Proceedings of International Conference on Technology and Social Science 2017
Invited Paper

Transmission Loss Analysis for Automotive Panel Laminated with Felt and Olefin Sheet

Yoshio Kurosawa[1, a], Naoyuki Nakaizumi[2, b] and Manabu Takahashi[2, c]

[1]Teikyo University, 1-1, Toyosatodai, Utsunomiya-shi, Tochigi 320-8551, Japan

[2]Parker Asahi Co., Ltd., 15, Kitane, Fukaya-shi, Saitama 369-1242, Japan

[a]< ykurosawa@mps.teikyo-u.ac.jp >, [b]< nakaizumi-n@parker-asahi.co.jp>,

[c]< takahashi-m@parker-asahi.co.jp>

Keywords: Porous Media, Transmission Loss, Biot theory, Transfer matrix method, Automobile

Abstract. To reduce the interior noise of cars in the high frequency region, sound proof materials are laminated onto the body panels and interior trims. The sound proof properties of the laminate play an important role in efficient acoustical design. In this study, we developed a program code for predicting both the sound absorption and sound insulation properties of laminates. This program code uses the transfer matrix method based on the Biot theory and involves the vibro-acoustic coupling of a laminated structure with an elastic body (panel, olefin sheet), porous body (felts), and air. First, we use a transfer matrix to express the properties of the individual layers (sound wave transmission inside the material and reflection properties on the surface). Then, we combine the individual properties in the actual lamination order to obtain the acoustic transmission properties of the entire lamination structure. In this report, we outline this program code and present our calculation results, which almost agree with the experimental results.

1. Introduction

In recent years, car comfort has been recognized as a valid car property, and a silent car environment is one important factor. For example, with respect to the toe board panel that separates the passenger compartment from the engine area, strict sound insulation measures are taken to prevent engine noise. When pressed, the toe board is molded into a steel sheet with a unique shape, and, in the conventional method, acoustic insulation (porous body) consisting of felt urethane is laminated onto the top of the epidermis (viscoelastic body), which consists of resin seats. Recently, to achieve a more lightweight substitution for the resin sheet, a film (an olefin sheet) is being used as the viscoelastic body. The soundproofing structure around the toe board comprises a compound soundproofing structure with a mixture of solid (elastic body, viscoelastic body) and porous bodies and air. Using the transmission matrix method, we can calculate an index to numerically predict the soundproofing property (sound insulation property), which is the sound transmission loss when a sound penetrates a wall. The drawback of the transmission matrix method [1,2,3] is that only shapes (assuming an infinite flat board) simpler than those addressed by the finite element method can be calculated, but it has a very short calculation time and is suitable for calculating laminate acoustic insulation with a simple shape.

In this paper, we introduce a numerical analysis technique [4] for a soundproofing structure using felt and an olefin sheet, which are laminated onto a panel. We compare the sound insulation measurement results of the acoustic insulation using a simple installed measurement device in a semi-soundproofed room.

2. Method for calculating sound absorption and insulation properties

In this section, we describe the calculation method of the developed tool. The insulator structure consists of a body panel, insulator material, and interior and is assumed to be a laminated structure consisting of *m* layers, as shown in Fig. 1. More specifically, this is a vibro-acoustic coupling of a laminated structure with an elastic body (panel), viscoelastic body (damping material, resin), porous body (felt, urethane form), and air. First, we use a transfer matrix to describe the properties of the individual layers (sound wave transmission inside the material and reflection properties on the surface). Then, we combine the individual properties in the actual lamination order to obtain the sound transmission properties of the entire lamination structure.

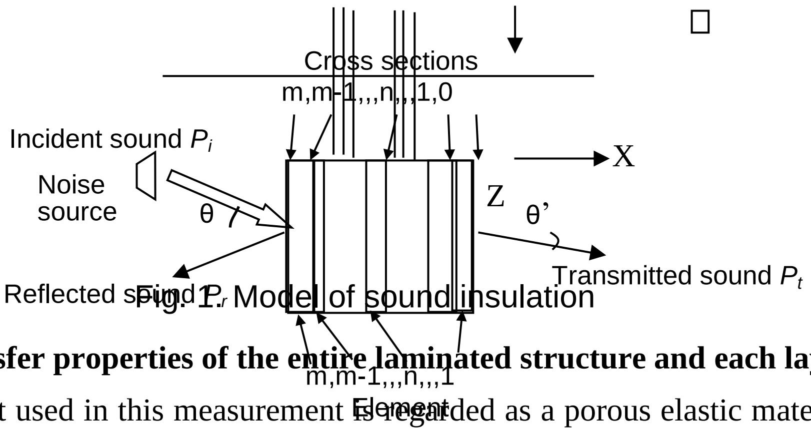


Fig. 1. Model of sound insulation

2.1 Acoustic transfer properties of the entire laminated structure and each layer

Because the felt used in this measurement is regarded as a porous elastic material [5], we defined it with respect to the sound pressures of the airborne and solid propagation sounds (a primary wave and side wave) in a 6*6 matrix [4,6], which can be considered to be particle velocity.

$$\begin{pmatrix} u_m^{sz} \\ u_m^{sx} \\ u_m \\ P_m^{sx} \\ P_m^{sz} \\ P_m \end{pmatrix} = \begin{bmatrix} T_{11} & T_{12} & T_{13} & T_{14} & T_{15} & T_{16} \\ T_{21} & T_{22} & T_{23} & T_{24} & T_{25} & T_{26} \\ T_{31} & T_{32} & T_{33} & T_{34} & T_{35} & T_{36} \\ T_{41} & T_{42} & T_{43} & T_{44} & T_{45} & T_{46} \\ T_{51} & T_{52} & T_{53} & T_{54} & T_{55} & T_{56} \\ T_{61} & T_{62} & T_{63} & T_{64} & T_{65} & T_{66} \end{bmatrix} \begin{pmatrix} u_{m-1}^{sz} \\ u_{m-1}^{sx} \\ u_{m-1} \\ P_{m-1}^{sx} \\ P_{m-1}^{sz} \\ P_{m-1} \end{pmatrix} \tag{1}$$

u_m^{sz} is the traverse wave of solid propagation sound (oscillating speed), u_m^{sx} is the longitudinal wave of the solid propagation sound (oscillating speed), P_m^{sx} is the longitudinal wave of the solid propagation sound (stress), and P_m^{sz} is the traverse wave of solid propagation sound (stress). The process procedure is shown in the equations below:

$$T_{11} = \frac{2N\beta^2\left(p_2D_1 - p_1D_2\right) - \left(p_3\left(C_1D_2 - C_2D_1\right)\right)}{\Delta} \tag{2}$$

$$T_{12} = j\beta \frac{\alpha_2 q_1\left\{\mu_2\left(\alpha_3^2 - \beta^2\right) + 2\beta^2\mu_3\right\} - \alpha_1 q_2\left\{\mu_1\left(\alpha_3^2 - \beta^2\right) + 2\beta^2\mu_3\right\} + 2\alpha_3 q_3 \alpha_1\alpha_2\left(\mu_1 - \mu_2\right)}{\alpha_1\alpha_2\left(\mu_1 - \mu_2\right)\left(\beta^2 + \alpha_3^2\right)} \tag{3}$$

$$T_{13} = j\beta \frac{\left(\alpha_1 q_2 - \alpha_2 q_1\right)}{\alpha_1\alpha_2\left(\mu_1 - \mu_2\right)} \tag{4}$$

$$T_{14} = \frac{\beta\omega}{\Delta}\left\{p_1 D_2 - p_2 D_1 - p_3\left(D_2 - D_1\right)\right\} \tag{5}$$

$$T_{15} = \frac{j\omega}{N\left(\alpha_3^2 + \beta^2\right)}\left(\frac{\beta^2 q_1\left(\mu_2 - \mu_3\right)}{\alpha_1\left(\mu_2 - \mu_1\right)} + \frac{\beta^2 q_1\left(\mu_1 - \mu_3\right)}{\alpha_2\left(\mu_1 - \mu_2\right)} + \alpha_3 q_3\right) \tag{6}$$

$$T_{16} = \frac{\beta\omega}{\Delta}\left(p_2 C_1 - p_1 C_2 - p_3\left(C_1 - C_2\right) + 2N\beta^2\left(p_2 - p_1\right)\right) \tag{7}$$

$$T_{21} = \frac{j\beta}{\Delta}\left(2N\left(\alpha_1 q_1 D_2 - \alpha_2 q_2 D_1\right) - \frac{q_3}{\alpha_3}\left(C_1 D_2 - C_2 D_1\right)\right) \tag{8}$$

$$T_{22} = \frac{p_2\left\{\mu_1\left(\alpha_3^2 - \beta^2\right) + 2\beta^2\mu_3\right\} - p_1\left\{\mu_2\left(\alpha_3^2 - \beta^2\right) + 2\beta^2\mu_3\right\} + 2\beta^2 p_3\left(\mu_1 - \mu_2\right)}{\left(\mu_1 - \mu_2\right)\left(\beta^2 + \alpha_3^2\right)} \tag{9}$$

$$T_{23} = \frac{p_1 - p_3}{\mu_1 - \mu_2} \tag{10}$$

$$T_{24} = -\frac{j\omega}{\Delta}\left(\alpha_1 q_1 D_2 - \alpha_2 q_2 D_1 + \frac{\beta^2 q_3}{\alpha_3}\left(D_2 - D_1\right)\right) \tag{11}$$

$$T_{25} = T_{14} \tag{12}$$

$$T_{26} = \frac{j\omega}{\Delta}\left(\alpha_1 q_1\left(C_2 + 2N\beta^2\right) - \alpha_2 q_2\left(C_1 + 2N\beta^2\right) - \frac{q_3\beta^2}{\alpha_3}\left(C_1 - C_2\right)\right) \tag{13}$$

$$T_{31} = \frac{j\beta}{\Delta}\left(2N\left(\alpha_1\mu_1 q_1 D_2 - \alpha_2\mu_2 q_2 D_1\right) - \frac{\mu_3 q_3}{\alpha_3}\left(C_1 D_2 - C_2 D_1\right)\right) \tag{14}$$

$$T_{32} = \frac{-\mu_1 p_1\left\{\mu_2\left(\alpha_3^2 - \beta^2\right) + 2\beta^2\mu_3\right\} + \mu_2 p_2\left\{\mu_1\left(\alpha_3^2 - \beta^2\right) + 2\beta^2\mu_3\right\} + 2\beta^2\mu_3 p_3\left(\mu_1 - \mu_2\right)}{\left(\mu_1 - \mu_2\right)\left(\alpha_3^2 + \beta^2\right)} \tag{15}$$

$$T_{33} = \frac{\mu_1 p_1 - \mu_2 p_2}{\mu_1 - \mu_2} \tag{16}$$

$$T_{34} = T_{26}, T_{35} = T_{16} \tag{17}$$

$$T_{36} = \frac{j\omega}{\Delta}\left(\mu_1\alpha_1 q_1\left(C_2 + 2N\beta^2\right) - \mu_2\alpha_2 q_2\left(C_1 + 2N\beta^2\right) - \frac{\beta^2}{\alpha_3}\mu_3 q_3\left(C_1 - C_2\right)\right) \tag{18}$$

$$T_{41} = \frac{2N\beta}{\omega\Delta}\left(C_1 p_1 D_2 - C_2 p_2 D_1 - p_3\left(C_1 D_2 - C_2 D_1\right)\right) \tag{19}$$

$$T_{42} = -j\frac{C_1 q_1\alpha_2\left\{\mu_2\left(\alpha_3^2 - \beta^2\right) + 2\beta^2\mu_3\right\} - C_2 q_2\alpha_1\left\{\mu_1\left(\alpha_3^2 - \beta^2\right) + 2\beta^2\mu_3\right\} - 4N\alpha_3\beta^2\alpha_1\alpha_2\left(\mu_1 - \mu_2\right)q_3}{\alpha_1\alpha_2\left(\beta^2 + \alpha_3^2\right)\left(\mu_1 - \mu_2\right)} \tag{20}$$

$$T_{43} = j\frac{\alpha_2 C_1 q_1 - \alpha_1 C_2 q_2}{\omega\alpha_1\alpha_2\left(\mu_1 - \mu_2\right)} \tag{21}$$

$$T_{44} = T_{22}, T_{45} = T_{12}, T_{46} = T_{32} \tag{22}$$

$$T_{51} = \frac{jN\beta^2}{\Delta\omega}\left(4N\alpha_1 q_1 D_2 - 4N\alpha_2 q_2 D_1 - q_3\frac{\alpha_3^2 - \beta^2}{\beta^2\alpha_3}\left(C_1 D_2 - C_2 D_1\right)\right) \tag{23}$$

$$T_{52} = T_{41} \tag{24}$$

$$T_{53} = \frac{-2N\beta}{\omega(\mu_1 - \mu_2)}\left(p_1 - p_2\right)$$

(25)

$$T_{54} = T_{21}, T_{55} = T_{11}, T_{56} = T_{31}, T_{61} = T_{53}, T_{62} = T_{43} \tag{26}$$

$$T_{63} = \frac{j}{\omega(\mu_1 - \mu_2)} \left(\frac{q_1 D_1}{\alpha_1} - \frac{q_2 D_2}{\alpha_2} \right) \tag{27}$$

$$T_{64} = T_{32}, T_{65} = T_{13}, T_{66} = T_{33} \tag{28}$$

3. Experimental and calculated results

Fig. 2 shows a schematic diagram of a simple sound insulation property measurement device (reverberation box) installed in a semi-soundproof room. A speaker is installed in the base of a cube, measuring about 1000 mm in length. Sound pressure can be input into a base panel (steel board 0.8 mm: thickness 0.8 mm) installed on the top of the cube with the laminated acoustic insulation on top. We installed a microphone 500 mm above the base panel and determined the sound insulation property by the difference (between the sound pressure level with and without the laminated acoustic insulation on the base panel) in the sound pressure levels for the same input sound pressure.

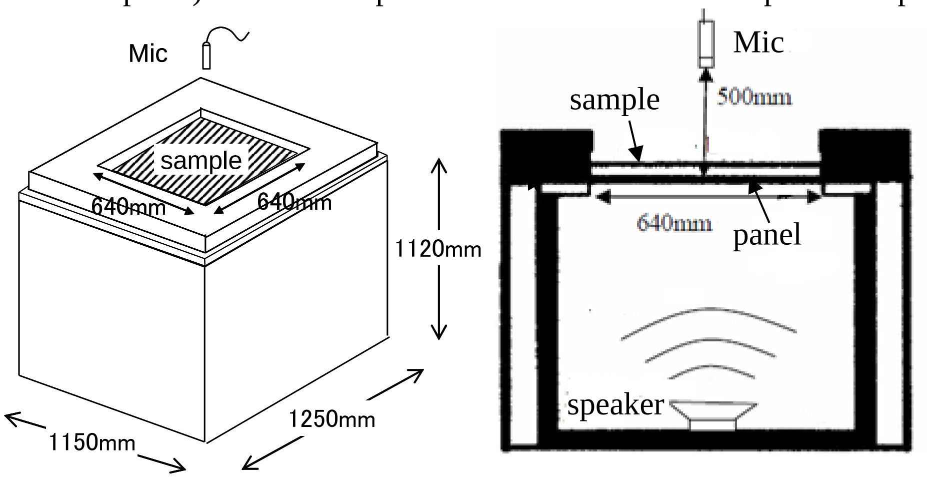
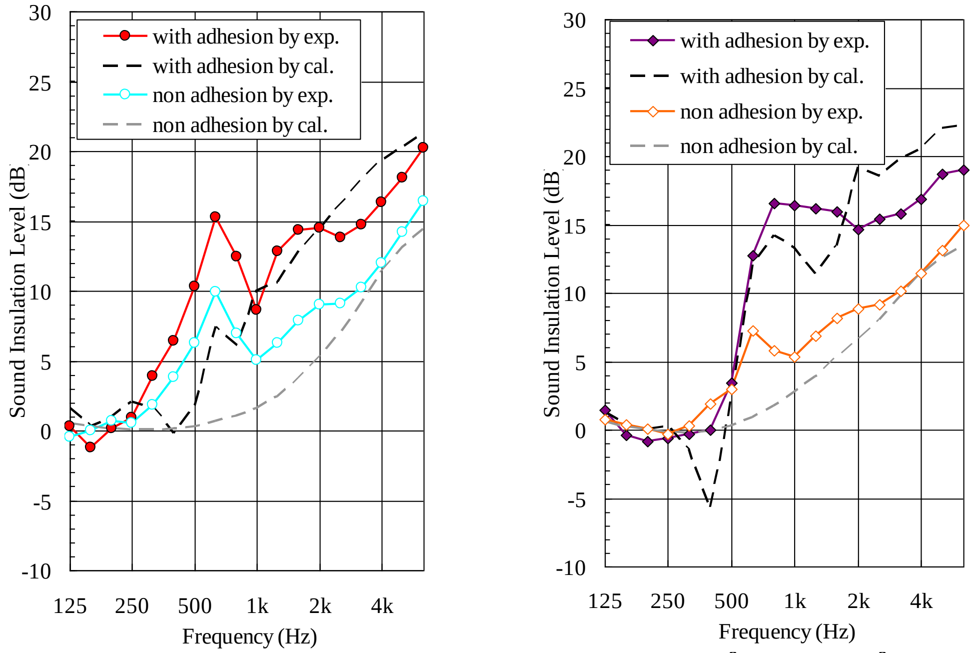


Fig. 2. Experimental setup of sound insulation level

Fig. 3 shows the experimental and calculated sound insulation results. Fig. 3a shows the experimental result (comparison between film adhesion with and without felt) with the acoustic insulation laminated on the base panel with 20 mm PET felt 800 g/m^2 + film 35 g/m^2 + 20 mm felt 1550 g/m^2, in that order, on the upper (mic) side, and Fig. 3b shows the experimental and calculated results for the acoustic insulation laminated on the base panel with 5 mm felt 550 g/m^2 + film 35 g/m^2 + 20 mm felt 1550 g/m^2, in that order. We found the sound insulation properties to be greatly influenced by film adhesion with and without felt. In addition, we know that the sound insulation property of the first felt order was good and that it differs when felt of different densities is used. The results indicate that thin felt that has high density (hard) and is more rigid improves adhesion.

When we used no adhesion in the calculation model, we set an atmospheric layer with a thickness of 0.1 mm between an olefin sheet and felt at the top and bottom. The calculation result could reproduce a qualitative difference in the influence of adhesion and we confirmed the calculation precision to be sufficient for predicting the sound insulation property. In addition, previous research confirmed that the influence of the adherent was not adequately expressed in the calculation model and did not consider the elasticity of the porous media [5], so the analysis precision is improved by this proposed model.

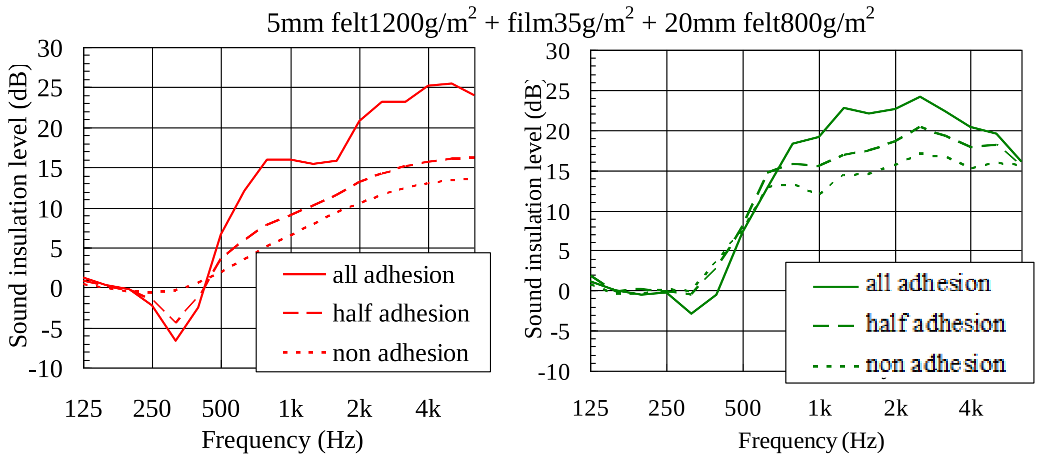


a. 20mm felt800g/m^2 + film35g/m^2 + 20mm felt1550g/m^2 b. 5mm felt550g/m^2 + film35g/m^2 + 20mm felt1550g/m^2

Fig. 3. Comparison of experimental and calculated results

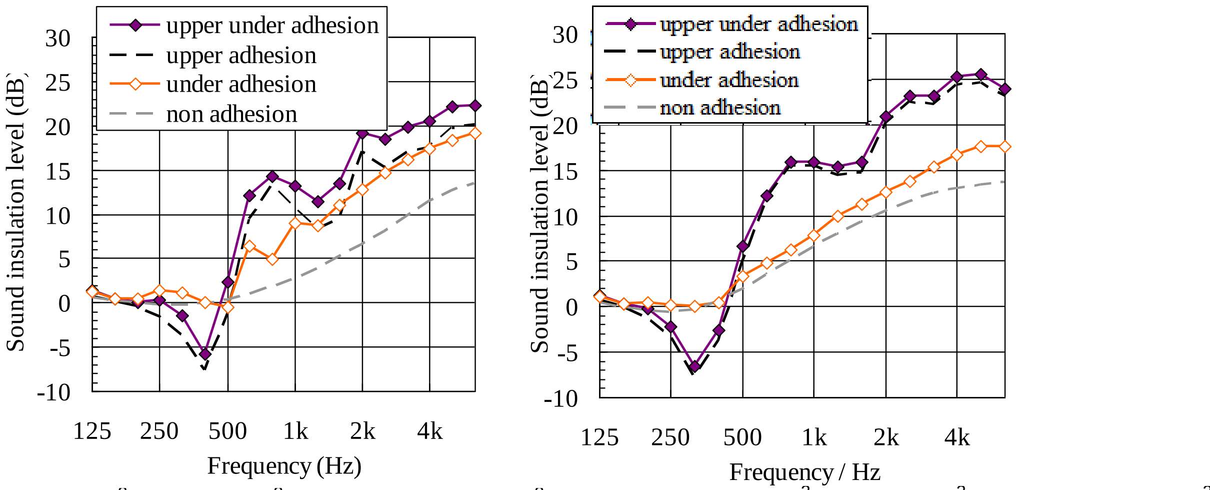
Fig. 4 shows a comparison of the calculated and experimental results when we changed the adhesion area lamination to 5 mm felt 1200 g/m^2 + film 35 g/m^2 + 20 mm felt 800 g/m^2. We can see that the sound insulation property takes the middle value without full-scale adhesion and when the adhesion area is half that of the experimental results.

In Fig. 5, we show a comparison between the 5 mm felt 550 g/m^2 + film 35 g/m^2 + 20 mm felt 1550 g/m^2 (Fig. 5a) and the 5 mm felt 1200 g/m^2 + film 35g/m^2 + 20 mm felt 800 g/m^2 (Fig. 5b). Using only both sides of the olefin seat, the top surface shows adhesion, and in comparison to the calculation results without adhesion, there is adhesion only on the undersurface. We can see that the 5 mm felt 1200 g/m^2 is harder than that shown in Fig. 5b and that the influence of the adherent with and without the film is great. Based on these results, we can conclude that we can greatly improve the sound insulation property by bonding hard felt to the film.



a. calculation results b. experimental results

Fig. 4. Comparison of all, half, and no adhesion



a. 5mm felt550g/m^2 + film35g/m^2 + 20mm felt1550g/m^2 b. 5mm felt1200g/m^2 + film35g/m^2 + 20mm felt800g/m^2

Fig. 5. Calculated sound insulation level results

4. Conclusion

Using a simple sound insulation property measurement device (reverberation box) installed in a semi-soundproof room, we measured the sound insulation properties of automotive acoustic insulation with felt and an olefin sheet laminated onto a panel. We confirmed that the sound insulation property of the panel greatly differed for a panel not adhered and when an olefin sheet was glued together with felt. In addition, we found it possible to greatly improve the sound insulation property by increasing the density of the felt (to make it hard) to improve adhesion with the olefin sheet.

Using the transmission matrix based on the Biot theory, we calculated the sound insulation property of the automotive laminated acoustic insulation. When comparing measurement results with the use of a reverberation box, we could reproduce qualitative differences by the presence or absence of adhesion and validated the precision adequacy of the analysis.

Because it is not possible to calculate the actual complicated acoustic insulation properties in a car with this technique, further program development by the finite element method represents a future challenge.

References

[1] A. London, " Transmission of reverberant sound through double wall", ***The Journal of the Acoustical Society of America,*** Vol. 22, No. 2, pp. 270, 1950.

[2] H. Utsuno, T. Tanaka, T. Fujikawa, " Transfer function method for measuring characteristic impedance and propagation constant of porous material", ***The Journal of the Acoustical Society of America***, Vol. 86, No. 2, pp. 637-643, 1989.

[3] T. Yamaguchi, "Numerical simulation of sound insulation and sound absorption properties for automotive multiple layered sound proof structures", ***Society of Damping Technology in Japan 1999 Symposium***, pp.91-94, 1999.

[4] J. F. Allad, N. Atala, Propagation of sound in porous media, ***A John Wiley & Sons, Ltd.***, 2009.

[5] Y. Kurosawa, N. Nakaizumi, M. Takahashi, and T. Yamaguchi, " Vibration and transmission loss for panels laminated with porous media", ***Dynamics and Design Conference***, No.262, 2013.

[6] D. Kato, " Predictive model of sound propagation in porous materials: Application of Biot theory in Kato model", ***The Journal of the Acoustical Society of Japan***, Vol. 64, No. 10, pp. 597-606,

2008.
[7] Yoshio Kurosawa, Naoyuki Nakaizumi, Manabu Takahashi, " Transmission Loss Analysis for Automotive Panel Laminated with Felt and Olefin Sheet", ***JTSS*** (accepted), 2017.