Numerical Analysis of Sound Absorbing Properties for Multi-layered Structures Including Perforated Plates, Plates with Slits, Porous Materials and Air Layers

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Abstract. In this paper, we proposed a multi-layered sound absorbing structure having a hard surface made of resin from middle frequency to high frequency range. This proposed structure includes perforated plates, a plate with slits, a porous material and air spaces. We investigate many kinds of multi-layered sound absorbing structures by experiment and numerical simulation using finite element method in consideration of damping in acoustic field. Further, we investigated and verified a computation method to obtain sound absorbing properties for structures including perforated plate and plates with slits. In this calculation we consider acoustic resistance for internal air in the pores and slits using increments in complex effective density.

1. Introduction

Many kinds of resin parts have been utilized as interior parts and exterior parts in automobiles. For these resin parts, it is necessary to have many kinds of performances, e.g., strength, rigidity, cost, light weight, recycle property, flame retardance, resistance to ultraviolet light, heat resistant property, low temperature impact resistance, thermal contraction/expansion characteristics. Further, beautiful appearances and surfaces in design with high quality are also important factors for the interior / exterior parts. Furthermore, recently, sound-proof performances such as sound absorption and sound insulation are required to these interior / exterior resin parts. Therefore, we have to obtain the sound-proof performances compatible with the other abovementioned various performances in the products.

Almost typical sound absorbing materials [1], such as fibrous materials or foam materials, do not always have beautiful surfaces or appearances to use for bare surface materials of the interior / exterior parts. Since these surface properties notably influence on the sound absorbing performances of the parts, it is important that surface treatments or surface finishing for the parts to be compatible between sound-proof performances and the other performances. On the other hand, conventional resin parts have comparatively hard surfaces, so that sound waves especially in mid and high frequency tend to reflect at the surface of the hard resin parts. This causes low sound absorbing performance. And this also means that it is difficult to obtain large sound absorption coefficients when typical sound absorbing materials are set behind these resin parts with the hard surfaces. Thus, openings (e.g., pores and slits) are sometimes formed in the hard resin parts, so that the waves reach adequately to the porous materials behind the resin parts through the openings.

If there exist back air spaces behind the plates with openings, Helmholtz resonances occur. Around these resonant frequencies, it is well known that external noise can be absorbed [1].

In this paper, to get high sound absorbing performance over mid and high frequency ranges, we propose multi-layered sound absorbing structure, which has a hard resin surface, containing perforated plates, a plate with slits, a porous material and air spaces. Both experiment [5], [12] and numerical simulation [1-19] are carried out to investigate. Moreover, in the numerical simulation, we try a new method to consider acoustic resistance for internal air in the pores and slits by using increments in complex effective density.

2. Measurement Method of Sound Absorbing Performances

By using an acoustic pipe (B&K corporation) as shown in Fig.1, we measure normal incident sound absorption coefficients. Test samples have circular shapes, and their diameters are 30 mm. Two microphones are set away from the positions (10mm and 30mm) at the surface of the samples in the pipe. Transfer functions are obtained between the sound pressures measured from these two microphones. From the functions, we compute normal incident sound absorption coefficients using the two microphones method [5], [12]. Pores and slits in test plates are formed by use of a 3D printer. The clearance is addressed between the side edges of the circular samples and the side wall of the acoustic pipe to keep loose boundary but minimum leak of sound waves. To obtain this condition, the circular samples in the pipe fall slowly and gradually due to their self-weight, when the pipe set vertically. When we measure sound absorption coefficients, the acoustic pipe is fabricated horizontally.

Fig.1. Experimental setup using acoustic pipe.

3. Numerical Method

Many researchers have been developed computation methods to obtain sound absorbing performances [1-16]. In this paper, we use an acoustic finite element method including complex effective density and complex volume elasticity for sound absorbing regions, which is proposed by Yamaguchi who is the one of the authors [9-10], [17-19].

3.1 Discrete Equation of Sound Fields Containing Gas and Sound Absorbing Materials

We discretized sound fields containing air and sound absorbing materials by using finite elements. Under the assumption of infinitesimal amplitude, equations of motion of an inviscid compressive perfect fluid undergoing periodic oscillation can be expressed as follows [3], [9-10], [17-19]:

$$-\operatorname{grad} p = -\rho\omega^2 \{U\} \tag{1}$$

The relation between sound pressure and volume strain are expressed as follows:

$$p = -E \operatorname{div}\{U\} \tag{2}$$

where p is the sound pressure, $\{U\}$ is the particle displacement vector, ω is the angular frequency, and E and ρ are the bulk modulus of elasticity and effective density of air, respectively.

By introducing shape functions $[N]^T$, the relationship between p in an element and nodal sound pressure vector $\{p_e\}$ at the nodal points can be approximated.

$$p = [N]^T \{p_e\} \tag{3}$$

where $[N]^T = [N_1, N_2, N_{3,...}]$, and T denotes transpose.

Next, the kinetic energy, strain energy and external work are derived from Eqs. (1)-(3). The following expressions are then obtained by applying the minimum energy principle.

$$([K]_e - \omega^2 [M]_e)\{p_e\} = -\omega^2 \{u_e\} \tag{4}$$

$$[K]_e = (1/\rho_e)[\widetilde{K}]_e \tag{5}$$

$$[M]_e = (1/E_e)[\widetilde{M}]_e \tag{6}$$

ρ_e and E_e are the effective density and bulk modulus of elasticity for gas in an element, respectively. $\{u_e\}$ is the nodal particle displacement vector. $[\widetilde{M}]_e$ is a matrix that contains the shape functions, whereas $[\widetilde{K}]_e$ is a matrix that contains the derivatives of the shape function. $\widetilde{M}_{eij}$ and $\widetilde{K}_{eij}$ are components of $[\widetilde{M}]_e$ and $[\widetilde{K}]_e$, respectively. These components are expressed as follows:

$$\widetilde{M}_{eij} = \iiint_e N_i N_j dxdydz \tag{7}$$

$$\widetilde{K}_{eij} = \iiint_e \{(\partial N_i / \partial x)(\partial N_j / \partial x) + (\partial N_i / \partial y)(\partial N_j / \partial y) + (\partial N_i / \partial z)(\partial N_j / \partial z)\} dxdydz \tag{8}$$

where i is the component of the ith row, and j represents the component of the jth column. In this paper, we refer to $[K]_e$ as the element stiffness matrix and $[M]_e$ as the element mass matrix.

A model employing complex effective density ρ_e^* and complex propagation speed c_e^* is used to analyze the sound fields inside the sound absorbing materials [6], [12]. In this paper, we use the following model having complex effective density ρ_e^* and complex bulk modulus of elasticity $E_e^* = \rho_e^* (c_e^*)^2$ for the elements in the sound absorbing materials [9], [11].

$$\rho_e \Rightarrow \rho_e^* = \rho_{eR} + j\rho_{eI} \tag{9}$$

$$E_e \Rightarrow E_e^* = E_{eR} + jE_{eI} \tag{10}$$

Where j is an imaginary unit. E_{eR} and E_{eI} are the real and imaginary parts of E_e^*, respectively. E_{eI} is related to hysteresis between sound pressure p and volume strain $\mathrm{div}\{U\}$. ρ_{eR} and ρ_{eI} are the real and imaginary parts of ρ_e^*, respectively. ρ_{eI} is related to flow resistance R. We have verified effectiveness of this model in our previous study [9].

The element mass matrix $[K]_e$ is obtained by substituting Eq. (9) into Eq. (5).

$$[K]_e = [K_R]_e (1 + j\eta_e) \tag{11}$$

$$[K_R]_e = (\rho_{eR} / (\rho_{eR}^2 + \rho_{eI}^2))[\widetilde{K}]_e, \quad \eta_e = -\rho_{eI} / \rho_{eR} \tag{12}$$

$[K_R]_e$ is the real part of the element stiffness matrix $[K]_e$. The imaginary part ρ_{eI} of the effective density is related to flow resistance of the sound absorbing materials. Hence $\eta_e = -\rho_{eI} / \rho_{eR}$ corresponds to material damping due to the flow resistance.

By substituting Eq. (10) into Eq. (6), the element stiffness matrix $[M]_e$ is obtained.

$$[M]_e = [M_R]_e (1 + j\chi_e) \tag{13}$$

$$[M_R]_e = (E_{eR} / (E_{eR}^2 + E_{eI}^2))[\widetilde{M}]_e, \quad \chi_e = -E_{eI} / E_{eR} \tag{14}$$

$[M_R]_e$ is the real part of $[M]_e$. χ_e is the damping effect due to hysteresis between pressure and volume strain in the sound absorbing materials.

All elements for the mixed sound field can be superposed using Eqs. (4)-(14). This results in the following discrete equation of a global system.

$$\sum_{e=1}^{e_{\max}} \left([K_R]_e (1 + j\eta_e) - \omega^2 [M_R]_e (1 + j\chi_e) \right) \{p_e\} = -\omega^2 \{u\} \tag{15}$$

The size of the matrix and vectors in Eq. (15) is modified to be concurrent with the degree of freedom of the global system. $\{u\}$ is the nodal particle displacement vector. Both the stiffness matrix and the mass matrix for the fields containing gas and sound absorbing materials have complex parameters. Equation (15) corresponds to simultaneous equations having complex parameters. If known values are assigned to the excitation angular frequency ω and the nodal particle displacement vector $\{u\}$, Eq. (15) can be solved to obtain unknown p for the frequency responses [17-19].

Material properties (i.e., complex effective density ρ_e^* and complex volume elasticity E_e^*) of porous materials are identified by the transfer functions of sound pressure in the acoustic pipe using Improved Two Cavity Method [5], [12].

3.2 Complex Effective Density Related with Acoustic Resistance in Pores/Slits in Plates

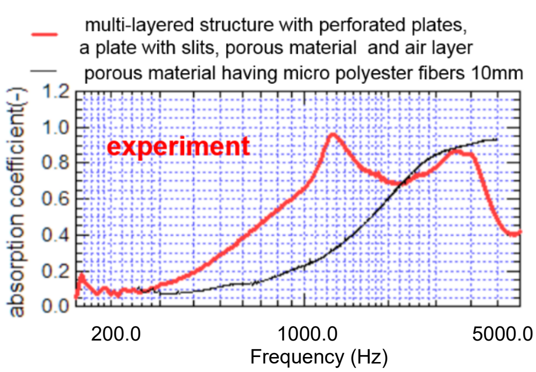
It is known that small pores or narrow slits affect the sound absorbing performances because the viscous resistance between these pores / slits and the side walls around the openings in plates is very large. For the openings formed into basic shapes such as pores having circular cross sections and rectangular slits, theoretical acoustic resistance was already derived and expressed by many previous researchers. However, we can use openings formed in other original shapes for the sound absorbing structures. Especially, nowadays, we can easily design and realize the plates with various openings by using 3D printers. Moreover, opening regions in actual resin parts for products are sometimes different from the ideal basic shapes due to restriction of design and workability and so on. In this paper, when we compute the sound absorption coefficients of multi-layered structures including perforated plates and plates with slits, we try to introduce damping term η_e (see Eq. (12)) in relation with complex effective density in consideration with acoustic viscous resistance around the openings. In Fig.2 (b), the elements in light blue color are an example of FEM model with the damping term η_e of air inside pores and slits.

4. Investigation Results

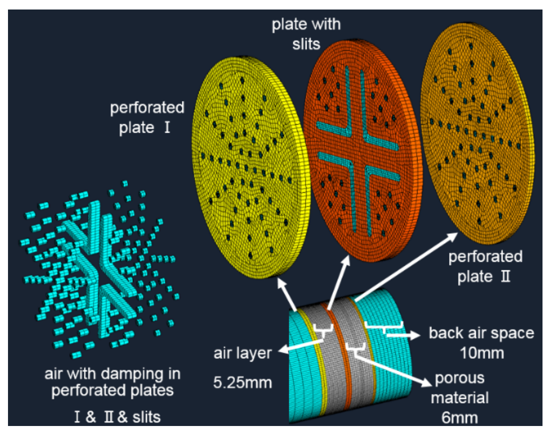
4.1 Measurement Results and a Proposition of Multi-layered Sound Absorbing Structure

To measure and evaluate normal incident sound absorption coefficients, we used circular plates made of hard resin as test samples. In these plates, many kinds of pores and slits are formed using 3D printer. Using these plates, we constructed more than 500 combinations of multi-layered structures including the plates with the pores and slits, porous materials and air spaces. And then we tested and evaluated for the multi-layered sound absorbing structures. To evaluate the absorbing performance for the samples, we focused on the sound absorption coefficient at 500Hz, 1lHz, 2kHz and 4kHz. To compare sound absorption performance, we selected a 10mm thickness sample of a typical porous material having polyester micro-fibers.

After we measured and evaluated on the sound absorption coefficients for many kinds of samples, we proposed a multi-layered sound absorbing structure having a hard surface made of resin from middle frequency to high frequency range as shown in Fig.2 (b). In this structure, a perforated plate Ⅰ + air layer 5.25mm + a plate with slits + porous material 6mm + perforated plate Ⅱ + back air space 10mm are set with a rigid wall in the pipe. The pores in the perforated plate Ⅰ has twice the depth than that in the perforated plate Ⅱ. Figure 2 (a) shows the measured results for the proposed multi-layered structure in comparison with the typical porous material having polyester micro-fibers. As can be seen, though the proposed structure has a hard surface, the structure can have good sound absorbing performances from middle frequency to high frequency range. From 2 kHz to 4 kHz, the proposed structure has the almost identical performance to the typical porous material, and from 500 Hz to 2kHz, the absorbing coefficients of the structure are superior to the typical material.



(a) sound absorption coefficients



(b) proposed multi-layered structure

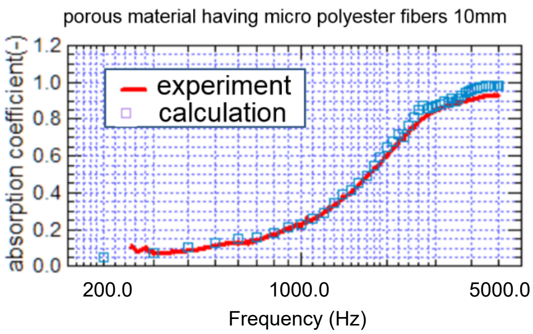
Fig. 2. Normal incident sound absorption coefficient for proposed multi-layered structure (experiment).

4.2 Calculation results of sound absorption coefficient

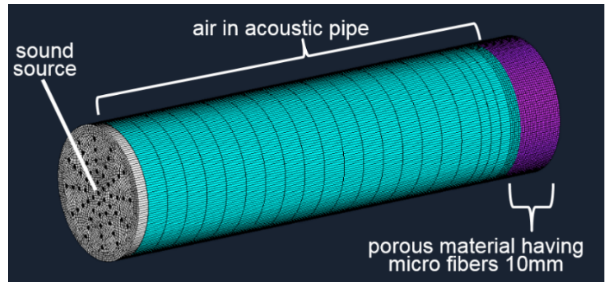
We computed the normal incident sound absorption coefficients for the proposed multi-layered structure as shown in Fig.2 (b).

In this proposed structure, there exists a porous material. First, we verified the sound absorption coefficients of this porous material from our numerical computation. Using FEM model in Fig.3 (b), we computed the absorption coefficients as shown in Fig.3 (a). From Fig.3 (a), the calculated results and the experimental results agree well.

Next, we show the validity of calculation models for the perforated plates.



(a) sound absorption coefficients

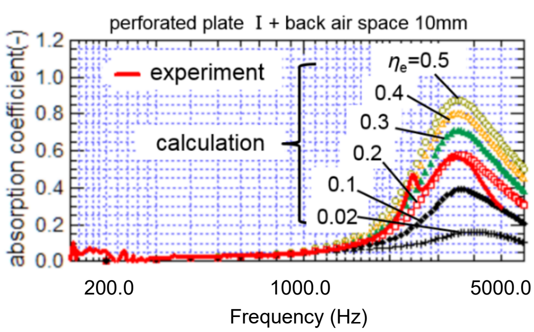


(b) FEM model

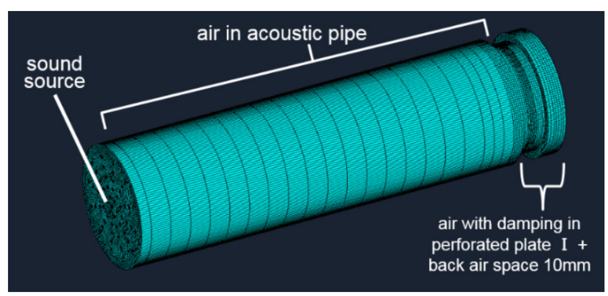
Fig. 3. Comparison between experimental and calculated results for porous material having micro fibers.

The perforated plate I with a back air space 10mm is evaluated on the sound absorbing performances between our computation and experiment. In this system, a Helmholtz resonance, in which the air in the pores behaves as mass and the back air become a pneumatic spring, will occur. At this resonant frequency, external noise will be absorbed. At that time, damping effects are generated from the acoustic resistance due to viscosity between the air in the pores and the walls in the plate at the pores.

By using numerical model in Fig. 4 (b), Fig. 4 (a) shows the calculated results in comparison with the corresponding experimental results. In this computation, the damping values η_e, related with the complex effective density due to the acoustic resistance around the pores, are changed.



(a) sound absorption coefficients



(b) FEM model

Fig. 4. Comparison between experimental and calculated results for perforated plate I with air space.

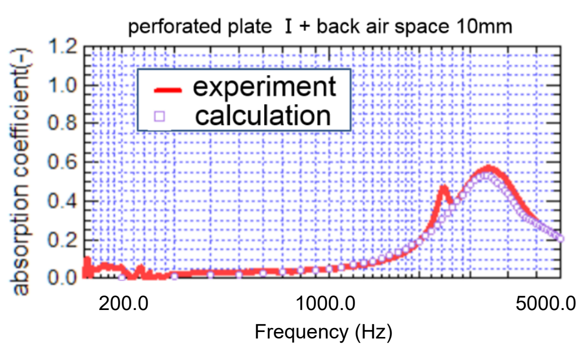
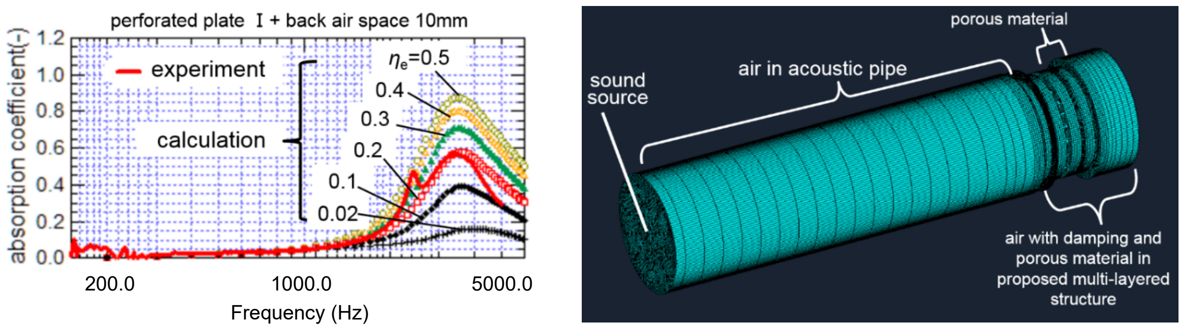


Fig. 5. Comparison between experimental and calculated results with frequency dependent damping η_e related with acoustic resistance for air in perforated plate I .



(a) sound absorption coefficients (b) FEM model

Fig. 6. Comparison between experimental and calculated results for perforated plates, plate with slits and air spaces.

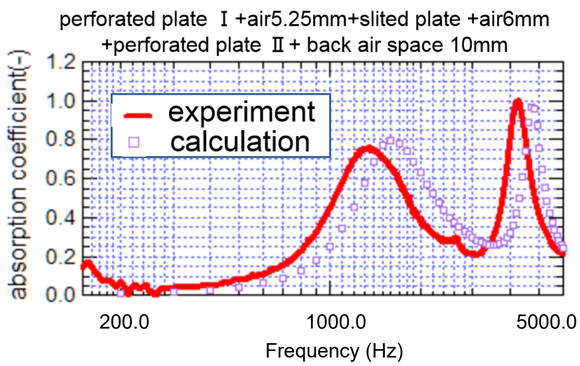


Fig .7. Comparison between experimental and calculated results with frequency dependent damping η_e related with acoustic resistance for air in perforated plate Ⅰ, Ⅱ and plate with slits.

From these results in Fig.4 (a), by using η_e=0.3 around 1.5kHz, η_e =0.2 around 3kHz and η_e =0.1 around 4kHz, the calculate values are consistent with the experimental ones. By giving these frequency dependences of the damping values η_e, we computed the absorption coefficients again as shown in Fig.5. Both the calculated and the experimental results are agreed very well.

Next, for a multilayered structure including the plate with slits (see Fig. 2(b)), we calculated the sound absorption coefficients in Fig. 6(a) using FEM model as depicted in the Fig. 6(b). In the acoustic pipe from the surface, we fabricated the perforated plate Ⅰ + air 5.25mm +the plate with slits +air 6mm + the perforated plate Ⅱ + air spaces10 mm + the rigid wall. In the Fig. 6(a), the damping values η_e due to the acoustic resistance are also varied. From the results in Fig. 6(a), if we use η_e = 0.3 around 1.5 kHz, η_e = 0.2 around 3 kHz and η_e = 0.1 around 4 kHz, the experimental values are well simulated. By using these frequency dependent η_e, the experimental sound absorption coefficients are almost reproduced by our computation as shown in Fig. 7. Nevertheless, the calculated peak frequencies are slightly shifted to the experimental ones.

Using the abovementioned investigations of the numerical parameters for the porous material, the two perforated plates and the plate with slits from Fig. 3 to Fig. 7, we construct a FEM model for the proposed sound absorbing multi-layered structure (see Fig. 2(b)). Fig.8 indicates the calculated result for this proposed structure. From this figure, the calculated absorption coefficient is well simulated to the experimental ones.

From these investigations, by considering the frequency dependent damping term η_e related with the complex effective density due to acoustic resistance, we confirm the validity of calculation method for normal incident sound absorption coefficients of multi-layered structures involving pores/slits in plates, porous materials and air spaces.

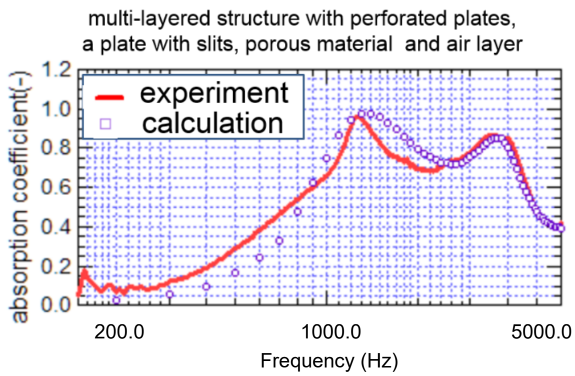


Fig. 8. Comparison between experimental and calculated results for proposed multi-layered structure.

This leads that the experimental results of the sound absorbing performance for the proposed multi-layered structure are also valid.

5. Conclusion

In this paper, we proposed a multi-layered sound absorbing structure having a hard surface made of resin from middle frequency to high frequency range. This proposed structure includes perforated plates, a plate with slits, a porous material and air spaces. We investigate many kinds of multi-layered sound absorbing structures by experiment and numerical simulation using finite element method in consideration of damping in acoustic field. Further, we investigated and verified a computation method to obtain sound absorbing properties for structures including perforated plates and plates with slits. In this calculation we consider acoustic resistance for included air in the pores and slits using increments in complex effective density.

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