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Dynamic behavior analysis of vehicle acoustic trim using FEM

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Multilayer vehicle acoustic trim containing poroelastic materials affects noise and vibration phenomena not only in the high-frequency range but also in the low and mid-frequency range. However, there are few established technology to create an accurate vehicle model which includes the characteristics of the trim. In this paper, a methodology for the analysis of multilayer acoustic trim within a vehicle FE model is derived based on Biot theory. Using this methodology, dynamic vibration, radiation and absorption behavior of three types of multilayer trim, a conventional isolative type, an absorptive type and an isolative type with absorptive top layer backed by rectangular metal plates are analyzed. The trim model is also applied to a vehicle FE model and its effects on interior sound pressure level are validated with experimental data.

1 Introduction

In the recent years, vehicle NV performances have been subjected more and more to weight and cost reduction. Therefore various investigations are being carried out to find the optimal solutions. One of the required technologies is to balance acoustic characteristics with weight and cost of acoustic trim like a floor carpet and a dash silencer.

Among the applicable CAE technologies to noise and vibration development of a vehicle, FEM (finite element method) is used mainly for the low frequency range below 200 Hz, and SEA (statistical energy analysis) is usually adopted for the high frequency range above 500 Hz. However, both methods are not sufficient for the mid frequency range from 200 Hz to 500 Hz. SEA decreases its accuracy in the mid frequency range because the prerequisite in which vibration modes are excited evenly cannot be satisfied below 500 Hz. On the other hand, when applied to the mid frequency range, FEM demands huge calculation time that increase proportionally to modal density, but this is to be solved in recent years with the improvement of computer power and calculation algorithm. The FEM technology which represent the accurate stiffness, mass and damping of trim parts such as the dash silencer and the floor carpet which has large influence on noise and vibration is essential in order to extend FEM to the road noise and the engine noise issue generated in the mid frequency range. Some studies of porous trim FE modeling have been already reported recently [1,2]. In this paper, based on analysis of the trim behavior and its mechanism, a technique of trim FE modeling is established. Furthermore, a system is developed to calculate this trim FE model that is applied to a conventional vehicle model in practical computational time. The results were compared with experimental data.

2 Theory and its verification of acoustic trim

Figure 1 shows a conceptual vibration mode of the insulative type trim that has the most basic acoustic trim construction. This sound insulative type trim consists of porous material mid-layer such as felt, urethane, etc, and impervious massive surface layer. Beyond the resonance frequency that is determined by combination of the surface layer mass and the porous layer spring, its transmissibility decrease in the mid frequency range. The damping effect of transmissibility is assumed to obtain by airflow friction through porous material.

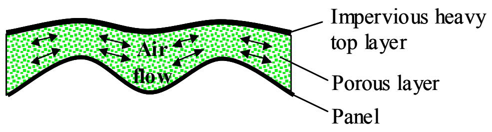

Fig. 1 Vibration mode of an insulative trim

Biot theory [3,4] is the only method published to represent the porous material behavior at this moment. In this theory, the equation of motion (Eq. (1)) has applied to the solid phase of porous material (Fig. 2) where frame is made of fiber or resin, and the wave equation (Eq. (2)) to the fluid phase that has communicating pores of porous material.

$$div\,\underline{\underline{\hat{\sigma}}}^{S}(\underline{u}) + \omega^2 \tilde{\rho}\underline{u} + \tilde{\gamma}\,\underline{\nabla p} = 0 \qquad (1)$$

$$\frac{h^2}{\tilde{\rho}_{22}}\Delta\, p + \omega^2 \frac{h^2}{\tilde{R}}\, p - \omega^2 \tilde{\gamma}\; div\,\underline{u} = 0 \qquad (2)$$

where $\underline{\underline{\hat{\sigma}}}^{S}$ is stress tensor of solid phase, $\tilde{\rho}$ solid phase density, $\tilde{\gamma}$ coupling factor between solid phase and fluid phase, $\tilde{\rho}_{22}$ fluid phase density, $\tilde{R}$ bulk modulus, h porosity, $\underline{u}$ solid phase displacement, p the fluid phase sound pressure and ω is angular frequency, respectively.

The first term of the left-hand side of Eq. (1) represents restoring force due to stiffness that includes structural damping, the second term inertial forces due to mass that includes coupling force with the fluid phase and the third term does coupling force with the fluid phase. Eq. (2) is derived from the equation of the equilibrium and the constitution equation of the fluid phase. The factor of the first term in the left-hand side stands for mass that includes the viscous damping and coupling inertia with the solid phase, the factor of the second term for bulk modulus which includes the attenuation with thermal loss and the third term does for the coupling force with the solid phase. The factors of each term of both equations are frequency dependent complex number because of the damping characteristic of porous material.

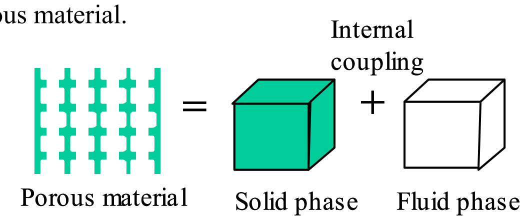

Fig. 2 A schematic view of Biot theory modeling

To prove the effectiveness of the theory, a Biot theory based trim FE model was built and verified. Figure 3 shows the panel excitation test setup with insulative type trim (Fig. 3). The result is shown in Fig. 4. The vibration of the panel surface and the trim surface shows good correlation up to 400 Hz and reveals this theory is effective.

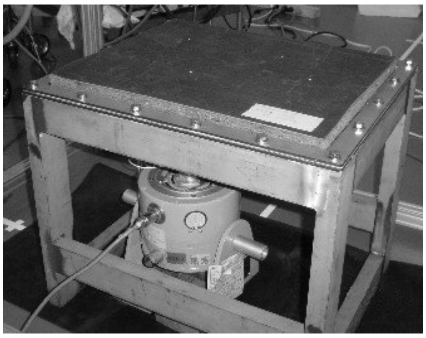

Fig. 3 Experimental setup

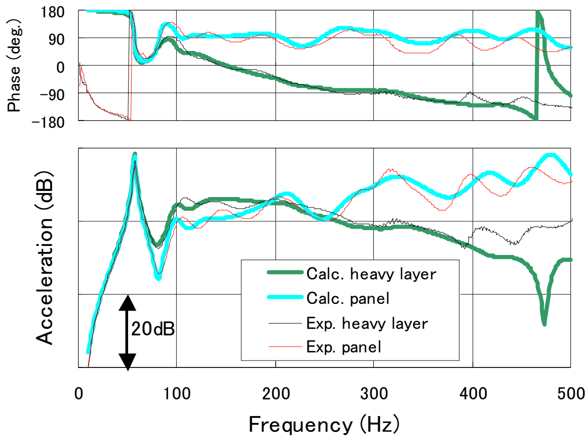

Fig. 4 Validation of Biot theory

3 Various constructions of acoustic trim

In the recent years, two more types of acoustic trims are used widely in vehicle. The one is called absorptive type and the other called absorptive and insulative type in addition to the insulative type used in the previous chapter. Their layered structures are shown in Fig. 5.

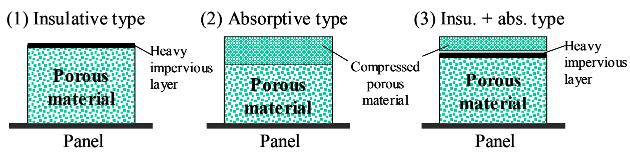

Fig. 5 Porous trim constructions

The absorptive type (2) has a layered structure where the mass surface layer of the insulative type (1) is replaced with compressed porous layer. In this absorptive type, high absorption performance is more expected than insulative one. And this is lighter in weight than insulative type because it does not have mass surface layer.

The absorptive and insulative type (3) has a layered structure where there is additional porous layer on the top of the insulative type (1). Its sound-absorbing effect in the high frequency range is strengthened more than that of original insulative type.

4 Dynamic behavior analysis of trims

The aforementioned trim FE modeling method established in the previous chapter is applied to each trim construction and their dynamic behaviors are analyzed and verified with the test. Figure 6 shows the model configuration. The trim of each type is laid on the 10 mm thick and 500 mm x 400 mm size aluminum panel. The plate edges are subjected to vertical excitation and the radiated sound pressure is calculated at a position of 20 mm upper from the surface center.

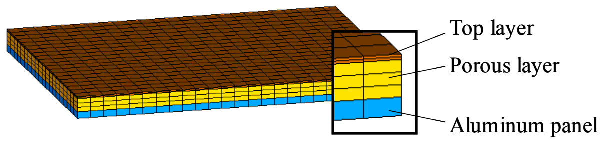

Fig. 6 Porous trim FE model

4.1 Sound-radiation characteristic

The calculated emission sound is showed in Fig. 7. The sound pressure peak around 350 Hz is caused by resonance of the base panel. The SPL of the sound insulative type (1) and the absorptive and insulative type (3) rise together around 150 Hz compared with the bare panel condition and it is obvious that spring mass resonance of the trim occurs at this frequency. Moreover, the SPL decreases above 200 Hz because of their insulating effect of vibration transmission in the frequency range that is sufficiently higher than spring mass resonance. On the contrary, the absorptive type (2) does not show definite spring mass resonance and decreasing effects of vibration level on higher frequency range is little. However, because the phase angles of all types have changed slowly according to the frequency as shown in Fig. 7, it is easily understood that the damping of the porous material is working.

Because of its surface layer permeability in the absorptive type (2) shown in Fig. 8, the air particle vibration occurs in this layer. Usually, high flow resistivity compressed felt layer is used for this porous surface layer and it generates the damping effect caused by viscous friction. Figure 9 shows the vertical acceleration response of the solid phase and the fluid phase of the compressed felt surface layer of this absorptive type trim respectively with comparison of the bare base panel response. It can be presumed that the high damping occurs around the base panel resonance at 350 Hz according to the vibration level difference of the solid phase and the fluid phase in the surface layer, and therefore, the vibration level of the base panel is decreased. On the same time, the vibration level of the surface fluid phase that has high correlation with the radiated sound is almost equal to that of the base panel. But their phase angle differs around 90 degree, and the phase angle of radiated sound changes according to that of the air phase.

It is usually impossible to represent this mechanism by using conventional elastic FEM element and acoustic FEM element. Therefore, introduction of the porous material FEM element that utilizes Biot theory is indispensable.

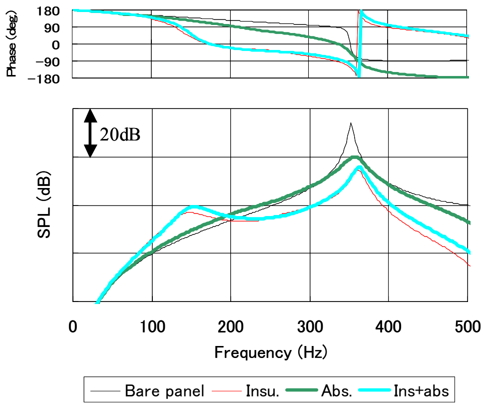

Fig. 7 Sound radiation of porous trims

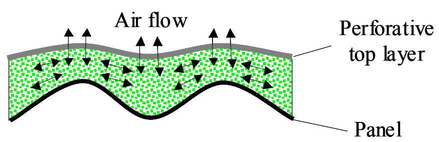

Fig. 8 Vibration mode of absorptive trim

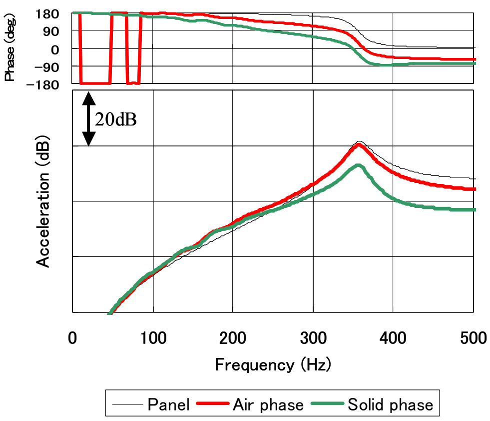

Fig. 9 Vibration of solid and air phase

4.2 Sound-absorbing characteristic

In order to affirm sound-absorbing characteristics, a FE sound field is set on the surface of the model in Fig. 6 and SPLs are calculated with acoustic excitation. The results are shown in Fig. 10. The absorptive type (2) has the sound-absorbing effect at the sound field resonance frequencies that have SPL peaks in the figure even in this low and mid frequency range. Contrary, such as the insulative type (1) and the absorptive and insulative type (3), have little sound absorbing effect. The absorptive type can obtain high sound absorbing effect because it has air transmittable surface layer as shown in Fig. 8 and therefore the fluid phase vibration of this trim reaches as far as the base panel and yields high damping effect caused by the high flow resistivity of the porous surface layer.

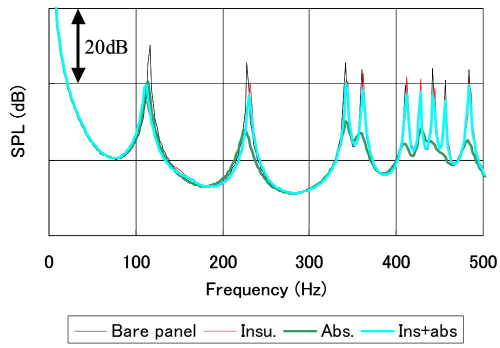

Fig. 10 Sound absorption of porous trims

5 Development of FEM system for acoustic trim

In the vehicle development using NVH-FEM, modal method is employed usually to reduce the computational time. Therefore, an FEM system is developed to insert an acoustic trim matrix into a vehicle modal matrix. Figure 11 shows an overview of the system procedure. This system is constituted of trim elements generation module, interface with commercial solver and coupled equation solver module.

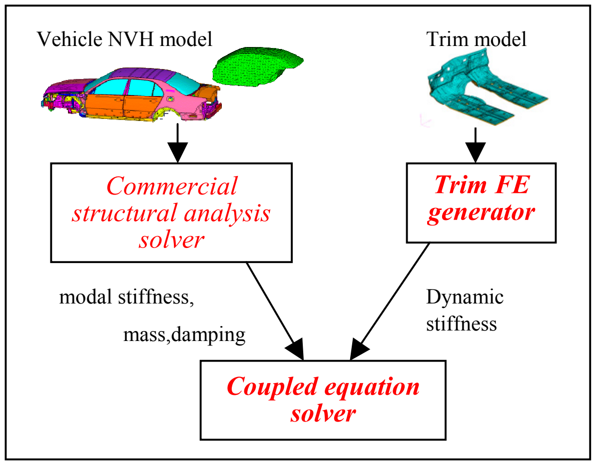

Fig. 11 Overview of trim FEM analysis system

In the trim elements generation module, the elastic surface layer and porous middle layer are discretized by isoperimetric structural elements and porous elements in which internal coupling between solid phase and fluid phase based on Biot theory are accomplished.

The characteristics of a body structure and a cabin air are obtained by commercial FEM solver and outputted in terms of stiffness, mass, damping and coupling matrixes. Equation solution module solves the equation of steady state frequency response of system (Eq. (3)) made of combination of structure-acoustic and trim matrices.

$$Z(\omega)u = \left[K + i\omega C - \omega^2 M\right]u = f \tag{3}$$

where $Z(\omega)$ is dynamic stiffness matrix, K , C , M , u , f are stiffness matrix, damping matrix, mass matrix, displacement (sound, pressure) vector, input force respectively, and i is the imaginary unit.

For a vehicle model that has over one million of physical DOF, direct solution of Eq. (3) incurs high cost of calculation. Also, calculation of acoustic trim stiffness matrix and mass matrix that have frequency dependent characteristics at every frequency of interest makes high computational cost drastically. Therefore in this system, to achieve reduction of the computational cost, a methodology that is explained bellow is developed.

First, dynamic stiffness matrix $Z(\omega)$ is separated into general part $Z_0(\omega)$ and acoustic trim part $Z_t(\omega)$. And reduction of DOF by normal mode method is applied.

$$Z(\omega) = Z_0(\omega) + Z_t(\omega) \tag{4}$$

$$u = \Phi\xi \tag{5}$$

where

$$\Phi = \begin{bmatrix} \Phi_s & 0 & 0 & 0 \\ 0 & I & 0 & 0 \\ 0 & 0 & I & 0 \\ 0 & 0 & 0 & \Phi_f \end{bmatrix}, \quad \xi = \begin{Bmatrix} \xi_s \\ u_t \\ p_t / i\omega \\ \xi_f / i\omega \end{Bmatrix}.$$

Φ_s and Φ_f are eigen-mode matrix of structure and cabin fluid respectively, ξ_s and ξ_f are corresponding modal DOFs, and I is a unit matrix. Furthermore, reduced dynamic stiffness matrix $\bar{Z}_0(\omega)$ is derived from modal decomposition using transfer matrix Φ .

$$\bar{Z}_0(\omega) = \Phi^T Z_0(\omega)\Phi \tag{6}$$

On the other hand, to reduce the computational cost, the dynamic stiffness of trim part is to be calculated at the sampled frequency rather than all interested frequency. Then the dynamic stiffness matrices are obtained by interpolation between each sampling frequency. Moreover, the DOF reduction and elimination are carried out as follows.

The dynamic stiffness matrix of acoustic trim $Z_t(\omega_j)$ derived at each sampling frequency j is reduced into modal coordinate.

$$\bar{Z}_t(\omega_j) = \Phi^T Z_t(\omega_j)\Phi \tag{7}$$

The DOFs of acoustic trim are reduced only at the part of connection with body structure and cabin air and the other parts are remaining as physical coordinate.

Then, remaining physical DOFs of acoustic trim should be eliminated. General part of reduced dynamic stiffness $\bar{Z}_0(\omega)$ is eliminated because concerned terms are zero. On the other hand, concerned DOFs of the reduced dynamic stiffness of acoustic trim $\bar{Z}_t(\omega_j)$ at sampling frequencies are eliminated by Gaussian reduction and then Schur decomposition matrix $S(\omega_j)$ is derived. This matrix can be understood contributions of acoustic trim added on modal dynamic stiffness matrix.

$$\bar{Z}_t(\omega_j) = \begin{bmatrix} Z_n & Z_{nb} \\ Z_{bn} & Z_b \end{bmatrix} \tag{8}$$

$$S(\omega_j) = Z_n - Z_{nb} Z_b^{-1} Z_{bn} \tag{9}$$

Finally, $S(\omega)$ is derived form interpolation of $S(\omega_j)$ and it will be added to modal dynamic stiffness $\bar{Z}_0(\omega)$, then Eq. (10) is solved.

$$\left[\bar{Z}_0(\omega) + S(\omega)\right]\begin{Bmatrix} \xi_s \\ \xi_f / i\omega \end{Bmatrix} = \begin{Bmatrix} \Phi_s^T f_s \\ -\Phi_f^T f_f / i\omega \end{Bmatrix} \tag{10}$$

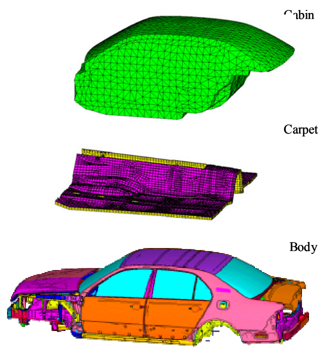

Fig.12 FE model for varidation

This methodology is applied to noise and vibration problem consisting of body structure, cabin air and carpet model showed in Fig. 12. The calculation time is about 23 hour using single 2.2 GHz CPU. To reduce the calculation time moreover, parallel computing with 4 CPU is introduced for Schur matrix $S(\omega_j)$ calculations, and then reasonable calculation time is achieved (Fig. 13).

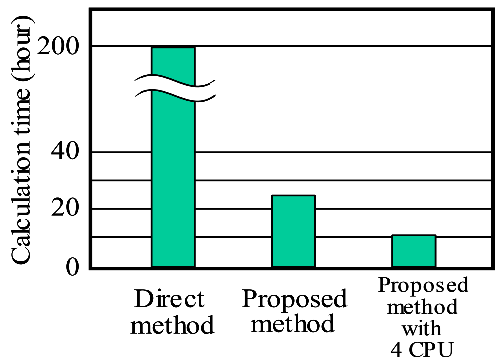

Fig.13 Comparison of calculation time

6 Verification of vehicle model

The developed trim FE model is applied to the vehicle model and the accuracy is confirmed. The model dealt in this chapter is a floor cut body from whole BIW (Fig. 14) because this FE model is more precise than that of full BIW FE model. To verify the accuracy of FE model, a SPL at a point in the cabin is calculated with point force excited at floor member and compared with experimental data. The result of bare floor panel is showed in Fig. 15 and agrees well with the experimental data.

Next, carpet FE model of insulative type and absorptive type were installed on the bare floor. The results are shown in Fig. 16 and Fig. 17 respectively. Because both results show good correlation from low frequency to mid frequency, it is confirmed that the developed system has good accuracy.

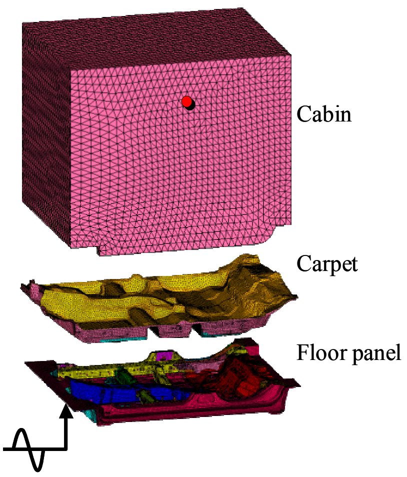

Fig.14 Floor panel with carpet FE model

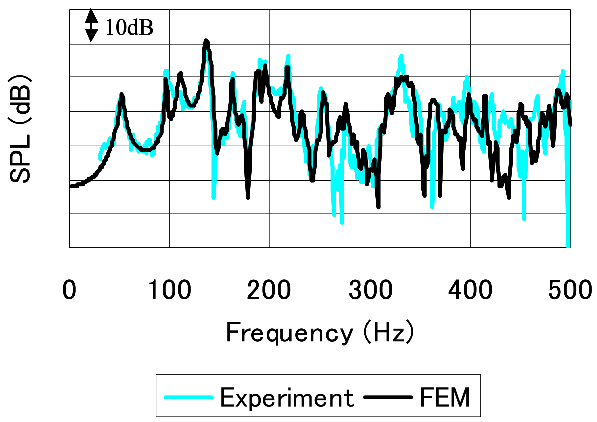

Fig.15 Validation of floor FE model (without carpet)

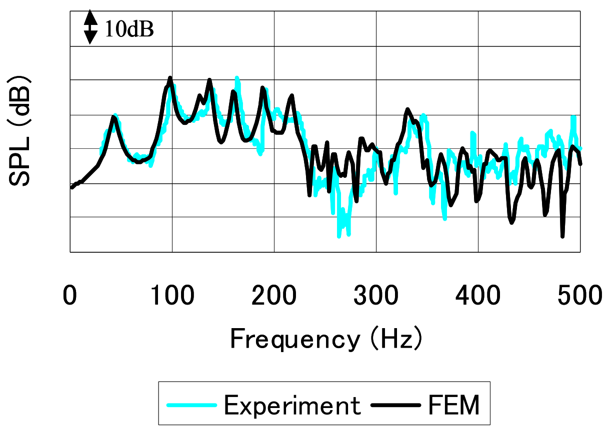

Fig.16 Validation of floor FE model (Insulative type)

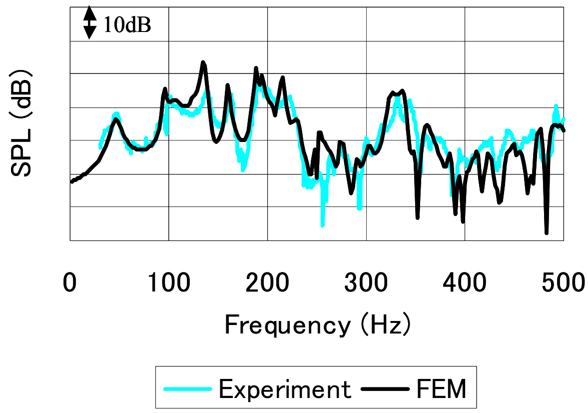

Fig.17 Validation of floor FE model (Absorptive type)

7 Conclusion

The dynamic behavior of the three types of acoustic trim including sound insulative, absorptive and absorptive and insulative type is analyzed by using developed trim FE model.

The system is developed to calculate a vehicle FE model with trim FE model in practical CPU time using modal decomposition and DOF reduction of trim matrix. The system is applied to the vehicle model and shows good correlation with experimental results.

Acknowledgments

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