Structural Modification for Vehicle Interior Noise Reduction Using Vibration Response Sensitivity Analysis

*Yong-Hwa Park, *Wan-Sup Cheung, **Youn-Sik Park

Abstract

A structural modification technique for reducing structure-borne noise of vehicles using a sensitivity analysis is suggested. To estimate the noises generated by the vibration response, a semi structure-acoustic coupling analysis was exploited. As a result of the coupling analysis, severe noise generating positions are identified whose vibrations should be cured through structural modifications. Formulation for the sensitivity analysis of those severe vibration responses with respect to the design changes is derived to enhance the vibration response. Special attention is given in this paper to the use of the experimentally measured vibration responses in the sensitivity analysis. As a result of the proposed method, the structural modifications can be performed accurately by using experimental data instead of using the finite element method though the higher vibration modes are considered as long as the vibration measurement and acoustic mode calculations are accurate. Effectiveness of this method was examined using an example model by experiments.

I. Introduction

Many researchers in automotive industry have studied the problem of noise and vibration reduction when designing vehicle structures for the past few decades[1]. As well known, vibration and noise in operating vehicles mainly come from unbalance forces in engine and tire forces caused by irregular road profiles[2]. These forces excite structure, propagate into car interior, and sometimes drive the system into the resonant states. Since an enclosed sound field has its own acoustic modes, when structural and acoustic modes are closely coupled, then any tiny vibration force can generate severe noise. Among structural vibration sources, the spectral components up to 200 Hz highly affect on the booming noises [3]. To deal with the structure borne noises, detailed analyses on both structural and acoustical systems are necessary to identify the energy propagation characteristics between the two media. Many numerical techniques, which deal with these coupled systems, have been developed [4,5].

To reduce the structure borne noises effectively, as one of appropriate intermediate procedures, the structural dynamics modification (SDM) is needed. Numerical SDM techniques using FEM have been popular in vehicle design since the methods can predict the noise and vibration level at the design stage and can examine the modification effects easily. But to do correct FE analysis, an accurate FE model that has dynamic characteristics close to its real structure up to medium and/or higher structural modes is essential because the structure borne noises are affected mostly by those vibration modes. However for most real complex structures, it is difficult to get the accurate FE structural model up to medium and/or higher modes because there are many uncertainties and limitations in modelling the structures.

To overcome this difficulty, a structural modification method using experimentally measured frequency response function (FRF) of a structure is suggested in this study to broaden the usage of SDM to medium and/or high structural modes. The sensitivity of vibration response, i.e. the variation of vibration response at the frequency of interest with respect to design changes is formulated. A simplified structure-acoustic coupling analysis is adopted to identify how the vibration response affects on the interior noise of vehicles. The vibration and acoustic characteristics of vehicle can be considered separately through the simplified coupling analysis. Finally, efficient structural modifications are determined effectively by

*Acoustics and Vibration Lab., Korea Research Institute of Standards and Science
**Dept. of Mechanical Engineering, Korea Advanced Institute of Science and Technology

Manuscript Received: March 22, 2000

combining the vibration response sensitivity and the structure-acoustic coupling analysis. As an example, experimental results for a half-scale model car were illustrated to show the effectiveness of the proposed method.

II. Vibration Response Sensitivity Analsys

In this section, the first order sensitivity of FRF with respect to design change is derived. Based on the FRF sensitivity, the change of vibration response due to design change is analyzed. From the frequency domain analysis, the relation between the vibration response $\{x(b,\omega)\}$ and the external force $\{f(\omega)\}$ of a structural system having n degrees of freedom at frequency ω can be written as follows:

$$\left[-\omega^2 M(b)+i\omega C(b)+K(b)\right]\{x(b,\omega)\}=\{f(\omega)\} \qquad (1)$$

$$\{x(b,\omega)\}=\left[-\omega^2 M(b)+i\omega C(b)+K(b)\right]^{-1}\{f(\omega)\} \qquad (2)$$

where $M(b)$, $C(b)$ and $K(b)$ are mass, damping and stiffness matrices (n-by-n) of the structural system, and b denotes a design variable such as mass, stiffness and thickness of the system of interest. From equations (1) and (2), dynamic stiffness matrix $D(b,\omega)$ and frequency response matrix $H(b,\omega)$ of the system can be defined as follows:

$$D(b,\omega)=-\omega^2 M(b)+i\omega C(b)+K(b) \qquad (3)$$

$$H(b,\omega)=\left[-\omega^2 M(b)+i\omega C(b)+K(b)\right]^{-1} \qquad (4)$$

The relationship between the dynamic stiffness matrix and the frequency response matrix can be expressed from equations (3) and (4) as

$$D(b,\omega)\cdot H(b,\omega)=I \qquad (5)$$

By differentiating equation (5) with respect to design variable b, we can have the sensitivity of frequency response matrix as

$$\begin{aligned}\frac{\partial H(b,\omega)}{\partial b}&=-D(b,\omega)^{-1}\frac{\partial D(b,\omega)}{\partial b}H(b,\omega)\\&=-H(b,\omega)\left[-\omega^2\frac{\partial M(b)}{\partial b}+i\omega\frac{\partial C(b)}{\partial b}+\frac{\partial K(b)}{\partial b}\right]H(b,\omega)\end{aligned} \qquad (6)$$

Hence the sensitivity of vibration response x_i (for $i=1,2,\ldots,n$) with respect to design variable b, denoted by $S_{xi}(b,\omega)$ can be obtained from equations (2), (4) and (6) as follows:

$$\begin{aligned}S_{xi}(b,\omega)&=\frac{\partial x_i(b,\omega)}{\partial b}=\frac{\partial(\lfloor H(b,\omega)\rfloor_i\{f(\omega)\})}{\partial b}\\&=-\lfloor H(b,\omega)\rfloor_i\left[\frac{\partial D(b)}{\partial b}\right]H(b,\omega)\{f(\omega)\}\\&=-\lfloor H(b,\omega)\rfloor_i\left[-\omega^2\frac{\partial M(b)}{\partial b}+i\omega\frac{\partial C(b)}{\partial b}+\frac{\partial K(b)}{\partial b}\right]\{x(b,\omega)\}\end{aligned} \qquad (7)$$

where $\lfloor H(b,\omega)\rfloor_i$ is the i-th row vector of $H(b,\omega)$.

$\lfloor H(b,\omega)\rfloor_i$ and $\{x(b,\omega)\}$ in equation (7) can be acquired from experimental measurements of the real systems. If the structural modifications are undertaken at n_m degrees of freedom, the derivative of dynamic stiffness matrix, i.e. $\frac{\partial D(b)}{\partial b}$ in equation (7) has n_m non-zero rows and columns obviously. Hence, FRFs between the vibration measuring position i and degrees to be modified, and vibration responses at degrees to be modified, which is usually very small number compared to the entire degrees of the system, are necessary to calculate the sensitivity in equation (7). Thus the sensitivity analysis can be performed accurately and also effectively using experimental data instead of using the finite element model up to higher vibration modal frequencies.

To obtain the sensitivity described in equation (7), the derivatives of mass, damping and stiffness matrices with respective to design variable are needed. There are two general approaches in calculating these derivatives. The first one is to use a closed form solution, which is feasible if any analytical expression of dynamic stiffness matrix is available. But its feasibility is limited only to simple structural modifications such as lumped mass, stiffness, beam and plate modifications. An alternative approach, which is always feasible but more complex, is calculating the sensitivities via the finite difference methods. Many different schemes for these finite difference calculations have been suggested for modifying complex structural components, but because of their high computational costs during the repeated calculations, it is recommended to use only the approximate first order differences [6]. In this paper, simple lumped mass modifications will be dealt as an example.

III. Semi Structure-Acoustic Coupling Analysis

In order to reduce car interior noise effectively, the study on the interactions between interior noise and structural vibration is important. Interior noises caused by structural vibrations, i.e. structure-borne noises, can be predicted through the structure-coupling analysis.

However a complete analysis on the coupling of structural and acoustic systems needs huge computations and sometimes it is not realistic, so a semi-coupling analysis is conducted in this work. Basically this analysis assumes that surface motions of a structure affect acoustical behaviors in an enclosed cavity, but the effects of the acoustical behaviors on the surface motions are negligible. This assumption is usually adequate in the case when the output impedance of acoustical system is large enough to assume the structural surface as a rigid one. Thus most vehicles, which have relatively stiffened surface, are good enough to apply this assumption. In this case, acoustic pressure can be expressed by using the Green's function approach[7,8]. Acoustic pressure at a specific receiving position ρ_r and at a frequency ω can be written as

$$P(\rho_r,\omega)=\rho_a\omega^2c_a^2\sum_k\left[\frac{\phi_k^{(a)}(\rho_r)}{\Omega_k^{(a)^2}-\omega^2}\int_S\phi_k^{(a)}(\rho_{r_0})x_N(\rho_{r0},\omega)ds\right] \quad (8)$$

where $P(\rho_r,\omega)$ is pressure level at position ρ_r in cavity and at frequency ω, ρ_a is the density of acoustic medium in cavity, and c_a is the speed of sound. $\phi_k^{(a)}(\rho_r)$ is the *k-th* acoustic mode shape at ρ_r, $\Omega_k^{(a)}$ is the k-th acoustic resonance frequency, S is the boundary surface, $x_N(\rho_{r0},\omega)$ is the vibration response orthogonal to the boundary surface, and ρ_{r0} denotes position vector defined on the surface.

To apply the theoretical result in equation (8) to experimental analysis of this paper, the surface integration in equation (8) is approximated to be the summation of small sub-integrations by adopting the Trapezoidal integration rule. Through this approximation, integration in equation (8) becomes the weighted inner product of two vectors. Considering design variable b together, the result is given as

$$P(\rho_r,\omega)=\rho_a\omega^2c_a^2\sum_k\left[\frac{\phi_k^{(a)}(\rho_r)}{\Omega_k^{(a)^2}-\omega^2}\{\phi_k^{(a)}\}^T W\{x(b,\omega)\}\right] \quad (9)$$

where $\{\phi_k^{(a)}\}$ is the *k-th* acoustic mode shape vector defined on n_S surface nodes, $x(b,\omega)$ is the measured vibration response orthogonal to the boundary surface defined on n_S surface nodes. W is area weighting matrix (n_S-by-n_S) whose components include the information on sub-areas adjacent to measurement nodes. In this paper, rectangular sub-areas are used to determine the weighting matrix W. Acoustic pressure described in equation (9) can be divided into two parts, i.e. acoustic and structural parts as follows:

$$P(\rho_r,b,\omega)=\{S_a(\rho_r,\omega)\}^T\{x(b,\omega)\} \quad (10)$$

where $$\{S_a(\rho_r,\omega)\}\equiv\rho_a\omega^2c_a^2\sum_k\left[\frac{\phi_k^{(a)}(\rho_r)}{\Omega_k^{(a)^2}-\omega^2}\{\phi_k^{(a)}\}^T W\right]^T. \quad (11)$$

$\{S_a(\rho_r,\omega)\}$ means the sensitivity of acoustic pressure with respect to the structural vibration and it depends only on the acoustic characteristics of cavity. It is interesting that through equation (10), the vibration behaviors and acoustical characteristics can be considered separately using the analyses of different two media to predict the resulting structure borne noises. Resultantly, the sensitivity of acoustic pressure with respect to the design change denoted by $\{S_a(\rho_r,\omega)\}$ can be obtained by combining the vibration response sensitivity in equation (7) and acoustic sensitivity in equation (11) as follows:

$$\begin{aligned}S_p(\rho_r,b,\omega)&\equiv\frac{\partial P(\rho_r,b,\omega)}{\partial b}=\{S_a(\rho_r,\omega)\}^T\frac{\partial}{\partial b}\{x(b,\omega)\}\\&=\{S_a(\rho_r,\omega)\}^T\{S_x(b,\omega)\}\end{aligned} \quad (12)$$

where $\{S_x(b,\omega)\}$ is the sensitivity vector whose components are $\{S_{xi}(b,\omega)\}$ (for $i=1,2,\dots,n_S$) defined in equation (7). The effect of structural modification on interior noise can be effectively examined using equation (12).

Ⅳ. Determination of Structural Modification

Structural modification for reducing interior noise is determined based on the sensitivity analysis described in equation (12). In the calculation of sensitivity $S_p(\rho_r,b,\omega)$ the sensitivities of vibration responses at all of n_S nodes on the boundary surface, i.e. $\{S_x(b,\omega)\}$, are needed to be calculated. It needs the measurement of n_S-by-n_m frequency response matrix as shown in equation (7), which is usually a burden work for experimenters[9]. However, each contribution of vibrations at n_S nodes to acoustic pressure is different from each other. It depends upon both of vibration response and the acoustic sensitivity $\{S_a(\rho_r,\omega)\}$ at the surface nodes. The contributions of the vibration responses to pressure can be written from equation (10) as follows:

$$PCF_i(\rho_r,b,\omega)=S_{a,i}(\rho_r,\omega)\cdot x_i(b,\omega) \quad \text{for } i=1,2,\dots,n_s \quad (13)$$

where $PCF_i(\rho_r,b,\omega)$ denotes the pressure contribution factor (PCF) of vibration response at node i, $S_{a,i}(\rho_r,\omega)$ is the i-th component of $\{S_a(\rho_r,\omega)\}$ and $x_i(b,\omega)$ is the i-th vibration response in $\{x(b,\omega)\}$ The resulting acoustic pressure is the summation of PCFs such that

$$P(\rho_r, b, \omega) = \sum_{i=1}^{n_s} PCF_i(\rho_r, b, \omega) \tag{14}$$

Since PCF shows the contribution of vibration response at the *i-th* node to the acoustic pressure, an efficient noise reduction can be achieved by suppressing the structural vibrations at the positions showing a relatively larger PCF values, instead of suppressing those of every surface nodes. Hence if the structural vibrations at n_p nodes having larger PCFs are selectively considered, the sensitivity of acoustic pressure in equation (12) can be simplified as

$$\hat{S}_p(\rho_r, b, \omega) = \{\hat{S}_a(\rho_r, \omega)\}^T \{\hat{S}_s(b, \omega)\} \tag{15}$$

where $\hat{S}_p(\rho_r, b, \omega)$ is the selective sensitivity of interior noise caused by n_p selected vibrations, $\{\hat{S}_a(\rho_r, \omega)\}$ is the acoustic sensitivity for selected n_p surface nodes and $\{\hat{S}_s(b, \omega)\}$ is the vibration response sensitivity of selected n_p vibrations. Considering N_b design variables, the selective sensitivity vector (N_b-by-1) can be written from equation (15) as follows:

$$\begin{aligned}\{\hat{S}_p(\rho_r, b_1, b_2, \ldots, b_{N_b}, \omega)\}^T &= \{\hat{S}_p(\rho_r, b_1, \omega), \hat{S}_p(\rho_r, b_2, \omega), \ldots, \hat{S}_p(\rho_r, b_{N_b}, \omega)\} \\ &= \{\hat{S}_a(\rho_r, \omega)\}^T [\{\hat{S}_s(b_1, \omega)\}, \{\hat{S}_s(b_2, \omega)\}, \ldots, \{\hat{S}_s(b_{N_b}, \omega)\}]\end{aligned} \tag{16}$$

Hence, pressure change ΔP due to design change $\{\Delta b\} = \{\Delta b_1, \Delta b_2, \ldots, \Delta b_{N_b}\}^T$ can be predicted by using the sensitivity in equation (16) as

$$\Delta P = \{\hat{S}_p(\rho_r, b_1, b_2, \ldots, b_{N_b}, \omega)\}^T \{\Delta b\} \tag{17}$$

Equation (17) shows the sensitivity relation when only a single frequency component as well as a single noise measuring position is considered. However, indeed many frequency components in the frequency range of interest should be considered simultaneously to achieve noise reduction more effectively. For this purpose, the necessary design change in equation (17) should be calculated by considering the multi-objectives desired. In general case when N_c frequency components, i.e. $\omega_1, \omega_2, \ldots, \omega_{N_c}$, are considered, equation (17) becomes a system of linear algebraic equation such that

$$\begin{Bmatrix} \Delta P_1 \\ \Delta P_2 \\ \mathrm{M} \\ \Delta P_{N_c} \end{Bmatrix} = \begin{bmatrix} \hat{S}_p(\rho_r, b_1, b_2, \ldots, b_{N_b}, \omega_1)^T \\ \hat{S}_p(\rho_r, b_1, b_2, \ldots, b_{N_b}, \omega_2)^T \\ \mathrm{M} \\ \hat{S}_p(\rho_r, b_1, b_2, \ldots, b_{N_b}, \omega_{N_c})^T \end{bmatrix} \{\Delta b\} \tag{18}$$

or $\{\Delta P\} = [\hat{S}_p]\{\Delta b\}$

where $[\hat{S}_p]$ is the N_c-by-N_b sensitivity matrix. The optimal design change $\{\Delta b\}_{opt}$ can be determined by considering the multi-objectives in equation (18), which is in the steepest decent direction of the pressure change as follows:

$$\{\Delta b\}_{opt} = k[\hat{S}_p]^+ \{\Delta P\} \tag{19}$$

where superscript + denotes the pseudo inverse of a matrix and $k (0 \le k \le 1)$ is a scaling parameter which is determined according to the allowed amount of design changes. The procedure for determining structural modification is summarized in Figure 1.

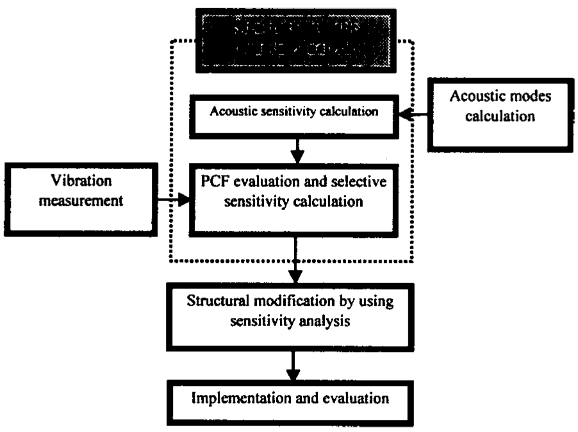

Figure 1. Flowchart of determining structural modification for interior noise reduction.

V. Example

The suggested structural modification technique is examined in this section with an example structure: a half-scaled model car built with steel plates and steel beams as

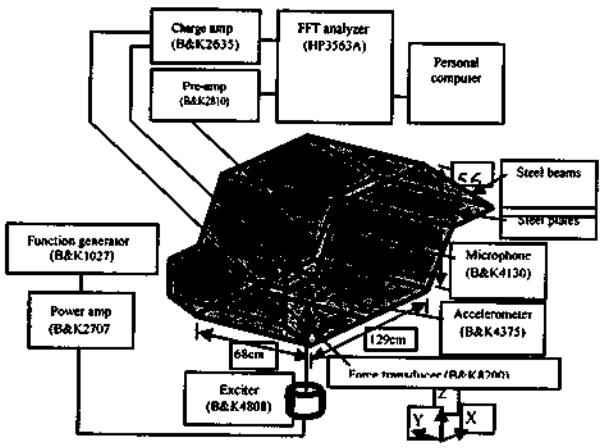

Figure 2. 1/2 scale vehicle model and experimental set-up.

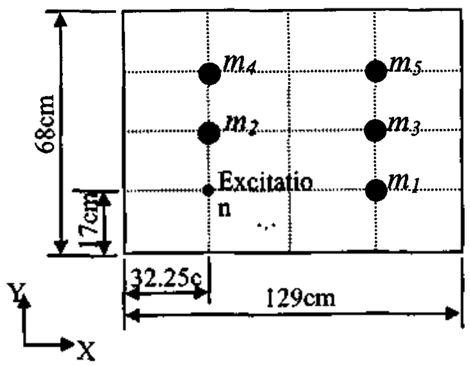

Figure 3. Layout of five lumped masses on the floor of vehicle model.

shown in Figure 2. The thickness of the plates is 1.2 mm and the beams have rectangular cross section with height 8.0 mm and width 8.0 mm. On the floor of the model, five lumped masses having identically 70 g are fixed to the designated nodes as shown in Figure 3. The lumped masses are approximately realized with fastened bolts and nuts: indeed, these cannot be regarded as exact lumped masses depending on their sizes compared to the thickness of the plate. The number of nuts are controlled to realize desired mass. The model car body was hanged by soft springs to realize the free-free boundary condition approximately. An excitation position was selected to be placed at the floor position under the driver seat as shown in Figure 3 and a banded random force up to 200 Hz was applied in the z direction to the model car through an exciter. 150 transnational vibration responses orthogonal to the surface of car body were measured by a roving accelerometer on the surface. To measure generated interior noise, a microphone was selected to be located at the ear position of a driver.

Figure 4 shows a measured vibration response at the excitation position. Measured sound pressure level at the

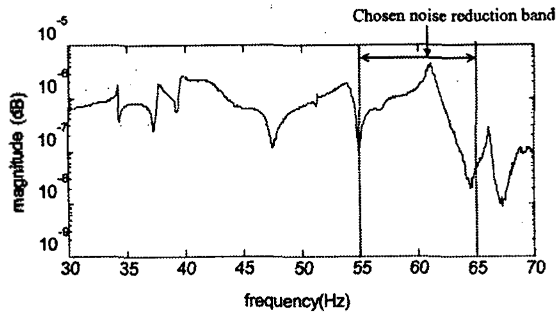

Figure 4. Measured FRF: point inertance at the excitation point on the floor.

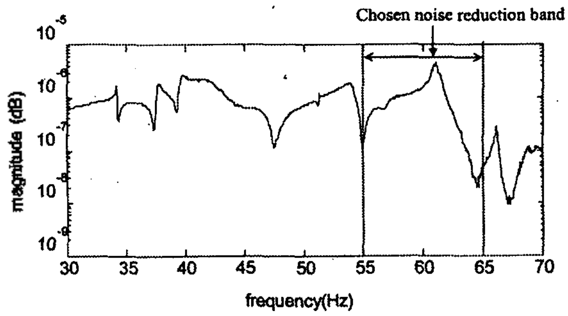

Figure 5. Measured sound pressure level at the receiving position.

receiving position is plotted in Figure 5. A loud noise around 60 Hz was observed in Figure 5. Comparing Figure 4 and Figure 5, one can see that this noise component is generated by the vibration response around 60 Hz. Thus in this example, the objective of structural modification is selected to be reducing the noise level in frequency range from 55 Hz to 65 Hz, which is mostly affected by the 60 Hz vibration component. To achieve the objective, the vibration response at a single frequency 60 Hz is selected to be modified in this example rather than considering many number of frequency components in the frequency range of interest, i.e. 55 to 65 Hz. In this case, a number of objectives N_C is equal to 1.

To investigate the effects of vibration response at 60 Hz on the interior noise, structure-acoustic coupling analysis was carried out. FE analysis was conducted to identify the structural modes of the model car shown in Figure 2. FE model having 1,227 nodes (7,362 DOFs) was made using the beam and shell elements built in ANSYS 5.0A. Table 1 shows the structural natural frequencies obtained from FEM and experiment. The relative error increases up to 13.2 % as the mode number increases. It indicates that FE analysis is not reliable for the vibration analysis in the frequency band of interest. Hence the vibration measurements were used rather than FEM for structure-acoustic coupling analysis in this example. The acoustic

Table 1. Comparison of structural natural frequencies of model car obtained from FEM and experiment.

Mode No.	FEM(Hz)	Experiment (Hz)	Percentage error (%)
1	32.3	34.7	6.9
2	34.3	37.5	8.5
3	42.4	39.1	8.4
4	48.0	43.0	11.6
5	51.3	46.5	10.3
6	56.6	51.2	10.5
7	60.1	54.2	10.9
8	67.9	60.0	13.2

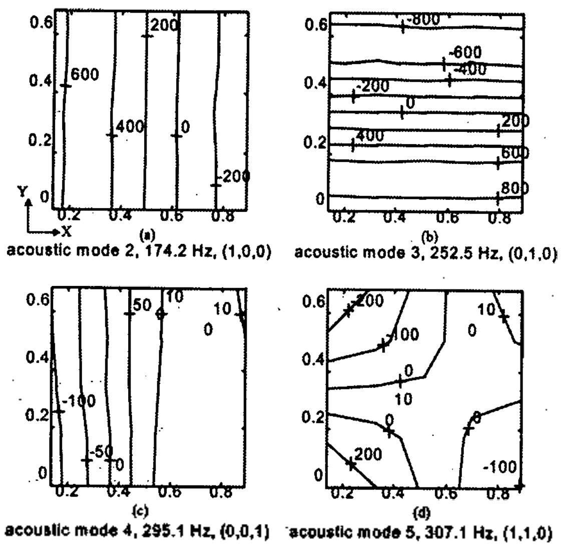

Figure 6. Acoustic modes on the floor calculated by BEM.

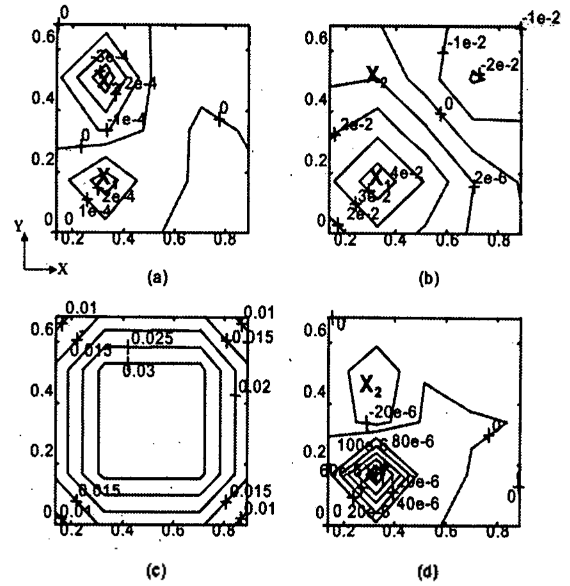

Figure 7. The results of coupling analysis on the floor at frequency 60 Hz. (a) Vibration response of the floor; (b) acoustic pressure sensitivity in equation (11); (c) area weighting matrix W in equation (9); (d) pressure contribution factor (PCF) in equation (13).

Table 2. Comparison of acoustic natural frequencies of model car interior obtained from BEM and experiment.

Mode No.	BEM(Hz)	Experiment (Hz)	Percentage error (%)
1	0.0	0.0	0
2	174.2	177.1	1.6
3	252.5	257.2	1.8
4	295.1	299.1	1.3
5	307.1	312.0	1.6

modes of interior cavity, which can be estimated accurately in aid of numerical methods [5] in this case, were calculated by the BEM. In the BEM calculations, the acoustic natural frequencies and mode shapes in equation (8) are calculated under the assumption that the cavity is enclosed by rigid walls. Figure 6 shows calculated acoustic modes on the floor. Table 2 shows that the resulting acoustic natural frequencies obtained from BEM are accurate within 1.8 % error when compared to the experimental results.

Vibration measurements indicated that the vibration of the floor panel is dominant compared to those of other panels. Thus in this experiment case, only the floor vibration is considered in the structure-acoustic coupling analysis for convenience. The results of coupling analysis on the floor at frequency 60 Hz are plotted in Figure 7. Contours of vibration responses on the floor and acoustic sensitivities at 60 Hz in equation (11) are plotted in Figures 7(a) and 7(b) respectively. Figure 7(c) shows the area weighting matrix W in equation (9). Area weighting becomes smaller as the node approaches to edge regions. The distribution of pressure contribution factor (PCF) for each node on the floor is also plotted in Figure 7(d). The deflection pattern of the floor in Figure 7(a) shows that the vibration level at position X_2 is larger than that of X_1. However, comparing Figures 7(a) and 7(d), the distribution of PCF values shows an opposite trend: PCF at X_1 is larger than that of X_2. This comes from the fact that the floor vibrations affect the interior noise through the acoustic sensitivity shown in Figure 7(b).

As described in equation (11), this acoustic sensitivity is only related to the acoustic modes of the cavity shown in Figure 6 and not depends on the structural motions. At the target frequency (60 Hz), the effects of the first few acoustic modes including rigid body mode are mixed so that the nodal line of the acoustic sensitivity on the floor looks like a biased diagonal as shown in Figure 7(b). The most sensitive position is placed at X_1 whereas the sensitivity is very small at X_2. Consequently, the smaller vibration of location X_1 generates more interior noise than that of X_2 as shown in Figure 7(d). As a result of coupling analysis, the vibration of node X_1 was identified to be the dominant source of measured noise around 60 Hz according to PCF shown in Figure 7(d). Hence the vibration at X_1 was selected to be suppressed by structural modifications to reduce the interior noise. In this case, a number of vibration suppression position n_p is equal to 1.

For the structural modification in this example, five lumped masses on the floor shown in Figure 3 were chosen to be design variables rather than more general structural modifications such as beams and plate modifications to show the modification procedure conveniently. In this case, the number of degrees of freedom to be modified (n_m) as well as that of design variables (N_b) is five. Hence design change defined in equation (17) can be written as

$$\{\Delta b\} = \{\Delta m_1, \Delta m_2, \Delta m_3, \Delta m_4, \Delta m_5\}^T \tag{20}$$

where Δm_i (for i=1,2,...,5) denotes the change of i-th mass on the floor. Total mass change is assumed not to exceed 220 g in this example. Since only the mass modifications are examined in this case, the derivative of dynamic stiffness matrix in equation (7) is simply given as

$$\frac{\partial D(m_i)}{\partial m_i} = -\omega^2 \frac{\partial D(m_i)}{\partial m_i} = -\omega^2 \begin{bmatrix} 0 & \Lambda & 0 & \Lambda & 0 \\ M & O & M & 0 & M \\ 0 & \Lambda & 1 & \Lambda & 0 \\ M & 0 & M & O & M \\ 0 & \Lambda & 0 & \Lambda & 0 \end{bmatrix}_i . \tag{21}$$

$\frac{\partial D(m_i)}{\partial m_i}$ has only one nonzero component at the i-th row and the i-th column corresponding to the node to which mass m_i is attached. Vibration response sensitivities at X_1 with respect to the floor masses m_i, i.e. $S_{x1}(m_i,\omega)$ for i=1,2,...,5, can then be calculated from equations (7) and (21) as

$$S_{x1}(m_i,\omega) = -\omega^2 \lfloor \hat{H}(m_i,\omega) \rfloor_{x1} \{\hat{x}(b,\omega)\} \tag{22}$$

where $\lfloor \hat{H}(m_i,\omega) \rfloor_{x1}$ (1-by-5) contains measured cross-FRFs between position X_1 and five mass-attached nodes, and $\{\hat{x}(b,\omega)\}$ is the measured vibration response shape at the five mass-attached nodes. The calculated acoustic sensitivity at node X_1, i.e. $\hat{S}_a(p_r,\omega)$ is $4.1 \times 10^{-2} Pa/m$. Finally, the selective sensitivity in equation (16) for vibration of X_1 was obtained as

$$\{\hat{S}_p(p_r, m_1, m_2, ..., m_5, \omega)\} = \hat{S}_a(p_r,\omega) \cdot [S_{x1}(m_1,\omega), S_{x1}(m_2,\omega), ..., S_{x1}(m_5,\omega)]^T$$
$$= [-0.00 \ \ -1.24 \ \ -0.01 \ \ -14.45 \ \ -0.01]^T \times 10^{-6} Pa/kg \tag{23}$$

The measured noise level at 60Hz was $68\mu Pa$, hence the desired pressure change ΔP was selected to be $-68\mu Pa$. The optimal mass changes were calculated by using equation (19) as

$$\{\Delta m\}_{opt} = \{\Delta m_1, \Delta m_2, \Delta m_3, \Delta m_4, \Delta m_5\}^T$$
$$= \{0.00, 0.02, 0.00, 0.20, 0.00\}^T kg \tag{24}$$

where the scaling parameter k was set to 0.04 in order that the total mass change does not exceed the fixed weight 220 g.

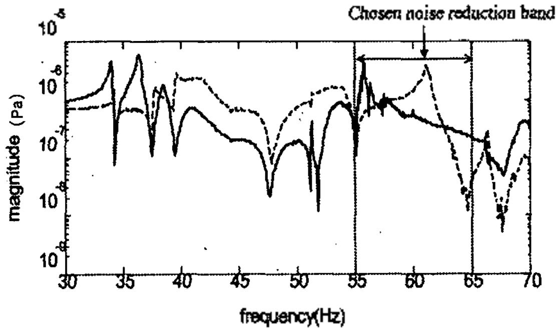

Figure 8. Measured pressure change due to floor mass modifications given in equation (24).---original measured pressure; —— pressure after mass modification.

Mass modifications shown in equation (24), i.e. m_4 increases to 270g and m_2 to 90g, were applied to the model car. Figure 8 shows spectral distributions of interior noises before and after the mass modifications. To evaluate the effectiveness of the modification, banded power spectrum of the interior noise from 55Hz to 65Hz, i.e. $\int_{55}^{65} P(f)^2 df$ was calculated, which can be an indicator of noise level in the frequency range of interest. The result showed that the banded power spectrum was reduced about 3.0 dB after the mass modifications. In this example, only the single frequency component was considered for illustration of the modification procedure (Nc = 1). However, Figure 8 shows that the modified vibration characteristics can give adverse effect on the generated noises out of the frequency range of interest. So it cannot be guaranteed that the noise reduction on the narrow band can give always the noise reduction on wider band. Hence the multiple frequency components in wider frequency range should be considered simultaneously to achieve noise reduction more satisfactorily in real structures. In this example, five lumped masses on the floor were chosen to be design variables. But when more general structural modifications such as beams and plate modifications are considered, the sensitivity matrix in equation (21) become difficult to be calculated since the number of degrees of freedom to be modified (n_m) as well as that of design variables (N_b) increase and hence lager number of measurements including rotational degrees of freedom are required.

Ⅵ. Conclusions

A structural modification technique for reducing structure-borne interior noise of vehicles using the vibration response sensitivity analysis was suggested. A semi structure-acoustic coupling analysis was exploited to estimate the structure-borne noises. As a result of the coupling analysis, severe noise generating positions can be identified whose vibration should be cured through the structural modifications. Formulations for the sensitivity analysis of vibration response with respect to the design changes were derived to cure harmful vibration identified from the structure-acoustic coupling analysis. Minimum number of measured FRFs are shown to be necessary to calculate the vibration response sensitivity. Hence the sensitivity analysis can be performed accurately by using experimental data instead of using the finite element method though the higher vibration modes considered.

Effectiveness of this method was examined using an example model. Loud interior noises in the frequency band chosen, which includes higher vibration modes, were tried to reduce. The experimental result indicated that by the mass changes the banded power spectrum in the frequency band of interest was reduced about in 3 *dB* compared to the original one. Although the structural dynamic modification using experimentally measured data can eliminate the burden work of numerical calculations such as finite element modeling, but sometimes it is impossible to apply this technique to real vehicle structures when the measurement is inaccessible and the feasible structure modification is limited.

References

1. E. R. Dowell, "Plan for prediction of Vehicle Interior Noise," AIAA Journal, Vol. 18, No. 4, pp. 353-366, 1980.
2. I. Morita, and K. Arai, "On Noise and Vibration from Engines Transversely installed in Front Wheel Drive Cars," JSME Review, Nov., pp. 24-31, 1983.
3. D. J. Nefske, J. A. Wolf and L. J. Howell, "Structural-Acoustic Finite Element Analysis of the Automobile Passenger Component: A Review of Current Practice," Journal of Sound and Vibration, Vol. 80, No. 2, pp. 247-266, 1982.
4. M. Petyt, J. G. Lea and H. Koopmann, "A Finite Element Method for Determining the Acoustic Modes of Irregular Shaped Cavities," Journal of Sound and Vibration, Vol. 45, No. 4, pp. 495-502, 1976.
5. S. Suzuki and S. Maruyama, "Boundary Element Analysis of Cavity Noise Problems with Complicated Boundary Conditions," Journal of Sound and Vibration, Vol. 130, No. 1, 1989.
6. E. J. Haug and K. K. Choi, Design Sensitivity Analysis of Structural Systems, Academic press, 1986.
7. E. H. Dowell, G. F. Gorman III and D. A. Smith, "Acoustoelasticity: General Theory, Acoustic Natural Modes and Force Response to Sinusoidal Excitation, including Comparisons with Experiment," Journal of Sound and Vibration, Vol. 52, No. 4, pp. 519-542, 1977.
8. L. E. Kinsler and A. R. Frey, Fundamentals of Acoustics, John Wiley, 1982.
9. D. J. Ewins, Modal Testing: Theory and Practice, Research Studies Press, John Wiley, 1984.

▲Yong-Hwa Park

Current Interests and Research Activities

- Structural Dynamics Analysis and Optimization
- System Identification and Inverse Problem
- Structure and Acoustics Coupling Analysis
- Human Vibration
- Dynamic Analysis and Design of Micro-electromechanical systems

Experience

9/2000 –present	*Visiting Researcher*, Department of Aerospace Engineering Sciences University of Colorado at Boulder
9/1999 –8/2000	*Post-Doc*, Acoustics and Vibration Lab, KRISS
3/1995 –8/1999	*Research Assistant*, Mechanical Engineering Research Institute, Korea Advanced Institute of Science and Technology (KAIST), Taejon, Korea.
3/1993 –2/1997	*Teaching Assistant*, Department of Mechanical Engineering, KAIST

Education

3/1993–8/1999	*Ph. D* in Mechanical Engineering, KAIST
3/1991–2/1993	*M. S.* in Mechanical Engineering, KAIST.
3/1987–2/1991	*B. S.* in Mechanical Engineering, KAIST.

▲ Wan-Sup Cheung

The Journal of the Acoustical Society of Korea Vol.17, No. 3, 1998.

▲ Youn-Sik Park

Experience

August 1972-July 1976 : Instructor, Department of Mechanical Engineering Korea Air Force Academy

August 1981-April 1984 : Sensoir Researcher, Division of Mechanical Engineering, KAIST

April 1984-February 1985 : Assistant Professor, Department of Mechanical Engineering, KAIST

March 1986-August 1991 : Assistant Professor, Department of Mechanical Engineering, KAIST

January 1988-December 1988 : Visiting Scholar, Department of Mechanical and Aeronautical Engineering, University of Virginia

September 1992-August 1994 : Chairman, Department of Mechanical Engineering, KAIST

September 1991-Present : Professor, Department of Mechanical Engineering, KAIST

September 1995-December 1996 : R&D Director, STEPI

April 1997-June 1998 : Dean of Research Affairs, KAIST

March 1999-January 2000 : Director, Center for Noise and Vibration Control Mechanical Engineering, KAIST

January 2000- : Dean of Planning Affairs, KAIST

Education

B.S.in Mechanical Engineering, Seoul National University, February 1972

M. S. in Engineering Mechanics, Iowa State University, May 1979

Ph. D. in Engineering Mechanics, Iowa State University, May 1981

Ph. D. Thesis : The Response and the Lift Force Analysis of a Cylinder Oscillating in Still Water.