Research on Prediction of Middle-frequency Ship Cabin Noise Based on Improved Statistical Energy Method

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Abstract. The ship vibration and noise calculation in middle-frequency region has always been a difficult point in engineering. The statistical energy method assumes that the coupling between structures is weakly coupled, which in turn limits its application in the calculation of ship's middle frequency vibration and noise. In order to improve the vibration and noise calculation accuracy of statistical energy method in middle-frequency region, the equivalent coupling loss factors (ECLFs) between ship sub-structures were calculated based on the energy flow method (EFM), and the ECLFs were substituted into the statistics energy balance equation. The ship's middle frequency cabin noise was calculated by the improved statistical energy method and compared with the measured values. The results show that the predicted values of the cabin noise are in good agreement with the measured values, which extends the application of the statistical energy method in the ship's middle frequency cabin noise prediction.

1. Introduction
During the voyage of a ship, its main power plants will generate vibration, the vibration will be transmitted to the accommodation cabin to cause cabin noise. The cabin noise will seriously affect the comfort and physical health of crew members and passengers. Various classification societies and the International Maritime Organization (IMO) have made more stringent regulations on the cabin noise. Therefore, it is of great practical significance to advance the prediction of cabin noise, which can improve the ship construction quality and reduce the construction cost.

There are usually empirical forecasting methods, analytical methods and numerical prediction methods for ship cabin noise prediction. Numerical forecasting methods are currently the most commonly used in engineering. Numerical prediction methods usually include acoustic finite element method (FEM), boundary element method (BEM), statistical energy method (SEA), gray theory method, ray tracing method and neural network method, etc[1-3].

Lima W D [4] et al. (2010) used the SEA method to predict the cabin noise, and the SEA model was validated using in-flight tests with the aircraft in various acoustic package configurations. Liang B N [5] et al. (2016) used FE-BEM method to calculate the surface velocity of the hull structure and sound pressure of acoustic cavity radiation, the noise reduction of the cabin was achieved by laying a damping coating and a floating floor. Zeng X Y[6] (2004) proposed an artificial neural network method for forecasting cabin noise, the rectangular cabins with different spatial dimensions were taken as the research object, an artificial neural network algorithm was used to achieve accurate prediction of high-frequency sound pressure distribution in the model cabin. Feng A J[7] (2017) used a sound-tracking method for noise prediction of ship cabins, the
calculated results were in good agreement with the SEA method, which verified the reliability of the sound-tracking method.

However, there are still deficiencies in the prediction of ship cabin noise. (1) How to divide the analysis frequency into low, middle and high frequency region is rarely involved. (2) There are more studies on cabin noise prediction for high frequency region, and less research on cabin noise prediction for a middle frequency band.

The statistical energy method predicts the energy transfer and energy response of each subsystem from the perspective of average statistics in time, frequency domain and space. The statistical energy method effectively solves the calculation of vibration and noise in the high frequency band of the structure. Nevertheless, the statistical energy method has many hypothetical conditions in its application. For example, the coupling between substructures is weak, and the substructures have enough resonance modes and a sufficiently high modal overlap coefficient. In the high frequency region, the hull structures and cabin acoustic cavities are all characterized by high density modality, which is suitable for solving with statistical energy method.

In the middle frequency region, the hull structure shows the mixed vibration characteristics of "low mode and high mode" simultaneously. In order to meet the calculation accuracy requirements of the finite element method, it is necessary to ensure that there are at least 6 elements within one wavelength of the discrete model, which leads to too many finite element elements of the discrete model. At the same time, the finite element method can only identify a limited number of low-order modes. In the middle-frequency region, some hull structures have dense modalities, and the finite element method cannot clearly identify them. When applying the statistical energy method, it is assumed that the subsystems are conservatively weakly coupled, and the excitations are not related to each other. These assumptions are generally difficult to hold in the middle frequency region, which limits the application of the statistical energy method in the calculation of middle-frequency vibration and noise.

This paper mainly considers extending the high-frequency statistical energy method to the middle-frequency band. The coupling loss factor is an important and difficult to determine parameter of the statistical energy method. The higher accuracy coupling loss factor is of great significance to the calculation accuracy of statistical energy method [8].

In this study, an oil tanker was taken as the research object. A new frequency division criterion was proposed. The power flow model and statistical energy model of the hull are established respectively. The power flow model is divided into coarse finite element grids and the corresponding mass and stiffness matrixes are formed. Based on the power flow theory, the structural power flow balance equation is established, and the energy influence coefficient matrix is obtained under unrelated excitation loads. Based on the statistical energy model, a structural coupling loss factor matrix is established, and the coupling loss factor is replaced with the off-diagonal elements of the energy influence coefficient matrix to form a revised statistical energy coupling loss factor matrix. Finally, the equivalent coupling loss factors were substituted into the statistics energy balance equation. The ship's middle frequency cabin noise was calculated by the improved statistical energy method and compared with the measured values.

The study object is an offshore oil tanker with a total length of 101.60 m, as shown in Figure 1. The main power plants in the engine room include the main diesel engine, gearbox and diesel generator unit, the power plants specific parameters are shown in Table 1 and Table 2.

Table 1. Main engine basic parameters.

| Type          | 6DKM-26       |
|---------------|---------------|
| Rated power $P_e$/kW | 1618          |
| Rated speed $n_e$/(r/min) | 750           |
| Number of cylinders $N_c$  | 6             |
Table 2. Diesel generator unit basic parameters.

| Type                      | Generator (DKBH4320/06) | Diesel engine (R6160ZC) |
|---------------------------|-------------------------|-------------------------|
| Rated power $P_e$/kW      | 160                     | 200                     |
| Rated speed $n_e$/(r/min) | 1 000                   | 1 000                   |
| Mass /kg                  | 780                     | 940                     |
| Amount                    | 2                       | 2                       |

2. Oil tanker simulation model

2.1. Frequency division

Different analysis frequency regions need to use different cabin noise prediction models and calculation methods, the analysis frequency regions need to be divided before cabin noise modeling and calculation. The division of low, middle and high frequency should be based on the dynamic characteristics of each subsystem at different excitation frequencies. When all subsystems of structures exhibit typical long-wave and low-mode characteristics, the frequency domain can be divided into low-frequency region. When all subsystems of structures exhibit typical short-wave and high-mode characteristics, the frequency domain can be divided into high frequency region. When the subsystems have both deterministic and statistical subsystems of a certain frequency range, the frequency domain is the middle-frequency region.

According to the basic theory of SEA, the modal density is a measure of the subsystem modal density in a certain frequency band. The greater the modal density, the more resonant peaks may be generated in the frequency domain. The modal overlap factor is used to characterize the contribution of the modal number to the system response and is used to measure the degree of similarity between modalities. When the modal overlap factor is large, there will be more modes that contribute to the response under certain excitation. The average effect of the response can be got.

In this paper, the new modal density of the subsystem is measured by the product of the modal density and the modal overlap factor. The modal overlap factor is related to the structural damping loss factor, in order to eliminate the influence of the structural damping loss factor on the frequency division, the modal overlap factor is divided with the structural damping loss factor. The deterministic subsystem is characterized by long-wave and low-mode, while the statistical subsystem is characterized by short-wave and high-mode. The bending wave wavelength in the structure has a certain influence on the frequency division, the wavelength is also divided. A characteristic number is proposed to divide the frequency range.

$$\Delta = \frac{M_r n(\omega)}{\eta \lambda}$$  \hspace{1cm} (1)

where, $M_r$ is the structural modal overlap factor, $n(\omega)$ is the structural modal density, $\eta$ is the structural damping loss factor, $\lambda$ is the structural bending wave wavelength.

The modal density of one-dimensional beam structure can be got by

$$n(\omega) = \frac{1}{\omega^{1/2}} \left[ \frac{\rho A}{EI} \right]^{1/4}$$  \hspace{1cm} (2)
where, $\omega$ is the frequency, $A$ is the cross-sectional area of the beam structure, $P$ is the mass density, $EI$ is the structural bending stiffness.

The modal density of two-dimensional flat structure can be got by

$$n(\omega) = \frac{\sqrt{3}A}{hc_i}$$

(3)

where, $h$ and $A$ are the thickness and area of plate structure respectively, $C_i$ is the structure bending wave velocity.

The three-dimensional acoustic cavity modal density can be got by

$$n(\omega) = \frac{\omega^2V}{2\pi^2c^3} + \frac{\omega^2A}{16\pi^2c^2} + \frac{\omega L}{16\pi c}$$

(4)

where, $V$, $A$, $L$ are the volume, total surface area and total length of each side of the sound cavity, respectively, and $c$ is the speed of sound.

The modal overlap factor can be calculated by

$$M_e = \frac{\pi}{2} \omega \eta n(\omega)$$

(5)

where, $\omega$ is the center frequency, $n(\omega)$ is the structural modal density, $\eta$ is the structural damping loss factor.

Combining equations (1) and (6):

$$\Delta = \frac{\pi \omega n^2(\omega)}{2 \lambda}$$

(6)

From equation (6), it can be seen that the characteristic number $\Delta$ is only related to the structural modal density and wavelength. The characteristic number $\Delta$ defined in this paper can be used to characterize the modal density within a unit wavelength. According to the size of the characteristic number, the frequency range is divided into low frequency region, middle frequency region and high frequency region:

- when $\Delta \leq 1$, it is defined as the low frequency region,
- when $1 < \Delta < 5$, it is defined as the middle frequency region,
- when $\Delta \geq 5$, it is defined as the high frequency region.

The characteristic numbers $\Delta$ of the hull structures and cabin acoustic cavities were calculated, which were shown in Figure 2.

**Figure 1.** Real ship of prediction.  **Figure 2.** Characteristic numbers of ship structures and sound cavities.

As can be seen from Figure 2, below 50 Hz, the hull structures and cabin cavities characteristic numbers are all basically less than 1, and 20-50 Hz can be divided into low frequency region for cabin noise prediction; above 630 Hz, the hull structures and cabin cavities characteristic numbers
are basically larger than 5, 630-8000 Hz can be divided into high frequency region for cabin noise forecast, and 63-500 Hz is divided into middle frequency region.

2.2. Cabin noise prediction of oil tanker in middle frequency region

When the power flow model system is uncorrelatedly excited, its energy balance equation is:

\[ E(\omega) = [EIC(\omega)]P_{in}(\omega) \]  \hspace{1cm} (7)

where, \( E(\omega) \) is the energy matrix of each subsystem in the power flow model, \( P_{in}(\omega) \) is the input power matrix of the external excitation to the subsystem, \( \omega \) is the center frequency of the analyzed frequency band, \([EIC(\omega)]\) is the energy influence coefficient matrix.

\[
[EIC(\omega)] = \begin{bmatrix}
    c_{11} & \cdots & c_{1n} \\
    \vdots & & \vdots \\
    c_{n1} & \cdots & c_{nn}
\end{bmatrix}
\]  \hspace{1cm} (8)

where, \( c_{mn} \) represents the energy of the mth substructure when unit power is input to the nth substructure.

The energy impact coefficient matrix is related to the type of excitation applied by the substructure, and it is usually better to aggregate the various possible excitations and then apply it to the power flow model substructure. The effect obtained by applying the "raindrop load" excitation on the power flow model and the set averaged excitation on the power flow model is equivalent. The energy impact coefficient matrix obtained under "raindrop load" can better characterize the power flow properties of each substructure. From equation (7), we can get:

\[
P_{in}(\omega) = [EIC(\omega)]^{-1} E(\omega)
\]  \hspace{1cm} (9)

For a conservative acoustic-structure weakly coupled system consisting of N subsystems, the expression of the statistical energy balance equation is:

\[
\omega \eta E(\omega) = P_{in}(\omega)
\]  \hspace{1cm} (10)

\[
\eta = \begin{bmatrix}
    \eta_1 & \cdots & -\eta_N \\
    \vdots & \ddots & \vdots \\
    -\eta_1 & \cdots & \eta_N
\end{bmatrix}
\]  \hspace{1cm} (11)

where, \( \omega \) is the center frequency of the analyzed frequency band, \( \eta_i \) is the internal loss factor of subsystem i, \( \eta_{ij} \) represents the one-way coupling loss factor of energy transfer from subsystem i to subsystem j, \( E(\omega) \) is the energy matrix of each subsystem in the statistical energy model, \( P_{in}(\omega) \) is the input power matrix of the subsystem for external excitation.

From formula (10), we can get:

\[
\eta = \frac{1}{\omega} P_{in}(\omega) E(\omega)^{-1}
\]  \hspace{1cm} (12)

Comparing equations (9) and (12), it can be seen that the non-diagonal elements of the inverse matrix can be equivalent to the coupling loss factor describing the energy transfer between the substructures. The equivalent coupling loss factor obtained by the power flow method is substituted
into the energy balance equation in the actual statistical energy model, and then the energy of each substructure and various acoustic vibration response quantities can be obtained.

According to the drawings, the ship statistical energy model is established. The hull structures and the cabin sound cavities statistical energy subsystems are established. The semi-infinite fluid subsystems are also required to simulate the effect of seawater on both sides of the ship. The established hull structures statistical energy subsystems and cabin sound cavities statistical energy subsystems are shown in Figure 3.

2.3. Determination of excitation source and internal loss factor

The cabin noise prediction only considers the structural noise and air noise generated by the main power plants. The structural noise is generated by the acceleration of the main diesel engine, the gearbox and the diesel generator set. The air noise includes the main diesel engine radiated sound power, intake and exhaust noise, generator radiated sound power, diesel radiated sound power and its intake and exhaust noise. The acceleration of the power plants is obtained by the actual ship test. The air noise of the power plants is got by estimation formulas.

According to the "Guide to Ship and Product Noise Control and Inspection" issued by China Classification Society[9], the radiated sound power level of diesel engine can be got by

\[ L_{w} = 10\log_{10} P_{e} + 58 + C_{w} \]  \hspace{1cm} (13)

where, \( P_{e} \) is the rated power of the diesel engine, \( C_{w} \) is the octave correction value of the air noise of the diesel engine.

Exhaust noise sound power of diesel engine can be got by

\[ L_{w} = 77 + 10\log_{10}(P_{e} \times n_{e}) + 30\log_{10}(n/n_{e}) - 10\log_{10}\left[\left(\frac{f}{f_{c}}\right)^{3} + \left(\frac{f}{f_{c}}\right)\right] \]  \hspace{1cm} (14)

where, \( f_{c} \) is the ignition frequency, \( n \) is the working speed of the diesel engine, \( r / \text{min} \).

The ignition frequency can be got by

\[ f_{c} = N_{e} \times n \times 2 / (60 \times S) \]  \hspace{1cm} (15)

where, \( N_{e} \) is the number of cylinders, \( S \) is the number of strokes.

Generator radiated sound power can be got by

\[ L_{w} = 13\log_{10} P_{e} + 15\log_{10} n_{e} + 6.6 + C_{w} \]  \hspace{1cm} (16)

where, \( P_{e} \) is the rated power of the generator, \( n_{e} \) is the rated speed of the generator, \( r / \text{min} \), \( C_{w} \) is the air noise correction value.

The airborne noise of the power plants is applied to the cabin acoustic cavity subsystem, while structural noise is applied to the base panel subsystem of power plants. The power equipment foot acceleration test sites are shown in Figure 4, the main power equipment excitation spectrums are shown in Figure 5.
The whole ship construction material is steel, and the internal loss factor of steel is obtained by empirical formula $\eta = 0.41 f^{-0.7}$. The loss factor in the acoustic cavity is obtained by the following formula.

$$\eta_c = \frac{2.2}{fT_R}$$

(17)

$$T_R = \frac{0.163V}{-S \ln(1 - \alpha) + 4mV}$$

(18)

where, $T_R$ is the sound cavity reverberation time, $f$ is the analysis of the center frequency of the band, $\alpha$ is the average sound absorption coefficient of the sound cavity, $m$ is the attenuation coefficient of sound intensity in the cavity medium, $V$ is the volume of the acoustic cavity subsystem, $S$ is the sound absorption area of the sound cavity.

The calculated loss factors of the hull structures and cabin acoustic cavity loss factor are shown in Figure 6.

![Figure 4. Vibration test sites.](image)

![Figure 5. Main power plants excitation spectrums.](image)

The acceleration level of the foot of each power equipment is imposed on the corresponding base panel subsystem in the form of constraints, as shown in Figure 7.

![Figure 6. Structure and acoustic cavity loss factor.](image)
Figure 7. Schematic diagram of the acceleration level applied on the base panel.

3. Middle frequency cabin noise prediction and experimental verification of oil tanker

The structural noise and air noise of the power plants are respectively loaded on the corresponding subsystems of different calculation models. The improved statistical energy method was used to calculate cabin noise in the middle frequency region (63 ~ 500 Hz). The calculated cabin sound pressure cloud maps of the oil tanker are shown in Figure 8.

(a) 63 Hz cabin noise cloud map  (b) 200 Hz cabin noise cloud map  (c) 400 Hz cabin noise cloud map

Figure 8. Sound pressure cloud maps of the tank cabins.

It can be seen from Figure 8 that the sound pressure levels of the cabins near the engine room are the largest. At different frequencies, the distribution and variation of cabin sound pressure are basically the same.

In order to verify the accuracy of cabin noise prediction, the Larson Davis Model 831 sound level meter is used to measure the ship's cabin sound pressure. The cabin noise test site is shown in Figure 9. The comparison of the simulated values and measured values are shown in Figure 10.

Figure 9. Noise test site.
Figure 10. Comparison of predicted and measured values of some cabin noise.

As can be seen from Figure 10, compared with the traditional statistical energy method, the cabin noise predicted values obtained by improved statistical energy method have a better
agreement with the measured values in the middle frequency, which shows that the calculation of the coupling loss factor between substructures by the power flow method can improve the vibro-acoustic calculation accuracy of statistical energy method in the middle frequency region.

The power flow method not only overcomes the statistical energy method's assumption of weak coupling between substructures, but also considers the effect of the non-diffusion field on energy transfer loss, so the coupling loss factor obtained by the power flow method has better accuracy.

The error between the predicted values and the measured values may be caused by the following reasons: (1) The internal loss factors of the hull structures and the acoustic cavities are obtained by the empirical formula, and may have an error with the actual. (2) Propeller excitation, chimney and air conditioning piping noise are ignored. (3) Structures such as doors and windows were ignored in the modeling process.

4. Conclusion
An oil tanker is taken as the research object, a new frequency interval division method is proposed to divide the low-frequency, middle-frequency and high-frequency region of ship cabin noise forecast. An improved statistical energy method is proposed to forecast ship middle frequency cabin noise. The effective coupling loss factors between ship substructures are calculated by the power flow method, thereby updating the relevant parameters in the traditional statistical energy balance equation. Based on the improved statistical energy method, the middle frequency ship cabin noise prediction is carried out and compared with the measured values. The results show that the equivalent coupling loss factor can effectively improve the statistical energy method calculation accuracy in the middle frequency cabin noise prediction.

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