

1 Basic theory

1.1 Structure-fluid coupling equation

The problem of underwater structural vibration and radiation noise is a fluid-solid coupling problem of fluid-structure interaction. A system of structure-fluid interaction, as shown in Fig.1, is considered: $S_0$ denotes an elastic thin-shell structure; $\Omega_0$ denotes an outer domain of fluid, which is filled with an acoustic medium with density $\rho_0$ and sound velocity of $c_0$. If the system enters a steady state and the angular frequency is $\omega$, the wave number is $k_0 = \omega^2/c_0$.

![Fig.1 Fluid-structure interaction system](image)

The structural domain is discretized by the FEM, and the boundary element method is used for the external fluid to obtain the additional mass and damping coefficient. Then, the additional mass and damping are superposed to the finite-element mass and damping matrices of the structure, in order to establish a finite-element-based structural dynamic response equation considering fluid-structure interaction, as shown in Formula (1), to realize the calculation of fluid-structure interaction. Finally, the radiation noise field of the structure is calculated by using the boundary element method.

$$\left[-\omega^2 \left( M_s + M_{oa} \right) - i \omega \left( C_s + N_{oa} \right) + K_s \right] \{ \ddot{a} \} = \{ f \} \quad (1)$$

where $K_s$ is a structural stiffness matrix; $M_s$ is a structural mass matrix; $C_s$ is a structural damping matrix; $\{ \ddot{a} \}$ is a nodal displacement vector; $\{ f \}$ is nodal force directly acting on the structure; $M_{oa}$ and $N_{oa}$ are respectively the additional mass and damping matrices generated by the external fluid acting on the structure. Once structural displacement is obtained, the nodal displacement on the interface between the structure and the external fluid can be extracted, and then, the normal displacement vector of the object plane can be obtained to calculate the noise pressure of the acoustic field (9).
1.2 Measurement indexes of noise radiation capacity

In order to measure the noise radiation capacity of a hull, the mean-square normal velocity of the hull surface and the radiated noise pressure are used as the main measurement indexes.

The mean-square normal velocity of a hull surface reflects the structural response of the hull in a flow field, and also represents the average velocity of the vibration of a noise source, which is defined as

\[
\langle \bar{V}^2 \rangle = \frac{1}{2} \text{Re} \left( \frac{1}{s} \int \bar{V} \bar{V}^* \, ds \right)
\]

Where \( \langle \bar{V}^2 \rangle \) refers to the mean-square normal velocity of a hull surface; \( \bar{V} \) is the complex amplitude of the normal velocity of the hull surface; \( s \) is the surface area of the hull; \( \ast \) denotes conjugation.

The level of a mean-square normal velocity is defined by

\[
L_{V} = 10 \log \frac{\langle \bar{V}^2 \rangle}{V_{ref}^2}
\]

Where \( V_{ref} = 5 \times 10^{-8} \text{ m/s} \), which is a reference velocity.

A submarine is located at 100 m underwater, and a calculation point of radiated noise pressure is arranged on the horizontal symmetrical plane of the parallel section at the middle of the submarine, with a distance of 15 m from both sides of the submarine; the noise pressure on the straight section along the submarine length is calculated and then arithmetically averaged, and the arithmetic average value is taken as the radiated noise pressure of the submarine. Fig.2 illustrates the calculation meshes of radiated noise pressure.

Suppose that \( p \) is the average of the radiated noise pressure of a submarine under unit force, and that the reference noise pressure is \( p_{ref} = 1.0 \times 10^{-6} \text{ Pa} \), then, sound pressure level is defined as

\[
L_p = 20 \log \left( \frac{p}{p_{ref}} \right)
\]

1.3 Structural vibration caused by longitudinal exciting force of a propeller

A conventional thrust bearing seating is of a seating type, located at the bottom of a submarine, as shown in Fig.3. The longitudinal propeller exciting force \( F \) works in such a process: propeller → thrust bearing → thrust bearing seating → pressure hull → non-pressure hull and whole submarine, thereby arousing the vibration of the hull. In the process, \( F \) acts on the hull through the thrust bearing seating, which is manifested by the longitudinal exciting force \( F_i \) and the exciting moment \( M_i \), so the control of \( F \)-induced structural vibration of the hull is to control \( F_i \) and \( M_i \).

Due to space limitation, this paper focuses on using structural acoustic design to control \( F_i \) that is transferred to the submarine.

The pressure hull is the main structure of a submarine, which can reflect the vibration characteristics of the whole submarine to a great extent, and the design of a pressure hull as a transversely and vertically symmetrical structure is helpful to the determination of its vibration law.

1.3.1 Calculation model

During the design of a pressure hull, the main dimensions referred to a French "Ruby"-class single-hull submarine, and the shape referred to the single-hull SUBOFF submarine model. Table 1 lists the main structural parameters of the pressure hull which is about 1/2 of the size of a real one, with a T-shaped frame section (Fig.4). The pressure hull was modeled by the FEM. Specifically, the hull was simulated by surface elements, while the frame and other stiffeners were simulated by beam elements; along the longitudinal direction of the submarine, one frame interval contains at least four rows of elements and five nodes to ensure the simulation of a complete waveform between frames, and the model.
was set in a free state.

Fig. 5 shows the finite element model of the pressure hull. Because of the bilateral symmetry of the hull, only the left part of the hull model is given.

1.3.2 Basic forms of natural low-frequency vibration of a pressure hull

The pressure hull is of a slender structure, with a length-to-diameter ratio of 8.7. The natural low-frequency vibration of the slender pressure hull has three types of basic forms: 1) overall bending vibration, for example, Nos. 1–3 in Table 2; 2) overall longitudinal vibration, for example, Nos. 4–5 in Table 2, which is accompanied by radial breathing vibration at the same time; 3) bending vibration of the cabin, for example, No. 6 in Table 2.

From Table 2, it can be seen that both overall

<table>
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<th>No.</th>
<th>Vibration form</th>
<th>Natural frequency/Hz</th>
<th>Vibration mode</th>
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<tbody>
<tr>
<td>1</td>
<td>First-order bending vibration, half a waveform</td>
<td>22</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Second-order bending vibration, one waveform</td>
<td>46</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Third-order bending vibration, two waveforms</td>
<td>100</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>First-order longitudinal vibration</td>
<td>70</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Second-order longitudinal vibration</td>
<td>126</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Bending vibration of cabin</td>
<td>92</td>
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</table>
bending vibration and overall longitudinal vibration show structural waves with a longer overall wavelength.

1.3.3 Characteristics of vibration response under longitudinal exciting force

The FEM was used to calculate the vibration response of the pressure hull in air, and the frequency range for calculation is 20 – 150 Hz, with a frequency interval of 1 Hz.

The longitudinal exciting force acts on one or more of the four points of A, B, C, and D on the pressure hull where a frame of the conical stern is located, as shown in Fig.6. Among the four points, points A and C are symmetric with respect to the Y axis, while points B and D are symmetric with respect to the Z axis.

The vibration response characteristics of the pressure hull under the following two cases are studied in this paper.

1) Case 1: longitudinal exciting force acts on the point A, with a magnitude of $F$ (Fig.7).

2) Case 2: longitudinal exciting force acts on the points A, B, C, and D, and the applied force at each point is $F/4$ in magnitude, which is equivalent to the uniform distribution of the exciting force in Case 1 among the four points (Fig.8).

Fig.9 shows the frequency response curves of the mean-square normal velocity of the pressure-hull outer surface varying with exciting-force frequencies under the two cases. The peak frequencies and the corresponding vibration modes of the vibration response of the pressure hull under both cases 1 (Table 3) and 2 (Table 4) are obtained according to the figure.

According to the comparison between Table 2 and Table 4, in Case 2, the first and second peak frequencies (70 and 126 Hz) of the low-frequency vibration response of the structure correspond to its natural first-order and second-order longitudinal vibration frequencies (70 and 126 Hz), and so do the vibration modes. Thus, following conclusions can be made: in Case 2, the longitudinal exciting force acts symmetrically on the cross section, and mainly excites the overall longitudinal vibration of the structure in the low-frequency band, showing a structural longitudinal wave with a longer wavelength in the whole range; vibration caused by such a structural wave has a higher level and determines the peak value of vibration response (11).

According to the comparison between Table 2 and Table 3, in Case 1, the longitudinal exciting force acts on a local part of the cross section, and simultaneously excites the overall bending vibration and the overall longitudinal vibration of the structure in the low-frequency band. When the exciting-force frequency is the same as or very close to the natural frequency of the structural vibration, even though the exciting force excites both the bending and longitudinal vibration of the structure at the same time, the natural vibration of the structure corresponding to this frequency dominates. Therefore, the vibration response will show a phenomenon similar to that in Case 2, that is, peak values of vibration response appear in the case of the exciting-force frequencies of 70 and 126 Hz. However, the difference from Case 2 is that peak values also appear at 22, 46 and 100 Hz in Case 1; the natural vibration of the structure corresponding to these frequencies is overall bending vibration, which is shown as a structural bending wave.
with a longer wavelength in the whole range; the vibration caused by such a structural wave has a higher level and determines the peak value of vibration response.\textsuperscript{[11]}

According to the comparison between the results of both cases, when longitudinal exciting force symmetrically acts on the hull, overall bending vibration can be avoided or weakened to a great extent, and overall longitudinal vibration is mainly excited, accompanied by radial breathing vibration; while frames and bulkheads have a strong constraint on breathing vibration, so the peak values of vibration are relatively small. In the low-frequency band, the structural vibration weakens. Therefore, it is necessary to make sure that the longitudinal exciting force of a propeller acts symmetrically on the submarine to avoid a structural bending wave with a longer wavelength in the whole range.

2 Structural design of a thrust bearing seating

According to the analysis in Section 1.3, in order to reduce structural vibration, it is necessary to redesign the structure of a thrust bearing seating, so that the longitudinal exciting force of a propeller can act symmetrically on the cross section of a submarine.

Two kinds of thrust bearing seating are used for a submarine, which are respectively of the traditional structure (Fig.3) and the symmetrical structure. Fig.10 shows the symmetrical structure of a redesigned thrust bearing seating, and the thrust bearing is supported by four pillars. The four pillars have the same I-shaped cross section, as shown in Fig.11.

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</table>

Fig.10 Structure of thrust bearing seating

Table 3 Peak frequencies and vibration modes of responses for pressure hull in case 1

Table 4 Peak frequencies and vibration modes of responses for pressure hull in case 2

Fig.11 Structural design of thrust bearing seating
The finite element model of the whole submarine is shown in Fig. 12. Fig. 13 and Fig. 14 respectively show the submarine stern with thrust bearing seating in the conventional and symmetrical forms.

Under the action of longitudinal propeller exciting force, the vibration response in air and radiated noise pressure in water for the two models are calculated respectively, and the results are shown in Fig. 15 and Fig. 16. Fig. 15 shows the frequency response curves of the mean-square normal velocity of the hull outer surface varying with exciting-force frequencies, while Fig. 16 shows the frequency response curves of the radiated noise pressure in water varying with exciting-force frequencies. It can be seen from both figures that compared with the traditional thrust bearing seating, the symmetrical thrust bearing seating, with longitudinal stiffness as low as possible, can reduce not only the vibration intensity of the hull, but also the radiated noise level of the hull.

3 Conclusion

From the perspective of structural acoustic design of transmission paths of longitudinal propeller exciting force, this paper studies the influence of the structural form of a thrust bearing seating on the vibration and acoustic radiation of a submarine, and draws the following conclusions:

1) With a traditional thrust bearing seating, longitudinal exciting force acts from one side on the hull, and in the low-frequency band, it excites a large-scale structural bending wave in the range of the whole submarine, resulting in a high level of vibration and radiation noise of the submarine.

2) With a symmetrical thrust bearing seating, longitudinal exciting force acts symmetrically on the hull, which can avoid or reduce a large-scale structural bending wave in the range of the whole submarine, thus lowering the level of vibration and radiation noise of the submarine.

References


推力轴承基座结构形式对潜艇振动噪声的影响

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关键词：推力轴承；结构振动；辐射噪声；峰值频率；振型

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