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Acoustic Design Sensitivity Analysis and Optimization for Reduced Exterior Noise

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A program, global acoustic design sensitivity analyzer, is developed that can perform a global acoustic design sensitivity analysis of exterior noise with respect to structural sizing design variables. A system for global acoustic design sensitivity is introduced and implemented numerically by employing the continuum sensitivity analysis. A half-scale automobile cavity model is considered as a numerical example. By using a continuum method, we obtained accurate and efficient sensitivities when the number of design variables was large. Also, the tendency plots of element sensitivities and energy contribution for global acoustic sensitivities are available. Finally, a design optimization was performed to simultaneously reduce the weight and sound pressure level at interesting points and frequencies.

I. Introduction

THE range of engineering problems that can be solved through numerical analyses has been greatly broadened with the advent of high-speed digital computers. In addition, the development of the finite element method (FEM) and the boundary element method (BEM) has also been important to engineering analysis. These two types of analyses are associated with analytical structure-borne noise prediction. FEM is commonly used to compute the vibration of the structure emitting noise, and BEM can be used to predict the generated noise. Other methods for acoustic prediction, such as statistical energy analysis, are also available. In this paper, the focus is on FEM and BEM.

In a gradient-based optimization scheme, it is important to have accurate gradients (sensitivities) of the objective function and constraints with respect to the design variables. Formulation of global acoustic sensitivity through chain-ruled derivatives using FEM and BEM has been studied and implemented by many researchers. Coyette et al.[1] have previously investigated the computation and utilization of acoustic sensitivities with respect to sizing design variables. Two types of sensitivities were considered. One was acoustic sensitivity with respect to the normal velocity of vibrating structure, and the other was structural sensitivity of structural velocities with respect to physical sizing design variables such as thickness. The acoustic sensitivities and structural sensitivities were calculated using BEM and FEM, respectively, and then these sensitivities were combined to obtain a global acoustic sensitivity. In addition, this approach was implemented to the commercial code, SYSNOISE. The same approach was presented in the articles by Cunefare et al.,[2,3] who focused on finding the best optimization formulation by comparing the relative performance and results obtained through the use of several different objective functions and constraints. They obtained acoustic sensitivities and structural sensitivities from the commercial codes NASTRAN and COMET. They also developed COMIN to combine the acoustic and structural sensitivities. Most researchers who studied global acoustic design sensitivity analysis (DSA) and acoustic optimization have used the structural sensitivity analysis module supported by commercial codes and have thus faced limitations in the accuracy and number of design variables.

There are several reasons for the limitations of the semianalytical method. First, when the semianalytical method is used for structural sensitivity analysis, a slight error could occur due to the amount of perturbation. For this paper, we used a continuum approach[4–6] to calculate the structural sensitivity combined with acoustic sensitivity. The continuum method is usually more efficient and accurate than the semianalytical method because the continuum method is independent of the design perturbation. Therefore, the global acoustic design sensitivity obtained by the global acoustic design sensitivity analyzer (GASA) program provides more accuracy in results than previous activities.

Second, it is difficult to use a large number of design variables because the semianalytical method requires too much calculation time when many design variables are used. GASA, however, can calculate the design sensitivity for all elements in a short time by use of the adjoint variable method.[6] The semianalytical method can calculate only part sensitivity, but the continuum method can calculate both element and part sensitivities. Although the part sensitivity result, rather than element sensitivity, is usually used for the optimization, it cannot help the engineer to understand the system characteristics. The element sensitivity plot is very helpful in giving the engineer intuition into understanding a system tendency. For example, Cunefare et al.[3] used two groups of design variables to compare the influence of variations in the design variables, circumferential and longitudinal types of element group, in the cylinder. The use of element sensitivity means that the best part or group of elements can be selected for optimization. This is discussed in detail in Sec. VI of this paper.

Furthermore, use of a continuum method can provide other advantages. Because the continuum method does not take differentiation to the discretized equation but to energy variational formulations, various energy contributions are also available (for instance, the membrane effect, bending effect, transverse shear effect, and mass effect). These energy contribution plots and related explanations are presented in Sec. V.

In Secs. II and III, the basic formulations to calculate acoustic sensitivity and structural sensitivity are introduced. In Sec. IV, the use of GASA to effectively obtain the global acoustic design sensitivity is explained. In Sec. V, a half-scale automobile cavity model is considered to confirm accuracy of the results of global acoustic design sensitivity obtained by GASA. Finally, in Sec. VI, the design optimization is performed to reduce the weight of structure and sound pressure level for a interesting point at three frequencies.

II. Global Acoustic Design Sensitivity Analysis

The discussion will be of externally semicoupled problems where the structural vibration generates noise but is not influenced by the acoustic medium. In this investigation, we concentrated on the global acoustic design sensitivity to structural design parameters, assuming that the shape is invariant. Field pressure sensitivity is evaluated by a two-step approach to combining acoustic and structural sensitivities.

When a finite body with surface S vibrates in an infinite area or volume V, as shown in Fig. 1, modeling radiation from a vibrating structure is usually performed using a boundary integral

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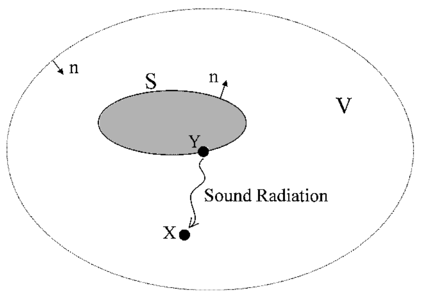


Fig. 1 Domain definition with finite surface S and infinite volume V.

formulation. The direct helmholtz integral equation can be obtained from the Helmholtz integral equation and the relationship between the normal derivative of pressure p and normal velocity v_n, $\partial_n p = -j\omega\rho \cdot v_n$, where ρ is the fluid density and ω is the circular frequency:

$$c(X)p_e(X) = \int_S \left\{ p_s(Y)\frac{\partial G(X,Y)}{\partial n_Y} + j\rho\omega v_n(Y)G(X,Y) \right\} \mathrm{d}S(Y) \tag{1}$$

In Eq. (1), $p_e(X)$ denotes the pressures at the field points X, and $p_s(Y)$ denotes the pressures at the structure surface Y. Green function $G(X, Y)$ is defined as the solution for a point source with unit source strength, and $c(X)$ is the coefficient vector with respect to the position of X. If field points X locate on the structure surface S, Eq. (1) leads to the derivation of boundary element equations that make it possible to relate the boundary-nodal pressures to the boundary-nodal normal velocities:

$$\boldsymbol{A}(\omega)\boldsymbol{p}_s = \boldsymbol{B}(\omega)\boldsymbol{v}_n \tag{2}$$

In Eq. (2), $\boldsymbol{A}$ and $\boldsymbol{B}$ are frequency-dependent matrices (full, complex, and unsymmetric), and $\boldsymbol{p}_s$ and $\boldsymbol{v}_n$ are vectors of nodal pressures and nodal normal velocities on the surface, respectively. Usually, the normal velocities are prescribed on the boundary surface so that nodal pressure can be computed from Eq. (2):

$$\boldsymbol{p}_s = \boldsymbol{C}(\omega)\boldsymbol{v}_n \tag{3}$$

where

$$\boldsymbol{C}(\omega) = \boldsymbol{A}(\omega)^{-1}\boldsymbol{B}(\omega) \tag{4}$$

Knowledge of boundary pressure and normal velocity values allows a discrete representation of pressure p_e at field points X with some distance from the structure by use of Eq. (1):

$$p_e(X) = \{\boldsymbol{A}_e(\omega)\}\boldsymbol{p}_s - \{\boldsymbol{B}_e(\omega)\}\boldsymbol{v}_n \tag{5}$$

$\{\boldsymbol{A}_e\}$ and $\{\boldsymbol{B}_e\}$ are row vectors of influence coefficients that result from the discrete form of Eq. (1). These coefficients are obtained from the discrete boundary surface and frequency. Sensitivity of field pressure p_e to a change of the design variable a_k at element k can be evaluated through the differentiation of Eq. (5) with respect to a_k:

$$\frac{\partial p_e}{\partial a_k} = \frac{\partial\{\boldsymbol{A}_e\}}{\partial a_k}\boldsymbol{p}_s + \{\boldsymbol{A}_e\}\frac{\partial \boldsymbol{p}_s}{\partial a_k} - \frac{\partial\{\boldsymbol{B}_e\}}{\partial a_k}\boldsymbol{v}_n - \{\boldsymbol{B}_e\}\frac{\partial \boldsymbol{v}_n}{\partial a_k} \tag{6}$$

If only sizing sensitivity is considered, the parameter a_k does not influence row vectors $\{\boldsymbol{A}_e\}$ and $\{\boldsymbol{B}_e\}$, and Eq. (6) is reduced to

$$\frac{\partial p_e}{\partial a_k} = \{\boldsymbol{A}_e\}\frac{\partial \boldsymbol{p}_s}{\partial a_k} - \{\boldsymbol{B}_e\}\frac{\partial \boldsymbol{v}_n}{\partial a_k} \tag{7}$$

Equation (7) shows that the sensitivity of the field pressures to a change in the structural sizing design variable requires the evaluation of the sensitivity of surface pressure vector $\boldsymbol{p}_s$ to this change. This can be obtained by differentiating Eq. (3) and substituting into Eq. (7):

$$\frac{\partial p_e}{\partial a_k} = (\{\boldsymbol{A}_e\}\boldsymbol{C} - \{\boldsymbol{B}_e\}) \cdot \frac{\partial \boldsymbol{v}_n}{\partial a_k} \tag{8}$$

The right-hand side of Eq. (8) is multiplied by two terms. The first term, in parentheses, represents acoustic sensitivity, and second term represents the structural sensitivities for the structural normal velocities with respect to the sizing design variable. In Part III, the structural sensitivity to multiply with acoustic sensitivity, as shown in Eq. (8), is derived.

III. Structural Sizing Design Sensitivity Analysis

The continuum structural sensitivities for a nodal velocity with respect to the structural design variable can be obtained from the variational equation of a fully coupled structural acoustic system[5]:

$$b_u(z,\bar{z}) - \iint\limits_{\Gamma^{as}} \boldsymbol{p}\bar{z}^{*T} n \, d\Gamma + d(\boldsymbol{p},\bar{\boldsymbol{p}}) - \omega^2 \iint\limits_{\Gamma^{as}} \bar{\boldsymbol{p}}^* z^T n \, d\Gamma = l_u(\bar{z}) \tag{9}$$

Equation (9) must hold for all kinematically admissible virtual states $(\bar{z}^*, \bar{\boldsymbol{p}}^*) \in Q$ where Q is a complex vector space:

$$Q = \{(z,\boldsymbol{p}) \in Z \otimes P | \boldsymbol{f}_p = \boldsymbol{p}n \text{ and}$$
$$\nabla \boldsymbol{p}^T n = \omega^2 \rho_0 z^T n, x \in \Gamma^a \equiv \Omega^s\} \tag{10}$$

where

$$Z = \{z \in \ \boldsymbol{H}^2(\Omega^s)]^3 \mid Gz = 0, x \in \Gamma^s\}$$

$$P = \{\boldsymbol{p} \in \boldsymbol{H}^1(\Omega^a) \mid \nabla \boldsymbol{p}^T n = 0, x \in \Gamma^{as}\}$$

and $\boldsymbol{H}^1$ and $\boldsymbol{H}^2$ are complex Sobolev spaces of orders one and two, respectively. In Eqs. (9) and (10), z and $\boldsymbol{p}$ are structural displacement and acoustic pressure in the closed structure and the asterisk denotes complex conjugate. In Eq. (9), the sesquilinear forms $b_u(\cdot,\cdot)$ and $d(\cdot,\cdot)$ and the semilinear form $l_u(\cdot)$ are defined, using complex L_2-inner product $(\cdot,\cdot)$ on a complex function space, as

$$b_u(z,\bar{z}) = -\iint\limits_{\Omega^s} \omega^2 m \bar{z}^{*T} z \, \mathrm{d}\Omega + j\omega c_u(z,\bar{z}) + a_u(z,\bar{z}) \tag{11}$$

where

$$c_u(z,\bar{z}) = \iint\limits_{\Omega^s} \bar{z}^{*T} C_u z \, \mathrm{d}\Omega, \qquad a_u(z,\bar{z}) = \iint\limits_{\Omega^s} \bar{z}^{*T} A_u z \, \mathrm{d}\Omega \tag{12}$$

$$d(\boldsymbol{p},\bar{\boldsymbol{p}}) = \iiint\limits_{\Omega^s} \left(-\frac{\Omega^2}{\beta}\boldsymbol{p}\bar{\boldsymbol{p}}^* + \frac{1}{\rho_0}\nabla \boldsymbol{p}^T \nabla \bar{\boldsymbol{p}}^* \right) \mathrm{d}\Omega \tag{13}$$

$$l_u(\bar{z}) = \iint\limits_{\Omega^s} \boldsymbol{f}^T \bar{z}^* \, \mathrm{d}\Omega \tag{14}$$

In Eq. (12), C_u is the linear differential operator that corresponds to the damping of the structure and A_u is the fourth-order symmetric partial differential operator for the structure. In Eq. (13), ω is forced frequency, and β and ρ_0 are bulk modulus and mass density of acoustic medium, respectively.

Equation (9) is solved using finite element analysis (FEA) codes such as MSC/NASTRAN. The structural displacement z and acoustic pressure $\boldsymbol{p}$ are approximated using shape functions, as:

$$\begin{bmatrix} [-\omega^2 M_{ss} + j\omega C_{ss} + K_{ss}] & K_{sf}] \\ [-\omega^2 M_{fs}] & [-\omega^2 M_{ff} + K_{ff}] \end{bmatrix} \begin{Bmatrix} z \\ \boldsymbol{p} \end{Bmatrix} = \begin{Bmatrix} \boldsymbol{f} \\ \boldsymbol{0} \end{Bmatrix} \tag{15}$$

Direct and modal frequency FEA methods are used to solve Eq. (15).

For the adjoint variable method, first consider the structural displacement at a point $\hat{x}$ in the domain Ω by the structure under harmonic excitation:

$$\Psi = \iiint\limits_{\Omega^s} \hat{\delta}(x - \hat{x}) z_i \, \mathrm{d}\Omega, \qquad i = 1,2,3 \tag{16}$$

The first variation of Eq. (16) is

$$\Psi' = \iiint_{\Omega^s} \hat{\delta}(x - \hat{x}) z_i' \, \mathrm{d}\Omega \tag{17}$$

Using the adjoint variable method, design sensitivity expression can be obtained, as:

$$\Psi' = l_{\delta u}'(\lambda) - b_{\delta u}'(z, \lambda) = \iint_{\Omega^s} f_u^T \lambda^* \delta u \, \mathrm{d}\Omega$$

$$+ \iint_{\Omega^s} \omega^2 m_u \lambda^{*T} z \delta u \, \mathrm{d}\Omega - i\omega c_{\delta u}'(z, \lambda) - a_{\delta u}'(z, \lambda) \tag{18}$$

where adjoint response λ is obtained from adjoint equation

$$b_u(\bar{\lambda}, \lambda) = \iint_{\Gamma^{\mathrm{as}}} \bar{\eta} \lambda^{*T} n \, \mathrm{d}\Gamma + d(\bar{\eta}, \eta) - \omega^2 \iint_{\Gamma^{\mathrm{as}}} \eta^* \bar{\lambda}^T n \, \mathrm{d}\Gamma$$

$$= \iint_{\Omega^s} \hat{\delta}(x - \hat{x}) \bar{\lambda}_i \, \mathrm{d}\Omega \tag{19}$$

which must hold for all kinematically admissible virtual states $(\bar{\lambda}, \bar{\eta}) \in Q$ where λ and η are adjoint structural displacement and acoustic pressure, respectively. Thus, the structural sensitivity can be calculated by substituting original responses and adjoint responses into Eq. (18). Original response can be obtained from commercial FEM codes such as MSC/NASTRAN. Adjoint response can be also calculated using MSC/NASTRAN by applying the adjoint load at the performance point.

If a system is not coupled or opened, then the coupled terms and acoustic term can be neglected. Use of Eq. (18) makes it possible to obtain the design sensitivity of interior acoustic pressure. For an exterior noise problem, however, the result of Eq. (18) has to be combined with acoustic sensitivity in Eq. (8). Structural velocity sensitivity in Eq. (8) can be obtained by multiplying a scale factor to the displacement sensitivity result of Eq. (18).

IV. Global Acoustic Design Sensitivity

The global acoustic design sensitivity can be determined by multiplying two sensitivities, acoustic sensitivity and structural sensitivity, using the chain rule. The number of calculations is usually dependent on the number of faced nodes with the interesting acoustic medium. For instance, when the number of design variables is k and the number of nodes is m, which are faced the acoustic medium in interesting section (interior or exterior), the global sensitivity results for a field point are going to organize the $1 \times \boldsymbol{k}$ row vector and acoustic sensitivity will be the $1 \times \boldsymbol{m}$ row vector. Thus, the structural sensitivity part has to be the $\boldsymbol{m} \times \boldsymbol{k}$ matrix:

$$[gs_1 \quad \cdots \quad gs_k]_{1 \times k} = [as_1 \quad \cdots \quad as_m]_{1 \times m}$$

$$\times \begin{bmatrix} ss_{1,1} & \cdots & ss_{1,k} \\ \cdots & \cdots & \cdots \\ ss_{m,1} & \cdots & ss_{m,k} \end{bmatrix}_{m \times k} \tag{20}$$

In Eq. (20), gs_i means the global acoustic sensitivity of sound pressure at the interesting field point with respect to the ith element, as_j means acoustic sensitivity of a normal velocity at the jth node, and $ss_{j,i}$ means structural sensitivity of normal velocity at the jth node with respect to the ith element. As shown in Eq. (20), the $\boldsymbol{m}$ times calculations are needed to obtain the required structural sensitivities. Moreover, because the performance of structural sensitivity is a nodal velocity in normal direction, the three velocities in x direction, y direction, and z direction are considered together. Although the adjoint variable method can calculate the sensitivities in one time regardless of the number of design variables, for the global acoustic sensitivity, $\boldsymbol{m} \times 3$ times adjoint analyses by MSC/NASTRAN are needed for the structural sensitivity because the number of performances is $\boldsymbol{m} \times 3$. When the first version of GASA was used to calculate the global acoustic design sensitivity, for the example in Sec. V a total of 178 minutes of CPU time was needed. Table 1 gives details of calculation times for the first and second versions for comparison. As shown in Table 1, most of the calculation time for the first version was used for adjoint analyses.

Table 1 Calculation time analysis for the first version and the second version of GASA

Analysis	Time, min: 1st version	2nd version
Acoustic sensitivity	4	4
Adjoint	127	39
Structural sensitivity	38	27
Global acoustic sensitivity	9	9
Total	178	79

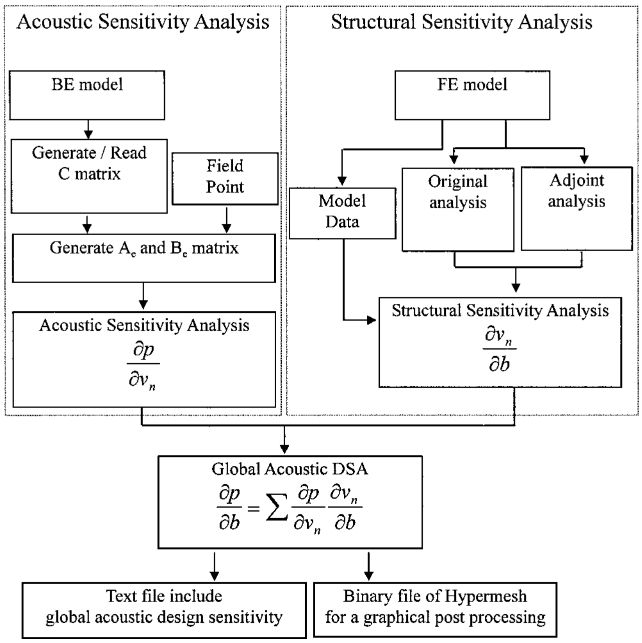


Fig. 2 Computation procedures of the global acoustic design sensitivity analysis.

Thus, in the second version of GASA, two modules were added to reduce the calculation time and cost. One was developed to apply the adjoint loads directly to the normal direction. The other module was developed to automatically run MSC/NASTRAN for adjoint analyses. Therefore, the total calculation times are reduced to 44.38%. The computation procedures of GASA and its two types of results are shown in Fig. 2. One result is a text file, and the other is Hypermesh, a commercial, pre-post program, binary file for a graphical postprocessing. Several formats of text file are available through use of options supplied by GASA: element sensitivity, part sensitivity, sort the results, and separate the result with regard to the element type.

V. Numerical Examples

The simplified vehicle cavity model shown in Fig. 3, which is a half-scale of an actual automobile, was employed for verification of global acoustic sensitivity analysis results. This model has 278 elements and 280 nodes. Because the interesting range of frequency is from 20 to 500 Hz for this example, the maximum element size has to be smaller than about 10 cm because the maximum element size must be smaller than one-sixth of the wavelength. The model used a 2-mm plastic panel to get a significant coupling effect between the structure and the interior acoustic medium. It is simply supported at four points and excited at the rear-bottom point as shown in Fig. 4. Usually, the interesting point to predict exterior noise is prefixed. For instance, to pass the standard for an exterior noise such as ISO, the measurement is performed at a stipulated point according to the standard. As shown in Fig. 4, one node (point 81) is considered as a field point shown for the exterior noise prediction. Figure 5 shows

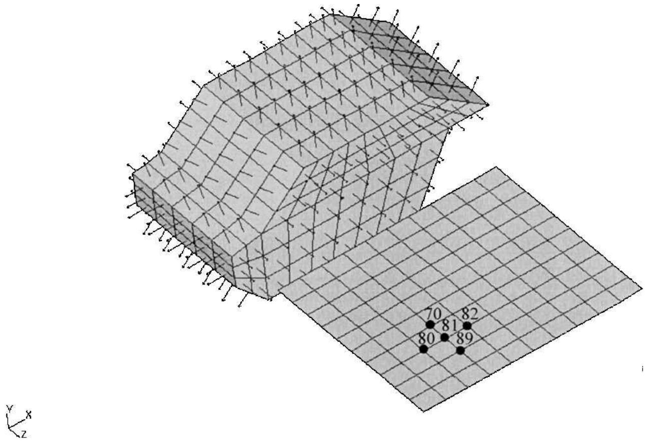


Fig. 3 Half-scale vehicle cavity model and field points mesh.

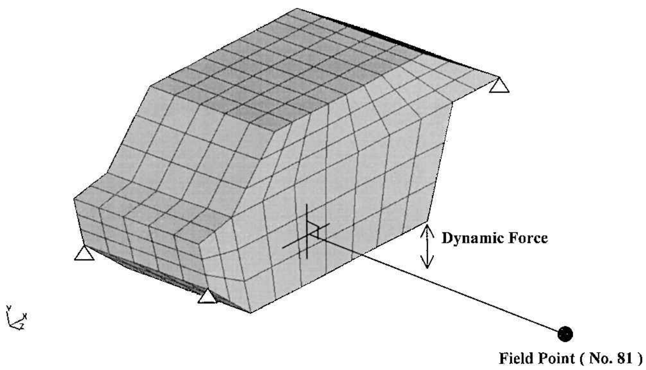


Fig. 4 Boundary conditions and a selected field point (point 81).

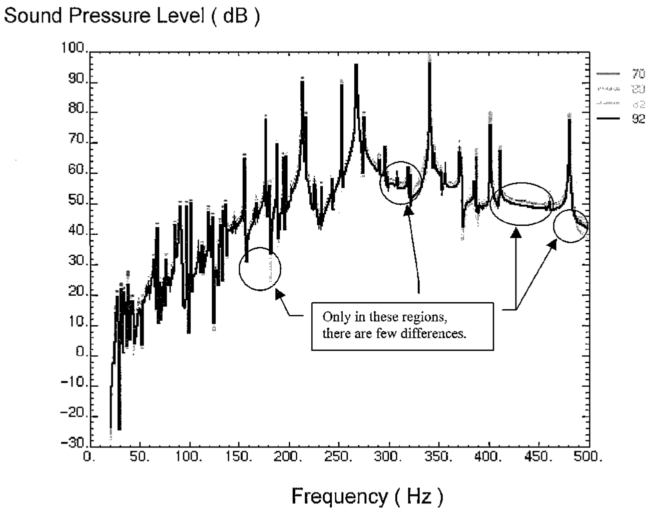


Fig. 5 Sound level plot at neighbored points of the selected field point (points 70, 80, 82, and 92).

that the neighboring points (79, 81, 82, and 90 in Fig. 3) have almost the same pressure distribution, except for four ranges of frequencies near 185, 330, 450, and 490 Hz. Thus, a field point in the middle of the interesting area is enough to predict its exterior noise quality.

The acoustic analysis is carried out after the structural analysis. As shown in Figs. 6 and 7, the pressure of field point 81 is greatest at 341 Hz. Thus, point 81 and 341 Hz are selected as the field point and the target frequency, respectively.

The global acoustic design sensitivity for the pressure at the field point with respect to the structural thickness of each element is shown in Figs. 8 and 9. In Fig. 9, the horizontal axis indicates the element identification number (EID), and the vertical axis indicates the

Table 2 Sensitivity verification for element 109 at 341 Hz

Perturbation, δd(mm)	$\psi(d+\delta d)$, Pa	$\psi(d-\delta d)$, Pa	$\Delta\psi$, Pa/mm	ψ', Pa/mm	$\psi'/\Delta\psi$, %
0.1	1.140	4.956	−19.08		168.60
0.01	1.798	2.436	−31.90	−32.1764	100.87
0.001	2.062	2.126	−32.00		100.55

Table 3 Sensitivity verification for element 266 at 341 Hz

Perturbation, δd(mm)	$\psi(d+\delta d)$, Pa	$\psi(d-\delta d)$, Pa	$\Delta\psi$, Pa/mm	ψ', Pa/mm	$\psi'/\Delta\psi$, %
0.1	6.274	1.032	26.21		88.43
0.01	2.319	1.866	22.65	23.1762	102.32
0.001	2.116	2.071	22.50		103.01

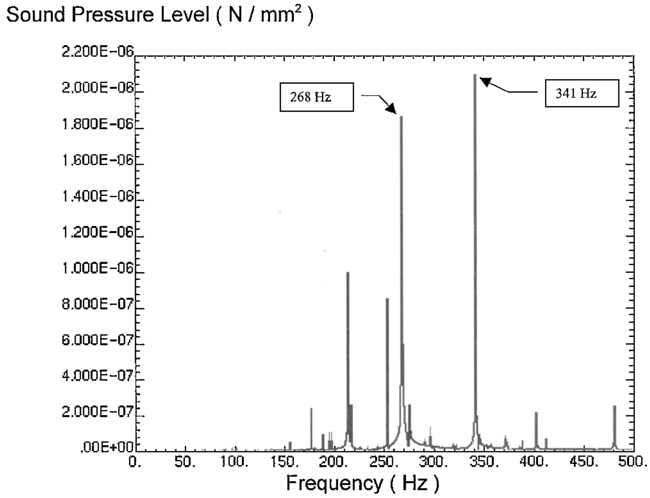


Fig. 6 Sound level plot at field point 81.

magnitude of global acoustic sensitivities. Two types of sensitivities can be observed in Fig. 9. Positive sensitivity means that the pressure at the field point will be increased when the thickness of element is increased. Negative sensitivity means that the pressure will be decreased when the thickness is increased. To verify the accuracy of the sensitivity coefficient, sensitivities of two elements are selected: element 266, which has maximum positive sensitivity, and element 109, which has maximum negative sensitivity. In Tables 2 and 3, the central finite difference method is used to compare the sensitivity with global acoustic sensitivity calculated by GASA. The central finite difference sensitivity is $\Delta\psi = \psi(d+\delta d) - \psi(d+\delta d)]/2\delta d$, and ψ' is a calculated global acoustic design sensitivity coefficient. These two tables show that the sensitivity coefficients calculated by GASA are accurate.

The energy contributions for the global acoustic design sensitivities are also available in GASA. Because the continuum method was employed in GASA, the derivative of energy variational equations with respect to the structural design variables has to be integrated according to the energy types, such as membrane energy, bending energy, transverse shear energy, and mass. As shown in Fig. 10a, the total global acoustic sensitivities were composed of the four kinds of energy effects. In the lower view of total sensitivity plot, the large negative sensitivities were distributed to the front and rear, lower dash, and back shelf, according to the definitions in Fig. 11. The large positive sensitivities were distributed to rear floor. Therefore, as shown in Fig. 10b–10e, the positive sensitivities in the rear floor occurred only as a result of mass effect. Because the positive sensitivities were leaded by mass effect, a decrease in thickness was the best way to decrease both sound pressure and weight. For the negative sensitivities in the lower dash and back shelf, an increase in thickness caused a simultaneous decrease of sound pressure and increase of weight. From the energy contribution plots, we can easily recognize that the main source of

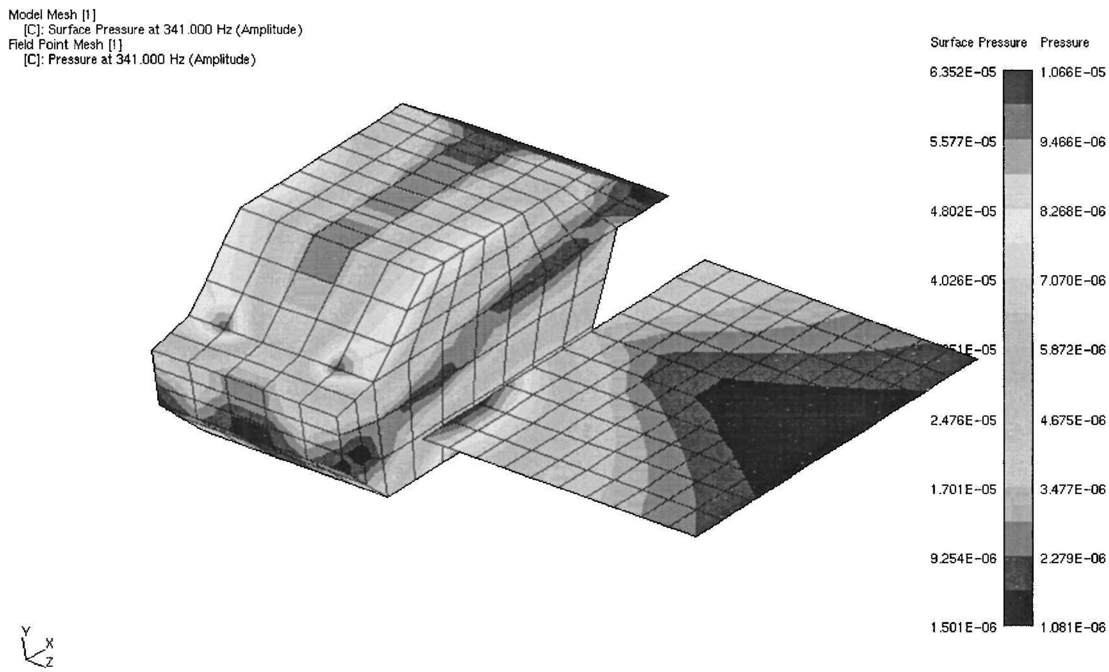


Fig. 7 Contour plot of the surface pressure and field points pressure at 341 Hz.

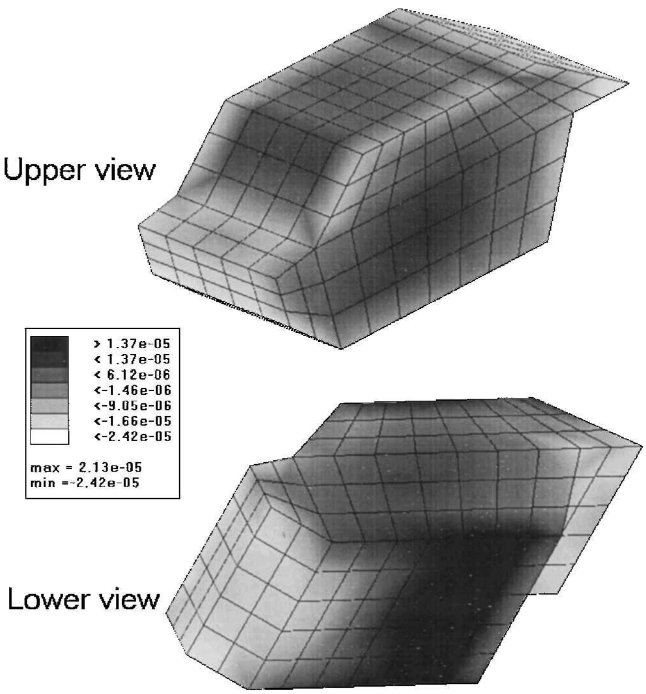


Fig. 8 Contour plot of the global acoustic design sensitivity results.

the negative sensitivities is the membrane energy for this case. Using this information, it is possible to decrease the sound pressure without weight increase by employing the rib to prevent membrane vibration. This information on energy contribution is very useful to engineers because it can help them to clearly understand the system characteristics.

VI. Acoustic Optimization

The optimum design of the half-scale vehicle cavity model is obtained by using sequential linear programming (SLP) with global acoustic design sensitivities obtained by GASA. The thickness of each part, which has the same properties, is selected as the design variable. Figure 11 gives the part identification (PID) numbers of

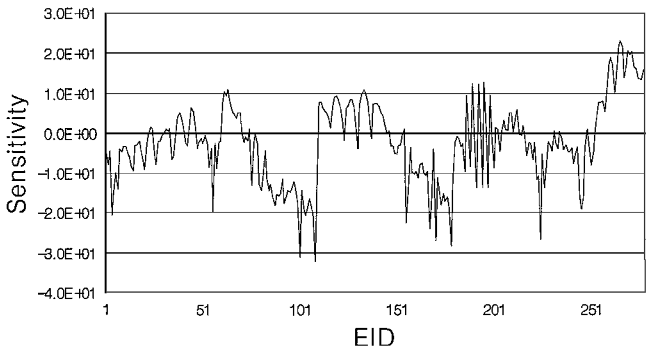


Fig. 9 Results of global acoustic design sensitivities for each element.

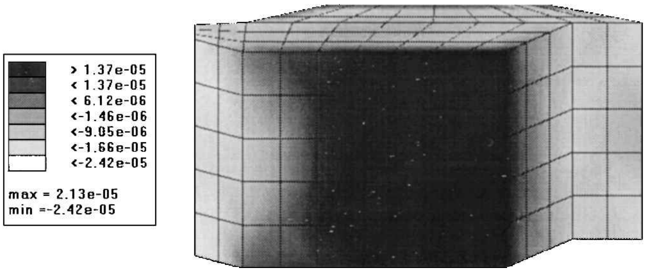
this model. Originally, there was only one floor part in the analysis model. However, the floor part of the original model was separated into two parts, front floor and rear floor, because the large sensitivities in the floor part had clustered on the rear section of the floor, as shown in Fig. 10. This is the advantage of the element sensitivity plot.

The optimization procedure with the SLP algorithm is shown in Fig. 12. In this example, an optimization problem is set to simultaneously reduce the weight of the structure and the sound pressure level (SPL) at field point 81 and at the target frequency of 341 Hz. Because it is possible that the peak point of pressure is just shifted without magnitude reduction, three frequencies, target frequency and side frequencies, are considered. For the side frequencies, however, a little increase of pressure is permitted because their initial pressures are very small. All of the design variables can be changed from 1.5 to 2.5 mm. The global acoustic design sensitivities of a field point pressure with respect to the thickness of the structure are used for the gradient of constraint functions. A standard form of the design optimization problem is written as follows:

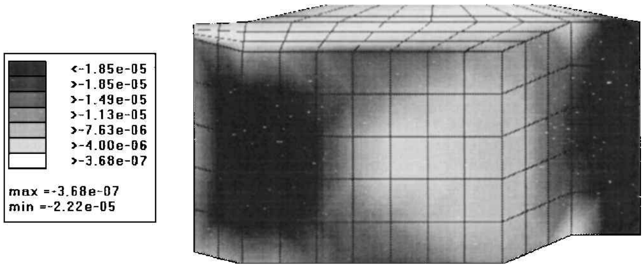
1) Minimize the weight (kilograms).

2) Subject to SPL at $340 < 0.5$ Pa, initial value: 0.161 Pa; SPL at $341 < 0.5$ Pa, initial value: 2.093 Pa; and SPL at $342 < 0.5$ Pa, initial value: 0.197 Pa.

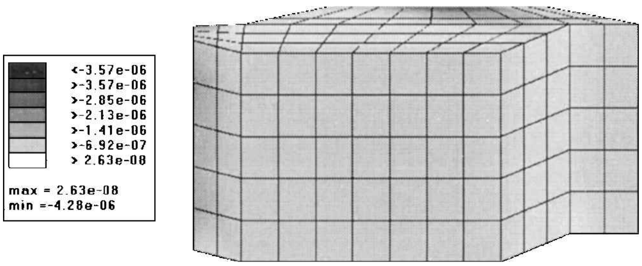
3) Side constraints are $1.5 \leq t_1 \leq 2.5$ mm, with an initial value of 2 mm, and where $i = 1$–14, number of design variables. The move limit was decided as 1% of initial value for each iteration, because of its nonlinearity: $-0.01 \leq d_t \leq +0.01$.



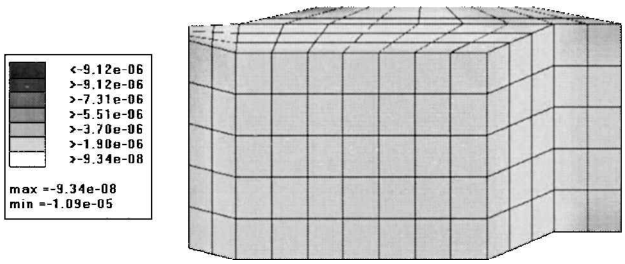
a) Total sensitivity (both positive and negative sensitivities)



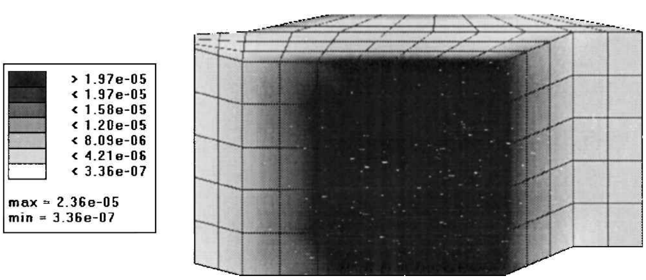
b) Membrane effect (large negative sensitivity)



c) Bending effect (small negative sensitivity)



d) Transverse shear effect (small negative sensitivity)



e) Mass effect (positive sensitivity)

Fig. 10 Energy contribution for global acoustic design sensitivity.

Optimum design was obtained after five iterations. The optimum design is shown in Table 4, and design iteration plots for cost, constraints, and design variables are shown in Figs. 13 and 14. Figure 15 is a frequency response plot, which includes the comparison of initial and optimum design.

After optimization procedures, the weight of the structure is reduced about 2%, and the sound pressure at 341 Hz is also decreased to 20% of initial value. Although the pressure at 342 Hz is increased to 150%, the final pressure is very small. Thus, the amount of

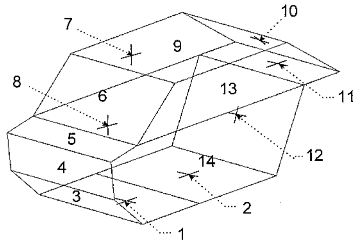


Fig. 11 Part identification number (PID) of the half-scale vehicle cavity model for optimization: 1, front floor; 2, rear floor; 3, lower dash; 4, dash; 5, upper dash; 6, wind shield; 7, side window; 8, door; 9, roof; 10, back window; 11, back shelf; 12, seat back; 13, side window; and 14, door.

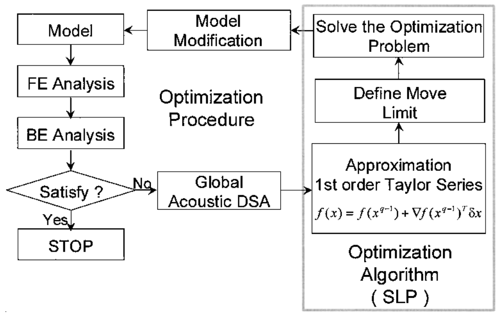
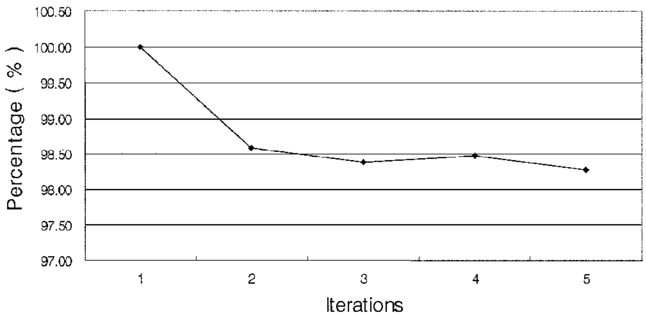
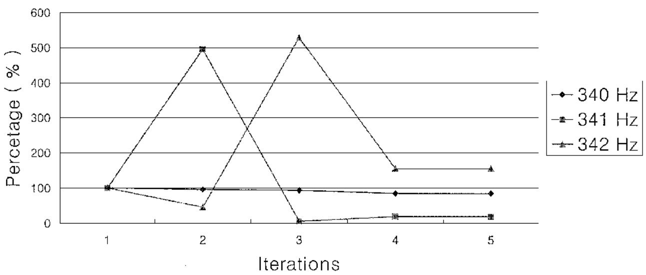


Fig. 12 Optimization procedure with SLP algorithm.



Optimization history of object function



Optimization history of constraints

Fig. 13 Design iteration plot.

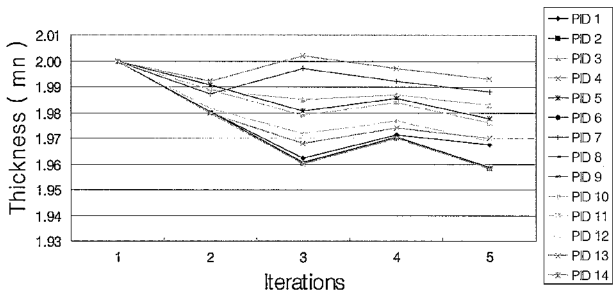


Fig. 14 Design iteration plot for the design variables.

Table 4 Design parameters at initial and optimum design

Design parameter, mm	Initial design, mm	Lower bound, mm	Upper bound, mm	Optimum design, mm
PID1	2.0	1.5	2.5	1.9675
PID2	2.0	1.5	2.5	1.9583
PID3	2.0	1.5	2.5	1.9830
PID4	2.0	1.5	2.5	1.9931
PID5	2.0	1.5	2.5	1.9778
PID6	2.0	1.5	2.5	1.9583
PID7	2.0	1.5	2.5	1.9882
PID8	2.0	1.5	2.5	1.9587
PID9	2.0	1.5	2.5	1.9584
PID10	2.0	1.5	2.5	1.9689
PID11	2.0	1.5	2.5	1.9760
PID12	2.0	1.5	2.5	1.9583
PID13	2.0	1.5	2.5	1.9700
PID14	2.0	1.5	2.5	1.9583

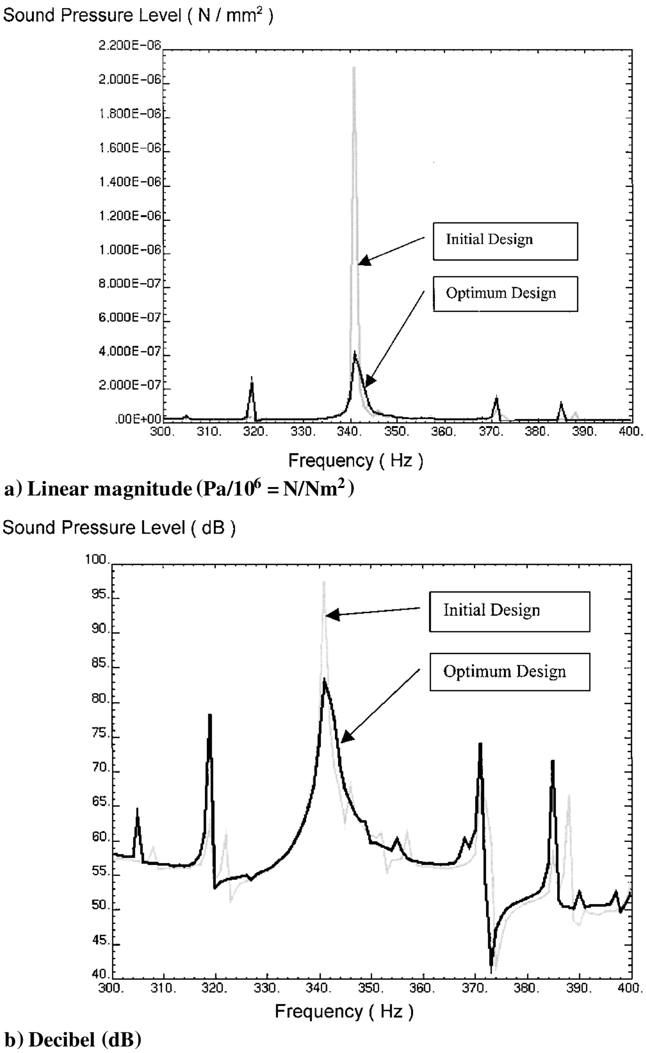


Fig. 15 Comparison of sound pressure of initial and optimum design.

increased pressure at 342 Hz can be ignored. As shown in Fig. 15b, all sound pressures at three frequencies (340, 341, and 342 Hz) are smaller than 85 dB. With the design optimization, a significant noise reduction has been achieved without increasing weight.

VII. Conclusions

The GASA program to calculate global acoustic design sensitivity was developed based on the continuum approach. Global acoustic design sensitivity comprises acoustic sensitivity for a field point pressure with respect to the normal velocity on the surface of the structure and structural design sensitivity for velocities on the surface of structure with respect to structural design variables. For an automated optimization process, the global acoustic design sensitivity is important as the gradient to reduce the structure-noise by modifying the structural design.

Use of the continuum method for GASA allows the acoustic sensitivity, structural continuum sizing design sensitivity, and global acoustic design sensitivity to be computed more accurately and efficiently than in the semianalytical approach. The continuum method can be used to obtain additional information in the energy contribution and detail distribution of the sensitivity results by using a element sensitivity coefficient. Using these results, design engineers can easily recognize the design intent and understand system characteristics. Also, design engineers can easily optimize structures when complex structures or systems are analyzed.

A half-scale vehicle cavity model shows that the global acoustic design sensitivity coefficients, calculated using GASA, are accurate. This global acoustic design sensitivity can be effectively applied to general sizing design problems. Optimization procedures are also performed to reduce both the weight and sound pressure by use of global acoustic design sensitivities and the SLP algorithm.

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