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LETTERS TO THE EDITOR

STRUCTURAL-ACOUSTIC MODAL COUPLING ANALYSIS AND APPLICATION TO NOISE REDUCTION IN A VEHICLE PASSENGER COMPARTMENT

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A practical method for noise reduction is proposed and is applied to the interior noise problem of a vehicle passenger compartment. The proposed method is based upon the structural–acoustic response model, in which the interior pressure is explicitly described in terms of the modal parameters and structural-acoustic modal coupling coefficients of the car body and compartment system. Considering only a few modes and modal coupling coefficients which have large contributions, the cause of the noise peak can be easily identified and the reduction procedure can be simplified compared with the conventional finite element analysis method. In addition, the use of experimental data reduces some inevitable errors in the numerical analysis and refines the numerical predictions in noise problems. These procedures are carried out by using a user-friendly computer program (ACSTAP) which has been developed during the course of this study.

1. INTRODUCTION

In the low-frequency range below 200 Hz, car interior noise is determined by system modal characteristics such as acoustic resonances, body vibration modes and structural-acoustic coupling characteristics, as well as the property of noise and vibration sources. When the proper design for low vibration and low noise is not prepared, severe noise problems like booming can occur, and the modification of the body structure is difficult once the model is manufactured. To avoid this problem, detailed analysis on the body compartment is required in the design stage and sufficient tests should be performed on the prototype car. Finite-element analyses have been performed using commercial programs such as MSC/

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NASTRAN, ANSYS, ABAQUS, in which vibration response of the car body is first analyzed and then pressure response is calculated [1–3], and for the structural modification, noise contribution of boundary panels or acoustic sensitivity of the design variable was introduced [4–6]. To get meaningful analysis result, however, it is very important to obtain the structural and acoustic modal parameters of the vehicle system. In real cars, however, body structure has tens of thousands or hundreds of thousands degrees of freedom in the finite element model and structure-borne noise is related to the high-frequency flexural modes of the car body. Furthermore, various damping elements render the acoustic response complicated. *This situation has made design engineers feel the* limitation of numerical analysis for the vehicle noise problem. *Authors* have introduced a noise analysis and reduction scheme assisted by experiments, in which the experimental modal data were effectively used in the numerical analysis. The efficiency of the analysis method has been verified by a simplified model of a vehicle structure [7]. As a continuing study, this paper shows an application of the method to a real car. All the analyses are carried out by using the special-purpose program (ACSTAP) which is developed for this study.

2. THEORETICAL CONSIDERATIONS

In a previous study, interior pressure $\{P\}$ has been described by the following analysis model [7]:

$$\{P\} = \omega^2 \left[\sum_j \frac{\{\phi_j\}^{\mathrm{T}}\{f_s\} \sum_i \{\phi_i\} C'_{ij}}{\{(\omega_{sj}^2 - \omega^2) + j\omega_{sj}^2 \zeta_{sj}\}} - \sum_i \frac{\{\phi_j\} \sum_j \{\phi_j\}^{\mathrm{T}}\{f_s\} C'_{ij}}{\{(\omega_{ai}^2 - \omega^2) + j\omega_{ai}^2 \zeta_{ai}\}} \right], \tag{1}$$

where the structural-acoustic modal coupling coefficient is defined as

$$C'_{ij} = \frac{\rho_a \{\phi_i\}^{\mathrm{T}} \int_s [N_a]^{\mathrm{T}} [N_s]' \,\mathrm{d}s \{\psi_j\}}{(1 + j\zeta_{sj})\omega_{sj}^2 - (1 + j\zeta_{ai})\omega_{ai}^2}. \tag{2}$$

Equation (1) explicitly explains the relation between pressure response and excitation condition, measurement position and modal parameters of the structural-acoustic system. The equation shows that strongly excited structural modes generate noise peaks and the level is determined by the structural-acoustic modal coupling coefficient as well as damping factors. In the analysis model, modal coupling coefficients play the role of natural parameters like natural frequency and mode. Usually, only a few structural modes or acoustic modes are involved in the standing wave problem like booming, and therefore the acoustic response analysis can be simplified by considering those modes within the frequency band of interest. To reduce noise level, large coupling coefficients should be reduced by proper structural modification, and contribution of the boundary panel to the large coupling coefficients is required. The contribution of kth boundary panel to C'_{ij} is defined as follows:

$$B_{ij}^k = \frac{C_{ij}'^k}{C'_{ij}}, \tag{3}$$

where C'^k_{ij} is the value calculated on the kth element of boundary panel. Configuration of B^k_{ij} is often very similar to that of panel sensitivity on the total noise, when a particular or small number of modes are involved in the noise peak as in booming [7, 8]. Panel contribution to the coupling coefficient B^k_{ij} is not dependent on the measurement or the excitation position. The use of B^k_{ij}, therefore, is useful for the noise problem, in which the excitation condition is too complex. The effectiveness has been verified on a simplified vehicle compartment model in previous research [7].

3. ACOUSTIC MODES AND STRUCTURAL MODES OF THE TEST CAR

3.1. ACOUSTIC MODES

For the medium-size test car used in this study, eight acoustic modes are calculated and measured below 200 Hz as shown in Table 1. Figure 1(a) shows the speaker excitation device for the measurement of acoustic mode. Under each acoustic resonance state, pressure distribution is measured by moving a microphone in the compartment and the nodal plane of each acoustic mode is obtained. Figure 1(b) is the finite element analysis model for the acoustic modes. Figure 1(c) shows pressure nodal planes of each acoustic modes. The first acoustic mode is uniform pressure mode and is usually excluded in the analysis. The nodal

TABLE 1

Structural and acoustic natural frequencies

Structural natural frequency (Hz)		Acoustic natural frequency (Hz)		
Mode number	Experiment	Mode number	FEM (ANSYS)	Experiment
1	25·1	1	0	
2	31·2			
3	41·6			
4	52·5			
5	64·2			
6	81·9	2	85·3	84
7	93·9			
8	97·2			
9	99·3			
10	104·7	3	113·2	110
11	117·2			
12	126·4			
13	139·5	4	144·9	142
14	148.9	5	151.7	148
15	166·5	6	154·9	
		7	192·3	
		8	194·1	

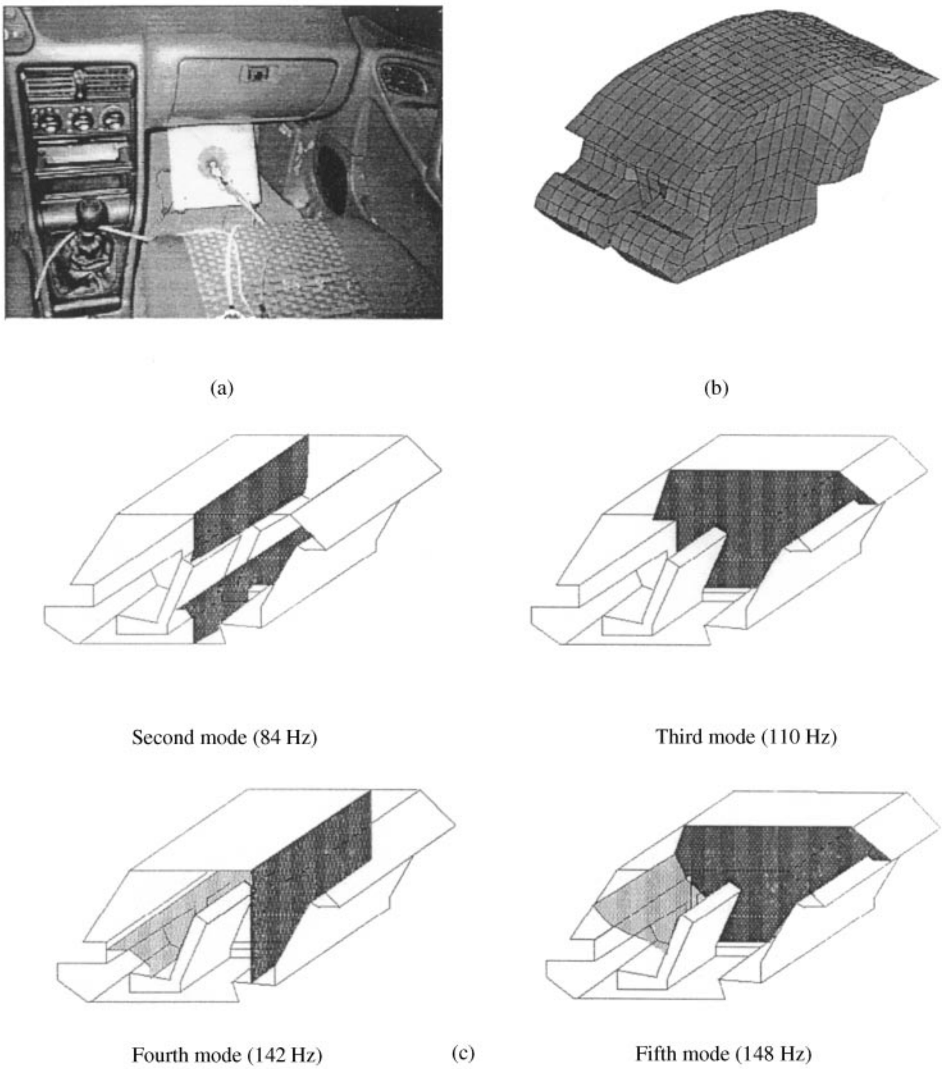


Figure 1. Acoustic nodal planes of the compartment cavity, (a) acoustic mode measurement by speaker excitation, (b) finite element analysis model, (c) nodal planes of acoustic modes by measurement.

plane of the second acoustic mode passes around the driver's ear position, which is desirable to avoid booming by the acoustic mode. Analysis results by ANSYS are in good agreement with the measured data in frequencies and modes. In the analysis, existence of the seats increased the effective length of the compartment cavity, which resulted in the decrease of the acoustic natural frequencies. In the finite element analysis, all the surfaces including seat and trim were modelled as rigid boundary for convenience. Impedance boundary did not have a significant influence on the values of natural frequencies or the position of nodal lines of the acoustic modes, while the mass density of seats could change the acoustic frequencies and modes. Therefore, it is practical to calculate the acoustic modes with rigid boundary and then to reflect the absorbent effect on the pressure level by using measured damping factors. Effect of the acoustic damping by the absorbent surface has been investigated in detailed in *another study* by the authors [9]. It is also interesting that acoustic frequencies and modes sometimes show quite different results from measured data when the compartment cavity is strongly coupled with the vibration of the roof, side door or when it is coupled with the trunk cavity through the rear seat. A detailed study by the authors is to be reported in other papers.

3.2. STRUCTURAL MODES

For the body vibration modes, going up to the high-frequency range, there is usually a large difference between calculated values and measured data. This limits the application of the finite element analysis to the noise problems. In this study, it was possible to enhance the accuracy of the analysis by using experimental modal data. To generate both symmetric and antisymmetric modes, a test car is excited below the front right-hand side member assembly as shown in Figure 2(a). Figure 2(b) shows the measurement points in a modal test. Acceleration is measured at about 200 points on the compartment wall including window and side doors. The measured modal data are transformed to the format compatible with the coupling analysis. The normal component of the modal displacement is required to investigate the structural acoustic interaction. In real cars, however, trim makes the measurement on the inside surface very difficult. Measurement in this study is performed on the outside surface of roof, floor, doors, dash panel and glass. Detailed effects of the trim on the coupling between body vibration and cavity acoustics is now under study by the authors. In the test, about 20 structural modes are measured below 200 Hz as shown in Table 1. Figure 3 shows the typical structural modes of the test car which will be involved in the strong coupling with acoustic modes and will generate noise peaks.

4. STRUCTURAL-ACOUSTIC MODAL COUPLING ANALYSIS

In equation (1), level of resonance noise is determined by the excitation condition, damping and coupling coefficients. Structural-acoustic modal coupling coefficients are calculated by equation (2) to find dominant structural and acoustic modes involved in noise peak. Figure 4 is the plot which compares the magnitude of the coupling coefficients between eight acoustic modes and 15 structural modes. In the figure, C'_{26}, $C'_{4\,13}$, $C'_{4\,14}$, $C'_{5\,14}$ and $C'_{6\,14}$ show large values compared with other coupling coefficients. This means that the second, fourth and fifth acoustic modes are well coupled with the sixth, 13th and 14th car body structural modes. The fourth and the fifth acoustic modes show especially strong couplings with the 13th and the 14th structural modes, since their frequencies are close to each other. These modes can contribute significantly to the interior noise, if they are excited.

5. NOISE REDUCTION SCHEME

To reduce the dominant coupling coefficients, the contributions of boundary panels to the large coefficients are investigated. Boundary contributions to C'_{26},

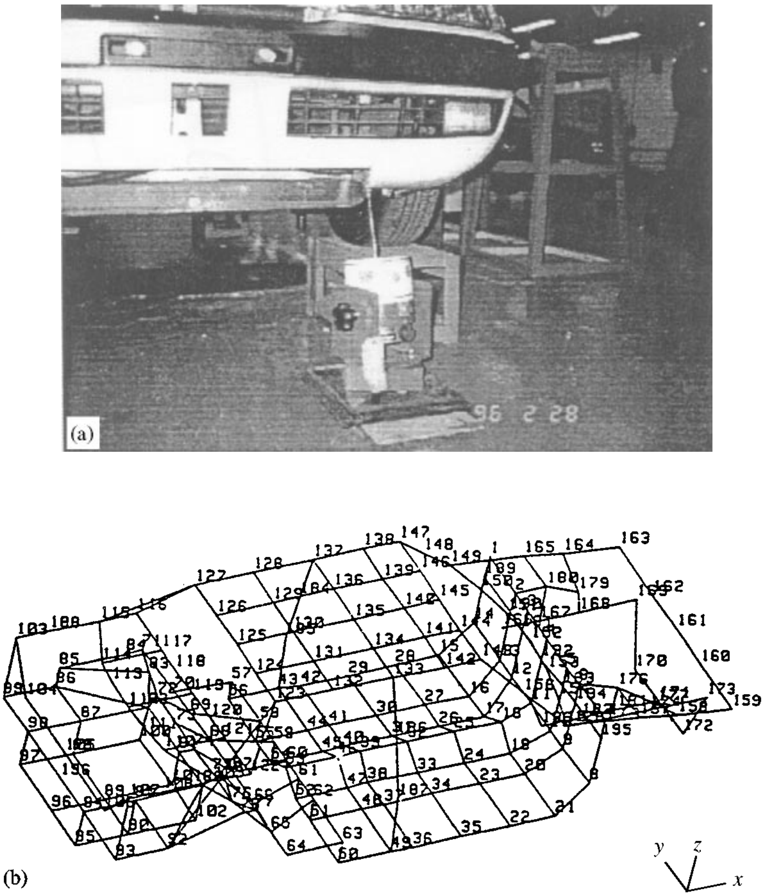


Figure 2. Excitation and measurement points in modal test. (a) excitation in modal testing, (b) measurement points.

$C'_{4\,13}$ and $C'_{4\,13}$ are plotted in Figure 5, in which darkness means the relative magnitude of the contribution. Front floor panel commonly shows large contribution to the dominant coupling coefficients. This position is also proper to increase modal damping since it shows large modal deformation in the involved structural modes in Figure 3. Damping treatment on that position is attempted to increase the modal damping factors involved in noise peaks and simultaneously, to decrease the dominant C'_{ij}. It was confirmed that stiffness modification required rather complex sensitivity analysis and produced noise peak at another speed by a frequency shift. Figure 6 shows the damping treatment on the front floor. Commercial damping sheet (asphalt pad) is attached on the front floor which was the most sensitive to the dominant coupling coefficients as shown in Figure 5. Increase of the structural damping and reduction of large C'_{ij} lessen the resonance effect by the involved modes and result in *a drop in* noise level. In Figure 7, compared with Figure 4, significant reduction is observed in the large coupling coefficients after damping treatment on the sensitive area. Some experiments are performed on the

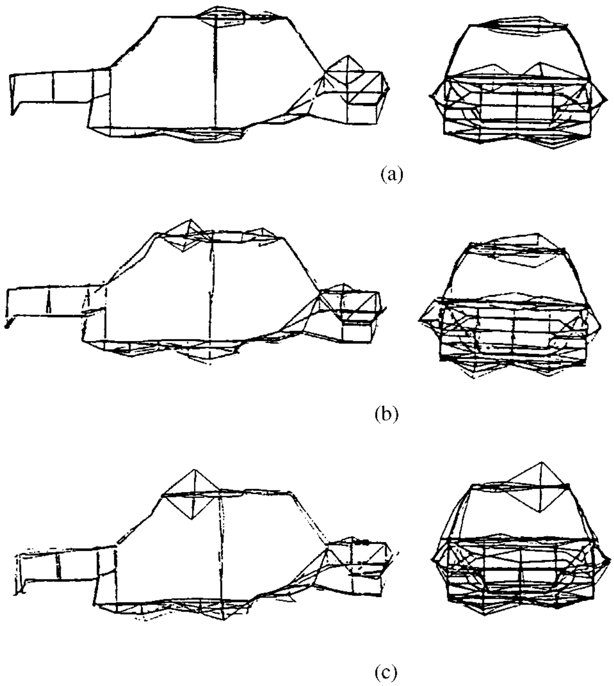


Figure 3. Typical structural modes of car body by experiment. (a) sixth mode ($f = 81{\cdot}9$ Hz), (b) 13th mode ($f = 139{\cdot}5$ Hz), (c) 14th mode ($f = 148{\cdot}9$ Hz).

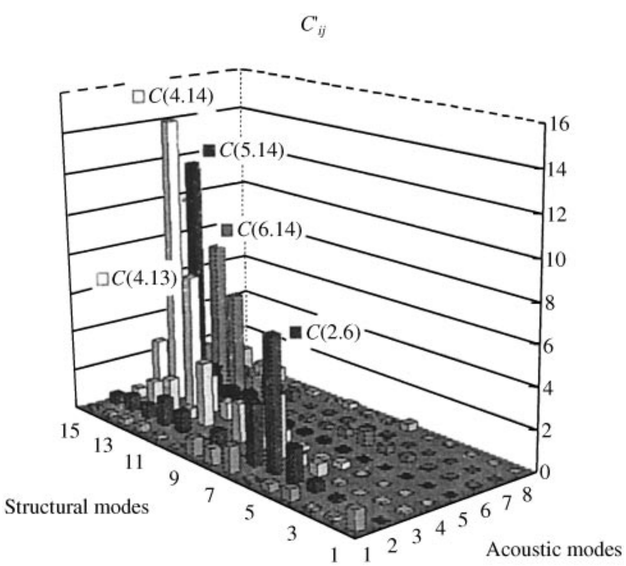
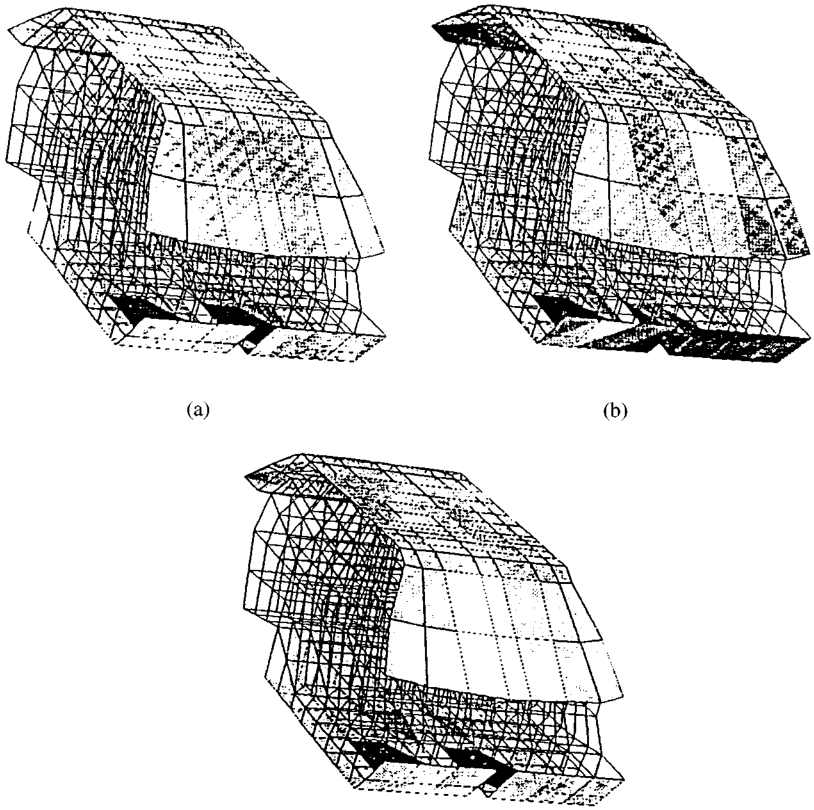


Figure 4. Comparison of structural-acoustic coupling coefficients (before damping treatment).

test car for trouble shooting and noise reduction. Figure 8 shows the acoustic frequency response at the driver's ear position under random excitation below the front right-hand side assembly as shown in Figure 2. A strong response is observed between 130 and 140 Hz and around 200 Hz. In Figure 8, the response (dotted line)



(a) (b)

(c)

Figure 5. Contributions of boundary panels to the dominant coupling coefficients (black: largest contribution, white: smallest contribution); (a) contribution of C'_{26}, (b) contribution to C'_{413}, (c) contribution to C'_{414}.

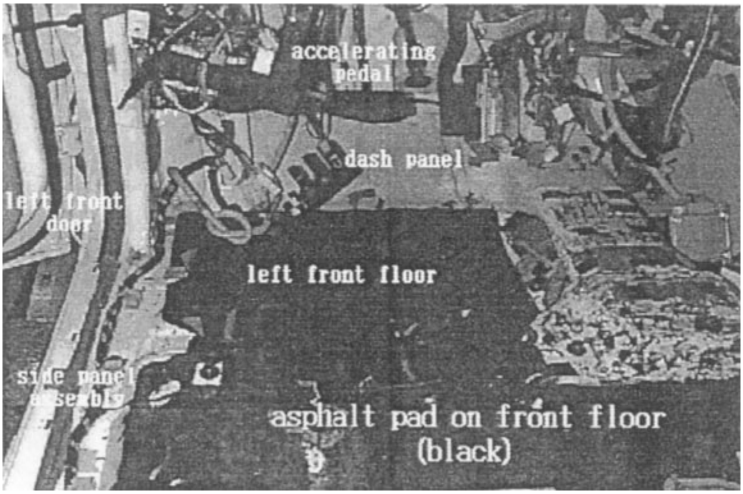


Figure 6. Damping treatment on the front floor panel (by attaching asphalt pad).

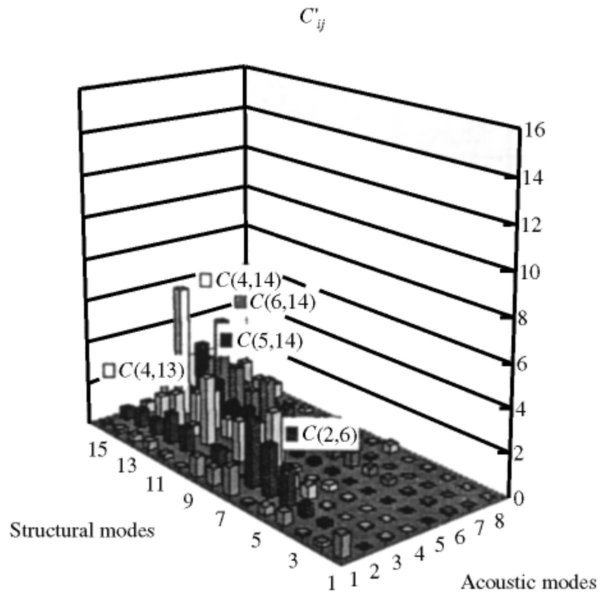


Figure 7. Reduction of the dominant coupling coefficients by damping treatment.

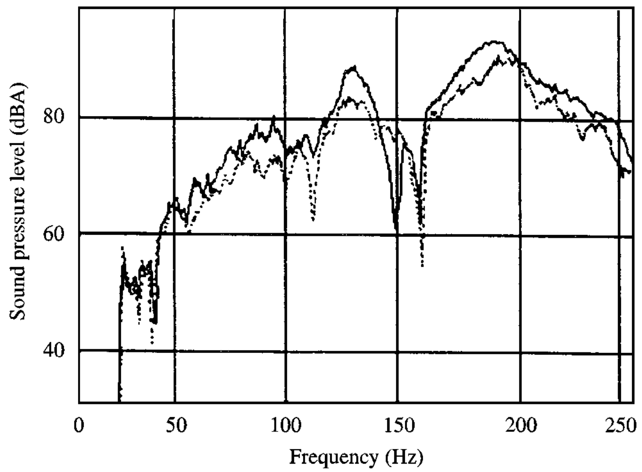


Figure 8. Structural-acoustic frequency response characteristics after damping treatment. —— Without damping sheet; ----- with damping sheet.

after damping treatment shows a reduced level around 80 Hz and 130–140 Hz, compared with that (solid line) before damping treatment. This means that the structural-acoustic sensitivity of the compartment system is reduced around this frequency component. Final effect of noise reduction is confirmed by a running test. Figure 9 shows noise level at the driver's ear position versus engine speed. The upper solid and dotted curves mean A-weighted overall level and the lower curves show the contribution of the engine's second harmonic component. This study

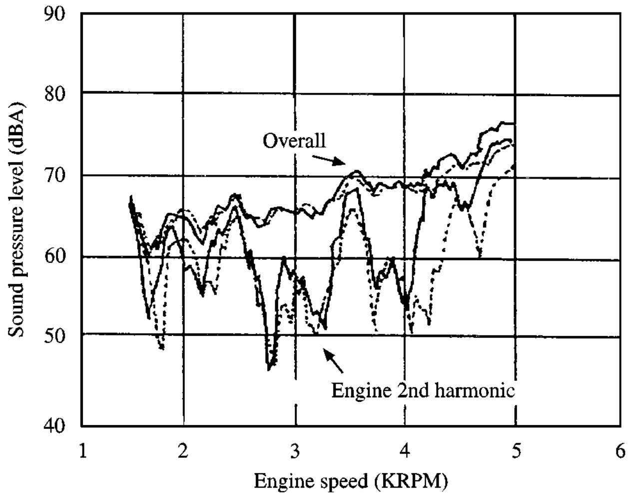


Figure 9. Noise level versus engine speed by running test after damping treatment. ——Without damping sheet; ----- with damping sheet.

focuses on the reduction of noise at 2400, 3600 and 4200 r.p.m. which generate high levels. In the figure, the engine's second harmonic component (firing frequency) contributes significantly to the overall level and the corresponding frequency components are 80, 120 and 140 Hz. As shown in Table 1 and Figure 4, there existed some structural and acoustic modes and also very large structural-acoustic modal coupling coefficients around those frequencies. As stated above, damping treatment on the sensitive area effectively increases the modal damping and decreases the large coupling coefficients, which results in the large reduction in the second harmonic component around 2400, 3600 and 4200 r.p.m. A small reduction in overall level means that various excitation sources have an influence on the level. In the analysis, by focusing on a few modes and coupling coefficients, the noise identification and reduction procedure could be simplified and the coupling concept was effectively used to understand the acoustic response characteristics of a vehicle compartment.

6. CONCLUSIONS

A practical method to reduce a car's interior noise was applied to a medium size test car, in which the structural-acoustic modal coupling coefficient was efficiently used. The trouble shooting and noise reduction procedure could be simplified by focusing on the small number of modes and modal coupling coefficients. A special-purpose program package was developed to carry out the proposed method, in which finite element analysis was assisted by experimental modal data. The use of the experimental data refined the result of the numerical analysis on the car's interior noise and the noise reduction effect was confirmed in the application to the real car.

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APPENDIX: NOMENCLATURE

$\{f_s\}$	excitation force vector
$[N_a]$	interpolation matrix for the acoustic finite element
$[N_s]'$	structural interpolation matrix to calculate the normal displacement component to the boundary surface
ζ_{ai}	damping factor of acoustic mode
ζ_{si}	damping factor of structural mode
ρ_a	mass density of air
$\{\phi_i\}$	acoustic mode
$\{\phi_j\}$	structural mode
ω	excitation frequency
ω_{ai}	acoustic natural frequency
ω_{si}	structural natural frequency