

FEA for an Effect of a Sail in Automotive Seat Including Poroelastic Materials and Metal Frames

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Abstract. This report deals with vibro-acoustic analysis using FEM for automotive seat having poroelastic materials and metal frames. There are two media (i.e. internal air and a resin block) in the poroelastic materials for the seat. By carrying out numerical analysis related, the poroelastic materials for the seat had sound absorption for cavity resonance. The poroelastic materials also had damping effects for the structural resonance of the metal seat frame under impact excitation. Under an acoustic excitation, vibration amplitudes of the metal frame increase by attaching the poroelastic materials to the metal frame. It is “an effect of a sail” of the resin block in the poroelastic material, which increases vibration level of the seat metal frame by receiving sound pressure. This effect can be simulated using Biot model for the poroelastic materials.

1. Introduction

It is known that interior parts of cabins in automobiles have influences on the interior noise [1]. The interior parts are constituted of elastic materials, viscoelastic materials and porous materials. Vibration and acoustic analysis in consideration of the materials are required. In low frequency, numerical analysis is effective since numerical methods under diffused sound field, cannot be applicable to the interior noise in the low or middle frequency. The seat structure for automobiles is mainly composed of metal frames, porous materials (e.g. urethane foams) and surface materials. In actual cars, the seat has a complicated structure for vibration and acoustic analysis because there exist many movable parts. The porous material for the seat has two media for wave propagation. One is a viscoelastic resin block of the seat foam. The other is an internal air of the foam. We mainly used Biot type poroelastic model. We use a simplified model for the automotive seat structures and perform finite element analysis to investigate vibro-acoustic coupling in the seat structure. Experiments are carried out to verify the numerical results.

2. Numerical procedure

We use Biot model for the foam of the seat as a poroelastic material. The coupling between sound pressure of the internal air and viscoelastic deformation of the resin block of the foam can be considered [2, 3, 4, 5]. The equations of motion for the internal air of the foam are as follows:

$$-\omega^2(\rho_{21}u_x + \rho_{22}U_x) + j\omega b(U_x - u_x) = -\Omega(\partial p / \partial x) \quad (1)$$

$$-\omega^2(\rho_{21}u_y + \rho_{22}U_y) + j\omega b(U_y - u_y) = -\Omega(\partial p / \partial y) \quad (2)$$

$$-\omega^2(\rho_{21}u_z + \rho_{22}U_z) + j\omega b(U_z - u_z) = -\Omega(\partial p / \partial z) \quad (3)$$

$\{u\} = \{u_x, u_y, u_z\}^T$ and $\{U\} = \{U_x, U_y, U_z\}^T$ are the displacement of the viscoelastic resin block and the particle displacement of the internal air, respectively. $\rho_{21} = -\rho_a$, $\rho_a = \rho_f(s-1)$, s is the tortuosity, $\rho_f = \rho_0\Omega$, ρ_0 is the mass density of the air, Ω is the porosity, $\rho_{22} = \rho_f + \rho_a$, $b = \Omega r$ and r is the flow resistance. The equations of motion for the viscoelastic resin block of the foam are as follows:

$$-\omega^2(\rho_{11}u_x + \rho_{12}U_x) + j\omega b(u_x - U_x) = (\partial\sigma_x/\partial x) + (\partial\tau_{xy}/\partial y) + (\partial\tau_{xz}/\partial z) \quad (4)$$

$$-\omega^2(\rho_{11}u_y + \rho_{12}U_y) + j\omega b(u_y - U_y) = (\partial\sigma_y/\partial y) + (\partial\tau_{yz}/\partial z) + (\partial\tau_{yx}/\partial x) \quad (5)$$

$$-\omega^2(\rho_{11}u_z + \rho_{12}U_z) + j\omega b(u_z - U_z) = (\partial\sigma_z/\partial z) + (\partial\tau_{zx}/\partial x) + (\partial\tau_{zy}/\partial y) \quad (6)$$

$\rho_{11} = \rho_s + \rho_a$, $\rho_{12} = -\rho_a$, $\rho_s = \rho_1(1-\Omega)$ and ρ_1 is the mass density of the resin skeleton. σ_x and σ_y are the normal stresses in x and y directions, respectively. τ_{xy} is the shear stress. Continuity equation of the internal air of the foam is as follows.

$$-\Omega p = Q_m \text{div}\{u\} + E_m \text{div}\{U\} \quad (7)$$

$E_m = \Omega E^*$, E^* is the bulk modulus of elasticity for the internal air. $Q_m = (1-\Omega)E^*$ is the coupling modulus of elasticity.

The relation between stress and strain for the viscoelastic resin block of the foam is as follows.

$$\tau_{xy} = \tau_{yx} = G_s^*((\partial u_x/\partial y) + (\partial u_y/\partial x)), \tau_{yz} = \tau_{zy} = G_s^*((\partial u_y/\partial z) + (\partial u_z/\partial y)), \tau_{zx} = \tau_{xz} = G_s^*((\partial u_z/\partial x) + (\partial u_x/\partial z)),$$

$$\begin{aligned} \sigma_x &= 2G_s^*(\partial u_x/\partial x) + A_s^* \text{div}\{u\} + Q_m \text{div}\{U\}, \quad \sigma_y = 2G_s^*(\partial u_y/\partial y) + A_s^* \text{div}\{u\} + Q_m \text{div}\{U\}, \\ \sigma_z &= 2G_s^*(\partial u_z/\partial z) + A_s^* \text{div}\{u\} + Q_m \text{div}\{U\}, \end{aligned} \quad (8)$$

Where, $G_s^* = E_s(1 + j\eta_s)/2(1 + \nu_s)$, $A_s^* = E_s\nu_s(1 + j\eta_s)/((1 + \nu_s)(1 + 2\nu_s))$. G_s^* is the shear modulus of elasticity of the resin block. E_s, η_s and ν_s are the storage modulus of elasticity, the material loss factor and Poisson's ratio, respectively. Deleting sound pressure p from Eqs. (1), (2) and (3), we obtain simultaneous differential equations using $\{U\}$ and $\{u\}$. Next, using a shape function $[N_U]^T$ for the internal air of the foam, the relation between the particle displacement $\{U\}$ in an element and the nodal particle displacement $\{U_e\}$ is approximated as $\{U\} = [N_U]^T \{U_e\}$. Using a shape function $[N_u]^T$ for the viscoelastic resin of the foam, the relation between the displacement $\{U\}$ in an element and the nodal displacement $\{U_e\}$ is approximated as $\{u\} = [N_u]^T \{u_e\}$. These equations are substituted into the differential equations with respect to $\{U\}$ and $\{u\}$. Applying Galerkin's procedure, the following equations can be obtained.

$$-\omega^2([M_{11}]\{u_e\} + [M_{12}]\{U_e\}) + j\omega([C_{11}]\{u_e\} + [C_{12}]\{U_e\}) + [K_{11}]\{u_e\} + [K_{12}]\{U_e\} = \{F_e\} \quad (9)$$

$$-\omega^2([M_{21}]\{u_e\} + [M_{22}]\{U_e\}) + j\omega([C_{22}]\{U_e\} + [C_{21}]\{u_e\}) + [K_{22}]\{U_e\} + [K_{21}]\{u_e\} = \{f_e\} \quad (10)$$

Where, $[M_{11}], [M_{12}], [M_{21}], [M_{22}]$ are the mass matrix and $[K_{11}], [K_{12}], [K_{21}], [K_{22}]$ are the stiffness matrix and $[C_{11}], [C_{12}], [C_{21}], [C_{22}]$ are the damping matrix of the Biot element. For the metal frame, we used the finite element model [6] having the complex stiffness matrix. For the sound field in a closed space, we used particle displacements as unknowns for the finite element [6, 7]. All elements are superposed. This yields the discrete equations in the global system.

3. Models for numerical analysis and test

Figure 1 shows an outline of the numerical model and the experimental device. We installed an aluminum frame and a urethane foam for the automotive seat structure into a small closed space.

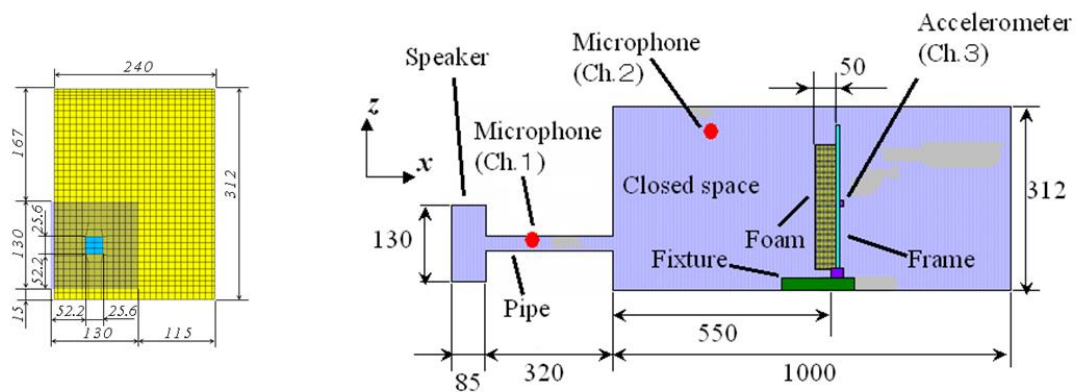
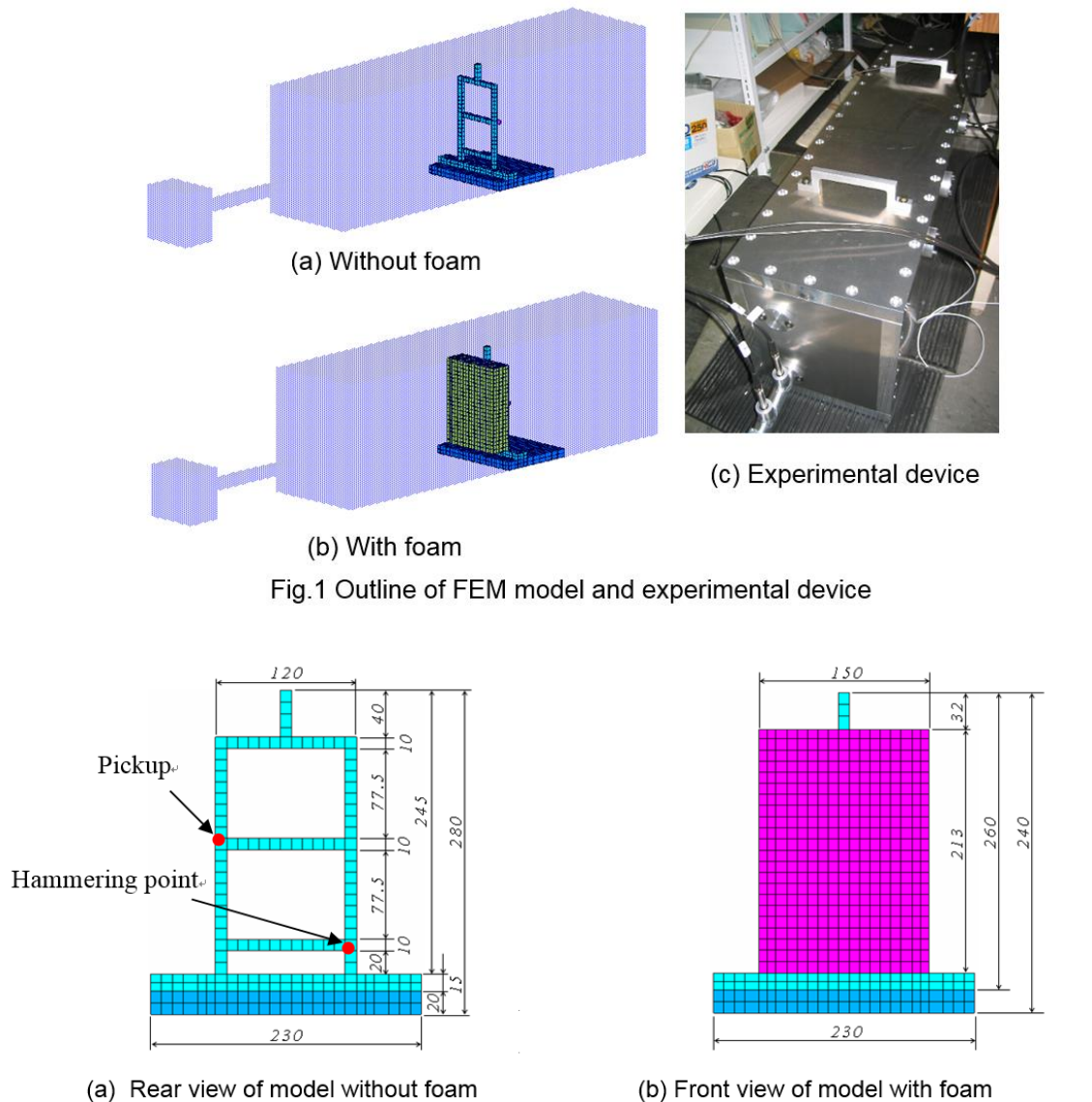


Fig.2 Detail geometry of seat model

Figures 1(a) and (b) show finite element models without / with the foam, respectively. Figure 1(c) is a photo of the experimental device. Figure 2 shows the detail geometry of the seat model composed of (a) the metal frame, (b) the foam and (c) the closed space. As shown in Fig. 2(a), the shape of the aluminum frame is like a ladder. The shape of the cross section of the beams in the frame is square (10[mm]×10[mm]). We attached the foam to the aluminum frame using adhesive tapes. As illustrated in Fig. 2 (b), the height and the width of the foam are 213[mm] and 150[mm], respectively. Thickness of the foam is 50[mm]. This foam is a typical material used in a seat structure in automobiles. The seat foam for this study is cut off from a larger block of the urethane foam. For the finite element model, we connect between the metal frame and the foam using complex spring elements to take the effects of the adhesive tape into consideration. Acoustic excitation is applied on this test model in the small closed space for the numerical analysis and experiment as shown in Fig. 2(c). The height and the width of the closed space are 312[mm]×240[mm]. The length of the closed space is 1000[mm]. Both sound pressure level of the space and acceleration level of the metal frame are observed and computed. The frequency range on this investigation is from 50[Hz] to 500[Hz]. This frequency range is corresponding to the range of road noise in cars where the coupling between the vibration of the automotive structures and the sound is strong. Note that the closed space cannot be regarded as diffused sound field, and the metal frame is also not regarded as diffused field because of their low modal density

By a hammering test, we observe transfer functions A/F between accelerations A of the frame and an input force F of the other point on the frame. Figure 2(a) shows the input point and the observation point. We compute the responses A/F using the finite element method under the same condition as the hammering test. Figure 3(a) is the experimental responses of the hammering test, and Fig. 3(b) is the corresponding calculated results. In the graphs of Fig. 3, there exist gains of A/F for the metal frame with / without the foam, respectively. According to Fig. 3(a), the first and the third peaks correspond to the first and the second bending modes in x -direction, respectively. The second peak is the first torsion mode around z -axis. The modal loss factor increases from 0.014[-] without the foam to 0.042[-] with the foam for the first peak. The foam improves the modal loss factor for the second peak from 0.0026[-] to 0.17[-]. The factor for the third peak increases from 0.030[-] to 0.086[-]. Thus, all peaks decrease due to vibration damping effect of the foam. The calculated results in Fig. 3(b) and the experimental ones in Fig. 3(a) agree well qualitatively.

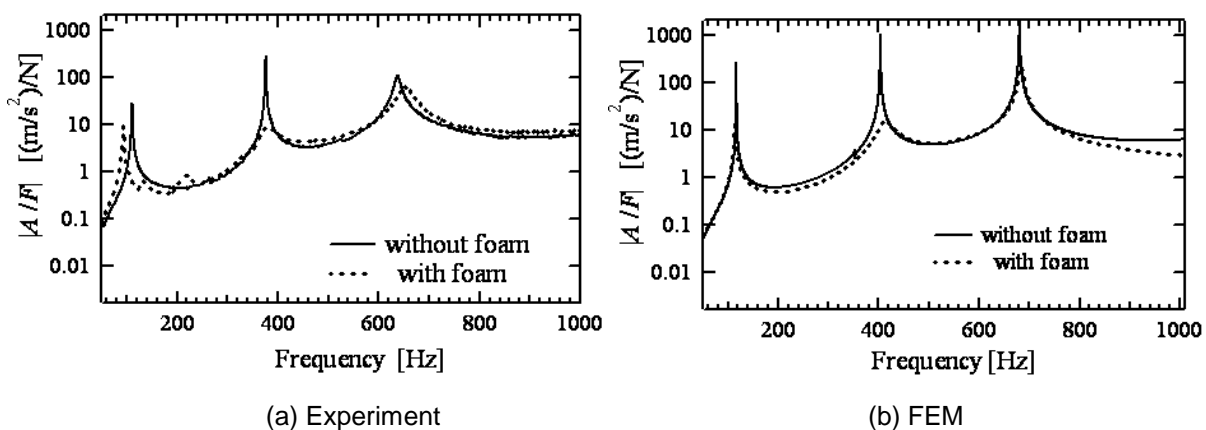


Fig. 3 Acceleration of frame under hammer excitation

As shown in Figs. 1 and 2, the metal frame with the frame with the foam is set in the closed space, and is excited by a speaker in the small box. This speaker box is connected to the closed space using a short pipe. Microphones are set at the connected pipe (i.e. Channel 1 as a reference point) and in the closed space (i.e. Channel 2 as an observation point) as illustrated in the Fig. 2(c). We observe transfer functions P_2/P_1 between sound pressure of the Channel 1 and sound pressure of the Channel 2. Figure 4(a) shows the experimental result. In this figure, solid line shows sound pressure $|P_2/P_1|$

without the foam, while the dotted line shows the result with the foam. The first and second peaks of $|P_2/P_1|$ are the first and second cavity resonance in x -direction, respectively. The foam increases modal loss factor of the first peak from 0.0072[-] to 0.031[-]. Therefore, all peaks are damped due to sound absorption effect of the foam. As can be seen in the Fig. 4(b), the computed sound pressure $|P_2/P_1|$ corresponds to the experimental one quantitatively.

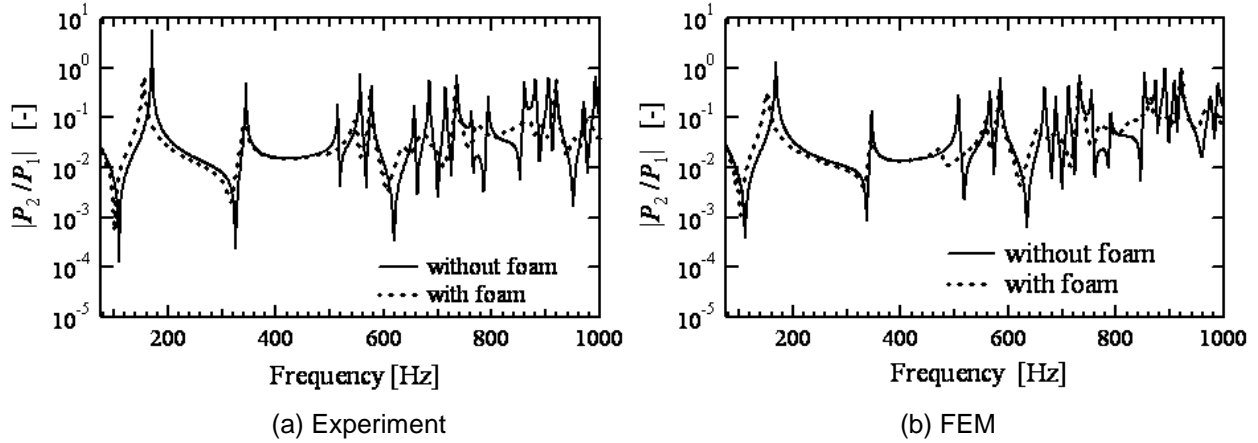


Fig. 4 Sound pressure in the closed space under acoustic excitation

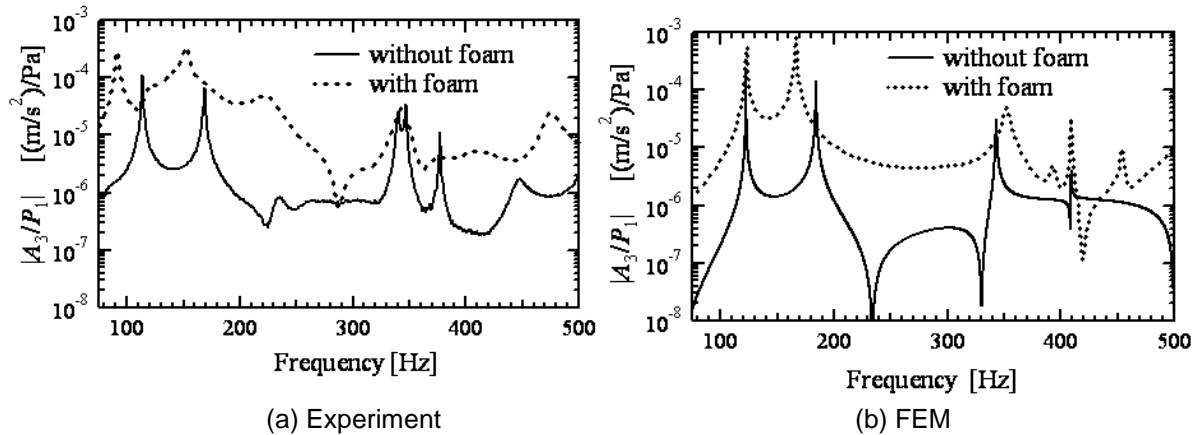


Fig. 5 An effect of a sail in acceleration of seat frame under acoustic excitation

An accelerometer (i.e. Channel 3) is attached to the same position as the hammering test on the metal frame. Thus, we evaluate transfer function A_3 / P_1 . Figure 5 (a) shows the experimental accelerations $|A_3 / P_1|$ with and without the foam. In this figure, the first peak corresponds to the first bending mode of the frame. This mode is the same mode as the first mode for the hammering test. With the foam, the modal loss factor of this mode increases from 0.0071[-] to 0.025[-] due to the damping effect of the foam. Thus, there exists the damping effect of the foam under the acoustic excitation. However, the amplitude of the acceleration increases, when the foam is attached to the metal frame. Namely, though damping increases due to the foam, the acceleration increases. The second peak in Fig. 5(a) is the first acoustic cavity mode in x -direction. This mode corresponds to the first mode in Fig. 4 (a). With the foam, the modal loss factor of this mode increases from 0.0068[-] to 0.034[-]. This implies that damping for this cavity mode improves due to the sound absorption effect of the foam. Nevertheless, the amplitude of the acceleration $|A_3 / P_1|$ is amplified due to the foam. For this mode, though the damping increases due to the foam, the amplitude increases. We think that it is “an effect of a sail” of the foam, which increases vibration level of the seat metal frame by receiving sound pressure. Figure 5(b) shows computed results of the acceleration $|A_3 / P_1|$ using the Biot model for the seat foam. As can be seen, the amplitude of the metal frame is also amplified

because of adhesion of the foam to the frame. This result is consistent with the experimental one. To simulate this effect, it is necessary to use the Biot model, which considers the coupling between sound waves in the internal air and the vibration of the viscoelastic resin block.

5. Conclusion

This paper deals with vibro-acoustic analysis using FEM for automotive seat structures having poroelastic materials and metal frames. There are two media (i.e. internal air and a resin block) in the poroelastic materials for the seat. By carrying out numerical analysis related to the model test, the porous materials for the automotive seat have sound absorption effects for cavity resonance in a room. This effect is mainly due to viscosity of internal air in the porous materials. Further the porous materials also have damping effects for the structural resonance of the metal seat frame under impact excitation at the metal frame. This effect is mainly due to viscoelasticity of the resin block in the porous materials. Furthermore, under an acoustic excitation from a speaker, vibration amplitudes of the metal frame increase by attaching the poroelastic materials to the metal frame. It is “an effect of a sail” of the resin block in the poroelastic material, which increases vibration level of the seat metal frame by receiving sound pressure. This effect can be simulated using Biot model for the poroelastic materials, which have coupling effects between the elastic deformations of the resin block and the acoustic waves in the internal air.

References

- [1] P. Chen and G. Ebbitt, "Noise absorption of automotive seats". *SAE paper*, 980659, pp.117-121, 1998.
- [2] M. A. Biot, "Theory of propagation of elastic waves in a fluid-saturated porous solid (1.Low-frequency range) ", *Journal of the Acoustical Society of America*, Vol. 28, No. 2, pp.168-178, 1955.
- [3] Y. J. Kang and S. Bolton, "A finite element model for sound transmission through foam-lined double panel structures", *Journal of the Acoustical Society of America*, Vol. 99, pp.2755-2765, 1996.
- [4] N. Attala, R. Panneton and P. Debergue, "A mixed pressure-displacement formulation for poroelastic materials", *Journal of the Acoustical Society of America*, Vol.104, No.3, pp.1444-1452, 1998.
- [5] J.F. Allard and N. Atalla, *Propagation of Sound in Porous Media(Second Edition)*, A John Wiley and Sons, Ltd., Publication, 2009.
- [6] T. Yamaguchi, Y. Kurosawa and S. Matsumura, "FEA for damping of structures having elastic bodies, viscoelastic bodies, porous media and gas", *Mechanical Systems and Signal Processing*, Vol.21, No.1, pp.535-552, 2007.
- [7] T. Yamaguchi, Y. Kurosawa and H. Enomoto, "Damped vibration analysis using finite element method with approximated modal damping for automotive double walls with a porous material", *Journal of Sound and Vibration*, Vol.325, pp.436-450, 2009.
- [8] T. Yamaguchi, T. Fukushima, T. Yamamoto, M. Fujimoto and I. Shiota, "Vibro-acoustic FEA for an effect of a sail in automotive seat structures including poroelastic materials and metal frames ", *Journal of Technology and Social Science*, (accepted).