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Visual Interaction in Acoustic-structural Analysis of Automobiles

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Abstract

This paper describes a graphical technique used for investigating the effects of structural design changes on the sound pressure levels generated in an automobile compartment. Low frequency noise in the passenger compartment (in approximately the 20-85 Hz frequency range) is of primary interest, and particularly that noise which is generated by the structural vibrations of the wall panels and various other smaller parts of the compartment. A designer faced with reducing the noise levels at various frequencies typically has several hundred design variables as candidates for change. This paper describes a computer interface that facilitates understanding of the effect various design changes have on the noise levels. This interface allows the designer to investigate the design space to determine which variables hold the most promise for achieving the objectives of noise reduction and allows the designer to set targets or constraints, to a desired confidence level, using the sensitivity information of these design variables. The design targets set in this manner are then input to an optimization algorithm which solves for the optimized sound pressure level.

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VISUAL INTERACTION IN ACOUSTIC-STRUCTURAL ANALYSIS OF AUTOMOBILES

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ABSTRACT

This paper describes a graphical technique used for investigating the effects of structural design changes on the sound pressure levels generated in an automobile compartment. Low frequency noise in the passenger compartment (in approximately the 20-85 Hz frequency range) is of primary interest, and particularly that noise which is generated by the structural vibrations of the wall panels and various other smaller parts of the compartment. A designer faced with reducing the noise levels at various frequencies typically has several hundred design variables as candidates for change. This paper describes a computer interface that facilitates understanding of the effect various design changes have on the noise levels. This interface allows the designer to investigate the design space to determine which variables hold the most promise for achieving the objectives of noise reduction and allows the designer to set targets or constraints, to a desired confidence level, using the sensitivity information of these design variables. The design targets set in this manner are then input to an optimization algorithm which solves for the optimized sound pressure level.

INTRODUCTION

Structural-acoustic analysis concentrates on reducing interior noise in transportation vehicles by modifying the design. Takata (1988, 1990) classified interior car noise into a high frequency component and a low frequency component (with 85 Hz as boundary). The high frequency component consists mainly of the air-borne noise, whereas the low frequency component consists primarily of structure-borne noise. This paper concentrates only on low frequency noise and how that level can be changed through structural redesign. Sophisticated numerical methods including finite element and finite difference schemes are readily available in commercial software, such as MSC/NASTRAN (1991), for modeling both the linear acoustic medium and the surrounding structure. In addition, linear sensitivities of the effect of design changes on the noise levels can also be calculated. The method presented in this paper combines sensitivity analysis with computer visualization that allows the effect of the design changes on the noise level in the passenger compartment to be investigated interactively.

STRUCTURAL/ACOUSTIC ANALYSIS

One approach to evaluating the sound levels in the passenger compartment is to compute the sound pressure levels (SPL) at the drivers and/or rear passengers ear as the design criteria. This computation has been applied during the advanced and development stage of vehicles. There is a strong contribution of the wall panel motion to the measured noise (or SPL in dBA). This noise may be amplified by the compartment cavity resonance. Control of this low frequency noise might require the use of large amount of absorptive material. A practical solution to the problem would seem to be modification of the vehicle structure. It has been confirmed that the best way to treat this type of problem is by a combined analytical- experimental method (Flanignan and Borders, 1984). The basic operations in a structural/acoustic analysis of an automobile are shown in Figure-1.

Nominal initial targets for the design variables are set by an initial experimental method and/or past experience. These variables are then used in computing design targets to the desired confidence level. A fully analytical approach, starts with generation of a finite element body model and acoustic cavity model and analysis is performed to obtain

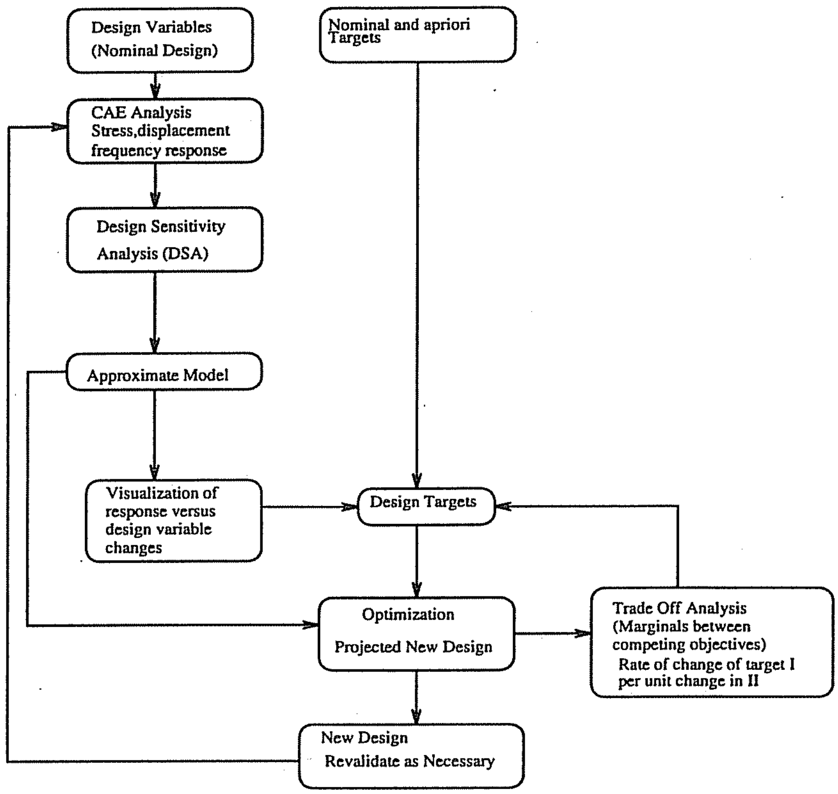

Figure 1: FLOW CHART FOR STRUCTURAL/ACOUSTIC ANALYSIS

necessary stress, displacement, velocity and frequency response information. This is followed by design sensitivity analysis (DSA) which computes the sensitivities of the frequency response with respect to each design variable affecting sound pressure. The sensitivities are obtained by solving the sensitivity equations for the fluid-structure interaction, the fluid being the acoustic medium in the passenger compartment. The sensitivity equations in turn are obtained from the general equations of motion for fluid-structure system as follows. Nefske, et.al. (1982) and Izadpanah, et.al. (1989) presented the following formulation of the finite element method for structural-acoustic analysis of an enclosed cavity.

Coupled Fluid-Structure Sensitivity

The general equations of motion for fluid-structure system can be written as

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{F} \tag{1}$$

The solution vector $\mathbf{x}$ consists of both structural displacements $\mathbf{U}$ and the fluid pressure $\mathbf{P}$.

$$\mathbf{x} = \left\{ \begin{array}{c} \mathbf{U} \\ \mathbf{P} \end{array} \right\} \tag{2}$$

The vector of external loads $\mathbf{F}$ is comprised of both structural as well as fluid forces, or

$$\mathbf{F} = \left\{ \begin{array}{c} \mathbf{F_s} \\ \mathbf{F_f} \end{array} \right\} \tag{3}$$

The mass matrix in Eq.1 is defined as

$$\mathbf{M} = \left[\begin{array}{cc} \mathbf{M_S} & \mathbf{0} \\ \mathbf{A} & \mathbf{M_f} \end{array} \right] \tag{4}$$

where :
$\mathbf{M_S}$ = structural mass
$\mathbf{M_f}$ = fluid mass (including compressibility effects)
$\mathbf{A}$ = fluid-structure coupling matrix

The stiffness matrix in Eq.1 consists of

$$\mathbf{K} = \left[\begin{array}{cc} \mathbf{K_S} & -\mathbf{A}^T \\ \mathbf{0} & \mathbf{K_f} \end{array} \right] \tag{5}$$

where :
$\mathbf{K_s}$ = structural stiffness
$\mathbf{K_f}$ = effective fluid stiffness

If the loads in Eq.1 are frequency dependent, a direct formulation can be written. Differentiating the direct frequency equation yields an expression which can be solved.

In very large problems, it is often desirable to apply the modal decomposition method in order to reduce the cost of the analysis. Modal formulations in coupled fluid-structure analysis are derived from a separate consideration of both the fluid and structural components.

The structural modes are computed for a structure in a vacuum, that is, in the absence of any fluid effects. The modal transformation for the structural degrees of freedom can then be written as

$$\mathbf{U} = \Phi_s \xi_s \tag{6}$$
$$\mathbf{m_s} = \Phi_s^T M \Phi_s \tag{7}$$
$$\mathbf{k_s} = \Phi_s^T K \Phi_s \tag{8}$$

where :
Φ_s = modes for structure in a vacuum

The modes for the fluid are computed under the effects of rigid wall boundary conditions, which eliminates the structural coupling effect. The resulting modal transformation for the fluid degrees of freedom is then:

$$\mathbf{P} = \Phi_f \xi_f \tag{9}$$
$$\mathbf{m_f} = \Phi_f^T M \Phi_f \tag{10}$$
$$\mathbf{k_f} = \Phi_f^T K \Phi_f \tag{11}$$

where :
Φ_f = modes of a fluid enclosed by a rigid container

Performing the dynamic reduction in both the fluid and the structure, by substituting the above transformations in the general equation of motion Eq.1 we get :

$$\begin{bmatrix} \mathbf{m_s} & 0 \\ \Phi_f^T \mathbf{A} \Phi_s & \mathbf{m_f} \end{bmatrix} \begin{Bmatrix} \ddot{\xi}_s \\ \ddot{\xi}_f \end{Bmatrix} + \begin{bmatrix} \mathbf{k_s} & -\Phi_s^T \mathbf{A}^T \Phi_f \\ 0 & \mathbf{k_f} \end{bmatrix} \begin{Bmatrix} \xi_s \\ \xi_f \end{Bmatrix} = \begin{bmatrix} \Phi_s^T \mathbf{F_s} \\ \Phi_f^T \mathbf{F_f} \end{bmatrix} \tag{12}$$

This equation, when differentiated with respect to each design variable, results in the coupled fluid-structure sensitivity equations which can be solved for design sensitivities.

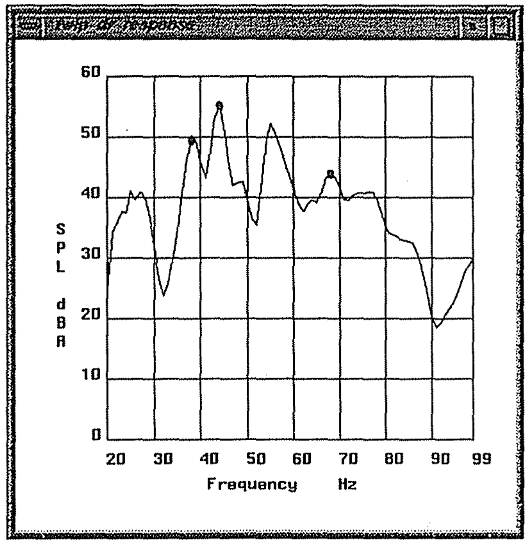

Figure 2: SPL AT THE DRIVERS EAR, FRONT WHEELS IN PHASE (*fwip*) CASE

This paper describes a method of displaying the results of the analysis, combined with the linear sensitivity information, to provide the designer with a powerful tool to aid in selecting target values for the optimization.

INTERACTIVE DESIGN

After the initial analysis has been completed sound pressure levels are available for both the front wheels in phase (fwip) and the front wheels out of phase(fwop) at the driver's ear position and the passenger's ear position. These values are converted to dBA to compensate for diminished human hearing response at low frequency. A baseline plot is drawn that shows the variation in SPL as a function of frequency. Figure-2 shows a baseline plot of the SPL at the drivers ear for the front wheels in phase. The multi-windowing capabilities of todays workstations allow several graphs to be displayed in a customized fashion on the computer at any one time. Windows and graphs are independently shaped and sized by the designer.

The challenge to the designer is how to choose reasonable design changes and targets to reduce the SPL over the frequency range from the hundreds of design parameters which are candidates. Visualization of the effect of multiple design changes on the SPL is a very valuable tool in this situation.

Once the initial analysis has been performed , sensitivities are available at various frequencies which correspond to peak SPL values. These coefficients are calculated from first-order information available from the finite element software, in this case MSC/NASTRAN (1991). This software generates the sensitivity coefficient of a design variable with respect to the pressure level by solving the coupled fluid-

Sensitivities	UNo.	Freq
1364.02	216	38
1467.64	147	38
1547.49	174	38
1637.85	178	38
1655.60	217	38
1800.96	180	38
2615.10	181	38
3217.25	259	38
11086.91	132	38
19189.59	191	38

Sensitivities	UNo.	Freq
-7632.69	140	41
-5803.98	182	41
-3565.69	141	41
-3137.08	155	41
-1898.21	146	41
-1844.28	20	41
-1733.07	157	41
-1467.64	36	41
-1234.24	40	41
-1217.19	42	41

Figure 3: WINDOWS DISPLAYING THE SORTED LIST OF SENSITIVITIES AT THE *fwip* DRIVERS AND PASSENGERS EAR

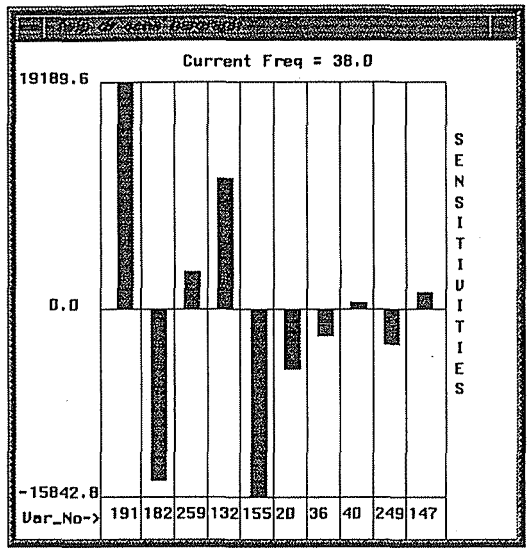

Figure 4: BAR GRAPH OF SENSITIVITIES AT PEAK FREQ. 38 Hz FOR THE *fwip* DRIVERS EAR CASE

structure sensitivity equation which in turn is obtained by differentiating Eq.12 with respect to this design variable.

The sensitivities are then ranked in decreasing order and displayed as a list (Figure-3). This list gives the designer a clear picture of which design variable changes have the largest effect on SPL values at a given peak. Note that the list of sensitivities displayed all correspond to a particular peak frequency. To further clarify the information about the design variables a bar graph showing upto ten sensitivities can be displayed (Figure-4). These sensitivities correspond to the design variables of interest at a certain frequency. The designer can select the design variable sensitivities to be displayed in the bar graph from a window which contains the list of all effective design variables (Figure-5). The designer can use the mouse to select a peak from the window displaying the baseline graph of SPL versus frequency (Figure-2) and change the corresponding peak frequency of the sorted list and the bar graph of sensitivities.

The next step in the visualization process is to choose a design variable, input a design change, and see how the response graphs change. Since the sensitivities are only available at discrete frequencies (peak SPL) and a line graph is needed, a linear interpolation of the sensitivities between peaks is performed:

$$s_{f_n} = s_{f_o} + \left(\frac{s_{f_l} - s_{f_o}}{f_l - f_o}\right)(f_n - f_o) \qquad (13)$$

where :
f_o = peak frequency
f_l = peak frequency where $f_l = f_o + \Delta f$
f_n = frequency of interest
s_{f_n} = new sensitivity at frequency f_n
s_{f_o} = sensitivity at peak frequency f_o
s_{f_l} = sensitivity at another peak frequency f_l

Not all of the design sensitivities for all the variables need to be calculated at frequencies between peaks, only the sensitivity of the variable of interest need be calculated.

The design change is input using a dial box, figure shown, in a window (Figure-5). Note that up to three variables can be picked and changed at any time. The designer can select these variables which correspond to a particular dial from the window displaying the sorted list of sensitivities (Figure-3) using the mouse. The resultant approximation at f_n, of the new SPL values would be a linear summation of the contribution of each design change.

$$P_{f_n} = P_f + \Sigma s_f(e_{f_n} - e_f) \qquad (14)$$

where :
e_{f_n} = new value of design variable at f_n
e_f = value of design variable at f
s_f = sensitivity at frequency f
P_f = SPL at frequency f
P_{f_n} = SPL at frequency f_n

This capability allows the designer to see the combined effects of changing three design variables.

Because the graphics display capabilities of the Silicon Graphics workstations are so fast, changes in the design variable are immediately reflected as changes in the graphs. The outcome is another line graph displayed with the baseline graph, Figure-6 shows the approximated SPL values as the

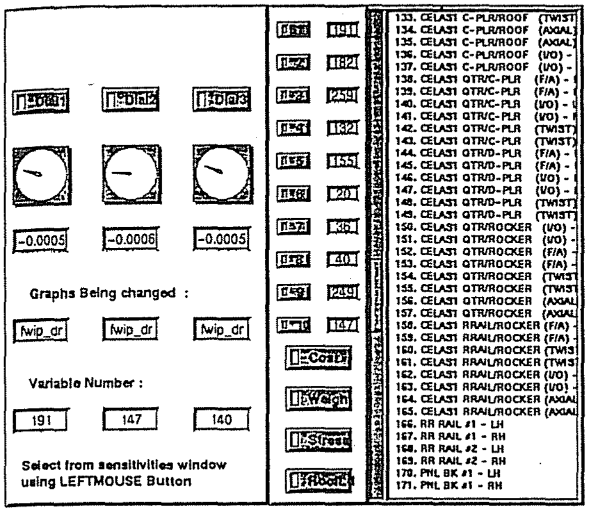

Figure 5: DIALS AND THE SCROLL LIST OF ALL EFFECTIVE DESIGN VARIABLES

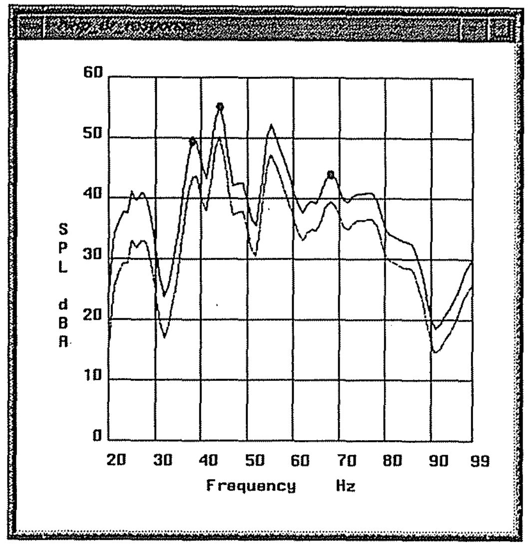

Figure 6: BASE LINE AND MODIFIED SPL

dashed line and the baseline values as the solid line. The designer here has used the dial set in order to achieve an appreciable decrement in SPL across the interval.

The designer can follow the above mentioned procedure for any of the four cases of, front wheels in phase and/or out of phase at drivers ear and/or passengers ear. The response changes for one graph as described above or any/combination of the four graphs at one time can be displayed by making changes to the selected design variables (Figure-7).

Using this visual interaction tool, the designer is able to vary several design variables at any one time and view the resulting changes in sound pressure levels.

CONCLUSIONS

The development, implementation and interpretation of acoustic sensitivities, their effect on design variables and the role of visual interaction in predicting variable change on sound pressure levels has been presented in this paper. The main advantage of visualization and interactive design is that they provide a means for the designer to sort through large amounts of data quickly and see the effects of selected design changes. This provides a way for the designer to set reasonable targets for optimization. One disadvantage of the method presented in this paper lies in the fact that prediction of design changes is carried out using a linear approximation, where higher order sensitivities are neglected. For small changes, linear sensitivities are generally sufficient.

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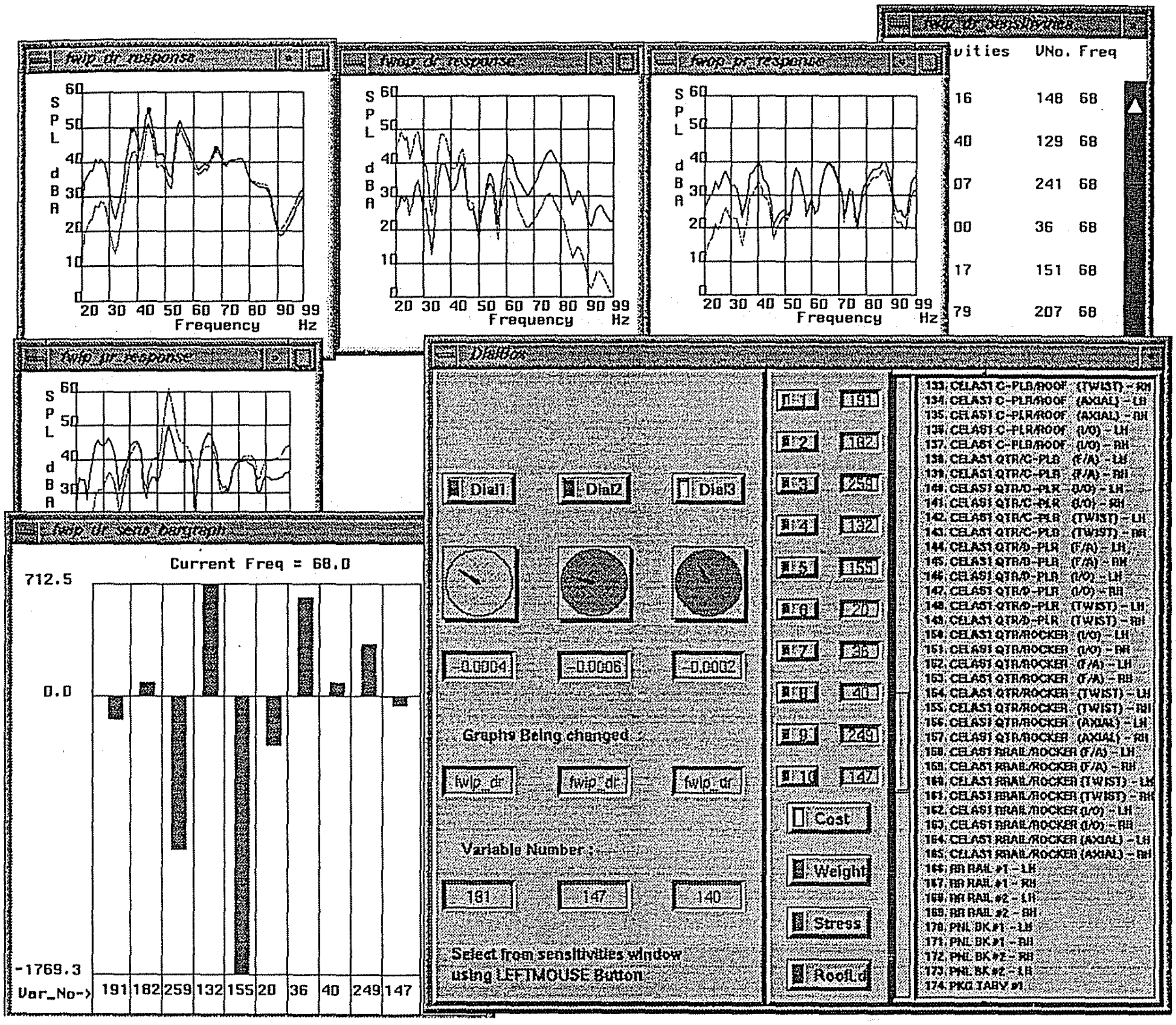

Figure 7: VISUAL INTERFACE SHOWING MULTIPLE GRAPHS, DIAL BOX INPUT AND SENSITIVITIES