Development of Multi-Layered Floating Floor for Cabin Noise Reduction

Jee-Hun Song1*, Suk-Yoon Hong2 and Hyun-Wung Kwon3

1 Department of Naval Architecture and Ocean Engineering, Chonnam National University, San 96-1, Dunduck-dong, Yeosu, Chonnam, 550-749, Korea
2 Department of Naval Architecture and Ocean Engineering, Seoul National University, San 56-1, Silim-dong, Kwanak-gu, Seoul, 151-742, Korea
3 Department of Naval Architecture and Ocean Engineering, Koje College, 91, Majeon 1-gil, Geoje, Gyeongnam, 656-701, Korea

* jhs@jnu.ac.kr

Abstract. Recently, regulations pertaining to the noise and vibration environment of ship cabins have been strengthened. In this paper, a numerical model is developed for multi-layered floating floor to predict the structure-borne noise in ship cabins. The theoretical model consists of multi-panel structures lined with high-density mineral wool. The predicted results for structure-borne noise when multi-layered floating floor is used are compared to the measurements made of a mock-up. A comparison of the predicted results and the experimental one shows that the developed model could be an effective tool for predicting structure-borne noise in ship cabins.

1. Introduction

Recently, ship cabins noise has become a big issue in ship yards as ship’s owner and operators are demanding quieter vessels and relevant noise regulations are getting more stringent. As a rule, however, a ship consists of complex steel structures where structure-borne noise is strong and there are a number of noise sources, such as main engine, diesel generator and auxiliary machinery. From the ship cabins noise tests, it was found that the combined noise level of a cabin was dominated by the radiated noise from the floor structure, which is a combined of a stiffened steel deck and deck covering [1].

Floor impact noise of floating floors has been investigated since the 1950s, and much effort has been directed toward the measurement of such floors. Cremer [2] analyzed the impact noise insulation of floating floors. Istvan [3] reported the characteristics of floor impact noise using a hard floor with an elastic surface material. Yago and Inoue [4] reported experimental results about the characteristics of floor impact noise. Buiten [5] and Jensen [6] reported experimental results of various types of floating floors for ships.

Until now, most research on floating floors has been conducted without a proper understanding of how such an installation works for sound insulation. In particular, research on floating floors with an inserted viscoelastic layer has never been properly conducted.

The aim of this work is to develop an adequate model for multi-layered floating floor that can be used to predict structure-borne noise in ship cabins. To verify the accuracy and validity of the model, the model predictions are compared with measurements made of a mock-up that simulates a typical ship cabin structure.
2. Analysis of multi-layered floating floor

As shown in Figure 1, multi-layered floating floor was investigated. To analyze this floor, we consider two regions: the first is the three-layered sandwich plate, which consists of two isotropic face plates sandwiching a viscoelastic core. The GHM method [7] is used to account for the frequency-dependent complex shear modulus of the viscoelastic layer. The second region is the single panel with high-density mineral wool. The Biot [8, 9] and Bolton [10] models can be used to describe the wave propagation in the wool. We assumed that the transverse displacement is compatible between the regions.

![Image of the multi-layered floating floor model](image)

**Figure 1.** The multi-layered floating floor model.

2.1. The first region: Three-layered sandwich plate

The model is formulated according to Kirchoff’s theory [11] for plates. The face plates are assumed to exhibit bending, in-plane shear, and extensional stiffness, and their lateral shear deformation is neglected. The viscoelastic layer is assumed to exhibit transverse shear stiffness alone. Furthermore, perfect continuity is assumed at the interface, with no slip occurring during the plate bending. Finally, every point of the three-layer cross section undergoes the same transverse displacement.

Under this set of assumptions, we can write the total kinetic energy $T$ and the potential energy $U$ of the sandwich plates as follows:

\[
T = \frac{1}{2} \int_{A} \left[ \rho h \left( \frac{\partial w}{\partial t} \right)^2 + \sum_{i=1,3} \rho h_i \left( \frac{\partial u_i}{\partial t} \right)^2 + \left( \frac{\partial v_i}{\partial t} \right)^2 \right] dA
\]  

(1)

\[
U = \frac{1}{2} \int_{A} \left[ \sum_{i=1,3} N_{x,i} \frac{\partial u_i}{\partial x} + \sum_{i=1,3} N_{y,i} \frac{\partial v_i}{\partial y} + \sum_{i=1,3} N_{xy,i} \left( \frac{\partial u_i}{\partial y} + \frac{\partial v_i}{\partial x} \right) - \sum_{i=1,3} M_{x,i} \frac{\partial^2 w}{\partial x^2} - \sum_{i=1,3} M_{y,i} \frac{\partial^2 w}{\partial y^2} - 2\sum_{i=1,3} M_{xy,i} \frac{\partial^2 w}{\partial x \partial y} + Q_{xz} \gamma_x + Q_{yz} \gamma_y \right] dA
\]  

(2)

$\rho h$ and $G$ are the mass per unit area for the entire total sandwich plate and the shear modulus of the viscoelastic layer, respectively. In order to discretize the system and generate the mass and stiffness matrices, the assumed mode method [12] was used to analyze the sandwich plate. Substituting the displacements for sandwich plate into Eqs. 1-2 yields

\[
[M_e] \ddot{\bar{x}} + [K_e] \bar{x} + [K_v] \bar{x} = [F]
\]  

(3)

where $[M_e]$ is the mass matrix, $[K_e]$ is the stiffness matrices from the face plates and $[K_v] = G [\bar{K}]$ is the stiffness matrix from the viscoelastic layer.
2.2. The second region: Single-panel

The single-panel model is based on Biot’s theory [8, 9] which has been adapted to describe wave propagation in high-density mineral wool. The elastic panel is assumed only to exhibit transverse displacement. The high-density mineral wool is assumed to experience solid and fluid longitudinal waves and a solid transverse shear wave. The solid and fluid stresses are defined to be forces acting on the frame:

\[ \sigma_i = 2N e_i + A e_s + Q e \]  
\[ s = R e + Q e_s \]

\( i = x, y, z \), where \( e_s = \nabla \cdot \mathbf{u} \) and \( e = \nabla \cdot \mathbf{U} \) are the solid and fluid volumetric strain, respectively. \( N = E_s / (1 + \nu) \) and \( A = \nu E_s / (1 + 2\nu) \) are the shear modulus and Lame constant, respectively. Also, the quantity \( Q = (1 - Y)E_f \) is the potential coupling coefficient [13]. \( R = Y E_f \) is a constant that depends on the fluid stress and strain [13]. \( Y \) is the porosity.

The longitudinal and transverse shear wave strains in the solid were assumed to be:

\[ e_s = e^{-j k_s x} (C_1 e^{-j k_{11} y} + C_2 e^{j k_{12} y} + C_3 e^{-j k_{21} y} + C_4 e^{j k_{22} y}) \]
\[ \Omega_{x,z} = e^{-j k_s x} (C_5 e^{-j k_{11} y} + C_6 e^{j k_{12} y}) \]

From the Helmholtz decomposition, we can obtain the \( x \)- and \( y \)-components of the displacement of the solid and fluid, respectively:

\[ u_x = jk_x e^{-j k_s x} (C_1 e^{-j k_{11} y} + C_2 e^{j k_{12} y} + C_3 e^{-j k_{21} y} + C_4 e^{j k_{22} y}) \]
\[ - j \frac{k_{1y}}{k_t} e^{-j k_s x} (C_5 e^{-j k_{11} y} - C_6 e^{j k_{12} y}) \]
\[ u_y = j e^{-j k_s x} (C_1 e^{-j k_{11} y} - C_2 e^{j k_{12} y} + C_3 e^{-j k_{21} y} - C_4 e^{j k_{22} y}) \]
\[ - j \frac{k_x}{k_t} e^{-j k_s x} (C_5 e^{-j k_{11} y} + C_6 e^{j k_{12} y}) \]

\[ U_x = jk_x e^{-j k_s x} (b_1 C_1 e^{-j k_{11} y} + b_2 C_2 e^{j k_{12} y} + b_3 C_3 e^{-j k_{21} y} + b_4 C_4 e^{j k_{22} y}) \]
\[ + j \frac{\rho_1^s k_{1y}^2}{\rho_2^s k_t^2} e^{-j k_s x} (C_5 e^{-j k_{11} y} - C_6 e^{j k_{12} y}) \]
\[ U_y = j e^{-j k_s x} (b_1 C_1 e^{-j k_{11} y} - b_2 C_2 e^{j k_{12} y} + b_3 C_3 e^{-j k_{21} y} - b_4 C_4 e^{j k_{22} y}) \]
\[ - j \frac{\rho_1^s k_{1y}^2}{\rho_2^s k_t^2} e^{-j k_s x} (C_5 e^{-j k_{11} y} + C_6 e^{j k_{12} y}) \]
By substituting Eqs. 6-7 and 8-11 into Eqs. 4-5, we can obtain the normal solid stress $\sigma_y$, the shear stress $\tau_{xy}$, and the fluid stress $s$. The unknown constants $C_1 - C_6$ in Eqs. 6-7 and 8-11 are determined by the boundary conditions at the interfaces of the high-density mineral wool.

3. Experiment Validation
To verify the accuracy and validity of the developed model, the predicted results have been compared with the corresponding experimental results, which comprise measurements of mock-up built that simulates a typical ship cabin structure.

The length and width of the sandwich plate are 3.76 and 2.94 m, respectively. The viscoelastic material modulus and loss factor were obtained from the product information provided by 3M Scotchdamp ISD-11217. A standard tapping machine was used to excite bending motion of the floating floor. The structure-borne noise transmitted by the tapping machine to the lower receiving room is measured in one-third-octave bands, from 31.5 Hz to 8,000 Hz. We measure structure-borne noise in dB defined as $20 \log (\text{rms acceleration (m/s}^2) / \text{ref. acceleration (10}^{-5}\text{m/s}^2))$.

![Figure 2](a) 25mm MW

![Figure 2](b) 50mm MW

Figure 2. The effect of high density mineral wool thickness in floating floor with an inserted viscoelastic layer structure-borne noise.
In Figures 2-3, we compared structure-borne noise measurements to predictions for floating floor with an inserted viscoelastic layer. Comparison shows that the results simulated by the developed floating floor predict the frequency dependency of experimental structure-borne noise relatively well.

In Figure 2, we show the effects of the high-density mineral wool (MW) thickness on structure-borne noise. The floating floor structure is: 6mm deck + MW + 1.6mm steel plate + 1mm viscoelastic layer + 1.6mm steel plate, in which MW thickness varies from 25mm to 50mm. It is found that as the thickness of MW increases from 25mm to 50mm, the structure-borne noise decreases by approximately 4-5 dB.

In the next example, Figure 3 illustrates the importance of the viscoelastic layer for damping a vibration in the medium-to-high frequency range. The floating floor structure is: Case 1: 6mm deck + 50mm MW + 3.2mm steel plate, Case 2: 6mm deck + 50mm MW + 1.6mm steel plate + 1mm viscoelastic layer + 1.6mm steel plate. By using a viscoelastic layer, the structure-borne noise is reduced by approximately 10-15 dB at medium-to-high frequencies. It provides also a more economically and technically effective solution to noise and vibration transmission problems.

4. Conclusions
In this paper, a model is developed of multi-layered floating floor to predict structure-borne noise in ship cabins. To verify the accuracy and validity of the developed model, the predicted results are compared with corresponding experimental results, which comprised measurements of a mock-up of a typical ship cabin structure. Various floating floor structures are studied, and the effects of variations of the thickness and viscoelastic layer. On the whole, the structure-borne noise can be controlled to some extent by varying any of the three prescribed parameters. Moreover, by using a viscoelastic layer, structure-borne noise is reduced by approximately 10-15 dB at medium-to-high frequencies. The developed model could be an effective tool for predicting structure-borne noise in ship cabins.

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References
[1] Joo W.-H, Kim S.-H, Kim D.-H, Bae J.-G. and Hong S.-Y. 2008 Quantitative Evaluation of Airborne Sound Insulation in Ship’s Accommodation using Large Scale Noise Test Facilities Noise Control Eng. J. 56(1) 45-51
[2] Cremer L 1952 Theorie des kolphschalles bei decken mit schwimmenden estrich *Acustica* **2**(4) 167-178
[3] Istvan L V 1971 Impact noise isolation of composite floors *Journal of the Acoustical Society of America* **50** 1043-1105
[4] Yago S, Inoue K 1983 An experimental study on the vibration characteristics of floor impact noise *Journal of the Acoustical Society of Japan* **332** 83-93
[5] Buiten J 1971 Acoustical investigations of asphaltic floating floors applied on a steel deck *Report 152S, TNO, Delft*
[6] Jensen J O 1975 *Noise in accommodation spaces onboard ships* The Technical University of Denmark
[7] McTavish D J, Hughes P C 1993 Modeling of Linear Viscoelastic Space Structures *J. Vib. Acoust.* **115**(1) 103-110
[8] Biot M A 1956 Theory of propagation of elastic waves in a fluid-saturated porous solid: I. low-frequency range *Journal of the Acoustical Society of America* **28** 168-178
[9] Biot M A 1956 Theory of propagation of elastic waves in a fluid-saturated porous solid: II. higher frequency range *Journal of the Acoustical Society of America* **28** 179-191
[10] Bolton J S, Shiau N M, Kang Y J 1996 Sound transmission through multi-panel structures lined with elastic porous materials *Journal of Sound and Vibration* **191** 317-347
[11] Rao Y V K S, Nakra B C 1974 Vibration of unsymmetrical sandwich beams and plates with viscoelastic cores *Journal of Sound and Vibration* **34**(3) 309-326
[12] Inman D J 1994 *Engineering vibration* Practice Hall Englewood Cliffs New Jersey
[13] Allard J F 1993 *Propagation of sound in porous media* Elsevier Applied Science London and New York