Sound insulation and radiation sound analysis for car floor carpet using FE method

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Abstract. The performance enhancement of the acoustic insulation is demanded for the high frequency for interior noise of automobile. A carpet having urethane adhered to a rubber skin is generally laminated on a floor panel of an automobile. In order to elucidate noise and vibration for this structure, the vibration transmission rate was measured using a test apparatus in which a carpet was laminated on a panel simulating the floor of an automobile. In addition, we modeled this experimental apparatus with finite elements (FE) and analyzed the vibration during displacement excitation. Biot-Allard model was used for felt and urethane. We calculate using separately identified Biot parameters and compare with experimental results and introduce the change of vibration and obtained findings when material is changed. In the low frequency range, the values changed significantly when the skin density, Young's modulus and loss factor of urethane were changed.

1. Introduction

In recent years, in-vehicle comfort has been emphasized as the performance of automobiles, and further quietness in the in-vehicle is required. The sound that the occupant hears while driving inside the vehicle is called the interior noise, and one of them is the phenomenon called road noise. This is because the tires are vibrated mainly by the unevenness of the road surface, and the vibration is transmitted to the wheels → suspension → body frame → body panel, and the sound radiated into the passenger compartment reaches the ears of the occupants. It occurs in a wide range of frequencies, but in any frequency range, it is jarring to the occupants and impairs the comfort inside the vehicle, so countermeasures are desired. The purpose of this study is to analyze the factors that can improve the sound insulation performance of automotive floor carpets using the calculation of FE models. The floor panel of an automobile contributes greatly to high-frequency road noise (200Hz to 500Hz), and vibration control and soundproofing measures are strictly taken. It is also important to take measures to insulate high-frequency (~ 8000Hz) tire pattern noise, engine noise, and wind noise. The floor panel is made by press-molding a steel plate into a required shape, and a damping material made of a viscoelastic body is laminated for the purpose of suppressing the vibration level. In addition, a carpet with urethane or felt bonded to the rubber skin is laminated. Although the authors have established an analysis method for predicting the vibration of the carpet skin of automobile panels on which damping materials are laminated [1], the damping materials are often reduced to reduce weight. Yamamoto et al. the sound field is simplified to the body sound field, and the sound and vibration effects of the laminated soundproofing material, which includes damping material and sound absorbing and insulating material provided at the boundary surface, are efficiently improved [2]. Although they considered methods for evaluating the transmission loss, they were unable to calculate the transmission loss. In this study, the vibration transmissibility was measured using a test device in which a carpet was laminated on a panel that imitated the floor of an automobile. We will introduce the results of measuring the vibration transmission of the soundproof material with a device that can vibrate the entire panel up and down.

In addition, this experimental device was modeled with FE, and the vibration and sound during displacement excitation were analyzed. In addition, the transmission loss was also calculated. For urethane, the Biot-Allard model was used [3][4]. Calculations are made using the Biot parameters identified separately, and comparisons with experimental results, changes in vibration when the panel thickness is changed, and the findings obtained are introduced.

2. Analytical method

This time, the calculation was performed using the Biot-Allard model, which handles the skeleton of the porous body and the internal air. The Biot-Allard model shows the airborne sound transmitted by the incident sound entering the material passing through the gaps of the porous elastic body (urethane, glass wool, etc.) in the material, and the displacement of the solid propagation transmitted inside the material. It is a theoretical formula to predict. The displacement of the skeleton considering the interaction between the solid propagation and the air propagation sound: $\vec{u}^s$ and the displacement of the fluid: $\vec{u}^f$ are expressed as equations (1) and (2), respectively.

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

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

ϕ : Porousness , ρ_s : Porous skeleton density , ρ_f : Fluid density (air in this paper) , ρ_a : Equivalent density of fluid considering viscous decay in the interaction between skeleton and fluid. ρ_a is shown in Eq. (3).

$$\rho_a=\alpha_\infty\rho_f\left(1+\frac{\phi\sigma}{j\omega\rho_f\alpha_\infty}\sqrt{1+j\frac{4a_\infty^2\eta\rho_f\omega}{\sigma^2\Lambda^2\phi^2}}\right) \tag{3}$$

η : Solid loss coefficient, σ : Flow resistance, α_∞ : Maze degree, and Λ : Viscous characteristic length. The elastic modulus P, Q, R is shown in Eq. (4).

$$P\approx\frac{4}{3}N+K_b+\frac{(1-\phi)^2}{\phi}K_f,\quad Q\approx(1-\phi)K_f,\quad R\approx\phi K_f \tag{4}$$

N : Skeleton shear modulus (in vacuum), K_b : Skeleton bulk modulus (in vacuum), and K_f : Equivalent rigidity of fluid considering thermal decay in the interaction between skeleton and fluid. N, K_b, K_f are shown in Eq. (5).

$$N = \frac{E(1+j\eta)}{2(1+\nu)}, \qquad K_b = \frac{2(1+\nu)}{3(1-2\nu)}N, \quad K_f = \frac{\gamma P_0}{\gamma-(\gamma-1)\left[1+\frac{8\zeta}{j\omega\Lambda'^2}\sqrt{1+\frac{j\omega\Lambda'^2}{16\zeta}}\right]^{-1}} \tag{5}$$

γ : Specific heat ratio, P_0 : Equilibrium pressure, ζ : Thermal diffusivity, and Λ' : Thermal characteristic length.

3. Analysis result

3.1 Experimental equipment and FE model

Figure 1 shows the test equipment used this time. It is a device that assumes the floor of an automobile, stacks a soundproofing material (assuming a floor carpet) with a porous body (urethane) and a rubber skin bonded to the panel, and vibrates up and down with the vibrator at the bottom. The measurement frequency range this time was set to 0Hz to 500Hz, assuming high-frequency road noise of automobiles. It has been confirmed that the resonance of the frame part of this measuring device is 500 Hz or higher. The panel and soundproofing material are not bonded. The outer shape of the panel is 600 mm×500 mm, and it is fixed with a large number of bolts from above and below with a frame. The internal shape of the frame is about 500mm×400mm. In this experiment, the panel thickness was 1.6 mm, the porous body thickness was 10 mm, and the rubber skin thickness was 2.8mm (5.0kg/m^2). Accelerometers were used to measure acceleration at points Acc0 to Acc2 in the figure.

Figure 2 shows the FE model used in the calculation of the radiated sound. The displacement of the lower surface of the frame where the exciter of the experimental device is connected was input. Since the panel and the soundproofing material are not bonded, the contact was modeled with a spring.

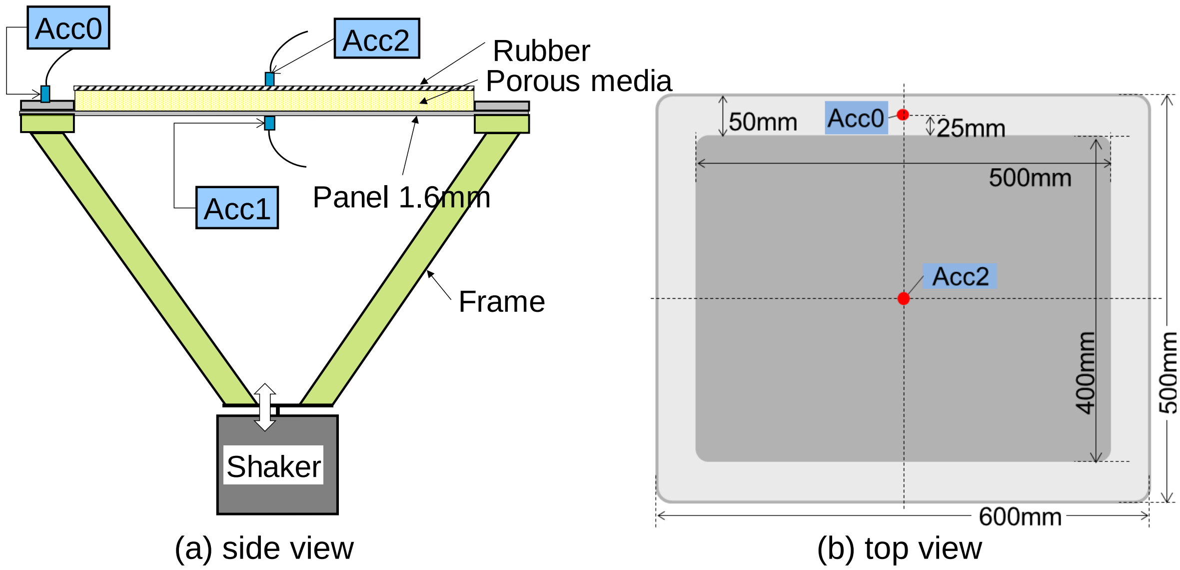


Fig. 1. Experimental setup. A device in which a panel and soundproofing material are set in a frame installed at the top of the frame, and the entire frame is vibrated up and down using an exciter at the bottom.

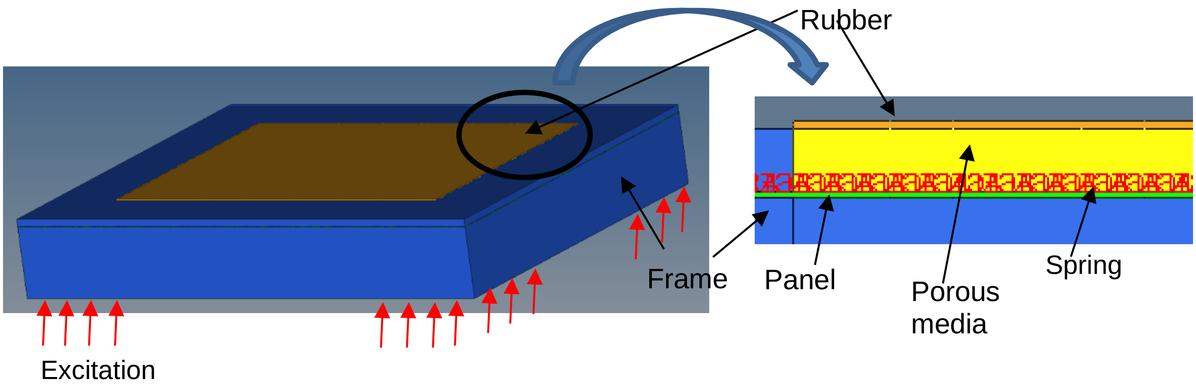


Fig. 2. FE model of the frame, panel, and soundproofing material in Fig. 1.

3.2 Experimental results and calculation results

Figure 3 shows a comparison of the experimental results and calculation results of the transfer function with the soundproofing material laminated. The graph on the left is the acceleration of the center of the rubber on the top surface divided by the acceleration of the bottom surface of the center of the panel (Acc2 / Acc1), and the graph on the right is the acceleration of the center of the rubber on the top surface divided by the acceleration of the frame (Acc2 / Acc0). By tuning the contact springs, the experimental results and the calculated results could be almost matched.

Next, the radiated sound was calculated. Figure 4 shows a comparison of the calculation results when the Young's modulus of urethane is halved (green line), doubled (blue line) and initials (red line). It was confirmed that the radiated sound became smaller when the Young's modulus was reduced at 250 Hz or higher. It was confirmed that this corresponds to the softening of the spring between them in the double wall structure, and the result is as theoretical. The radiation sound was calculated by changing other Biot parameters of urethane, and it was found that Young's modulus had the greatest effect.

Next, the radiated sound when the Young's modulus of rubber was changed was calculated. Figure 5 shows a comparison of the calculation results when the Young's modulus of rubber is increased 10 times (green line), when it is made considerably smaller than PE (polyethylene) (blue line) and when the initials (red line) are reduced. It was confirmed that when the Young's modulus was increased, the peak value of 60Hz, which is an out-of-plane resonance, increased, and the resonance frequency of 160Hz increased. When the Young's modulus was reduced, almost no change in the radiated sound was observed. This is thought to be due to the balance between the thickness of the rubber and the Young's modulus and hardness of the urethane.

Figure 6 shows a comparison between the calculation results of the insertion loss and the experimental results when the value of the spring constant of the spring between the urethane and the panel used in the FE modeling is changed. In the figure, the orange line is the experimental result (every 1/3 octave band), the green line is the calculation result with a 0.1mm air layer between the urethane and the panel, and the blue line is the spring constant tuned by the vibration acceleration response this time. This is the calculation result when the yellow line is shared (continuous) between the urethane and the panel. It was confirmed that the insertion loss decreases as the spring constant increases. Since the panel and the soundproofing material are not adhered this time, the calculation result of node sharing does not match the experimental result. In addition, when calculating with the transmission matrix method, it was common to model with an air layer of about 0.1mm when not bonded, but in FEM it can be said that the calculation accuracy is higher when modeling with a spring.

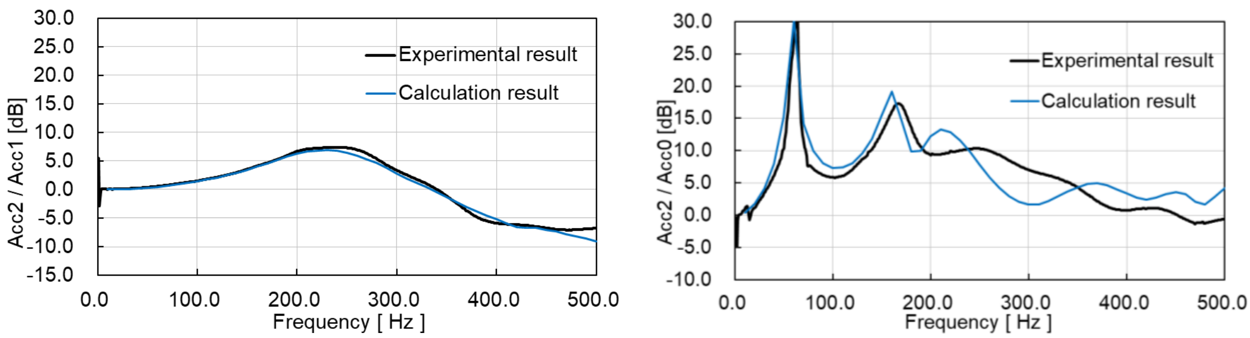
It is considered that the set value of the spring constant this time is appropriate from the comparison result of the vibration experiment and the calculation (see Figure 3). Figure 7 shows a comparison of the experimental and calculated results of insertion loss when the urethane thickness is changed to 20mm. In the figure, the orange line is the experimental result (for each 1/3 octave band), the blue line is the calculation result using the spring constant tuned by the vibration acceleration response this time, and the gray line is the calculation result when the spring constant is changed to 1.5 times. is there. At 4000Hz and above, multiplying the spring constant by 1.5 times was closer to the experimental results, but when compared over the entire frequency range, it was judged that there was no problem with this tuning value.

Figure 8 shows a comparison of transmission loss when the Young's modulus of urethane from 0Hz to 500Hz is reduced to 1/10 (orange line), when it is multiplied by 10 (yellow line), and when the initial result (gray line) is used. It was confirmed that the resonance transmission frequency (valley of transmission loss: about 300Hz in the initial) decreases when the Young's modulus is changed, and decreases (larger) when the Young's modulus is decreased (increased). The radiation sound and transmission loss were calculated by changing other Biot parameters of urethane, and it was found that Young's modulus had the greatest effect.

Figure 9 shows the results of the same calculation performed for radiated sound (left figure) and transmission loss (right figure) at 500Hz to 8000Hz. There are many peaks in acoustic radiation, but basically the order and difference in magnitude was the same as the transmission loss (the graph is upside down because of the radiation sound and transmission loss). In the high frequency range, when Young's modulus was increased, a valley of transmission loss occurred at about 2500Hz at the initial due to resonance in the thickness direction of urethane. Hereafter, it is called the second valley of transmission loss. It did not change much at Young's modulus 1/10, but the frequency changed significantly at Young's modulus 10 times. This is thought to be due to the balance between the thickness of the rubber skin and the Young's modulus of urethane. The details of the mechanism will be discussed in the future.

Figure 10 shows the calculation results of transmission loss when the porosity of urethane is changed. When the porosity of urethane is increased, the frequency of the second valley of transmission loss shifts to the higher side. At 0.999, the valley disappears. It was found that there was an inflection point between 0.95 and 0.999.

Figure 11 shows the calculation results of transmission loss when the urethane density is changed. It was found that when the bulk density of urethane was reduced, the frequency of the second valley of transmission loss moved to the higher side.



(a) Ratio of acceleration Acc2/Acc1 (b) Ratio of acceleration Acc2/Acc0

Fig. 3. Experimental results and calculation results for Ratio of acceleration. The experimental results and calculation results were roughly in agreement.

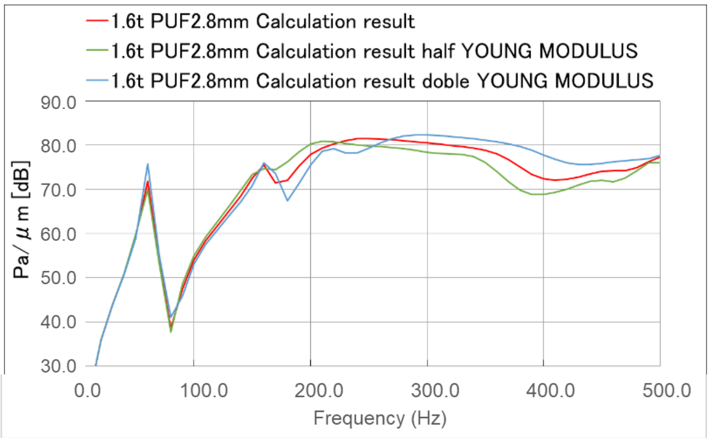


Fig. 4. Calculation results of radiation sound (changed Young modulus for PUF).

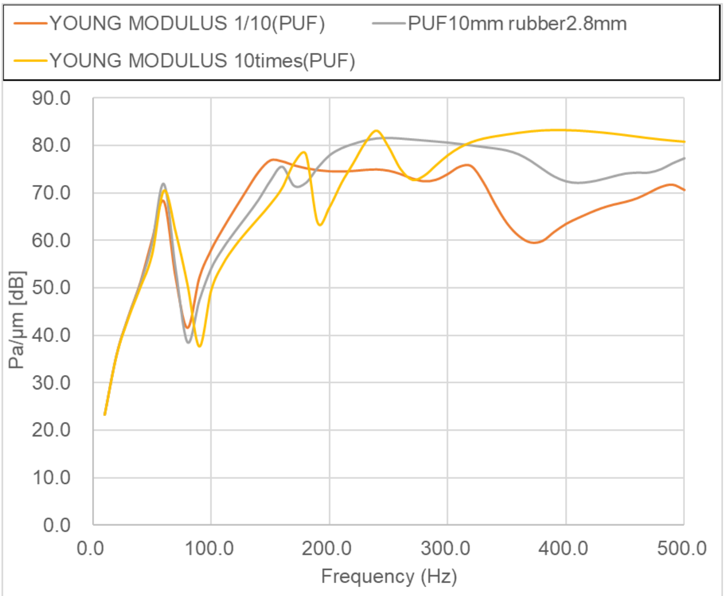


Fig. 5. Calculation results of radiation sound (changed Young modulus for rubber).

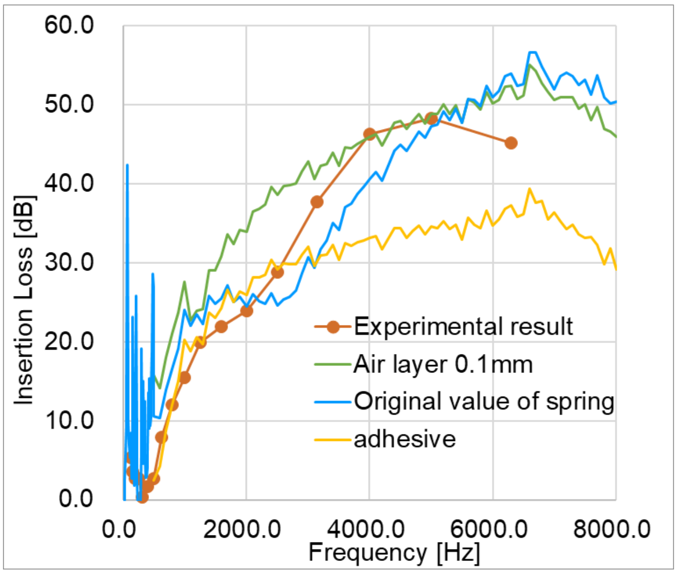


Fig. 6. Comparison between experimental results and calculation results of insertion loss (changed spring constant).

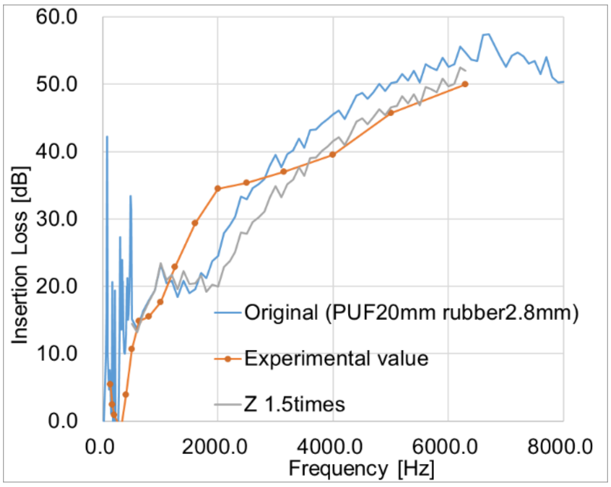


Fig. 7. Comparison between calculation results and experimental results of insertion loss.

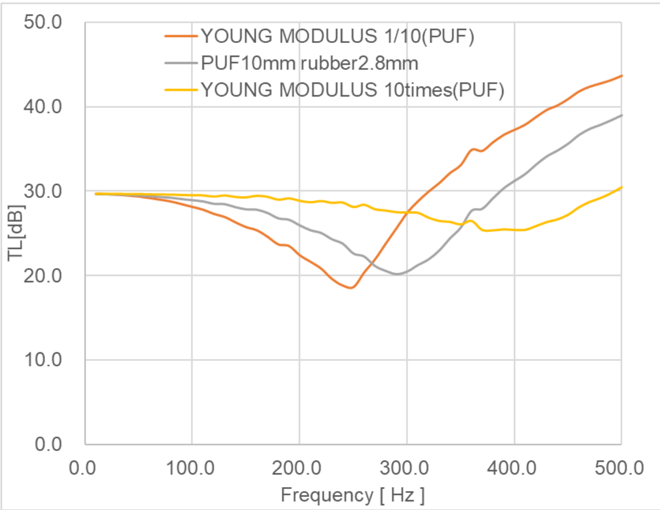
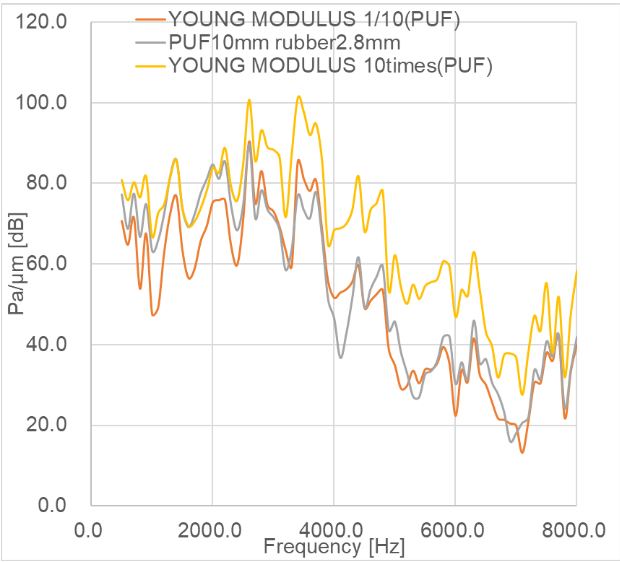
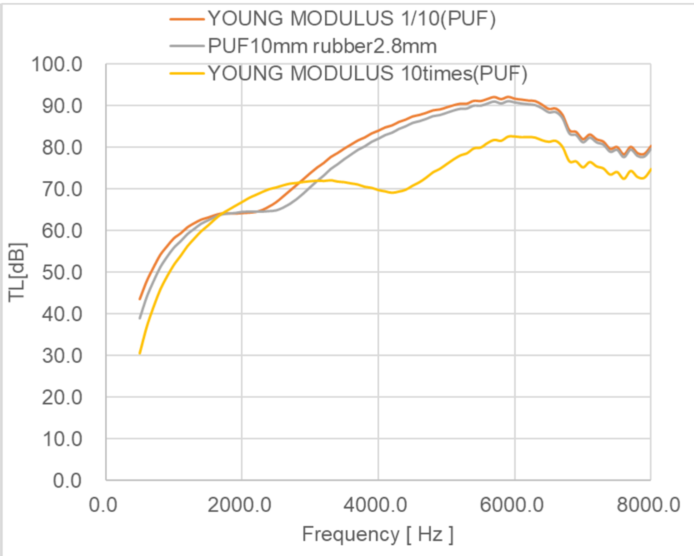


Fig. 8. Calculation results of transmission loss (changed Young modulus).



a. Radiation sound



b. Transmission loss

Fig. 9. Calculation results of radiation sound and transmission loss (changed Young modulus)

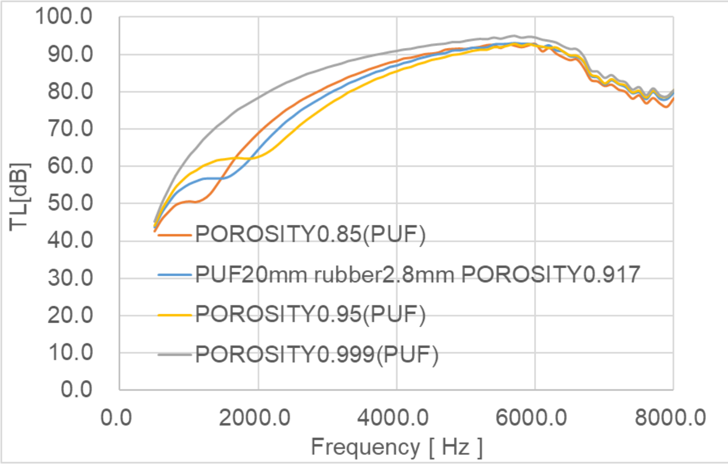


Fig. 10. Calculation results of transmission loss (changed porosity: 0.85~0.999).

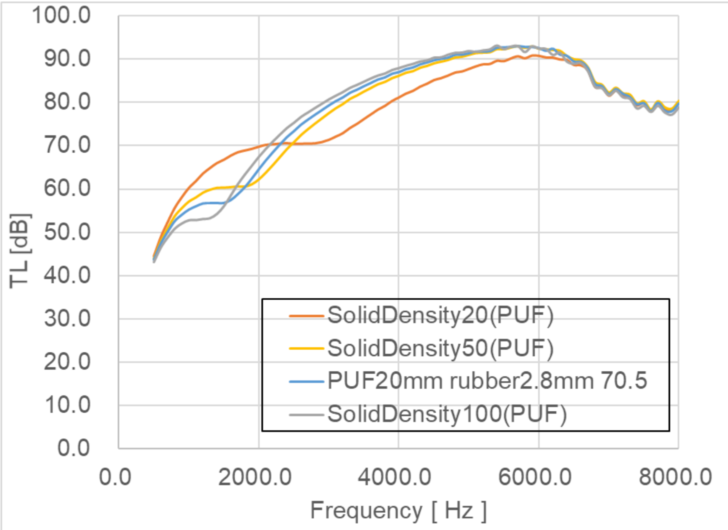


Fig. 11. Calculation results of transmission loss (changed density).

4. Conclusion

The vibration transmissibility was measured using a test device in which a carpet was laminated on a panel that was restrained all around with a frame that imitated the floor of an automobile. In addition, this experimental device was modeled with FE, and the vibration and radiated sound during displacement excitation were calculated. For urethane, the Biot-Allard model was used. Calculations were made using the Biot parameters identified separately, and the changes in radiated sound and transmission loss were calculated when compared with the experimental results and when the material (Biot parameter) of the double wall sound insulation structure was changed. As a result, the following conclusions were obtained.

- In the low frequency range, the values changed significantly when the skin density, Young's modulus of urethane, and loss coefficient were changed. In the high frequency range, the values changed significantly when the density / Young's modulus of the epidermis and the porosity / Young's modulus / density of urethane were changed.
- If the spring constant of the connection spring between the panel and urethane is reduced (floating image), the transmission loss increases. The larger the connecting spring, the smaller the transmission loss, which is the smallest value for bonding.

- It was found that the epidermis had no effect on the second valley of transmission loss. When the porosity of urethane is increased, the frequency of the valley moves to the higher side, and when it reaches 0.999, the valley disappears. There is an inflection point between 0.95 and 0.999.
- For the second valley of transmission loss, if the Young's modulus of urethane is increased, the frequency of the valley moves to the higher side. The smaller the Young's modulus, the larger the transmission loss.
- When the bulk density of urethane is reduced, the frequency of the second valley of transmission loss moves to the higher side.

From this study, it was found that it is important to reduce the Young's modulus of the urethane foam between them by using a double-wall structure in order to improve the sound insulation performance of automobile floor carpets.

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