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LOW TO MIDDLE VIBRO-ACOUSTIC NOISE PREDICTION IN SHIP CABIN BY USING PLATE-CAVITY COUPLING MODEL

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ABSTRACT

A plate-cavity coupling method based on modal expansion technique in the closed sound cavity was introduced, aiming at ship cabin structural noise. Using this method, a coupled equation was established. The structural vibration acceleration of the target cabin was extracted from a ship vibration response calculation, applied to the model. Then the target cabin noise value was obtained through numerical calculation. The effectiveness and reliability of the method were validated through experiments. The coupled model predicts noise in the cabin does not require fluid finite element model of the cabin air, thus greatly reducing the calculation time compared with the pure finite element method. It was shown that the method is suitable for the calculation of noise in a single ship cabin; the method has a high calculation efficiency. Furthermore, the calculated result is a continuum. On the one hand, it can be conveniently converted to an octave or 1/3 octave according to the specification. On the other hand, the form of the continuum also provides a corresponding response to the subsequent vibration and noise control.

Keywords: low to middle frequencies; vibration; ship noise; analysis model; rapid calculation method

INTRODUCTION

The revised draft of "Rules on Ship Noise Levels" was officially approved in the 1st meeting of Maritime Safety Committee (MSC) of the International Maritime Organization (IMO), in which, the noise level requirements for ships above 10,000 GT in habitats, medical districts, restaurants and entertainment sites are lower by 5 dB as a compulsory standard[1]. With the implementation of new standards for ship noise, ship noise has become an increasingly difficult problem for shipyards. Ship noise affects ship cabin comfort, lowers workers' work efficiency, easily causes cardiovascular diseases and damages human health[2]. For ships, it may cause acoustic vibration and fatigue damage to ship structure, affecting normal operation of cabin instrument and equipment. To reduce ship noise, a current common practice in engineering is to achieve targeted reduction of measured ship noise after the completion of ship building, but the actual effect of this practice is unsatisfactory. According to scholar statistics, installation of acoustic equipment on built ship costs about three times more than that done in ship design and building stage[3]. Therefore, it is necessary to accurately and quickly predict cabin noise during the ship design.

The methods for noise prediction usually include engineering estimation, theoretical analysis and numerical calculation[4]. Engineering estimation method is a semi-empirical and semi-analytic method mainly characterized by convenient use and fast estimation speed. The examples are more studied transmission path analysis[5,6] and grey forecasting methods[7,8]. Engineering estimation method makes prediction based on measured data of parent ship, which provides high prediction accuracy when the predicted ship and parent ship share similar structural characteristic parameters (ship structure, equipment type, general arrangement, etc.). The method depends more on engineering personnel experience when the predicted ship differs greatly from the parent ship and engineering accuracy requirements will not be met in case of inaccurate parameter selection. Theoretical analysis method directly establishes coupling dynamic equations on structure and air to obtain acoustic radiation values of the structure through modal expansion and integral transformation[9-11]. The method has high solving efficiency and good calculation accuracy, but only applicable to geometric structures with simple analytical solutions and thus less applied in ship noise prediction. Numerical methods include statistical energy method, finite element method, boundary method, etc. Statistical energy method can only be applied to frequencies with higher modal density, merely able to calculate the average response level of the entire modal set in the statistical sense, and unable to obtain exact response of each modal, while finite element, boundary element methods are more applied to low frequencies owing to limited computer performance.

Noise is generally categorized into low frequency, intermediate frequency and high frequency noise according to the modal density or the number of vibration modes within the bandwidth[12]. There is no clear boundary between low-frequency and intermediate-frequency and between intermediate-frequency and high-frequency noise, and the division varies for different structures and different fields. Because of its short wavelength, high-frequency noise can be easily eliminated by common passive noise control measures such as widely used cotton felt materials. Reduction of low-middle-frequency noise requires sufficient cotton felt thickness and weight, thus inapplicable to ships with narrow cabins, making low-middle-frequency noise more harmful than high-frequency noise. Given large volume and complex structure of the ship, direct calculation of low to middle frequency noise using finite element/boundary element method requires establishment of finite element models for ship structure and sound field to make coupling calculation, which has such disadvantages of large-scale calculation and low solution efficiency in view of the current complex ship structure[13]. It often takes several days to seek solution in coupling model calculation of a cabin, which is time consuming, laborious, and costly.

Ship cabin can be studied as a closed cavity. Owing to its generally regular shape, solution can be directly obtained using theoretical analysis. The ship cabin noise is affected by structural noise, air noise and electromagnetic noise. According to superposition principle of acoustics, total sound pressure can be obtained via linear superposition of values of various noise types. Air noise and electromagnetic noise are generally produced by equipment on ships, which involves simple estimation in which ship noise is mainly solved by calculating ship structural noise. The structure is strongly coupled to the air, so structural vibration will largely affect the noise level of the cavity to which it is coupled, while sound pressure in the cavity exerts little influence on structural vibration. The study on ship noise transmission path shows that structural noise is the main source of noise for cabins without indoor air noise sources, and the impact of air noise of the main engine and propulsion system is almost negligible for the distant living cabins. Therefore, structural noise prediction of the ship is the key to ship noise prediction. It is generally difficult to determine coupling of air and structure, and the total noise can be obtained by solving structural noise and then adding it with air noise and electromagnetic noise.

In view of the impact of the ship structural noise, this paper studies rapid calculation method targeted at single ship cabin. The ship noise is divided into two types: hull structure noise and equipment noise. Air impact on structure is limited, so structural vibration noise is the main source of indoor noise when there is no noise source in the cabin. The vibration acceleration value of the target cabin is extracted from the ship's finite element calculation model. For cabin that requires noise estimation, there is no need to establish a fluid finite element model for complex fluid-structure interaction calculation, and noise value of the target cabin can be obtained by directly establishing and solving the cavity-plate coupling model.

COUPLING MODEL

PLATE-CAVITY COUPLING MODEL

The cabin was studied as a closed chamber, which was surrounded by six thin plates. The source of sound pressure in the chamber was divided into two parts: the radiation of the plate and the noise source inside the cavity. Here only the coupling equation between the cavity and the single plate needs to be deduced. According to the principle of linear superposition, the coupling of the cavity and multiple plates is a linear superposition of the coupling of the cavity and the single plate. For the cavity, the control equation is:

$$\nabla^2 p - \frac{1}{c^2}\ddot{p} = -2\rho_0 \ddot{w}\delta\left(z - z_0\right) \tag{1}$$

In Eq. (1):

c – Sound speed in air;
ρ_0 – The density of air;
p – Sound pressure in cavity;
w – Vibration displacement of the plate.

For the plate, the control equation is:

$$D\nabla^4 w + m\ddot{w} = -p\big|_{z=z_0} \tag{2}$$

In Eq. (2):

D – Bending stiffness;

E – Young's Modulus;

h – Thickness of the plate;

v – Poisson's ratio;

m – Plate mass per unit area.

The modal expansion of the vibration displacement of the plate and the sound pressure of the cavity are write respectively by:

$$p = \sum_{n=0}^{\infty} \Phi_n P_n e^{i\omega t}, \quad w = \sum_{n=1}^{\infty} \varphi_n W_n e^{i\omega t} \tag{3}$$

Each mode of acoustics and structures meets the following characteristic equations, respectively:

$$\nabla^2 \Phi_n + \frac{\alpha_n{}^2}{c^2}\Phi_n = 0, \quad D\nabla^4 \varphi_n - \beta_n{}^2 m\varphi_n = 0 \tag{4}$$

In Eq. (4):

α_n, β_n – cavity and plate nth mode angular frequencies, respectively.

Substituting Eq. (4) into Eq. (1) and using the modal orthogonality, the decoupled equation can be obtained:

$$\frac{N_j^{\mathrm{I}}}{\rho_0 c^2}\left(\omega^2 + 2i\xi_j^C \omega\alpha_j - \alpha_j{}^2\right) P_t = \omega^2 \sum_{n=1}^{\infty} C_{jn} W_n \;, \tag{5}$$

$$\left(\omega^2 + 2i\xi_j^P \omega\beta_j - \beta_j{}^2\right) M_j W_j = \sum_{n=0}^{\infty} C_{nj} P_n \,. \tag{6}$$

In Eq. (6) and Eq. (6):

$$N_j^I = \iiint_\Omega \Phi_j^2 d\Omega, \;\; M_j = m\iint_s \varphi_j^2 ds, \;\; C_{jn} = \iint_s \Phi_j \varphi_n ds$$

Note that the air damping $2i\xi_t^C\omega\alpha_t$ and structural damping $2i\xi_n^P\omega\beta_n$ have been considered in equations (7) and (8), respectively, where ξ_n^C is the air damping of the nth order, and ξ_n^P is the nth order structural damping. In the calculation, the values of n need to be convergent learning to be truncated, and then the modal responses in the cavity can be obtained by the simultaneous expressions (5) and (6). The calculated result can be obtained by modal superposition to obtain the sound pressure value at any point in the cavity.

POINT SOURCE

In addition to the noise generated by the hull structure, noise sources inside the cabin, such as indoor small mechanical vibration, noise generated by rotation, and air noise generated by air conditioning and ventilation, will also have an impact on the total noise value inside the cabin, and it is related to structural noise. Superimposed to get the total noise of the cabin. The source of the sound is used to simulate the noise source inside the cabin, and the source strength of the sound source is set to Q. The control equation in the cabin is:

$$\nabla^2 p - \frac{1}{c^2}\ddot{p} = -\rho_0 \dot{q}\delta\left(x - x_s\right)\delta\left(y - y_s\right)\delta\left(z - z_s\right) \tag{7}$$

Where (x_s, y_s, z_s) is the coordinates of the sound source in the cabin, and $q=Qe^{i\omega t}$ is the intensity of the sound source. Using the modal expansion method to solve equation (7) and get the result:

$$p = -i\omega\rho_0 c^2 Q \sum_{j=0}^{\infty} \frac{\Phi_j\left(x_s, y_s, z_s\right)}{\left(\omega^2 - \alpha_j{}^2\right) N_j^I} \Phi_j \tag{8}$$

SYNTHETIC SOUND PRESSURE LEVEL

The sound field in a cavity is usually a combination of multiple noise sources, and the total sound pressure is a vector superposition of multiple noise sources. Assuming that there are n noise sources, and each noise source has an independently generated sound pressure of $\mathbf{p}_i$, i=1, 2, …, n, the expression for the total sound pressure $\mathbf{p}$ is

$$\mathbf{p} = \sum_{i=1}^{n} \mathbf{p}_i \tag{9}$$

The sound pressure of each frequency under the same position can be obtained by formula (9). The formula for the total sound pressure level decibel number for all frequencies is:

$$L_p = 10\lg\left(\frac{1}{p_0^2}\sum_{j=1}^{u} p_j^2\right) = 10\lg\left(\sum_{j=1}^{u} 10^{0.1L_j}\right) \tag{10}$$

In Eq. (10):

L_p – Synthetic sound pressure level;

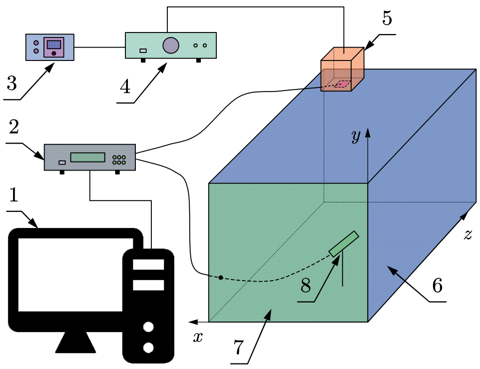
p_0 – Reference sound pressure.

VALIDATION

Some simplifications have been made in the previous derivation process, and it is necessary to verify the stability and effectiveness of the procedure.

EXPERIMENTAL SETUP

Figure 1 shows a schematic of the experiment. A reverberation box made by high-density board with 25mm thick was used. The dimensions were: length(x) × height(y) × depth(z) = 0.6m × 0.8m × 1.0m. The reverberation box is open on one side and coupled to an aluminum plate. The four edges of the aluminum plate are fixed on the box by a bead. The modes are calculated according to the fixed boundary conditions. Calculations of the plate madal can use either analytical solutions[14–17] or finite element or measured results. This paper uses analytic solutions to perform the calculation[16,18,19]. The dimensions of the aluminum plate are: length (x) ×width (y) = 0.6m×0.8m, the Young's modulus of the aluminum plate is 69GPa, the Poisson's ratio is 0.3, and the plate thickness is 4mm. As shown in the coordinate system, the plane where the aluminum plate is located is the *xy* plane.



1. computer 2. dynamic signal analyzer 3. signal generator 4. amplifier 5. speaker 6. reverberation box 7. aluminum plate 8. microphone

Fig. 1. Experimental setup and related equipment connection

At the top of the box, there is a square hole 0.1m by 0.1m in the *xz* plane. The center of the square hole is [0.2, 0.8, 0.85], in meters, and a 6.5-inch speaker is installed as a sound source. There is a small cavity behind the horn to separate the horn sound from the outside world so that the horn produces only a forward plane wave. The interior of the small cavity is provided with sound-absorbing cotton, which reduces the volume of the cavity and makes the cavity not coupled with the large cavity actually tested. The actual excitation is a plane wave that enters the cavity from a square hole. Because the coupling of the plane wave and the cavity is more complicated, in the numerical simulation calculation, a point sound source is used instead of the plane wave for excitation. For the difference between the experimental model and the experimental device, corrections need to be made to ensure the accuracy of the calculation result. Cummings gives a correction formula using a point source instead of a square hole [20]:

$$\bar{Q}(\mathbf{r}_c) = Q(\mathbf{r}_c)\prod_{i=1}^{3}\sin\left(n_i\pi\varepsilon_i / 2l_i\right)\Big/\left(n_i\pi\varepsilon_i / 2l_i\right) \quad (11)$$

Through the correction of this formula, the point source can be instead by the plane wave in the reverberation box.

MODE TRUNCATION

Since the modalities of the cavity and the plate in the formula are all extended to infinite order, the actual calculation needs to be truncated when the calculation accuracy is satisfied. In order to ensure the accuracy of the truncation, convergence learning should be performed. Choosing the fewest number of modalities involved in the coupling calculations, the whole system has higher accuracy and computational efficiency. It is a key issue for the modal expansion method to calculate. Assume that the frequency range of interest is $[f_{min}, f_{max}]$, f_{min} and f_{max} are the minimum and maximum values of the frequencies concern. The general truncation method is based on the principle of frequency truncation in modal superposition, and a certain multiple of f_{max} is used to truncate the main mode. In the coupled system, each rectangular plate and each cavity can be regarded as a subsystem. The method of truncating the frequency of each subsystem is to adopt a certain multiple of the f_{max} of the subsystem to perform the mode of the substructure. Truncated [21]. In general, taking 2 to 3 times of f_{max} for truncation can achieve better accuracy [22]. In order to find the best cutoff frequency, convergence learning is performed and truncation is calculated by taking 1 to 3 times of f_{max} respectively. In order to ensure that the calculated frequency covers all low frequencies with low modal densities, and that the comparison result has higher accuracy, the highest frequency of interest in the modal truncation calculation is taken as a slightly larger value, which is taken as 1200 Hz. Far more than the frequency of interest in actual calculations, it is used only for verifying the frequency truncation accuracy, and the smaller value actually used in the actual calculation is also guaranteed to be satisfied. See Table 1 for the order of the board and cavity.

Tab. 1. Convergence Learning mode of each structure

substructure	P_t	R	H
Plate	1	54	1202.3
	2	115	2399.7
	3	175	3592.1
Cavity	1	126	1207.6
	2	846	2399.9
	3	2671	3600

In Tab. 1:
P_t – Multiple of f_{max};
R – Modal order retained;
H – The highest order modal frequency(Hz).

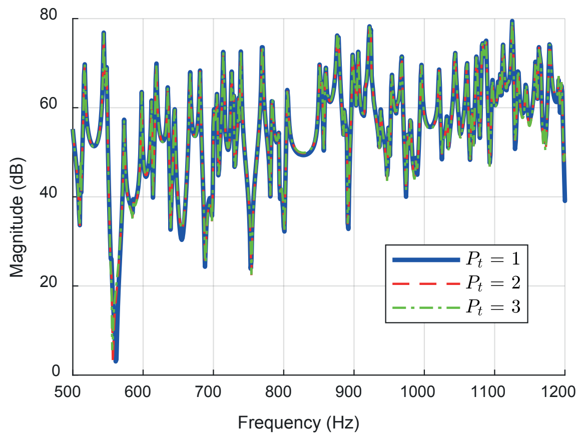


Fig. 2. Convergence Learning

Randomly select one point for the response calculation and compare the calculated results cut off at different frequencies. The coordinates of the arbitrary point taken are [0.1, 0.15, 0.05] in units of meters. In the frequency range of interest, when the error of the higher cutoff frequency calculation and the lower cutoff frequency is lower than the error, the lower cutoff frequency can be considered to meet the calculation accuracy requirement. The result of the response calculation is shown in Fig. 2. It can be seen from the figure that the response lines at Pt=2 and Pt=3 have completely coincided, which is slightly different from Pt=1. The difference between the calculations of Pt=2 and Pt=1 is a maximum of 6dB, and the average is 0.258 dB; the difference between Pt=3 and Pt=2 is a maximum of 0.87 dB, and the average value is 0.029 dB. The difference between Pt=3 and Pt=2 is very small, and the average difference of 0.029 dB is already within the calculated error range. Therefore, it can be considered that the calculation has converged when Pt=2. When Pt=2, the moduli of the plate remain the first 115 orders, and the cavities retain the first 846 orders, which can not only satisfy the calculation accuracy requirements, but also have high calculation efficiency.

VALIDATION OF SIMULATION MODEL

This section compares the measured and calculated noise. A microphone was installed inside the reverberation box to test the response of the cavity. The coordinates of the measured point is [0.12m,0.15m,0.05m]. White noise input was used for the speaker. To ensure the accuracy of the experiment, the white noise bandwidth should be more than twice the maximum frequency of interest, which is 2.5 kHz here. Since the actually used speaker is not an ideal device, the speaker's excitation of the air cavity is slightly different at each frequency. To eliminate the effect of sound source intensity, calculations and tests are performed using transfer functions instead of responses. An accelerometer was mounted on the speaker used as the excitation source, and the test data of the accelerometer and the microphone are simultaneously input to the dynamic signal analyzer to obtain the transfer function.

For the coupled model, the magnitude of the point source is Q, then the transfer function T is calculated by:

$$T = P / \mathrm{i}\omega Q \tag{12}$$

The comparison between the calculated results of this section and the experimental results is shown in Fig. 3. It can be seen from Fig. 3 that the predicted noise value of the model agrees well with the measured value. In low frequency bands with low modal densities, formants with very high discrimination can be seen.

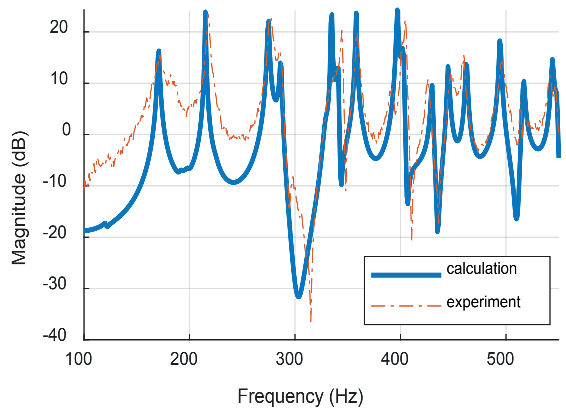


Fig. 3. Compare Experimental and Calculate Results

Tab. 2 shows the comparison of the first 10 measured peaks with the calculated peak position of the simulation. It can be seen from the table that, with the exception of the first peak occurring at 32 Hz, the other peak frequencies are all greater than 170 Hz, which is far from the 100 Hz region where the data in the experiment is not reliable. At the same time, the frequency difference between the calculation and the experiment is basically between 1 and 3 Hz and the maximum is not more than 5 Hz. From the viewpoint of the amplitude, the experimental results and the calculation results also basically coincide, and it can be considered that the results of this calculation model are reliable.

Tab. 2. Resonant Position Comparison

Peak No.	Model(Hz)	Experiment(Hz)	Error
1	170	171.5	-1.5
2	215	217	-2
3	275	277	-2
4	285	285	0
5	335	338	-3
6	359	360	-1
7	396	401	-5
8	430	426	3
9	445	447	-2
10	462	460	-2

The experimental device uses the air noise source as the excitation. In order to further understand the influence of the indoor sound source on the plate-cavity coupling model, the six-sided rigid wall single cavity and the plate cavity coupling structure were compared, and the other settings remain unchanged. Fig. 4 is a comparison of the calculation results of the single cavity model and the plate-cavity coupling model. It can be seen from the figure that the single cavity model does not show a peak at 32 Hz. For the convenience of observation, the first 15 modes of the plate are marked with a vertical dotted line in Fig. 4 and the first 10 modes of the cavity are marked with a vertical solid line. It can be seen that in the case of the excitation source inside the cavity, the peak is mainly controlled by the cavity mode, and the first mode of the plate has the greatest influence on the cavity, and the effects of other modes on the response in the cavity are compared. Small, basically negligible. It can be seen that the coupling of air to the plate in the cavity is weakly coupled, the noise excitation inside the cavity has very little effect on the plate, and only a small resonance is excited at the first mode frequency of the plate. From Fig. 4, the resonance is located at a very low frequency, where the frequency is approximately 35 dB attenuated under the common A frequency weighting, so the total sound pressure value for the A weighting is almost negligible, in practical calculations. The influence of the coupling can be ignored directly, and it can be considered that the sound source inside the cavity and the sound source for the vibration of the plate structure can be calculated separately and independently without affecting the accuracy of the final result.

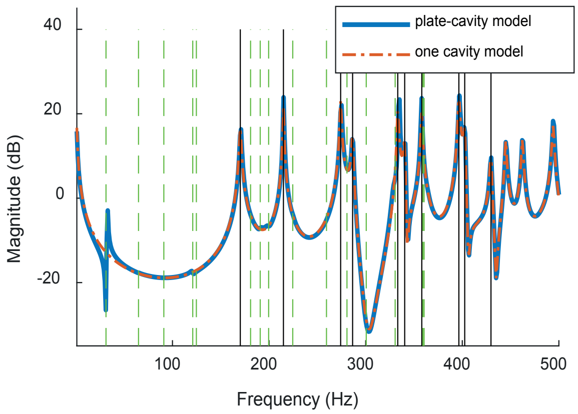


Fig. 4. Compare of Single Cavity and Cavity-Plate Model

In summary, it can be seen that it is completely feasible to divide the noise sources into cabin sound sources and structured sound sources and separate them. The method of analytical calculation is used to calculate the noise in the cavity, which has a very high calculation accuracy and can meet the requirements of ship noise prediction. The following will take a container ship as an example to calculate the noise of a ship in an upper cabin.

SHIP CABIN NOISE CALCULATION

STRUCTURAL NOISE SOURCES

The acceleration value of the structure extracted from the target cabin is used as the noise source. Fig. 5 shows the model of the target cabin. The model is part of a full-ship model for a container ship with a total length of 94.8 meters, a width of 15.4 meters and a depth of 6.9 meters. The main engine is a four-stroke inline six-cylinder diesel engine with a rated power of 1,618 kW at a speed of 600 rpm. Propeller diameter of 2.7 meters, 4 blades, speed of 200 rpm, the actual ship received power of 1522 kilowatts. According to China Ship Classification Society "Guide on Shipboard Vibration Control", ISO 6954-2000 Standard and ISO 6954-2000 (IDT) Standard, propeller excitation is considered and frequency response method is used to analyze the forced vibration response of the ship.

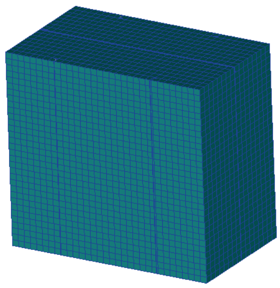


Fig. 5. Model of Target Cabin

The target cabin is located in the middle of the upper layer of the ship, with a height of 16.5m. There is no mechanical equipment that generates noise in the room. The noise source is structural vibration noise. The average acceleration curve of the extracted target cabin structure is shown in Figure 6. After obtaining the vibration acceleration of each wall of the target cabin, this acceleration is used as the excitation source of our coupling model to calculate the cabin vibration and noise.

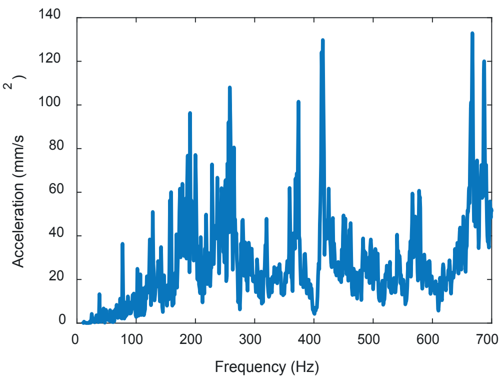


Fig. 6. Average Acceleration of the Target Cabin Structure

In order to apply the coupling model developed in the previous paragraph to the target cabin, the coordinate axes shown in Fig. 7 are established for the target cabin, where the x-axis is along the width of the ship, the y-axis is along the height of the ship, and the z-axis is along the length of the ship. The dimensions are: length (x) × height (y) × depth (z) = 2.5m × 2.3m × 1.5m. A coupled model was created and calculated for the chamber and the six plates coupled to this chamber. The boundary conditions of the six panels were all considered fixed. Using the acoustic superposition principle, the total sound pressure in the cabin is the sum of the acoustic pressure generated by the six plates coupled to the cavity.

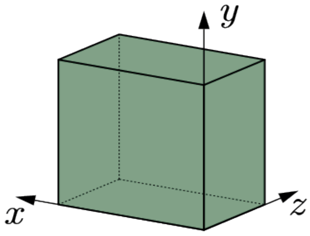


Fig. 7. Axis of Target Cabin

CALCULATION RESULTS

The result of the structural noise calculation is shown in Fig. 8. The positions of the two nodes are node 1: [1700mm, 500mm, 600mm] and node 2: [600mm, 1100mm, 300mm]. This result is the result of the continuous spectrum sound pressure level. According to the Chinese Society of Shipbuilding's "Guidelines for Control and Inspection of Ship and Product Noise Control and Detection", the method for calculation of cabin noise has a frequency interval of 1 Hz and meets the specification requirements. In order to better evaluate the impact of noise on the ship's environment and harm to the human body, according to the requirements in the guidebook, the continuum is converted into an octave and weighted by A level. The results are shown in Tab. 3.

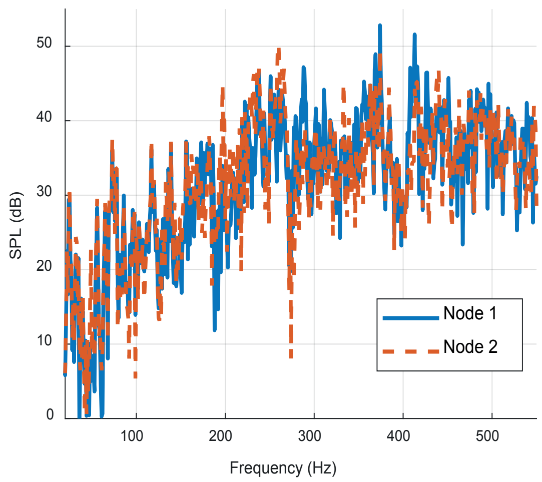


Fig. 6. Average Acceleration of the Target Cabin Structure

Tab. 3. Octave sound pressure and A – weighted total sound pressure

Octave (Hz)	Node 1 (dB)	Node 2 (dB)
31.5	32	28
63	38.5	36
125	41.4	41.7
250	49.2	50.8
500	56.9	57
total	59.2	59.6

When the ship goes straight along the design speed during the sea trial, the noise of node 1 and node 2 is measured using a sound level meter (BK 2250). The measurement frequency range was selected to be 20-550 Hz, and the total sound pressure measured was 61 dB (A) and 62 dB (A), respectively. Compared with the calculated value, the error of the two values does not exceed 3 decibels, which indicates that the calculation results using this method are more reliable and the precision required by the project is achieved.

CONCLUSIONS

In the acoustic coupling between air and structure, structural vibration generates noise in the air, and air vibration exerts very weak influence on the structure. The experiments in this paper show that when the structure encloses as a closed cavity, modal is produced in the cavity and cavity air exerts a very small effect on the vibration which is completely negligible in ship noise calculation. Therefore, in the calculation of ship cabin noise, noise falls into two categories, one is air noise, which generates noise in the cavity, and has no significant impact on structural vibration of the ship. The impact can be ignored as long as there is no such noise in and around the cabin. The other type of noise is structural noise which is generated by structural vibration of the ship. The plate-cavity coupling model developed in this paper is targeted at structural noise of the ship.

Most ship cabins are in relatively regular hexahedral shape, so it is very appropriate to apply this coupling model in cabin noise prediction. The prediction excitation derives from the calculation of the ship's finite element dynamic response. If such calculation is made for the ship in advance, direct data extraction is possible with strong adaptability. There is no need to establish finite element modeling of the fluid in the cabin. For a specific target cabin, workload of data extraction can be effectively controlled if data directly coupled to the cabin is obtained. With the finite element calculation results of ship vibration as the excitation, direct use of plate-cavity coupling model in calculation has low requirement for computer resource, and calculation only takes a few minutes if it is done on a computer with 16G memory. Nevertheless, it will take hours if air is modeled using full finite element calculation method for fluid-structure coupling calculation. The results were consistent with measurement index of real ship sailing tests when the coupling model was used to predict the cabin noise. This method is a good solution for forecast analysis of low-middle-frequency noise. Meanwhile, continuous spectrum is obtained as the calculation result of the method. On the one hand, it can be easily converted to an octave and compared with the specification. On the other hand, the continuous spectrum also provides corresponding data basis for the subsequent vibration and noise control work.

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