

FEM System Development for Dynamic Response Analysis of Acoustic Trim

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ABSTRACT

The multilayer vehicle trim is well known for its effective influence upon noise and vibration characteristics not only in the high-frequency range but also in the low and mid-frequency ranges. FEM technologies which represent the accurate stiffness, mass and damping of trim parts such as the dash silencer and the floor carpet are essential in order to extend current body FEM capability to the road noise and the engine noise issues generated in the mid-frequency range. Conventional modeling methodologies such as local impedance and/or spring-mass modeling that express absorption and insulation properties of acoustic trim contain limitations in the mid-frequency range. There are few reliable FEM technologies to create practical vehicle models that represent the precise characteristics of the trim. In this paper, poroelastic modeling of acoustic multilayer trim was established by employing Biot theory. The trim model effectiveness proposed in this paper was confirmed by applying it to various types of trim; for example conventional isolative type trim and absorptive one. But the trim model usually demands direct frequency response analysis that consumes excessive computation time. Therefore an FEM system was developed to reduce this computation time by using reduction of degree of freedom (DOF) of trim model, modal decomposition of structure attached to trim model, and parallel processing. The system was applied to a conventional vehicle model and a vehicle body cut out panel and showed good correlation with experimental data.

INTRODUCTION

In recent years, vehicle NV performance has been conflicted more and more with weight and cost

reduction. Therefore various investigations have been carried out to find the optimal solutions. One of the required technologies is to balance acoustic characteristics with weight and cost of acoustic trim such as a floor carpet and a dash silencer. Among the current applicable CAE technologies for developing the noise and vibration attributes 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, neither method is sufficient for the mid frequency range from 200 Hz to 500 Hz. SEA decreases in accuracy in the mid frequency range because its requirement for even excitation of vibration modes cannot be satisfied below 500 Hz. On the other hand, when FEM is applied to the mid frequency range, it requires huge amount of calculation time, which increases proportionally to modal density. However, this has been solved in recent years with increases in computer power and improvement of calculation algorithms. Nevertheless, FEM technology for representing the accurate stiffness, mass, and damping of trim parts such as a dash silencer and a floor carpet, which have a large influence on noise and vibration, is essential in order to extend the application range of FEM to the road noise and the engine noise issues generated in the mid frequency range. Some studies of porous trim FE modeling have been reported recently [1,2]. This paper describes a technique of trim FE modeling based on analysis of trim behavior and its mechanism. Furthermore, a system is developed to calculate this trim FE model applied to a conventional vehicle model in practical computational time. The results of the system were compared with experimental data.

THEORY AND VERIFICATION WITH ACOUSTIC TRIM FE MODEL

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

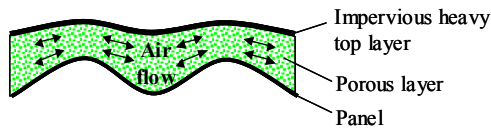


Fig. 1 Vibration mode of an insulative trim

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

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

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

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

The first term of the left-hand side of Eq. (1) represents the restoring force due to stiffness that includes structural damping, the second term represents the inertial forces due to mass that includes coupling force with the fluid phase, and the third term represents the coupling force with the fluid phase. Eq. (2) is derived from the equation of the equilibrium and the constitutive equation of the fluid phase. The factor of the first term in

the left-hand side represents the mass that includes the viscous damping and coupling inertia with the solid phase, the factor of the second term represents the bulk modulus that includes the attenuation due to thermal loss, and the third term represents the coupling force with the solid phase. The factors of each term of both equations are frequency dependent complex numbers because of the damping characteristics of the porous material.

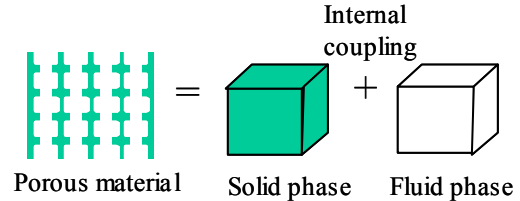


Fig. 2 A schematic view of Biot theory modeling

To prove the effectiveness of the theory, a trim model based on Biot theory was built (Fig. 3) and verified. Figure 4 shows the panel excitation test setup used for the validation. The results are shown in Fig. 5. The vibration of the panel surface and the trim surface shows good correlation up to 400 Hz, which indicates that this theory is effective.

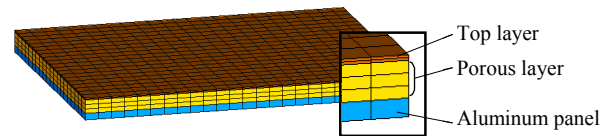


Fig. 3 Porous trim FE model

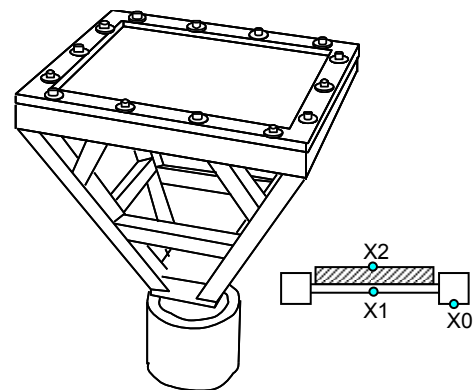
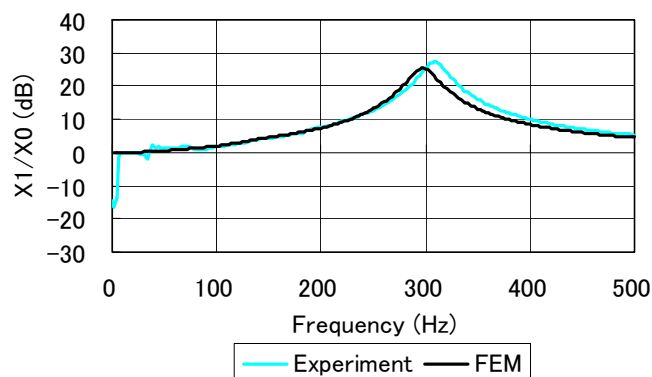
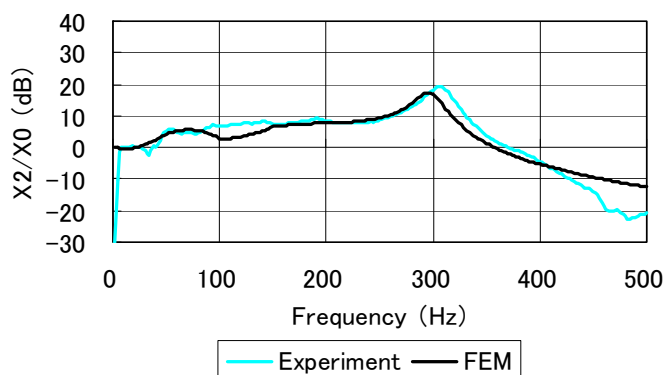


Fig. 4 Experimental setup



(a) Acceleration ratio of the center of panel surface with trim to frame



(b) Acceleration ratio of the center of top layer surface of trim to frame

Fig. 5 Validation of Biot theory

VARIOUS STRUCTURES OF ACOUSTIC TRIM

In recent years, two types of acoustic trims have been used commonly in vehicles in addition to the insulative type used in the previous chapter. One is called the absorptive type and the other is called the absorptive and insulative type. Their layered structures are shown in Fig. 6.

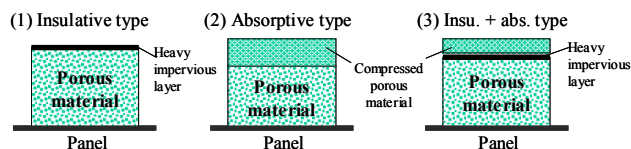


Fig. 6 Porous trim constructions

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

The absorptive and insulative type (3) has a layered structure with an additional porous layer on the top of

insulative type (1). Its sound-absorbing effect in the high frequency range is stronger than the original insulative type.

DYNAMIC BEHAVIOR ANALYSIS OF TRIMS

The trim FE modeling method established in the previous chapter was applied to each trim structure and their dynamic behaviors were analyzed. The FE model shown in Fig. 3 was used for this analysis. The trim of each type was laid on a 10 mm thick, 500 mm x 400 mm aluminum panel. The plate edges were subjected to vertical excitation and the radiated sound pressure was calculated at a position 20 mm above the surface center. The calculated sound pressure is shown in Fig. 7.

Sound-radiation characteristics

The sound pressure peak around 350 Hz is caused by resonance of the base panel. The SPL of sound insulative type (1) and 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 the insulating effect with regard to vibration transmission in the frequency range that is sufficiently higher than the spring mass resonance. On the contrary, absorptive type (2) does not show definite spring mass resonance and it has little vibration reduction effect in higher frequency ranges. However, because the phase angles of all types change slowly according to the frequency as shown in Fig. 7, it is easily understood that the damping of the porous material is working.

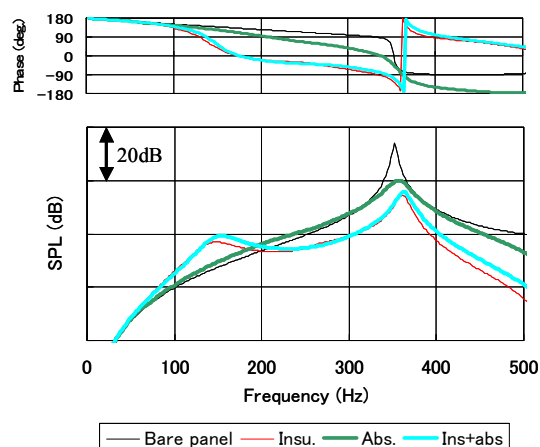


Fig. 7 Radiated sound pressure from trims

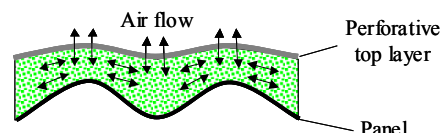


Fig. 8 Vibration mode of absorptive trim