Sound Insulation Analysis of Automotive Soundproof Material using Urethane and Rubber

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Abstract. Enhancement is necessary in the acoustic insulation performance with respect to the high-frequency interior noise of automobiles. In this study, to improve the sound insulation in the overall acoustic insulation of a rubber sheet laminated to urethane, we used a waffle shape on the urethane surface and then examined the resulting acoustic change. We performed the test using a simple sound insulation performance measurement device and compared its results with the vibration sound analysis results obtained using a finite element model. We used the Biot model for the urethane using individually calculated parameters. In addition, with respect to the sound and vibration inputs, we calculated the sound performance changes resulting from differences in the waffle shape (rib width, rib interval). As a result, in a certain frequency band, we confirmed that the sound insulation performance had improved.

1. Introduction

In today’s automobiles, more than ten kilograms of sound-absorbing and insulation materials are typically used in each car. These materials play an important role in preventing noise from entering the car, including road noise, engine sounds upon acceleration, and wind noise during high-speed operation. In hybrid and electric vehicles, new noises, such as motor noises, have been introduced, which will require further attention to ensure in-vehicle quietness and comfort. However, to lower the price of cars and improve fuel economy, increasing the number of sound-absorbing and insulation materials is not necessarily the best strategy. Rather, we must improve soundproofing effects while reducing the weight and optimally arranging the sound insulation material.

In this study, we used a finite element model (FEM) to model sound-absorbing and insulation material and, using Biot theory and the finite element method, we then analyzed the acoustic characteristics of the sound-absorbing and insulation material of a laminated structure with various shapes. Specifically, we studied a sound-insulation material in which rubber was laminated onto urethane, and in which we changed the urethane its conventional flat shape to a waffle shape with ribs at regular intervals (Fig. 1). With respect to the sound insulation performance, we analyzed the resulting acoustic vibration and vibration excitation.
2. Analysis models and analysis results

In this study, we performed our calculations using the Biot-Allard model to handle the frame of the porous body and the internal air. The Biot model is a theoretical formula for predicting the air-propagation sound transmitted by incident sound coming into the material and passing through the gap between the porous elastic bodies in the material and the resulting displacement of the solid inside the material. We used the displacement $u^i$ of the frame and the displacement $u^f$ of the fluid to consider the interaction between the solid displacement and the air-propagation sound, which can be expressed as shown in Equations (1) and (2):[1][2]

$$
((1 - \phi) \rho_s + \rho_a) \frac{\partial^2 u^i}{\partial t^2} - \rho_a \frac{\partial^2 u^f}{\partial t^2} = (P - N)\mathbf{\nabla} (\mathbf{\nabla} \cdot u^i) + Q \mathbf{\nabla} (\mathbf{\nabla} \cdot u^f) + N \mathbf{\nabla}^2 u^i - \sigma \phi^2 G(\omega) \frac{\partial}{\partial t} (u^i - u^f)
$$

(1)

$$
(\phi \rho_f + \rho_a) \frac{\partial^2 u^f}{\partial t^2} - \rho_a \frac{\partial^2 u^i}{\partial t^2} = R \mathbf{\nabla} (\mathbf{\nabla} \cdot u^f) + Q \mathbf{\nabla} (\mathbf{\nabla} \cdot u^i) + \sigma \phi^2 G(\omega) \frac{\partial}{\partial t} (u^i - u^f)
$$

(2)

$\phi$: porosity, $\rho_s$: density of porous frame, $\rho_f$: air density, $\rho_a$: equivalent density of the fluid with respect to viscous damping in the interaction between the porous frame and the fluid.

$\rho_a$ is expressed as shown in Equation (3):

$$
\rho_a = \alpha_o \rho_f \left[ 1 + \frac{\phi \sigma}{j \omega \rho_f \alpha_o} G(\omega) \right] \left[ 1 + j \frac{4 a_o \eta \rho_f \alpha_o}{\sigma^2 \Lambda^2 \phi^2} \right]^{1/2}
$$

(3)

$\eta$: solid loss factor, $\sigma$: flow resistivity, $\alpha_o$: tortuosity, $\Lambda$: characteristic length of the viscosity. The elastic modulus for A, B, and C is shown in Equation (4):
$P = \left(1 - \phi \right) \left(1 - \phi - \frac{K_b}{K_s} \right) K_s + \frac{\phi K_s K_f}{K_f} \left(1 - \phi - \frac{K_b}{K_s} + \phi \frac{K_s}{K_f} \right) + \frac{4}{3} N$ 

$Q = \left(1 - \phi - \frac{K_b}{K_s} \right) \phi K_s \left(1 - \phi - \frac{K_b}{K_s} + \phi \frac{K_s}{K_f} \right)$ 

$R = \frac{\phi^2 K_s}{1 - \phi - \frac{K_b}{K_s} + \phi \frac{K_s}{K_f}}$

$K_s$: bulk modulus of the porous frame (in vacuum), $K_b$: bulk modulus of the frame material, and $N$: shear modulus of the porous frame (in vacuum). Equation (5) for A and Equation (6) for B are as follows:

$N = \frac{E(1 + j\eta)}{2(1 + \nu)}$ 

$K_b = \frac{2(1 + \nu)}{3(1 - 2\nu)} N$ 

$K_f = \frac{\gamma \rho_0}{\gamma - (\gamma - 1) \left[1 + \frac{8\zeta}{j\omega \lambda^2} H(\omega)\right]^{-1}}$ 

$H(\omega) = \left(1 + \frac{j\omega \lambda^2}{16\zeta}\right)^{1/2}$

$\gamma$: specific heat ratio, $P_0$: atmospheric pressure, $\zeta$: thermal diffusivity, $K_f$: equivalent bulk modulus of fluid considering thermal attenuation in interaction between porous frame and fluid, and $\lambda^*$: characteristic heat length.

3. Model and analysis results

With respect to the laminated absorbing and insulating material of the FEM, we analyzed four acoustic input models and three vibration input models. In this section, we first introduce the acoustic and vibration models. Then, we describe the physical properties of the rubber, urethane, air, and iron plate and the acoustic incidence and vibration input conditions used in our analysis.

3.1 Sound input model

The sound-absorbing and insulation material we used in the analysis is a laminate of urethane and rubber on a steel panel. We modeled a waffle shaped rib with widths of 60 [mm] (Fig. 1) and 20 [mm] (Fig. 2). We then prepared one with a layer with air (Fig. 3) and another with ribless material...
(Fig. 4), which is used in conventional automobiles. We used the following combinations of stacking arrangements:

Rib width 60 [mm] (Fig. 2) : rubber (thickness 2 [mm]) → urethane with one side waffle shaped (rib thickness 2 [mm], including the whole rib 20 [mm]) → air layer (thickness 5 [mm]) in the gap of the rib → steel panel (thickness 0.8 [mm])

Rib width 20 [mm] (Fig. 3) : rubber (thickness 2 [mm]) → urethane shaped (rib thickness 5 [mm], including ribs as a whole 20 [mm]) → air layer (thickness 5 [mm]) in the gap of rib → steel panel (thickness 0.8 [mm])

With air layer (Fig. 4) : rubber (thickness 2 [mm]) → urethane (thickness 15 [mm]) → air (thickness 5 [mm]) → steel panel (thickness 0.8 [mm])

No ribs (Fig. 5) : rubber (thickness 2 [mm]) → urethane (thickness 20 [mm]) → steel panel (thickness 0.8 [mm])

3.2 Vibration input model

The models for the vibration inputs are as follows:

Rib width 4 [mm] (Fig. 6) : from above rubber from the top (thickness 1.6 [mm]) → urethane with waffle shape on one side (rib thickness 2 [mm], including ribs as a whole 20 [mm]) → air layer (thickness 2 [mm]) → steel panel (thickness 0.8 [mm]) in the rib gap
With air layer (Fig. 7): from above rubber (thickness 1.6 [mm]) → urethane (thickness 18 [mm]) → air layer (thickness 2 [mm]) → steel panel (thickness 0.8 mm)

No rib (Fig. 8): from above rubber (thickness 1.6 [mm]) → urethane (thickness 20 [mm]) → steel panel (thickness 0.8 [mm])

3.3 Analysis conditions
We conducted the acoustic incidence experiment using the simplified sound insulation measurement device shown in Fig. 9. We performed the vibration input experiment in an APAMAT testing machine.

The sound incidence and vibration input conditions were as follows. We analyzed all models at frequencies from 100 [Hz] to 7100 [Hz] and compared the results at 1/3 octave band intervals. We analyzed the set model under the following two conditions.

Sound incidence (random incidence): We generated sounds that randomly entered the whole frequency range from the lower surface of the steel panel and then analyzed the sound emitted from the rubber surface. Thereafter, to obtain the insertion loss, we subtracted the sound pressure level of the steel plate from the sound pressure level of the whole model. The observation point in the model is 500 mm above the center of the lower surface of the steel panel.

Vibration input (vertical input): We calculated the sound pressure level coming from the rubber surface when subjecting the entire lower surface of the steel plate to vertical vibration at all frequencies with 1 [N]. The observation point of this model is also 500 mm above the center of the lower surface of the steel panel.
3.4 Sound input analysis results (random incidence / free boundary condition)

Here, we provide the calculation results (insertion loss) for the sound incidence. Figure 10 shows the analysis conditions. The graph of Fig. 11 shows the actual experimental results, which indicate better sound insulation performance as the frequency increases. The calculation results of the graphs with rib widths of 60 [mm] and 20 [mm] are between that for the presence of an air layer and that with no ribs, and similar tendencies are shown in the calculated and experimental results. Based on these results, we consider the calculation results for both rib widths (60 [mm] and 20 [mm]) to be valid. The insertion loss for the shape of urethane between about 250 [Hz] and 2000 [Hz] was above that for the material without ribs, which shows that the sound insulation performance increased. However, this result differed greatly from the experimental result in the vicinity of 250 [Hz] and 1000 [Hz] to 2000 [Hz]. The reason for this may be the lack of a boundary condition, the fact that the urethane and steel panel are consecutive in our analysis model, or differences in the physical property values.

In the calculation results (Fig. 10), the insertion loss in the resonance frequency range (about 250 [Hz]) decreased, but at rib widths of 20 [mm], 60 [mm], and no ribs, the rubber and steel panel vibration values are large. Since the rubber and steel panel were in opposite phases (approaching and separating), the urethane acted as a spring, so we believe that the vibration of the rubber and steel plate was promoted. Therefore, since the movement of the rubber against the steel panel increased, we consider the insertion loss to have decreased.
As the frequency becomes higher than that of the resonance frequency, the movement of the rubber gradually decreases, and when the frequency reaches 2000 [Hz] or more, the rubber hardly moves at all. Also, the motion of the steel panel becomes small and finer.

### 3.5 Sound input analysis results (random incidence / fixed boundary condition)

To improve the accuracy of our analysis, we performed our calculations using the model boundary condition as a complete constraint. Figure 10 shows the analysis results when the constraint condition is applied to the periphery of the model. Figure 11 shows a graph comparing the experimental results for a rib width of 60 [mm], a rib width of 60 [mm] without constraint, and a rib width of 60 [mm] with restraint (rubber, urethane, steel panel). We found that when the restraint condition (rubber, urethane, steel panel) is applied, the results approached those of the experiment better than did those for no restraint. However, in the actual experimental apparatus, the periphery of the rubber surface is pressed by a steel panel and is not completely restrained. As such, it seems that at about 250 [Hz] to 300 [Hz] and about 1000 [Hz] to 2000 [Hz], different levels were obtained than those in the experiment.

With respect to the displacement distribution (Fig. 12) for a rib width of 60 [mm] with a boundary condition at a resonance frequency of about 300 [Hz], we observed a local displacement that did not occur for the rib width of 60 [mm] without the constraint condition. Although the local displacement increased, the rubber and the steel panel vibrated in opposite phases overall, and we consider that the insertion loss decreased due to the large amplitude of the rubber with respect to the steel panel.

![Fig. 12. Calculation results of insertion loss (fixed boundary condition)](image1)

![Fig. 13. Comparison of Calculation and Experiment (Rib width : 60mm)](image2)

![Fig. 14. Displacements of rubber and panel for rib width 60 mm](image3)
3.6 Analysis results (vibration input)

Figure 15 shows the calculation result (sound pressure level) for the vibration input described in A. The rib width of 4 [mm] is hardly improved at all because of the drop in the sound pressure level due to the resonance frequency shifts to a frequency lower than that without ribs. However, we found that a rib width of 4 [mm] in the middle frequency range from around the resonance frequency to about 2000 [Hz] increases the sound insulation performance more than that without ribs.

The resonance frequency of the rib width of 4 [mm] was around 220 [Hz], but at this time the displacement of the model without ribs and that with a rib width of 4 [mm] plays the role of the urethane and the steel panel and rubber are in reversed vibration phases (Fig. 16). We consider that the sound pressure level decreased in the resonance frequency range because of the large amplitude of the rubber with respect to the steel panel. As the frequency approaches 6300 [Hz], the amplitude of the rubber decreases.

![Fig. 15. Calculation results of sound pressure](image1)

![Fig.16. Displacements of rubber and panel for rib width 4 mm](image2)

4. Summary

In this paper, we analyzed the two insertion loss conditions when sound was input from under the steel panel and the sound pressure when vibration was applied to the steel plate over the entire surface with 1.0 [N]. We obtained the following results:

· Both the sound incidence and vibration input improved the insertion loss and sound pressure level performances due to the waffle shape, except at low and high frequencies.

· With respect to the acoustic input, the calculation results for rib widths of 20 mm and 60 mm were between those for no rib and the presence of an air layer.

· We consider that if the waffle shape is changed, the insertion loss also changes for the calculation results of no rib and the presence of an air layer.

· We found that the calculated acoustic incidence approaches that of the experimental result when a constraint condition is applied. However, since the experiment is not perfectly constrained, this must be taken into consideration when constructing a model.
In future work, to further improve analysis accuracy, we will examine the restraint conditions of the sound-absorbing and insulation material. Also, to improve the sound insulation performance, we will consider a new shape and structure.

References
