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Effects of Inplane Modes in SEA on Structure-Borne Noise Transmission in Ship Structures

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Abstract

It is normal practice to consider bending wave modes only, when one applies SEA (Statistical Energy Analysis) to ship structures because of complexities in SEA modeling and evaluation of coupling loss factors for inplane modes. According to the result of Tratch[1], the inplane wave modes becomes important for the analysis of a foundation structure as the distance from the source and receiver increases.

In this paper, the effect of inplane wave modes on structure-borne noise propagation in ship structures is presented. It is shown that the inplane wave could increase the noise level more than 10 dB compared with the results without inplane wave modes at high frequency bands for compartments far from the source location.

1 Introduction

SEA[2] is known to be one of the practical methods available for the prediction of noise onboard ships based on theoretical background. It has been a common practice to consider bending mode only, when one deals with SEA modeling of ship structures[3][4]. This is because the inclusion of inplane modes makes the problem difficult to handle. For example, SEA model of a ship structure, in general, comprises several hundreds of SEA elements and thousands of structural joints for the analysis of bending wave component only. When one include inplane wave component, the number of SEA elements increases about three times and the number of joints grows even more rapidly.

The other reason could be found in the assumption that the inplane wave mode has little effect on the result. But, Tratch[1] showed that the inclusion of inplane vibration to SEA model of a machinery foundation could increase the vibration level of bottom plate significantly. His conclusion implies that the usual assumption mentioned above could not be valid.

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In this paper, the effect of the inplane wave on the transmission of structure-borne noise in ship structures is presented. The result shows that the exclusion of inplane wave could under-estimate noise level as much as 20 dB for compartments located far apart from noise sources in the high frequency ranges even though the overall dB(A) levels remain within ranges unnoticeable.

2 SEA Equation

SEA equation for a system composed of N sub-systems is given as follows.

$$\Pi_i = \omega\eta_i N_i\left(\frac{E_i}{N_i}\right) + \omega \sum_{j=1,j\neq i}^{N} \eta_{ij} N_i\left(\frac{E_i}{N_i} - \frac{E_j}{N_j}\right), \quad i = 1, 2, \ldots, N \tag{1}$$

where, Π_i, E_i denote external input power and total energy of i-th sub-system, η_i, η_{ij}, and N_i are internal loss factor of i-th sub-system, coupling loss factor between i-th and j-th sub-system, and mode count of i-th sub-system, respectively. Rewriting the above equation in matrix form yields

$$\vec{\Pi} = \omega[C]\vec{E} \tag{2}$$

where, $\vec{\Pi} = [\Pi_1, \Pi_2, \ldots, \Pi_N]^T$, $\vec{E} = [E_1/N_1, E_2/N_2, \ldots, E_n/N_N]^T$. Note that loss factor matrix $[C]$, whose elements are combinations of internal and coupling loss factors, is symmetric and banded-matrix when considering the reciprocity relation of coupling loss factors, i.e. $\eta_{ij}N_i = \eta_{ji}N_j$. Because of these properties of loss factor matrix, we can use the standard FEM solution algorithm when implementing the SEA equation into computer program to reduce the amount of required memories and computing times[4].

In order to include inplane wave effects, we have to idealize a structural element by 3 SEA elements in contrast to the analysis considering bending wave only. Besides SEA element with bending resonators, we should model additional two SEA elements each representing longitudinal and transverse shear wave resonators for every structural members of ship structures. Therefore, the number of total sub-systems is 3 times of the case without inplane components for structural members. For complicated structures with several hundreds of physical elements such as ships, the solution scheme of the SEA equation is even more important when one includes inplane wave effects in the analysis.

3 Coupling Loss Factors

Coupling loss factors are related with transmission coefficients as follows[2].

$$\eta_{ij} = \frac{c_g L_{ij}}{2\pi^2 f A_i}\langle\tau_{ij}\rangle \tag{3}$$

where, f, c_g and A_i are frequency, group velocity of the wave considered and the area of the i-th element, respectively. L_{ij} and $\langle\tau_{ij}\rangle$ represent length of the joint and

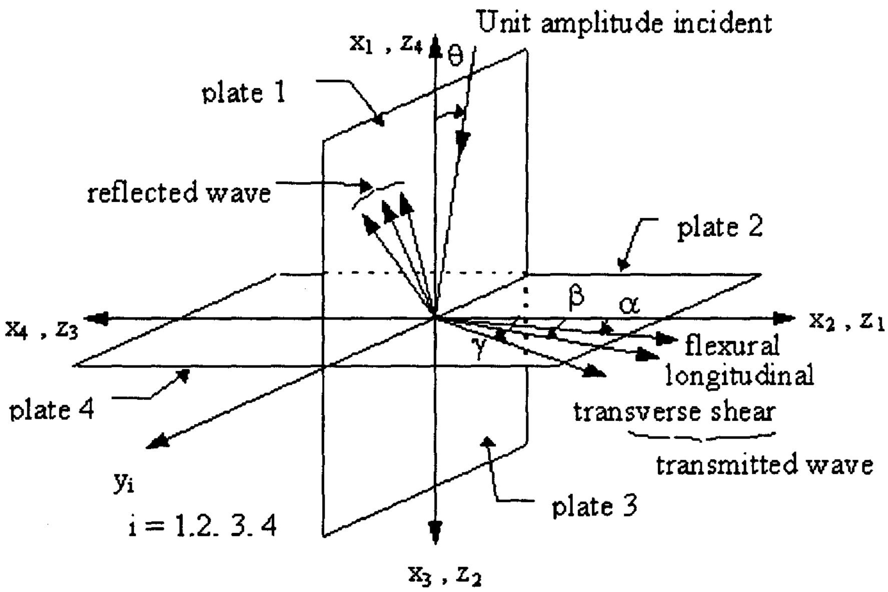


Figure 1: Waves on a plate junction

average power transmission coefficient at the joint between the i-th and j-th element, respectively.

3.1 Waves at Joint

When any type of structural wave is incident on a plate joint, reflected and transmitted waves are generated and they are consisted of 3 types of waves, i.e. bending, longitudinal and transverse shear waves, as shown in Figure 1. And the wave fields on each plates are given as follows[6].

Bending wave:

$$W_n = \{a_n \exp(-ik_n x_n \cos\alpha_n) + a_n^{'} \exp(-k_n x_n \sqrt{1+\sin^2\alpha_n})\} \exp(-ik_n y \sin\alpha_n)$$

Longitudinal wave:

$$L_n = b_n \exp(-il_n x_n \cos\beta_n) \exp(-il_n y \sin\beta_n) \tag{4}$$

Shear wave:

$$T_n = c_n \exp(-it_n x_n \cos\gamma_n) \exp(-it_n y \sin\gamma_n)$$

In the above expressions, x_n, $y_n(=y)$, and z_n are coordinates systems of plate n as defined in Figure 1, and k_n, l_n, and t_n represent bending, longitudinal and shear wave number of plate n, respectively.

$$k_n = \sqrt{\omega}(\frac{D_n}{\rho_n h_n})^{-1/4}, \quad l_n = \omega(\frac{E_n}{(1-\nu_n^2)\rho_n})^{-1/2}, \quad t_n = \omega(\frac{G_n}{\rho_n})^{-1/2} \tag{5}$$

Where, ω denotes angular frequency. ρ_n, ν_n, h_n, D_n, E_n, and G_n are density, Poisson's ratio, thickness, bending rigidity, Young's modulus, and shear modulus of plate n, respectively.
The reflected or transmitted angles, α_n, β_n, and γ_n, are obtained from Snell's law at the joint.

$$l_n \sin\beta_n = t_n \sin\gamma_n = k_n \sin\alpha_n \tag{6}$$

The unknown wave amplitudes are a_n, $a_n^{'}$, b_n, and c_n, where $n = 1$ is for reflected waves and $n = 2, 3, 4$ is for transmitted waves.

Displacements of each plates into 3 principal directions are given as follows. Note that the direction of displacement is the same as that of the wave propagation for longitudinal waves while the two directions are perpendicular to each other for transverse shear waves.

$$\begin{aligned} x_n - \mathit{direction} &: U_n = L_n \cos\beta_n + T_n \sin\gamma_n \\ y\ - \mathit{direction} &: V_n = L_n \sin\beta_n - T_n \cos\gamma_n \\ z_n - \mathit{direction} &: W_n = W_n \end{aligned} \tag{7}$$

The boundary conditions at the joint ($x_n = z_n = 0$) must satisfy the continuity of displacements and slopes as well as the equilibrium of forces and moment. They are given as follows.

$$\begin{gathered} W_1 = U_2 = W_3 = -U_4,\ \ U_1 = -W_2 = -U_3 = W_4,\ \ V_1 = V_2 = V_3 = V_4 \\ \frac{\partial W_1}{\partial x_1} = \frac{\partial W_2}{\partial x_2} = \frac{\partial W_3}{\partial x_3} = \frac{\partial W_4}{\partial x_4} \\ Q_1 - Q_3 - F_2 + F_4 = 0,\ \ Q_2 - Q_4 + F_1 - F_3 = 0,\ \ S_1 + S_2 + S_3 + S_4 = 0 \\ M_1 + M_2 + M_3 + M_4 = 0 \end{gathered} \tag{8}$$

where, Q_n, F_n, S_n, and M_n represent shear force due to bending waves, longitudinal force and shear force due to inplane waves, and moment due to bending waves, respectively.

$$Q_n = D_n\left\{\frac{\partial^3 W_n}{\partial x_n^3} + (2-\nu_n)\frac{\partial^3 W_n}{\partial x_n \partial y^2}\right\},\ \ M_n = D_n\left\{\frac{\partial^2 W_n}{\partial x_n^2} + \nu_n\frac{\partial^2 W_n}{\partial y^2}\right\}$$

$$F_n = \frac{E_n h_n}{1-\nu_n^2}\left(\frac{\partial U_n}{\partial x_n} + \nu_n\frac{\partial V_n}{\partial y}\right),\ \ S_n = G_n h_n\left(\frac{\partial U_n}{\partial y} + \frac{\partial V_n}{\partial x_n}\right)$$

From 16 boundary conditions shown in (8), the unknown 16 amplitudes of transmitted or reflected waves appeared in (4) could be obtained. For plate junctions with less than 4 plates such as T or L joint, one could obtain the same results just eliminating components from plates removed.

3.2 Power Transmission Coefficient

When a wave is incident on a joint with angle θ, the incident power carried by the wave is

$$\Pi = \frac{1}{2}\rho_s c_g |v|^2 \cos\theta \tag{9}$$

where, ρ_s, c_g, and v are surface density, group speed of the wave and velocity of the plate, respectively. The same relation holds for reflected or transmitted wave. Therefore, the power transmission coefficients at a plate junction, that is the transmitted energy divided by the incident energy, are obtained as follow when bending wave is incident.

$$\begin{aligned} \tau_{1n}(BB) &= \frac{\Pi_B^n}{\Pi_B^1} = \frac{|a_n|^2 D_n k_n^3 \cos\alpha_n}{D_1 k_1^3 \cos\alpha_1} \\ \tau_{1n}(BL) &= \frac{\Pi_L^n}{\Pi_B^1} = \frac{\rho_n h_n \omega c_L^n |b_n|^2 \cos\beta_n}{2D_1 k_1^3 \cos\alpha_1} \\ \tau_{1n}(BT) &= \frac{\Pi_T^n}{\Pi_B^1} = \frac{\rho_n h_n \omega c_T^n |c_n|^2 \cos\gamma_n}{2D_1 k_1^3 \cos\alpha_1} \end{aligned} \tag{10}$$

Where Subscript $1n$ stands for from incident(1) to transmitted(n) plate, B and L denote bending and longitudinal waves, and so on. c_L^n and c_T^n are phase speeds of longitudinal wave and transverse shear wave of plate n, respectively. And, ρ_n, h_n, and D_n represent density, thickness, and bending stiffness of plate n, respectively.

Transmission coefficient appeared in (3) is the average value for all incident angles. It is calculated from the relations given below provided that the probability of incident angle is uniform from 0 to 90 degrees. Similar relationships hold for the other cases.

$$\langle \tau_{1n}(BB) \rangle = \frac{\int_{-\pi/2}^{\pi/2} \Pi_B^n d\theta}{\int_{-\pi/2}^{\pi/2} \Pi_B^1 d\theta} \tag{11}$$

One could check the validity of the calculation of transmission coefficients by evaluating the sum of powers of all transmitted and reflected waves on the basis of energy conservation.

$$\sum_{n=1}^{4} \{\langle \tau_{1n}(BB) \rangle + \langle \tau_{1n}(BL) \rangle + \langle \tau_{1n}(BT) \rangle\} = 1 \tag{12}$$

3.3 Reciprocity of Coupling Loss Factors

Reciprocity relation holds between coupling loss factor and modal density as follows. The relationship could be implemented in the computer code so that one could save the time required to calculate coupling loss factors. The reduction is significant especially for the case when one includes inplane wave effects. Roughly speaking, more than half of the computing time was spent to evaluate coupling loss factors using the method stated according to the experience of the authors.

$$\eta_{ij}(f) n_i(f) = \eta_{ji}(f) n_j(f) \tag{13}$$

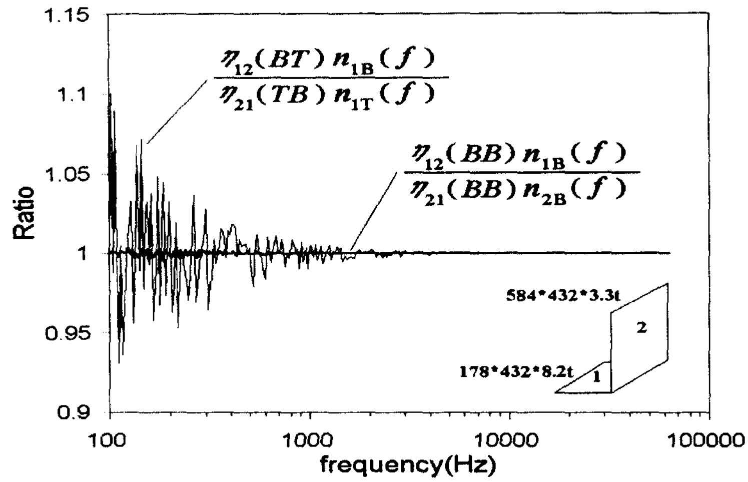


Figure 2: Reciprocity relation for a L-type joint.

Figure 2, which was numerically calculated based on the method described in this paper, shows the validity of the reciprocity relations for L-type connections. In the figure, the ratio approaches to one as the frequency becomes higher, which confirms that the reciprocity relation holds exactly at high frequencies.

4 Example of Applications

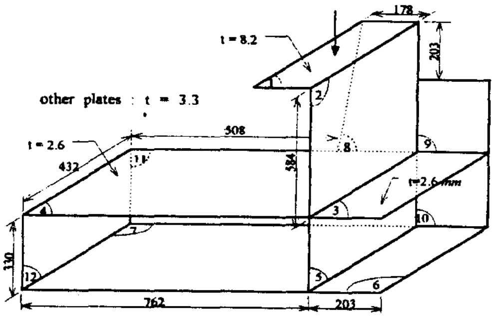
A computer code to calculate coupling loss factors including the effect of inplane wave is developed and merged into 'NASS'[5], a noise prediction program onboard ships using SEA with bending waves only. The updated version of 'NASS' was named as 'SEANV(Statistical Energy Analysis of Noise and Vibration)'.

Two applications of 'SEANV' are presented here together with the discussions of the result.

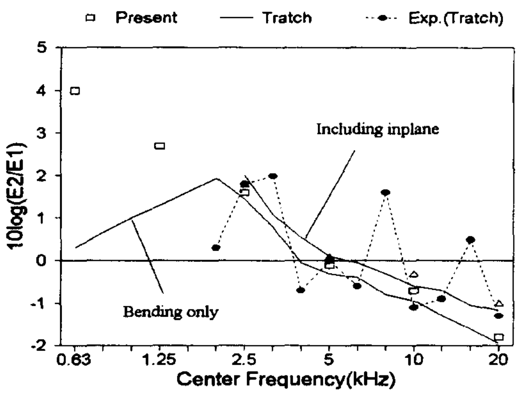
4.1 Machinery Foundation

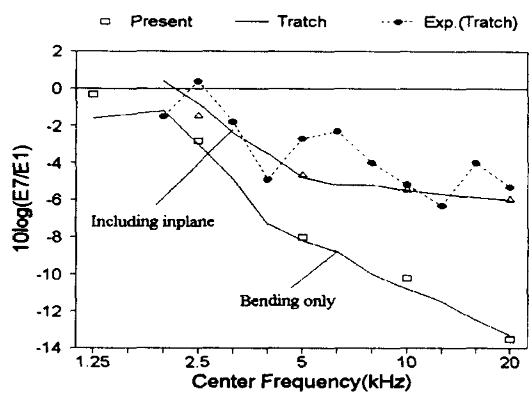
The first example, shown in Figure 3.(a), is a 12-plates SEA model of an engine foundation which is the same as Tratch[1]. In Figure 3.(b) and (c), structure-borne noise energy ratios between source and receiver plates are given together with the result of Tratch. Energy ratios between source plate(E1) and neighbouring plate(E2) are compared in (b), and bottom plate(E7) in (c). The specific locations of each plates are found in (a). The figures show that the present prediction results are in good agreements with those of Tratch except frequency ranges below 1kHz. One possible reason of the discrepacies at those low frequency bands is the uncertainty of SEA parameters such as loss factors between the two models.

It is quite impressive that the structure-borne noise levels increase up to 7 dB when including inplane vibration modes for bottom plate in the high frequency bands



(a) SEA model of a foundation





(b) Ratio of vibration energy (E2/E1) (c) Ratio of vibration energy (E7/E1)

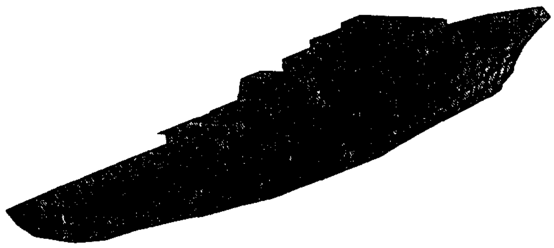
Figure 3: Result of a foundation model

as shown in (c). But, the increase is less than 1 dB for plates adjacent to the source plate. This implies that the effect of inplane waves cannot be neglected for remote locations from major noise sources especially for high frequencies.

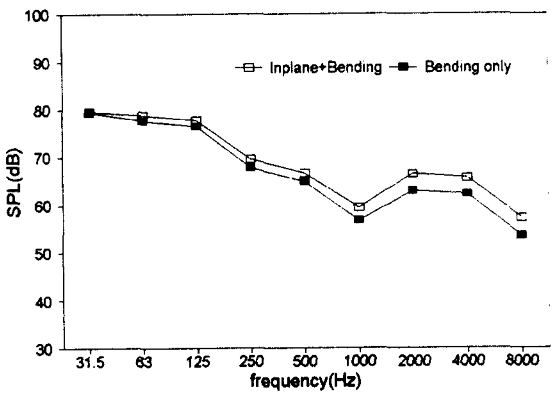
4.2 Shipboard Noise Prediction

The SEA model of a ship structure is shown in Figure 4.(a). It was consisted of 799 plates and 104 air spaces so that the total number of sub-systeme is 2501. Noise sources considered are main propulsion engines and generators installed in the engine room located at lower deck of the aft-ship.

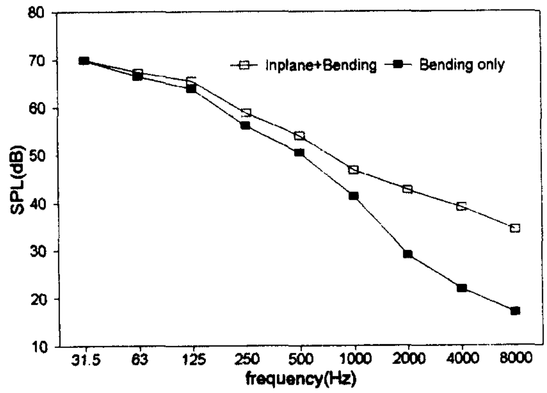
The prediction results are compared in Figure 4.(b) and (c). The specific locations of compartment shown in (b) is on the same deck as the engine room. Noise prediction result of pilot house, 5 decks higher than the engine room, is shown in Figure 4.(c). The figure shows the effect of inplane wave on the compartment noise prediction clearly. According to the experiences of the authors', the noise levels on higher decks predicted by SEA were usually too low to be accepted when comparing with measurement data in the high frequency bands. The discrepancies were thought to be due to the other local



(a) SEA model of a ship



(b) SPL near source area



(c) SPL far from source area

Figure 4: Result of a ship model

noise sources such as HVAC system noise. But, Figure 4 suggests that the neglection of inplane wave components might be partially responsible for those discrepancies.

This example shows that the noise level of compartments far from the main noise sources could be increased up to 20 dB for high frequency bands, when one includes inplane wave modes in SEA model of ship structures. Nevertheless, note that the increase of overall dB(A) level is expected to be around 5 dB(A).

5 Conclusions

The increase of sound pressure level reaches up to 20 dB at 8000Hz octave band when including inplane vibrations in SEA model of ship structures. The result implies that inplane wave effect increases as the frequency and the distance between source and receiver increases. And the validity of reciprocity relations are checked numerically to find out the reciprocity of coupling loss factors holds better as frequency becomes higher.

Acknowledgments

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