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Sound and Vibration Analysis of the Acoustic Fields Involving Felt when the Panel is Inputed

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Abstract. In automobiles, noise may be generated in the passenger compartment, which is a closed space with a panel that is the outer wall and a breathable trim. We created a simple model based on these assumptions and performed experiments and vibration acoustic analysis. We placed a partition plate with a hole in an acrylic pipe that has an inner diameter of 100 mm and vibrated the iron plate (panel) placed on the bottom with hammer. Then we measured the vibration and sound pressure level when the hole shape of the partition plate was changed and felt was placed. In addition, we create a similar FE model and report the results of the analysis of the sound pressure level when the acoustic characteristics of felt placed in the holes of the partition are changed.

1. Introductions

In recent years, comfortability inside vehicle is emphasized as the performance of automobiles, and quietness in the car is required from the design concept stage. Particularly, noise around 1 to 5 kHz is classified as high-frequency noise of automobiles interior's sounds, and the main sound sources are the noise generated from the engine transmission during acceleration, tire pattern noise, and wind noise during high-speed driving. Although these noises are advanced measures on the sound source side, there is a limit and measures (sound insulation and sound absorption) on the vehicle body side are important when considering the efficiency of cost and weight. In addition, weight reduction of the vehicle is inevitable in order to comply with the fuel consumption regulations that will become more and more stringent in the future. The interior materials such as apron trim (trim vertical wall of the cargo compartment) and door trim (Fig. 1) were resins like polypropylene that is conventionally without gas-permeability, gas-permeable materials (hard felts, etc.) began to be used. Although the weight and strength and rigidity of these interior materials have been evaluated, the sound absorption and sound insulation performance have not been sufficiently evaluated. When noise enters the car, apron trim becomes a wall (or passage) and the contribution to the noise in the car is large, so the evaluation and prediction of sound absorption and sound insulation performance are important.

The space in the car interior and the back of the trim have a closed space, so standing waves occur while driving. When sound absorbing material (porous body) is placed in space, the acoustic energy of standing waves (acoustic mode) is attenuated. In the frequency range where the influence of acoustic mode is large, SEA (statistical energy analysis) has poor prediction accuracy, and analysis including sound transmission in closed space is not suitable in BEM (boundary element method). In order to clarify the characteristics of the sound field when sound absorbing material (attenuation element) is placed in space, numerical calculation [1-3] using finite element method is effective.

A simple model was created that simulated the car door panel, door space, trim and car compartment space, using gas-permeable trim as a sound absorbing material (porous body) that separates the vehicle compartment space and the panel. We placed a partition plate with a hole in an acrylic pipe that has an inner diameter of 100mm and vibrated the iron plate (panel) placed on the bottom with hammer. Then we measured the vibration and sound pressure level when the hole shape of the partition plate was changed and felt was placed. In addition, we create a similar FE model and report the results of the analysis of the sound pressure level when the acoustic characteristics of felt placed in the holes of the partition are changed.

Fig.1. Door trim for car.

2. Experimental test pieces and calculation models

Fig. 2 shows the test piece used in this experiment. Acrylic pipes with total length of 150 mm, inner diameter 100 mm and thickness 10 mm were used and created. A steel plate (panel) with a thickness of 1.0 mm was placed on one side of the pipe, and the entire circumference was fixed with eight hexagonal bolts with an iron ring with a thickness of 10 mm and a width of 10 mm. The tightening torque of the hex bolts was all 1.0 Nm. In addition, a partition plate with a thickness of 10 mm and a diameter of 100 mm was installed at a position of 40 mm from the panel. The small space on the panel side of the partition plate is Space1, and the space on the other side is Space2. A hole was drilled to the partition plate so that the felt could be attached. The hole shape is shown in Fig. 3. The partition plate has four holes, and two different types of hole size. The one with a small hole is Type1, and the large one is Type2. In both cases, the hole position starts at 18 mm away from the center, the hole shape 1 has a hole with a width of 8 mm and the hole shape 2 is 60 degrees open, respectively. The other side of the panel was covered with acrylic with a thickness of 20 mm and a 1/4 inch microphone was installed at the center. In addition, the partition plate assumes as the door trim, and the acrylic part expresses the trim of the conventional resin type, and the weld part expresses the trim which is gas-permeable. In addition, Space1 represents the space in the door panel to the door trim, and Space2 represents the passenger compartment space.

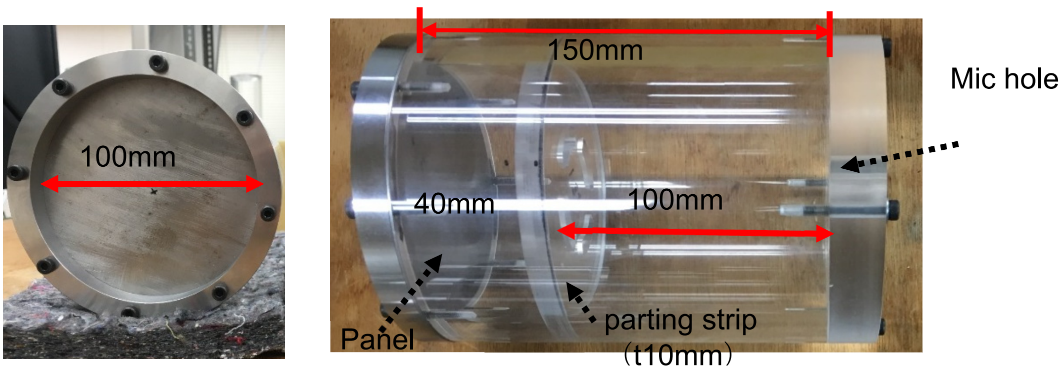


Fig. 2. Test piece.

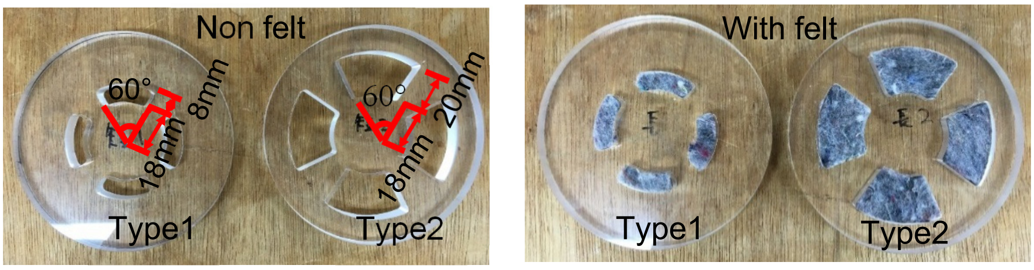


Fig.3. Parting strip.

The point away from the center of the iron plate is performed by the vibration input with an impulse hammer, and the vibration of the point which deviates 90 degrees from the vibration position 10 mm away from the center of the iron plate is measured by a laser Doppler vibration meter (VMS-100, product of NEOARK CO., Ltd.). A microphone was attached to a hole in the center of the acrylic plate on the opposite side of the panel, and the sound pressure was measured (see Fig. 4). For FFT analyzer, OR24, the product of OROS CO., Ltd, is used.

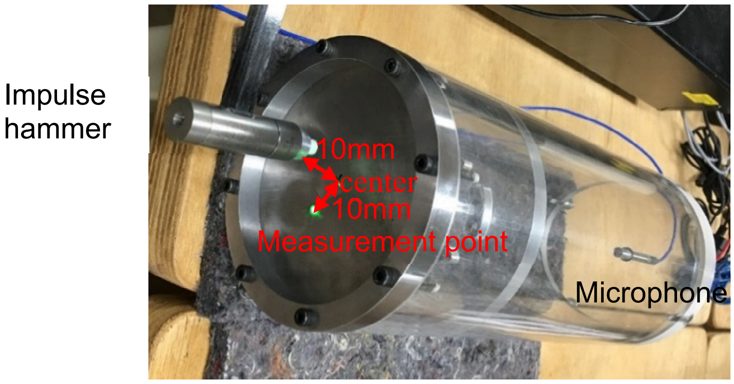


Fig. 4. Excited point and measurement point

FE model (see Fig. 6) was created the same as the experimental model. From left to right is the panel, air layer 1, hole or felt, and air layer 2, mesh size is 2 mm, and the number of elements is 117000 elements. A spring was installed on the circumference of the panel and the spring constant from the vibration measurement results was tuning. In this study, the acrylic part was assumed as a model that does not vibrate. As in the experiment, the sound pressure and panel amplitude when 1.0 N was applied to the vibration input position of the impulse hammer was calculated. Since the flow Resistivity was 42268 Ns/m^4, we used a Miki model [4] (modeled only felt internal air) that can calculate the sound absorption rate from only flow resistance. In the Miki model, characteristic impedance and propagation constants are expressed by experimental equations like in equation (1).

$$\begin{aligned} Z_c &= \rho_0 c_0 \left[1 + 0.070 \left(\frac{f}{R_f} \right)^{-0.632} \right] - j \frac{\omega}{c_0} \rho_0 c_0 \left[0.107 \left(\frac{f}{R_f} \right)^{-0.632} \right] \\ \gamma &= \frac{\omega}{c_0} \left[0.160 \left(\frac{f}{R_f} \right)^{-0.618} \right] + j \frac{\omega}{c_0} \left[1 + 0.109 \left(\frac{f}{R_f} \right)^{-0.618} \right] \end{aligned} \tag{1}$$

ρ_0: Air density 1.205 kg/m^3, c_0 : sound velocity 343.5 m/s, f: frequency Hz, R_f: flow resistance Ns/m^4, j: imaginary units, ω: angular frequency 1/s. Fig. 5 shows the measurement result of the vertical incident sound absorbing rate by the acoustic tube and the comparison of the calculation result of each model calculated from the flow resistance. It can be seen that the Miki model roughly matches.

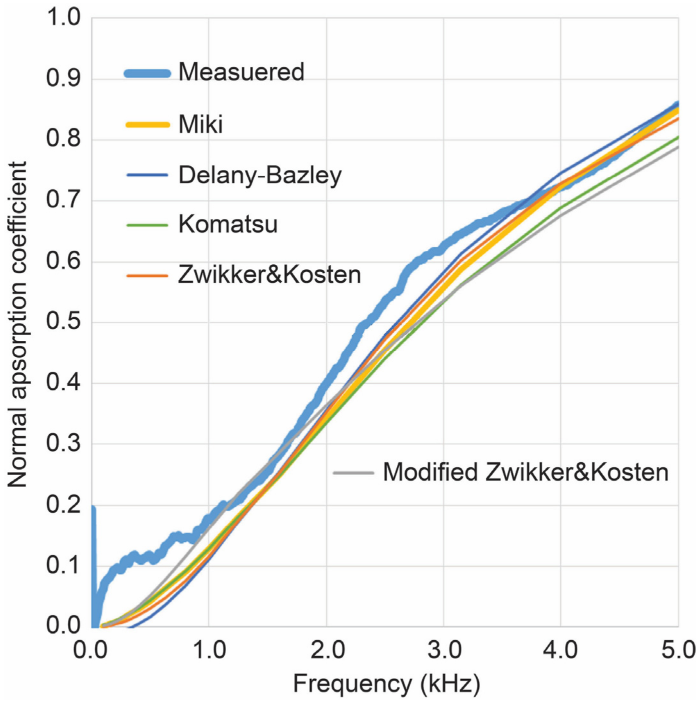


Fig. 5. Comparison between measurement results and calculation results.

The characteristic impedance and propagation constants obtained in Equation (1) were used to determine the density of felt internal air ρ kg/m^3 and stiffness K Pa, and a FE model as shown in Fig. 6 was created, and the vibration acoustic coupling problem was calculated.

$$\gamma = j\omega\sqrt{\frac{\rho}{K}} \qquad Z_c = \sqrt{\rho K} \tag{2}$$

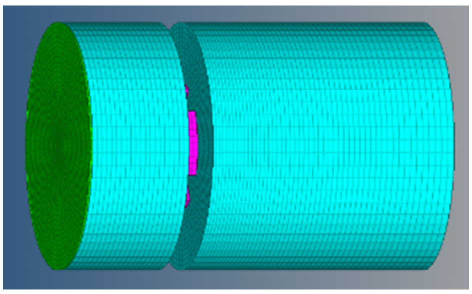


Fig. 6. FE model.

3. Experimental results and calculation results

3.1 Analysis results when felt is not attached to the partition plate

Fig. 7 shows the comparison of the experimental results and the calculation results of the amplitude when the hole of the partition plate does not attach to the felt for Type1. Four vibration modes appeared up to 5000 Hz. The other peaks are influenced by coupling. The resonant frequency of the 2nd to 4th mode was slightly higher, but the experimental results were roughly reproduced.

Fig. 8 shows the comparison results of sound pressure in the same case. The first peak is Helmholtz resonance. The second is the vibration mode, the third is the acoustic mode of Space2, and the next major peak is the acoustic mode of Space1. Since the 2nd and 4th vibration modes are symmetrical, no peak appeared in the calculation result of sound pressure, but it slightly appears in the experimental results. This is probably because the microphone position was slightly off-center. The level difference was observed around 3000 Hz, but other than that, the experimental results and the calculation results showed a good agreement.

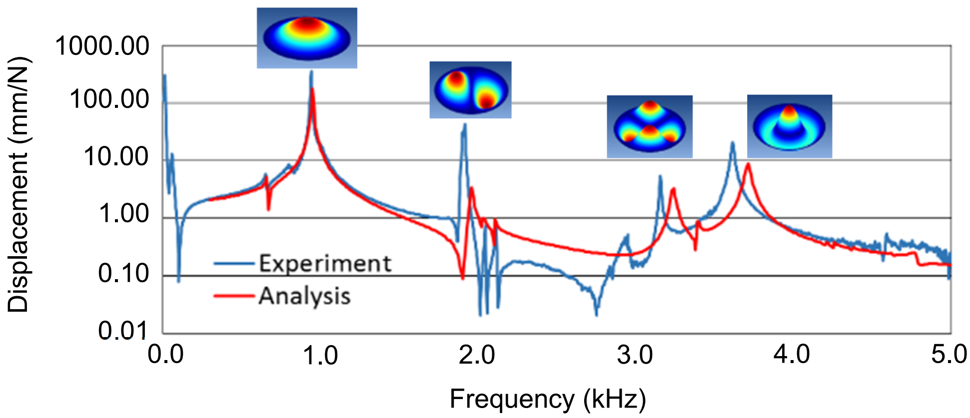


Fig. 7. Displacement for type1 (non felt).

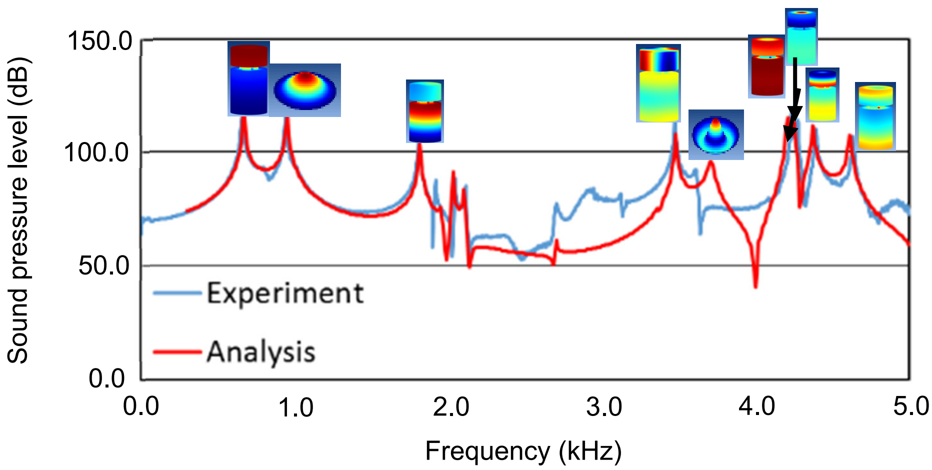


Fig. 8. Sound pressure for type1 (non felt).

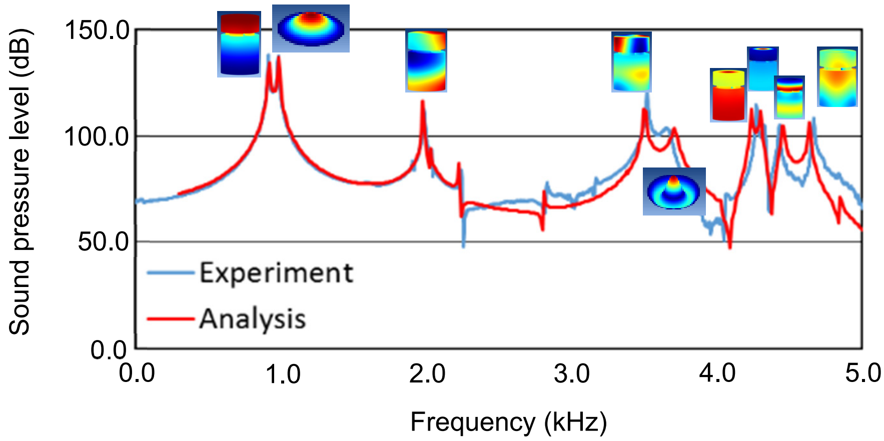


Fig. 9. Sound pressure for type2 (non felt).

Fig. 9 shows the comparison result of the sound pressure for Type 2 to the hole of the partition plate. Because the hole became large, it can be seen that the acoustic mode of Space2 and Space1 is not completely separated. The difference in the level of experiments and calculations is small around 3000 Hz, and the experimental results and calculation results show a good agreement for Type 2. This difference can also be thought of as a possibility of the influence of viscosity when passing through a thin hole. We would like to make it as a problem and will be considered in the future study.

3.2 Analysis results when felt is attached to the partition plate

In Fig. 10 shows the comparison of the experimental and calculated results of sound pressure when the hole in the partition plate is Type 1 and felt is attached. It can be seen that the Helmholtz resonance that appeared when there was non felt disappears when there is with felt. The peak by the vibration mode is small influence of the presence or absence of felt, but it can be seen that the peak by the acoustic mode is attenuated greatly by felt. It was also noticed that the acoustic mode itself did not change much. It can be observed that there is a level difference around 3000 Hz, but other than that, the experimental results and the calculation results showed a good coincidence. In Fig. 11 shows a graph comparing the non felt or with felt between experimental results. And in Fig. 12 shows a graph comparing the non felt or with felt between calculation results. It can be seen that the above results appear clearly. In addition, the calculation result catches the tendency of felt presence of the experimental result well. It could be confirmed that FE model created in this study, material data, and felt were appropriate.

In Fig. 13 shows the comparison of the experimental and calculated results of sound pressure when the hole in the partition plate is Type 2 and felt is attached. Since the amount of felt has increased, it can be seen that the peak by acoustic mode is greatly attenuated than Type1 (Fig. 9). As with Type1, the level difference can be seen around 3000 Hz, but besides that, the experimental results and the calculation results show a good match.

The amplitude of the panel was omitted in this paper because there was little influence with or without felt.

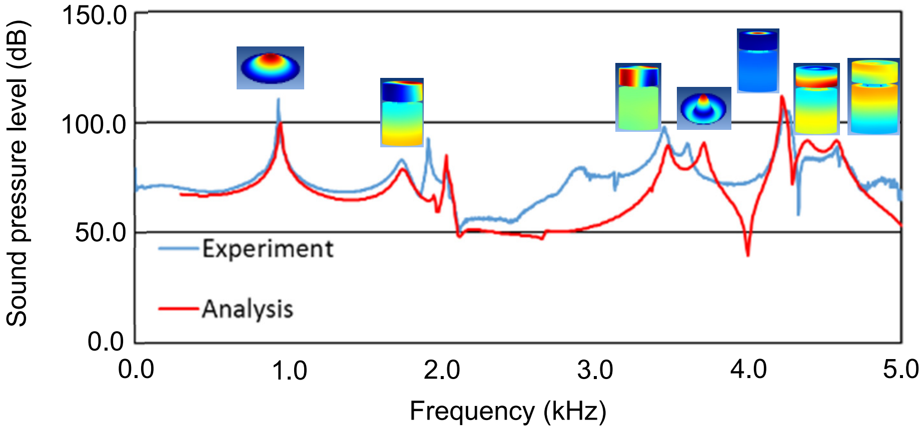


Fig. 10. Sound pressure for type1 (with felt).

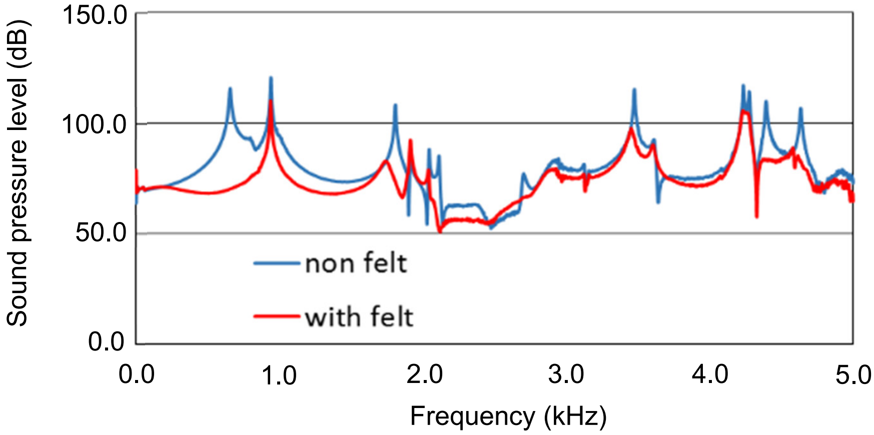


Fig. 11. Sound pressure for type1 (Experiments).

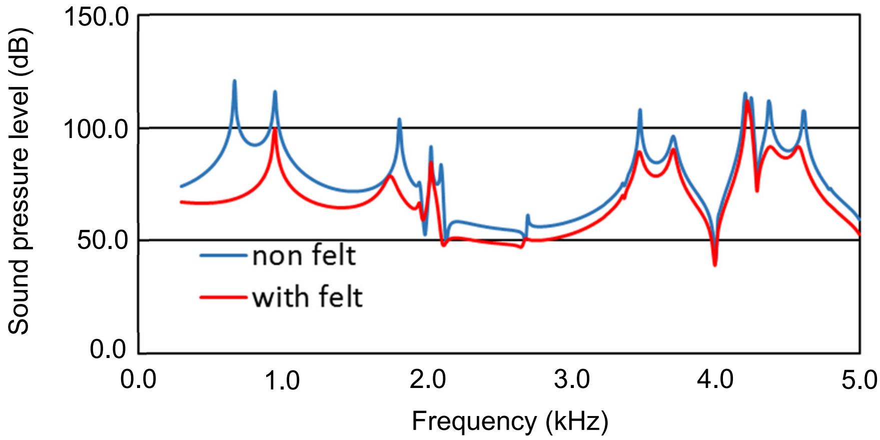


Fig.12. Sound pressure for type1 (Analysis).

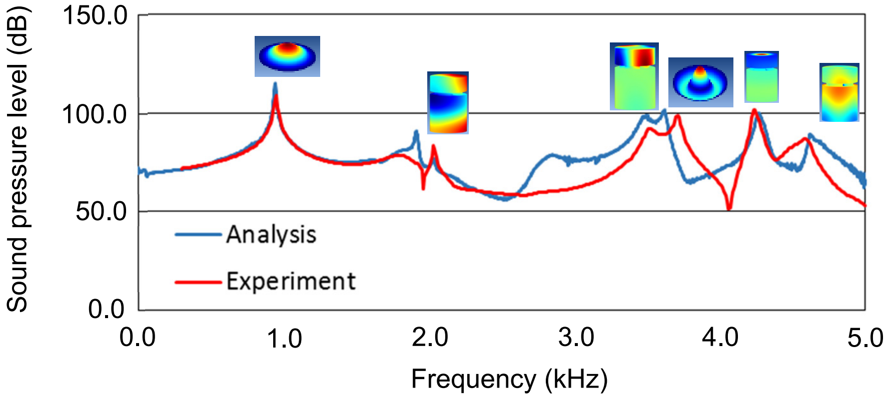
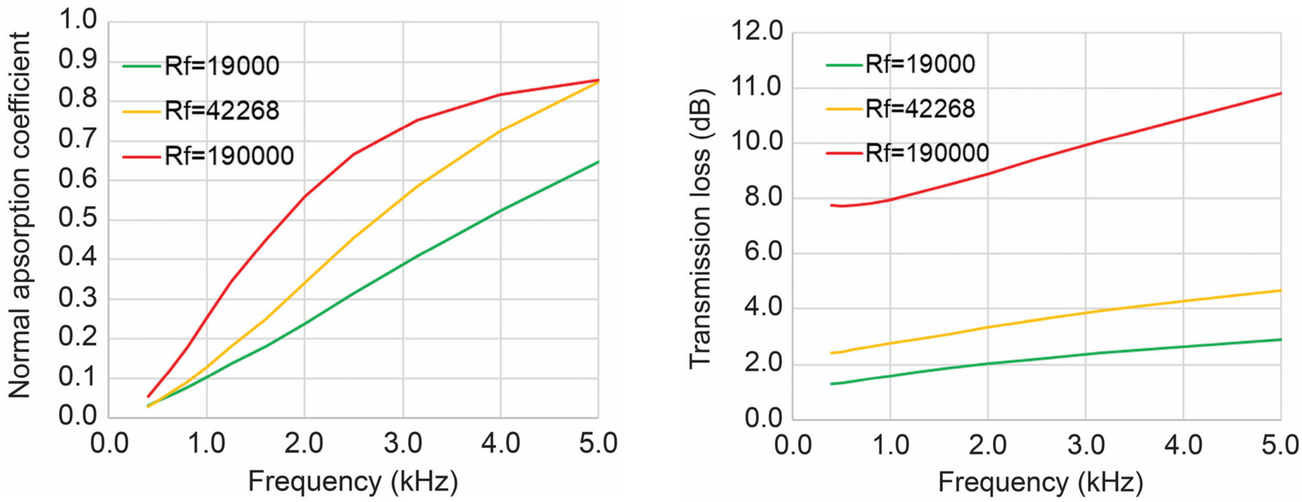


Fig.13. Sound pressure for type2 (with felt).

3.3 Changing felt flow resistance

In the actual door trim, what kind of trim is suitable for lowering the sound pressure at the time of panel vibration was examined.

In this study, when the flow resistance was about half (19000 Ns/m^4) and about 5 times (190000 Ns/m^4), the vertical incidence absorbance and transmission loss were calculated using the transmission matrix method [5]. In Fig. 14 shows the comparison result with the initials (42268 Ns/m^4). It can be seen that for the sound absorption rate, the initials show almost intermediate value, while the transmission loss (sound insulation performance), the performance is greatly improved to 190000 Ns/m^4 in the whole area. Using this flow resistance value, we calculated this FE model.



(a) Normal sound absorption coefficient (b) Sound transmission loss

Fig.14. Sound absorption and transmission loss (felt 10mm).

Fig. 15 shows a comparison of the calculation results of sound pressure when Type 2 is used, without felt, with felt, and when the flow resistance value of felt is 19000 Ns/m^4 and 190000 Ns/m^4. When the flow resistance of the felt is increased, the sound pressure is reduced over the entire frequency range, but in some areas the peaks are sharp. In addition, the sound pressure tends to be relatively reduced in the low frequency range, and it is difficult to predict from the sound absorption and sound insulation performance of felt alone.

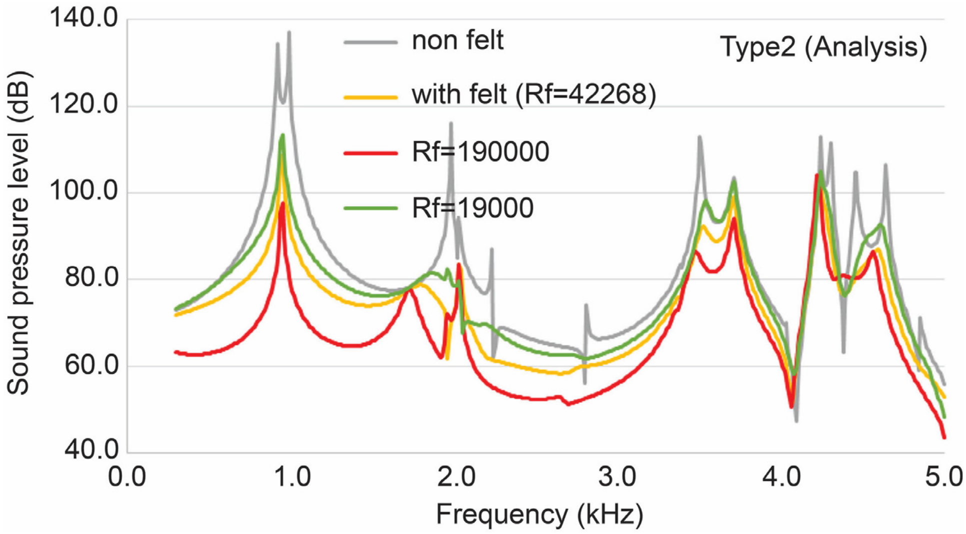


Fig.15. Sound pressure level for type2 (analysis).

4. Conclusion

Using gas-permeable trim as a sound absorbing material (porous body) that separates the chamber space and panel, we created a simple model and FE model that simulated the car door panel, the door space, the trim and the car compartment space, the following findings were obtained by measuring and calculating the sound pressure at the time of panel vibration.

- It was confirmed that the FE model, material data, and felt used in this study were appropriate.
- Depending on the presence or absence of felt, the change in the peak of the sound pressure by the vibration mode is small, but the peak of the sound pressure by the acoustic mode varies greatly in attenuation. The acoustic mode itself does not change much.
- It was found that it was difficult to predict the sound pressure level only by the sound absorption performance and sound insulation performance of the felt alone.

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