Damped Vibration Analysis of Automotive Panels Laminated Porous Materials

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NOMENCLATURE

s : force per unit area, ρ : effective density of the internal air in porous media, ω : angular frequency, $\{u_f\}$: particle displacement vectors in the region of an element for porous media, E : volume elasticity of the internal air in porous media, p : pressure, $[N_f]^t$: matrix comprised of shape functions for porous media, $\{u_{fe}\}$: particle displacement vectors at nodal points in the element for porous media, $\tilde{T}_f$: kinetic energy for porous media, $\tilde{U}_f$: strain energy for porous media, $\tilde{V}_f$: external work for porous media, $[K]_{fe}$: element stiffness matrix for porous media, $[M]_{fe}$: element mass matrix for porous media, $\{f_{fe}\}$: nodal force vector for porous media, ρ_e , E_e : effective density and volume elasticity for media in the region of the elements for porous media, $[\tilde{M}]_{fe}$, $[\tilde{K}]_{fe}$: matrix consisted of shape functions and their derivatives, ρ_e^* : complex effective density, j : imaginary unit, ρ_{eR} , ρ_{eI} : real part and imaginary part of ρ_e^*, E_e^* : complex volume elasticity, E_{eR} ,E_{eI} : real part and imaginary part of E_e^*, $[M_R]_{fe}$: real part of $[M]_{fe}$, χ_e : material damping due to flow resistance, $[K_R]_{fe}$: real part of $[K]_{fe}$, η_e : damping effect concerning hysteresis, $\{\sigma\}$: stress vector for solid bodies, $[D]$: matrix including modulus of elasticity and Poisson's ratio for solid bodies, $\{\varepsilon\}$: strain vector for solid bodies, $[A]$: matrix comprised of differential operators, $\{u_s\}$: displacement vector for solid bodies, $[N_s]^t$: shape functions of solid bodies, $\{u_{se}\}$: displacements at nodal points for solid bodies, $\tilde{T}_s$: kinetic energy for solid bodies, $\tilde{U}_s$: strain energy for solid bodies, $\tilde{V}_s$: external work for solid bodies, $[K]_{se}$, $[M]_{se}$: element stiffness matrix and element mass matrix for solid bodies, $\{f_{se}\}$: nodal force vector in an element e for solid bodies, $[K_R]_{se}$: real part of element stiffness matrix for the solid bodies, $e_{\max}$: total number of elements, $\{F\}$: nodal force vector, $\{u_e\}$: nodal displacement vector in global system which is consisted of $\{u_{se}\}$ and $\{u_{fe}\}$, $[K_R]_e$:

matrix containing $[K_R]_{fe}$ and $[K_R]_{se}$, superscript(n) : the n-th eigenmode, $\omega^{(n)}$: real part of complex eigenvalue, $\{\phi^{(n)*}\}$: complex eigenvector, $\eta_{tot}^{(n)}$: modal loss factor, $\mu = j\eta_{max}$: small parameter, η_{max} : maximum value among the elements' material loss factors χ_e and η_e, $S_{se}^{(n)}$: share of strain energy of each element to total strain energy, $S_{ke}^{(n)}$: share of kinetic energy of each element to total kinetic energy, $\{\phi^{(n)}\}_0$: approximated eigenmodes, $\{A\}$: acceleration vector at the response point, $m^{(n)}$: modal mass

ABSTRACT

We have developed a technique for estimating damped vibration of automotive body panels with sound-proof structures. It calculates damping properties for three – dimensional sound-proof structures involving elastic body, viscoelastic body and porous media. For elastic and viscoelastic body displacement are modeled using conventional finite elements including complex modulus of elasticity. Both effective density and volume elasticity have complex quantities to represent damped sound fields in the porous media. Particle displacement in the media is discretised using finite element method. Displacement vectors as common unknown variables are solved under coupled condition between elastic body, viscoelastic body and porous media. Further, explicit expressions of modal loss factor for the mixed structures are derived using asymptotic method. Frequency responses were calculated for automotive test panel laminated with viscoelastic and porous materials using this technique. The results calculated almost agreed with the experimental results.

1 INTRODUCTION

Automotive body panels are laminated with damping materials and sound insulation materials to prevent noise in the cabin. An automotive body panel, which is made of steel sheet press-molded into a required form, is laminated with damping materials to reduce the vibration level. Furthermore, porous media, and PVC sheet (surface material) are laminated on the damping materials. Sandwiching the porous media between the panel and the PVC sheet realizes a double-walled sound insulation structure (Fig.1). In this way, solid materials (elastic and viscoelastic materials), porous media and gas (air) coexist in the sound isolation structure for the automotive body panel. Fig.2 shows the results of vibration level measurement of the front (hereafter called Ft) floor (acceleration response). In this measurement the mounted at part of Ft suspension was selected as vibration excitation point to estimate road noise. The vibration was measured under the panel ("Panel" in the figure) and on the PVC sheet ("PVC" in the figure). The difference of the vibration levels of the two areas is small until 200 Hz, while it becomes greater at larger noises.

From the above, for predicting the high-frequency road noise (200 – 500 Hz), it is essential to predict the vibration noise characteristics of the sound-proof structure, especially the surface material (PVC sheet) which emits in-vehicle noise, and numerical calculation is a possible technique for this. This study proposes a numerical analysis method for a sound-proof structure where an elastic material, a viscoelastic, porous media and air, designed with complicated sound-proof structures of automotive body panels. The finite element method is used to handle any shapes and boundaries. The method is designed to solve coupled problems of solid materials, porous media and air. In addition, an approximate calculation method is proposed for the modal loss factor of the complicated sound-proof structure. With this new technique, a vibration analysis of a panel modeling the automotive panels was performed and the results were compared with experimental results for accuracy verification.

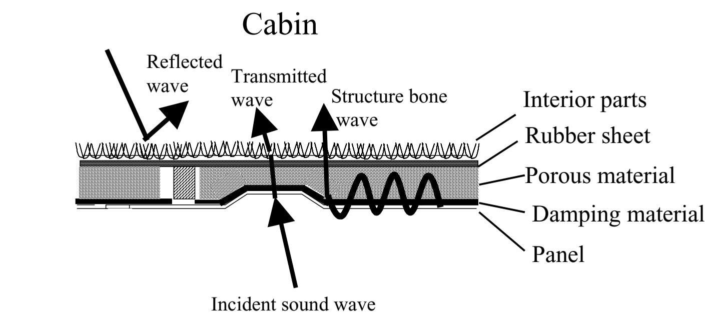


Fig.1 Automotive panel laminated with viscoelastic body and porous material.

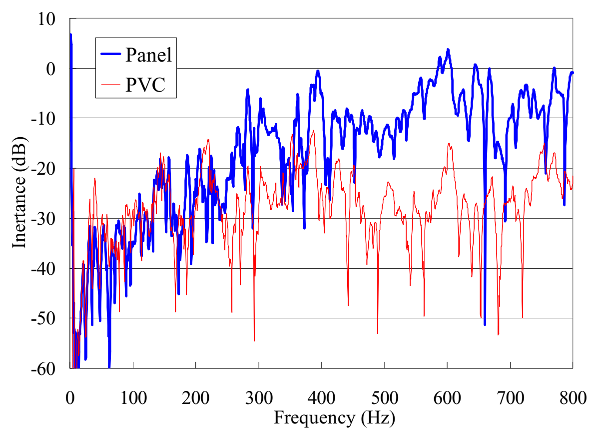


Fig.2 Effect of porous media to reduce vibration of front floor. (Driving point : front cross member)

2 ANALYSIS METHOD

This chapter introduces a numerical analysis method for vibration damping characteristics of coupled problems of vibration and acoustics where an elastic, a viscoelastic, porous media and gas. These components are expressed as finite elements and superposed tacked in consideration of coupling in order to handle any structures having arbitrary shapes. First, the section 2.1 will suggest a numerical analysis method by discretizing particle displacement in a damped sound field. Then the section 2.2 will explain finite elements of the displacement field for solid bodies (elastic and viscoelastic). The section 2.3 will explain the discrete equation for the global coordinate where solid bodies, porous media and gas coexist. In the section 2.4, an equation will be derived that approximately calculates the modal loss factor of the global coordinate by applying the asymptotic method. Finally, the section 2.5 will introduce an equation for damped vibration response using the MSKE method.

2.1 DISCRETIZED EQUATION FOR INTERNAL GAS IN POROUS MEDIA

First, the sound field of internal air in porous media is discretized using finite elements. Assuming infinitesimal amplitude, the equation of motion of inviscid compressive perfect fluid can be expressed under periodic oscillation as follows [5]-[7].

$$\mathrm{grad}\, s = -\rho\,\omega^2 \{u_{\mathrm{f}}\} \tag{1}$$

The equation of continuity is represented by the following equation.

$$s = E\,\mathrm{div}\,\{u_{\mathrm{f}}\} \tag{2}$$

Where, $\{u_f\}$ is the displacement vector of particles. *s* is the force per unit area. This *s* is a parameter that is introduced to adjust force when the structure and the sound filed are joined. It has a relationship with the pressure p as $s=-p$. *E* is the volume elasticity and *ρ* is the effective density of the internal air in porous media. *ω* is the angular frequency.

Conventional acoustic analysis often eliminate the particle displacement in equations (1) and (2), and derive an equation of motion which treats the pressure as unknown. In this study, however, the pressure is eliminated from the two equations and the particle displacement is retained as unknown. An advantage of this technique is that the displacement can be used as a common unknown for solid bodies. This allows easy superposition of elements between solid bodies and the sound field [7]. And makes the calculation method more suitable for complicated problems where solid bodies, porous media and gas are divided into many regions. On the other hand, while the unknown for the pressure is a scalar variable, the unknown for the particle displacement is a vector variable, which requires a larger number of calculations.

Relations between $\{u_f\}$ and the particle displacement $\{u_{fe}\}$ at nodal points in the element can be approximated as follows.

$$\{u_{\mathrm{f}}\} = [N_{\mathrm{f}}]^{\mathrm{t}} \{u_{\mathrm{f}e}\} \tag{3}$$

Where , $[N_f]$ represents a matrix comprised of appropriate shape functions.
Irrotational condition is :

$$rot\,\{u_f\} = \{0\}$$

The kinetic energy $\tilde{T}_f$, the strain energy $\tilde{U}_f$, and external work $\tilde{V}_f$ are obtained from equation (1), (2) and (3). The following expressions can be derived by applying the minimum energy principle $\delta\left(\tilde{U}_f - \tilde{T}_f - \tilde{V}_f\right) = 0$.

$$([K]_{\mathrm{f}e} - \omega^2 [M]_{\mathrm{f}e})\{u_{\mathrm{f}e}\} = \{f_{\mathrm{f}e}\} \tag{4}$$

$$[M]_{fe} = \rho_e [\tilde{M}]_{fe} \quad (5)$$

$$[K]_{fe} = E_e [\tilde{K}]_{fe} \quad (6)$$

Where, $\{f_{fe}\}$ is the nodal force vector, $[K]_{fe}$ is the element stiffness matrix, and $[M]_{fe}$ is the element mass matrix. ρ_e and E_e are the effective density and the volume elasticity for media in the region of element. $[\tilde{M}]_{fe}$ and $[\tilde{K}]_{fe}$ are the matrix consisted of the shape functions and their derivatives.

Equations (4), (5) and (6) are kinetic equations for the element that is linear compressible perfect fluid. These equations can be used as element equations for acoustic problems of gas under undamped conditions. For expressing the sound in the porous media, a model is proposed which converts the complex effective density and the acoustic velocity or complex volume elasticity, and its effectiveness is confirmed [4][5][10]-[12]. Based on this method, the following equations are obtained.

$$\rho_e \Rightarrow \rho_e^* = \rho_{eR} + j\ \rho_{eI} \quad (7)$$

$$E_e \Rightarrow E_e^* = E_{eR} + j\ E_{eI} \quad (8)$$

This model is mainly applied to textile materials such as glass wool. It ignores the influence of the elastic wave which transmits the frames of porous media, and assumes that the motion of gas is the dominant determiner. The model effectiveness is verified for porous media when their frame materials have adequate flexibility and large damping [4][5][9], and automotive sound-proof materials are often the case. On the other hand, when the frame materials of porous media are made of rigid materials such as metal, the elastic wave transmitting through the frames have larger influence than the air wave. In this case other models such as Biot's model will be required [13][14].

The element mass matrix $[M]_{fe}$ is obtained as follows by substituting equation (7) into equation (5).

$$[M]_{fe} = [M_R]_{fe}\,(1 + j\ \chi_e) \quad (9)$$

$$\chi_e = \rho_{eI} / \rho_{eR} \quad (10)$$

Where, $[M_R]_{fe}$ is the real part of $[M]_{fe}$. The imaginary part of the effective density ρ_{eI} is a term related to the flow resistance of the porous media, and $\chi_e = \rho_{eI} / \rho_{eR}$ corresponds to the material dumping caused by the flow resistance.

In the same way, the element stiffness matrix $[K]_e$ is obtained by substituting equation (8) into equation (6).

$$[K]_{fe} = [K_R]_{fe}\,(1 + j\ \eta_e) \quad (11)$$

$$\eta_e = E_{eI} / E_{eR} \quad (12)$$

Where, $[K_R]_{fe}$ is the real part of the $[K]_{fe}$. η_e is the material damping corresponding to the hysteresis in the relationship between the pressure and the volume strain (loss factor; all the damping values below are loss factors).

From the above, among the elements for the sound field in the porous media, the element stiffness matrix $[K]_{fe}$ and the element mass matrix $[M]_{fe}$ are both expressed with complex quantities. Gas such as air can be expressed by lowering their damping parameters, χ_e and η_e . The parameters χ_e , ρ_{eR} , η_e and E_{eR} can be identified by an experiments using a impedance tube [5][10].

Prior to this study, we proposed an analysis method of the field using the fine element method where solid bodies are not included but porous media and air coexist. This method consists of the procedure similar to the above, except that the pressure is treated as unknown instead of the particle displacement. The effectiveness of the method is confirmed regarding the damped response and the modal loss factor [4][11][12]. The proposed method in this new study is an enhanced version of the previous method to apply to coupled problems which also include solid bodies.

2.2 DISCRETIZED EQUATION FOR VIBRATION IN DAMPED SOLID BODIES

The vibration field of a solid bodies is discretized conventionally with the finite element method [15], using the following equations (13) – (17).

The relationship between the stress and the strain, and the relationship between the strain and the displacement are expressed as follows.

$$\{\sigma\} = [D]\{\varepsilon\} \tag{13}$$

$$\{\varepsilon\} = [A]\{u_s\} \tag{14}$$

Where, $\{\sigma\}$ is the stress vector, $\{\varepsilon\}$ is the strain vector, and $\{u_s\}$ is the displacement vector of the solid bodies. $[D]$ is the matrix including modulus of elasticity and Poisson's ratio, and $[A]$ is the matrix comprised of differential operators.

By using the matrix comprised of shape functions $[N_s]^t$, the relationship between the element displacement $\{u_s\}$ and the nodal displacements $\{u_{se}\}$ is approximated as follows.

$$\{u_s\} = [N_s]^t\{u_{se}\} \tag{15}$$

The following equation is obtained by obtaining the kinetic energy $\tilde{T}_s$, the strain energy $\tilde{U}_s$, and the external work $\tilde{V}_s$, and applying the minimum energy principle $\delta\left(\tilde{U}_s - \tilde{T}_s - \tilde{V}_s\right) = 0$

$$\left([K]_{se} - \omega^2[M]_{se}\right)\{u_{se}\} = \{f_{se}\} \tag{16}$$

Where, $\{f_{se}\}$ is the nodal force vector in an element *e* for solid bodies, $[K]_{se}$ and $[M]_{se}$ are the element stiffness matrix and the element mass matrix for solid bodies, respectively.

In order to express the viscoelastic material with hysteresis damping as a finite element, it is necessary to convert the elasticity $[D]$ in equation (13) into a complex modulus [3][16]. By doing this, the element stiffness matrix in equation (16) is also represented by a complex quantities as follows.

$$[K]_{se} = [K_R]_{se}(1 + j\eta_e) \tag{17}$$

Where, η_e is the material loss factor corresponding to each element *e*, and $[K_R]_{se}$ is the real part of the element stiffness matrix for solid bodies.

2.3 DISCRETIZED EQUATION IN GLOBAL SYSTEM

At the boundary of a solid bodies and gas or a solid bodies and porous media, only the displacement in the normal direction toward the boundary is continuous. By taking this into account and using equations (4) – (17), all the elements in an intended field (the complicated space of gas, porous media and solid bodies) are stacked to obtain the following discrete equation for the global coordinate [7].

$$\sum_{e=1}^{e_{max}} \left([K_R]_e (1 + j\eta_e) - \omega^2 [M_R]_e (1 + j\chi_e) \right) \{u_e\} = \{F\} \tag{18}$$

Where, e_{max} is the total number of elements and $\{F\}$ is the external force vector. $\{u_e\}$ is the nodal displacement vector in global system, which consists of $\{u_{fe}\}$ and $\{u_{se}\}$. Similarly, $[K_R]_e$ consists of $[K_R]_{fe}$ and $[K_R]_{se}$, while $[M_R]_e$ consists of $[M_R]_{fe}$ and $[M_R]_{se}$. In this equation, χ_e of the solid elements must be 0.

From the above, for the system where solid bodies, porous media and gas coexist, the stiffness matrix and the mass matrix are both expressed as complex quantities.

2.4 APPROXIMATE CALCULATION OF MODAL DAMPING (MSKE METHOD)

This section explains the approximate calculation of the mode damping of the global coordinate. The complex eigenvalue problem of equation (18) is represented by the following equation.

$$\sum_{e=1}^{e_{max}} \left([K_R]_e (1 + j\eta_e) - \left(\omega^{(n)}\right)^2 \left(1 + j\eta_{tot}^{(n)}\right) [M_R]_e (1 + j\chi_e) \right) \{\phi^{(n)*}\} = \{0\} \tag{19}$$

Where, $\left(\omega^{(n)}\right)^2$ is the real part of the *n*'th order complex eigenvalue, $\{\phi^{(n)*}\}$ is the *n*'th order complex eigen mode, and $\eta_{tot}^{(n)}$ is the *n*'th order modal loss factor.

Among the material damping χ_e, η_e (*e*=1,2,3,…, e_{max}), the largest number is expressed as η_{max}. In addition, the following value is defined and introduced.

$$\beta_{se} = \eta_e / \eta_{max}, \beta_{se} \le 1, \; \beta_{ke} = \chi_e / \eta_{max}, \beta_{ke} \le 1 \tag{20}$$

Here, by assuming $\eta_{max} << 1$ and introducing the small parameter $\mu = j\eta_{max}$, equation (19) is asymptotically expanded as follows.

$$\{\phi^{(n)*}\} = \{\phi^{(n)}\}_0 + \mu\{\phi^{(n)}\}_1 + \mu^2\{\phi^{(n)}\}_2 +,... \tag{21}$$

$$(\omega^{(n)})^2 = (\omega_0^{(n)})^2 + \mu^2(\omega_2^{(n)})^2 + \mu^4(\omega_4^{(n)})^2 +,... \tag{22}$$

$$j\eta_{tot}^{(n)} = \mu\eta_1^{(n)} + \mu^3\eta_3^{(n)} + \mu^5\eta_5^{(n)} + \mu^7\eta_7^{(n)} +,... \tag{23}$$

$\beta_{ke} \le 1$ and $\beta_{se} \le 1$, therefore if $\eta_{\max} << 1$, then $\eta_{\max}\beta_{ke} << 1$ and $\eta_{\max}\beta_{se} << 1$, and $\mu\beta_{se}$ and $\mu\beta_{ke}$ are also negligible amounts like μ . In addition, $\{\phi^{(n)}\}_0, \{\phi^{(n)}\}_1, \{\phi^{(n)}\}_2, \cdots$, $(\omega_0^{(n)})^2, (\omega_2^{(n)})^2, (\omega_4^{(n)})^2, \cdots$, and $\eta_1^{(n)}, \eta_3^{(n)}, \eta_5^{(n)}, \cdots$ are real quantities.

Then by substituting Equation (21) – (23) into equation (19) , the orders μ^0 and μ^1 are respectively combined as the following equations.

μ^0 order:

$$\sum_{e=1}^{e\max}\left(\left[K_R\right]_e - \left(\omega_0^{(n)}\right)^2\left[M_R\right]_e\right)\left\{\phi^{(n)}\right\}_0 = \{0\} \tag{24}$$

μ^1 order:

$$\sum_{e=1}^{e_{mav}}\left(\mu\beta_{se}\left[K_R\right]_e - \mu\eta_1^{(n)}\left(\omega_0^{(n)}\right)^2\left[M_R\right]_e - \mu\beta_{ke}\left(\omega_0^{(n)}\right)^2\left[M_R\right]_e\right)\left\{\phi^{(n)}\right\}_0 + \sum_{e=1}^{e_{\max}}\left(\mu\left[K_R\right]_e - \mu\left(\omega_0^{(n)}\right)^2\left[M_R\right]_e\right)\left\{\phi^{(n)}\right\}_1 = \{0\} \tag{25}$$

Furthermore, by arranging equations (24) and (25), equation (26) is obtained.

$$\eta_{tot}^{(n)} = \eta_{se}^{(n)} - \eta_{ke}^{(n)} \tag{26}$$

$$\eta_{se}^{(n)} = \sum_{e=1}^{e\max}\left(\eta_e S_{se}^{(n)}\right) , \quad \eta_{ke}^{(n)} = \sum_{e=1}^{e\max}\left(\chi_e S_{ke}^{(n)}\right)$$

$$S_{se}^{(n)} = \frac{\left\{\phi^{(n)}\right\}_0^t\left[K_R\right]_e\left\{\phi^{(n)}\right\}_0}{\sum_{e=1}^{e_{\max}}\left\{\phi^{(n)}\right\}_0^t\left[K_R\right]_e\left\{\phi^{(n)}\right\}_0} , \quad S_{ke}^{(n)} = \frac{\left\{\phi^{(n)}\right\}_0^t\left[M_R\right]_e\left\{\phi^{(n)}\right\}_0}{\sum_{e=1}^{e_{\max}}\left\{\phi^{(n)}\right\}_0^t\left[M_R\right]_e\left\{\phi^{(n)}\right\}_0}$$

According to these expressions, modal loss factor $\eta_{tot}^{(n)}$ can be approximately calculated using material loss factors η_e of each element *e* concerning elasticity, share $S_{se}^{(n)}$ of strain energy of each element to total strain energy, material loss factors χ_e of each element *e* concerning effective density and share $S_{ke}^{(n)}$ of kinetic energy of each element to total kinetic energy. The eigen modes in equation (26) are real, which is easily obtained by solving equation (24), which is obtained by ignoring all the damping terms, as real eigenvalue problem. Equation (26) is an extended method of the MSE method, which calculates the modal loss factor of a structure where an elastic and a viscoelastic coexist, and Modal Strain and Kinetic Energy Method (MSKE method), which calculated the modal loss factor of a sound field where porous media and gas coexist [4][11][12].

2.5 DAMPED VIBRATION RESPONSE USING MSKE METHOD

The acceleration response that uses the modal loss factor obtained from equation (26) and the modal parameter obtained from real eigenvalue analysis is represented by the following equation.

$$\{A\} = \sum_{n=1}^{\max} \frac{-\omega^2 \{\phi^{(n)}\}^T \{F\}\{\phi^{(n)}\}}{m^{(n)}\left[\left(\omega^{(n)}\right)^2 - \omega^2 + j\left(\omega^{(n)}\right)^2 \eta_{se}^{(n)} - j\omega^2 \eta_{ke}^{(n)}\right]} \tag{27}$$

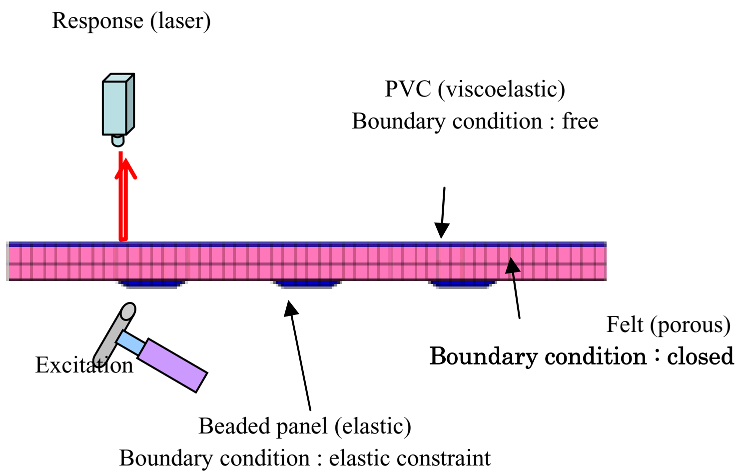
Where, $\{A\}$ is the acceleration vector at the response, $\{F\}$ is the external force vector at the excitation point, $\{\phi^{(n)}\}$ is the *n*'th order mode vector at the excitation point, and $m^{(n)}$ is *n*'th order modal mass.

3. ANALYSIS RESULTS AND TEST VERIFICATION

3.1 DAMPED VIBRATION ANALYSIS AND TEST (ELASTIC + POROUS MEDIA + VISCOELASTIC MATEZRIAL)

A vibration analysis was performed with a FE model. A beaded panel, porous media (felt) and a viscoelastic material (PVC) were laminated to model the automotive panel, as shown Fig. 3. Bead is a groove for reinforcing stiffness and the beads on the panel used in this study are 4 mm high. With a similar test piece, the vibration response (hereafter “response” means acceleration response) was also measured as shown in Fig. 4. The beaded panel was made of 0.7 mm thick steel sheet and constrained by a peripheral jig with bolts. The felt was 20 mm thick. Around the felt was the wall of the jig, which prevented the leakage of air in the felt and closed boundary condition could be assumed. The boundary condition of the surface material (PVC : 1.8 mm thick) was free. The surface material and the felt were adhered. Although the panel and the felt were not adhered, there was no clearance between them. The FE model was modeling using solid elements with the mesh pitch of 10 mm (except for thickness direction). For the boundary condition of the beaded panel, springs were installed in the X, Y, and Z directions to account for stiffness of the jig. The boundary conditions of the sides of the felt were rigid wall in the normal direction and free in the tangential direction. Displacement of the surface material and the particle displacement in the internal air are continuous only in the normal direction towards the boundary surface. This continuous condition is applied to the particle displacement in the air in the felt and the panel displacement.

First of all, the vibration response of the beaded panel was measured without felt and PVC. And the spring constants in the X, Y and Z directions which were set on the boundary of the FE model were identified so that the resonance frequency and the acceleration response agree with the measurement results. Fig. 5 shows the identified results: the calculated value agrees well with the experimental value, far to 500 Hz. Secondly, analysis and measurement were performed for the laminated model shown in the Fig. 3. The material data of the air in the felt was identified by improved two-cavity method [8]. Specifically, the real part of the effective density ρ_{eR}, the imaginary part ρ_{eR}, the real part of the volume elasticity E_{eR} and the imaginary part η_e were respectively set to 2.12kg/m^3, -1.97, 1.15×10^{5}N/m^2, and 0.111. Fig. 6 shows the experiment and analysis results. In the top graph, a measured response at the surface material for the laminates is depicted. The measured response of the beaded panel without felt and PVC is also shown in the same figure. The middle graph compares the analytical results of the same conditions. The analytical response shows smaller damping values than the experimental one, and there were also some discrepancies in their damping effect. The modal loss factor of the analysis did not consider the damping due to friction between the fibers of the porous media and the panel, or between the fibers and the surface material. Therefore, the modal loss factor was adjusted by adding 0.05, and the response was newly calculated. The bottom graph shows the new results. A tendency for difference of the vibration levels (decline in the response level at resonant peaks) is reproduced well. Fig. 7 and Fig. 8 show the vibration modes resulting from the eigenvalue analysis. The three vibration modes, which appear in the case of the panel only, still appear at the laminated panel although the level is lower. In these modes, the surface material and the panel move in-phase. In addition, the surface material and the panel sometimes show many completely different vibration modes, as shown in Fig. 8. This indicates the sound insulation effect of the sound-proof materials.



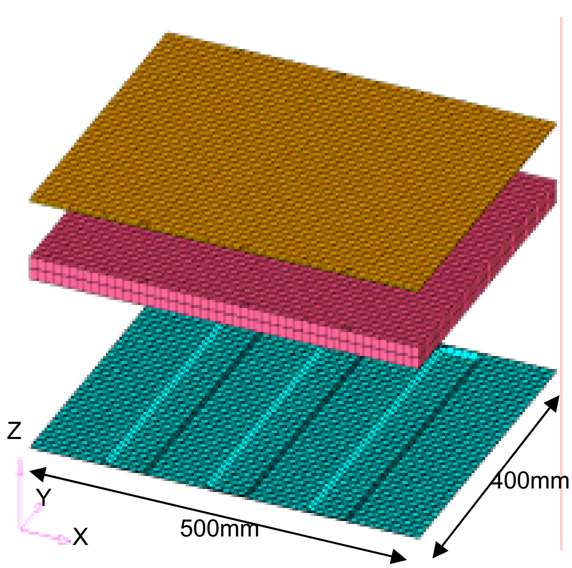


Fig3 FE model of test piece.

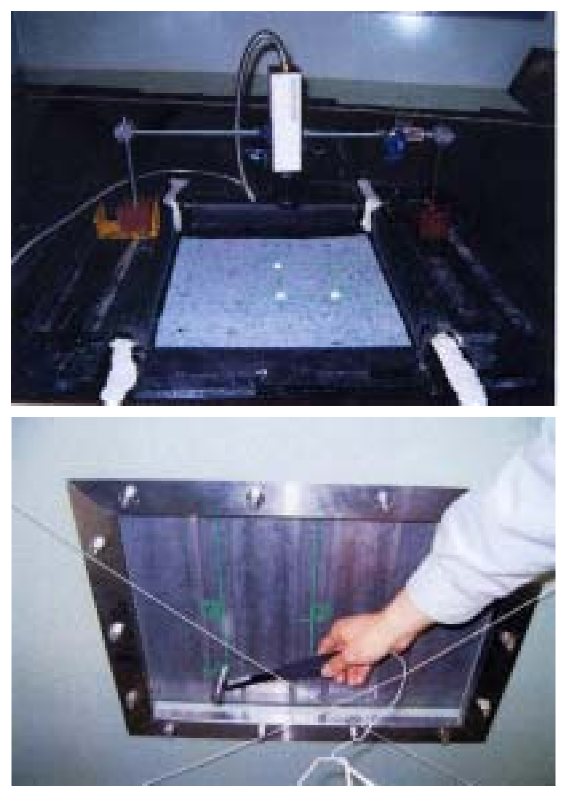


Fig.4 Experimental setup (test piece).

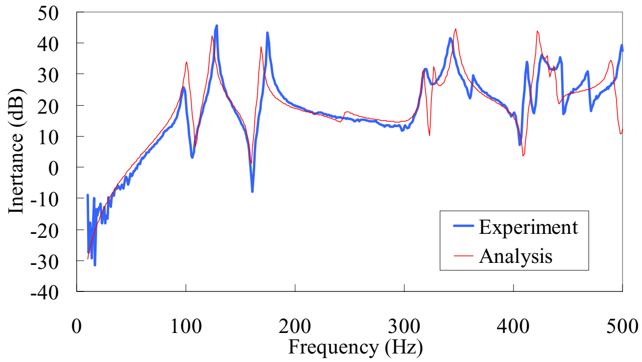


Fig.5 Response level (panel only).

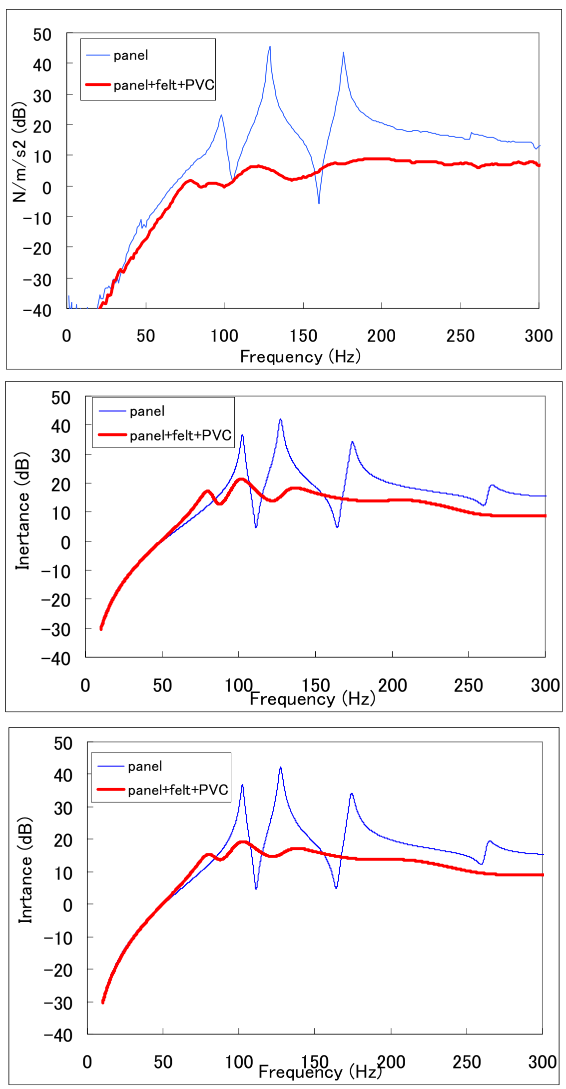


Fig.6 Response level.

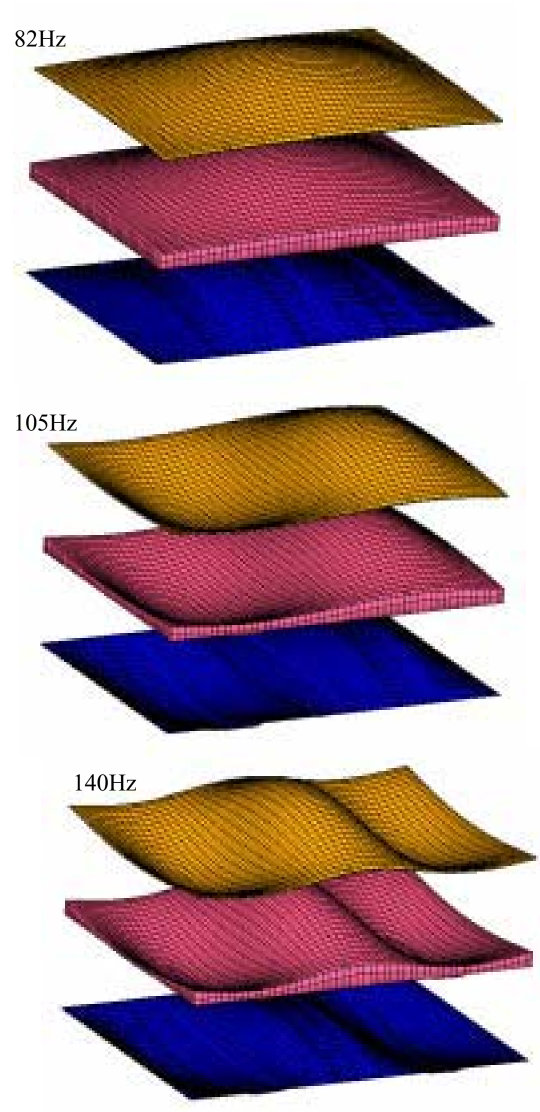


Fig.7 Vibration mode ①

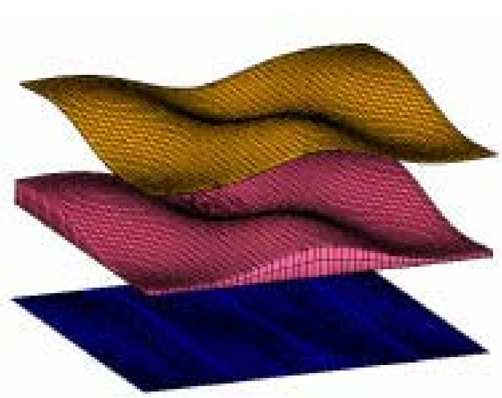


Fig. 8 Vibration mode ②.

3.2 MEASUREMENT OF DAMPED VIBRATION (ELASTIC + POROUS MEDIA + VISCOELASTIC)

Under the test conditions shown in Fig. 4, the vibration response was measured. Fig. 9 shows the test pieces used in the experiment. From the top, beaded panel only (CASE 1); beaded panel and a damping material (CASE 2);. Multi-layered sound-proof structure in which beaded panel, damping material, porous media and surface material (CASE 3) are shown. The test results reproduce fairly well the difference of the vibration levels of the panel and the surface material on an actutal vehicle such as shown in Fig. 2. In addition, the vibration level of each lamination was clarified.

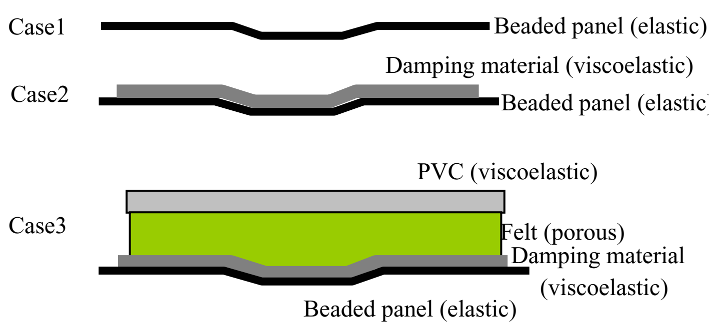


Fig.9 Test piece.

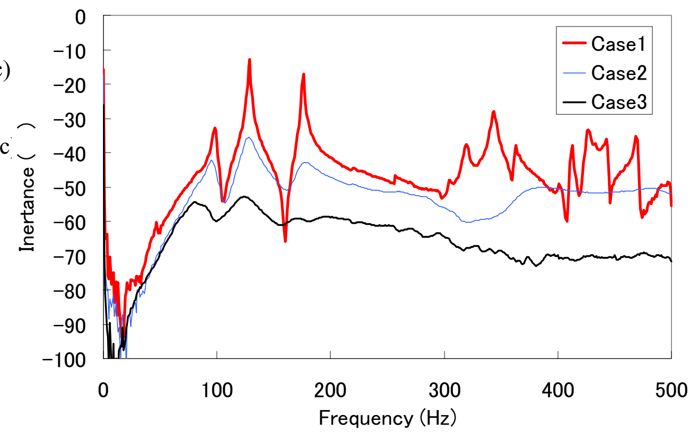


Fig.10 Response level of test piece.

4. CONCLUSION

An analysis method based on the three – dimensional finite element method was proposed for the analyzing the vibration characteristics of the structure in which an elastic, a viscoelastic, porous media, and gas coexist in order to analyze by CAE the vibration damping problem of complicated sound-proof structures used for automotive panels. The obtained results are summarized as follows:

1) Porous media was formulated by the model which expressed the sound field of the internal air with complex effective density and the complex volume elasticity. And the medial was discretized with the element which treats particle displacement as unknown. Elastic and viscoelastic materials were discretized and formulated by the element which treats displacement as unknown. By combining these, the coupled problem where an elastic, a viscoelastic, porous media and gas coexist in any shape was modeled with three – dimensional finite elements and formulated with displacement as common unknown.

2) A Modal Strain and Kinetic Energy Method (MSKE method) was developed to apply the approximate calculation of the modal loss factor by based on the asymptotic method to complicated sound-proof structures. With this method, the modal loss factor was obtained from the results of real eigenvalue analysis. Therefore, the number of calculations required was considerably reduced.

3) A calculation method of the vibration response was developed using the modal loss factor obtained by the MSKE method. An accuracy test was performed with the test pieces which mimic the automotive floor panel, and the sufficient calculation accuracy was confirmed. In addition, test pieces reproduced the damped vibration of the automotive complicated sound-proof structures.

REFERENCE

[1] Zwikker,C.and Kosten,C.,A.,*Sound Absorbing Materials*,(1949),Elsevier Press,Amsterdam.

[2] Kurosawa, Y. and Matsumura,S., Enomoto, H. and Yamaguchi,T. ,High Frequency Vibration Analysis of Automotive Nodies with Panels that Have Attached Viscoelastic Layers,Proceedings of IMECE2003-43839,25-30,2001.

[3] Yamaguchi,T.,Kurosawa,Y.,Sato,N.and Matsumura,S., Vibration Characteristics of Damped Laminates Having Three-dimensional Shapes in Automotive Body Panels, Proceeding of the 17th International Congress on Acoustics, (2001).

[4] Yamaguchi, T., Kurosawa, Y., Matsumura, S. and Nomura,A.,Finite Element Analysis for Vibration Properties of Panels in Car Bodies Having Viscoelastic Damping layer,Transactions of Japan Society of Mechanical Engineers, Vol.69,678C,297-303,2003.

[5] Sato, S., Fujimori, T. and Miura,H., Sound Absorbing Wedge Design Using Flow Resistance of Glass Wool, Journal of the Acoustical Society of Japan,Vol.33,11,628-636,1979.

[6] Ejima,K.,Ishii,T.and Murai,S.,The Modal Analysis on the Acoustic Field, Journal of the Acoustical Society of Japan,Vol.44,6,460-468,1988.

[7] Yuge,K.,Ejima, R., Udagawa,R., Kishikawa,Y. and kasai,K., Sound Insulation Analysis of a resin using Viscoelastc Constitutive Equations,Transactions of Japan Society of Mechanical Engineers, Vol.60,570A, 535-552,1994.

[8] Utsuno,H.,Tanaka.T. and Fujikawa,T., Transfer Function Method for Measuring Characteristic Impedance and Propagation Constant of Porous Materials, Journal of the Acoustical Society of America, Vol.86,2,637-643,1989.

[9] Utsuno,H.,Wu.T.W.,Seybert,A.F. and Tanaka,T., Prediction of Sound Fields in Cavities with Sound Absorbing Materials, AIAA Journal,Vol.28,11,1870-1875,1990.

[10] Utsuno,H., Tanaka,T.,Morisawa,Y. and Yoshimura,T.,Prediction of Normal Sound Absorption Coefficient for Multi Layer Sound Absorbing Materials by Using the Boundary Element method, Transactions of Japan Society of Mechanical Engineers,Vol.56,532C, 3248-3252,1990.

[11] Yamaguchi,T.,Approximated Calculation to Damping Properties of a Closed Sound Field Involving Porous Materials(Proposal of a Fast calculation Procedure for Modal Damping and Damped Response), Transactions of Japan Society of Mechanical Engineers,Vol.66,648C, 2563-2569,2000.

[12] Yamaguchi,T., Kurosawa, Y. and Matsumura,S., Damped Analysis of 3D Acoustic Fields Involving Sound Absorbing Materials using FEM, Transactions of Japan Society of Mechanical Engineers, Vol.66,646C,1842-1848,2000.

[13] Biot,M.A., Theory of Propagation of Elastic Waves in a Fluid-Saturated Porous Solid, Journal of the Acoustical Society of America, Vol.28,2,168-178,1955.

[14] Kang,Y.J.and Bolton,S.,Finite Element Modeling of isotropic Elastic Porous Materials Coupled with Acoustical Finite Elements, Journal of the Acoustical Society of America, Vol.98,1,635-643,1995.

[15] Kagawa,Y., Yamabuchi,T. and Mori,A.,Finite Element Simulation of an Axisymmetric Acoustic Transmission System with a Sound Absorbing Wall, Journal of Sound and Vibration,Vol.53,3, 357-374,1977.

[16] MA,B.A. and HE,J.F.,A Finite Element Analysis of Viscoelastically Damped Sandwich Plates, Journal of Sound and Vibration,Vol.152,1,107-123,1992.