

Boundary Condition Effect on the Correlation of an Acoustic Finite Element Passenger Compartment Model

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ABSTRACT

Three different acoustic finite element models of an automobile passenger compartment are developed and experimentally assessed. The three different models are a traditional model, an improved model, and an optimized model. The traditional model represents the passenger and trunk compartment cavities and the coupling between them through the rear seat cavity. The improved model includes traditional acoustic models of the passenger and trunk compartments, as well as equivalent-acoustic finite element models of the front and rear seats, parcel shelf, door volumes, instrument panel, and trunk wheel well volume. An optimized version of the improved acoustic model is developed by modifying the equivalent-acoustic properties. Modal analysis tests of a vehicle were conducted using loudspeaker excitation to identify the compartment cavity modes and sound pressure response to 500 Hz to assess the accuracy of the acoustic models. The optimized acoustic model is also coupled with a structural finite-element model of the trimmed body to evaluate the effect of body panel flexibility on the interior sound pressure response. The optimized acoustic model is found to exhibit the best correlation in terms of the predicted sound pressure FRF response at the passenger compartment interior locations and at the compartment boundary surfaces.

INTRODUCTION

Vehicle engine operating loads and road induced vibrations can excite the passenger compartment acoustic cavity resonance modes and result in high noise levels at specific

frequencies and interior locations in the compartment [1-2]. To diagnose this type of potential noise problem, a structural-acoustic finite element model of the vehicle can be implemented in the early vehicle design stage [3]. The structural-acoustic model couples a structural finite element model of the vehicle with an acoustic finite element model of the passenger compartment. In a previous paper, an acoustic finite-element model of an automobile passenger compartment that includes equivalent-acoustic models of the front and rear seats, front and rear doors, heating, ventilation and air conditioning (HVAC) system, instrument panel (IP), instrument panel (IP) volume, trunk wheel well, and parcel shelf was developed and experimentally assessed using loudspeaker excitation [4]. The equivalent-acoustic model represents the acoustic volumes modeled with acoustic finite elements with modified properties to represent the mass, stiffness, and damping [5, 6].

In the present paper, traditional, improved, and optimized acoustic finite element models of the passenger compartment are developed and experimentally evaluated. The traditional acoustic model represents the passenger and trunk compartment cavities and the coupling between them through the rear seat cavity. The improved acoustic model includes traditional acoustic finite element models of the passenger and trunk compartments, but includes equivalent-acoustic models of the front and rear seats parcel shelf, door volumes, IP and HVAC volumes, and wheel well volume. An optimized version of the improved acoustic model is developed by modifying the equivalent-acoustic properties to improve the accuracy of the predicted sound pressure

response. Modal analysis tests [4] of a vehicle were conducted using loudspeaker excitation to assess the accuracy of the three different acoustic finite element models. The optimized acoustic model is also coupled with a structural finite-element model of the trimmed body to assess the effect of body panel flexibility on the interior sound pressure response. Finally, the predicted sound pressure responses at six passenger compartment boundary surfaces are compared with the measured sound pressure responses to further assess the accuracy of the optimized acoustic model.

FINITE ELEMENT FORMULATION

The passenger and trunk compartments are modeled using the acoustic finite element method where the finite element equation-of-motion that determines the sound pressure response is [5, 6]

$$\left([H_o] + i\omega[D_o] - (\omega/c_o)^2[Q_o]\right)\{p_o\} = i\omega\rho_o\{q_o\} \quad (1)$$

Here p_o is the sound pressure response at the grid points of the model, $[H_o]$, $[D_o]$, $[Q_o]$ are stiffness, damping and mass matrices, q_o is the volume velocity of sound sources, $f = \omega/2\pi$ is the frequency, ρ_o is the air density, c_o is the speed of sound. The seats, doors, IP and HVAC system, trunk wheel well, and parcel shelf are considered as boundary conditions of the passenger and trunk compartment models and are represented as acoustic finite element models with equivalent-acoustic properties to account for their mass, stiffness, and damping. The governing equation-of-motion of the boundary condition volumes is therefore of a similar form to Eq. (1) [5-6],

$$\left([H_s] + i\omega[D_s] - (\omega/c_s)^2[Q_s]\right)\{p_s\} = i\omega\rho_s\{q_s\} \quad (2)$$

where p_s is the sound pressure response of the air interior to the volume, and the subscript s refers to the effective equivalent-acoustic properties. Equations (1) and (2) can be extended to include modal and material damping and expressed as,

$$[S_{oo}]\{p_o\} = i\omega\{q_o\}, \quad [S_{ss}]\{p_s\} = i\omega\{q_s\} \quad (3)$$

Where

$$[S_{oo}] = \frac{1+i\eta_o}{\rho_o}[H_o] + \frac{i\omega}{\rho_o}[D_o] - \frac{\omega^2}{B_o}[M_o],$$

$$[S_{ss}] = \frac{1+i\eta_s}{\rho_s}[H_s] + \frac{i\omega}{\rho_s}[D_s] - \frac{\omega^2}{B_s}[M_s] \quad (4)$$

Here, η_o , η_s are the interior and effective damping loss factors, $B_o = \rho_o c_o^2$, $B_s = \rho_s c_s^2$ are the air and effective acoustic modulus. By requiring identical sound pressures and mass flow at the boundaries, the coupled acoustic and equivalent-acoustic equations-of-motion become [5-6],

$$\begin{bmatrix} S_{oo} & S_{ob} & 0 \\ S_{bo} & S_{bb}^o + S_{bb}^s & S_{bs} \\ 0 & S_{sb} & S_{ss} \end{bmatrix} \begin{Bmatrix} p_o \\ p_b \\ p_s \end{Bmatrix} = i\omega \begin{Bmatrix} q_o \\ 0 \\ 0 \end{Bmatrix} \quad (5)$$

Here the subscript b refers to the common interface boundary grids and q_o represents volume velocity loads, such as loudspeaker excitation. In the present study, loudspeaker excitation is prescribed as simple source strength representation as,

$$q_o = A_e a_o / i\omega \quad (6)$$

where a_o is the loudspeaker diaphragm acceleration and A_e is the effective surface area of the diaphragm. Equation (5) can then be solved for the sound pressure frequency response functions (FRFs) as p/a_o by prescribing $a_o=1$ and A_e at the grid point representing the center of the diaphragm.

MODAL ANALYSIS TEST

Modal analysis tests of a subcompact automobile with 4 side doors were conducted using loudspeaker excitation to identify the compartment cavity modes and sound pressure response to 500 Hz to assess the accuracy of the acoustic models. Figure 1 shows the 222 points in the passenger and trunk compartments at which the sound pressure frequency response functions (FRFs) were obtained using loudspeaker excitation. Figure 2a shows the special 6-microphone array that was used to make the sound pressure measurements, and Fig. 2b shows the loudspeaker device used in the tests.

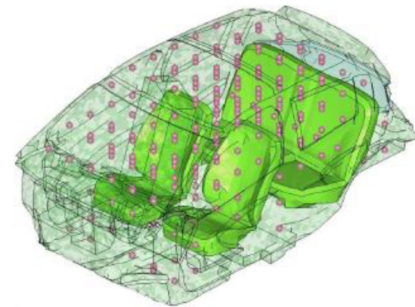


Figure 1. Microphone locations used in modal analysis tests.

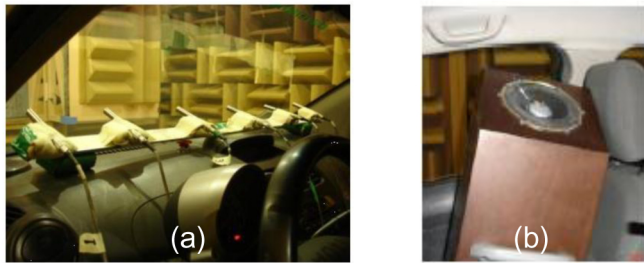


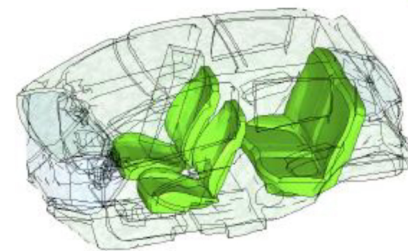
Figure 2. Microphone array and loudspeaker setup used in modal analysis tests.

The 6-microphone array in Fig. 2a had a 200 mm microphone spacing and was made to fit laterally inside the passenger and trunk compartments as in Fig. 2a and sequentially moved to the 37 locations to obtain the sound pressure FRFs at the 222 locations. In addition, microphone measurements were also made at the 4 front and rear occupant ear locations for a total number of 226 sound pressure FRF measurements. The loudspeaker device in Fig. 2b consisted of a speaker mounted in one wall of a 24×35×75 mm rigid-wall box that was internally damped and created a baffled condition of the speaker. An accelerometer was attached to the speaker diaphragm as a reference source to obtain sound pressure FRFs relative to the speaker diaphragm acceleration. Random loudspeaker excitation to 500 Hz was utilized, and the loudspeaker device was located in the various test locations either on the right front seat as in Fig. 2b or on the left rear seat. An effective speaker diaphragm diameter of 150 mm was identified from the tests and used to represent the effective surface area of loudspeaker diaphragm.

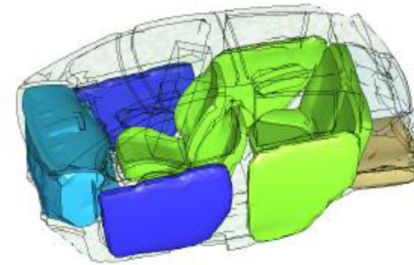
ACOUSTIC FINITE ELEMENT MODELS

Three different acoustic finite element models of the subcompact vehicle were developed for validation. The three different models are shown in Fig. 3 as a traditional acoustic model, an improved acoustic model, and an optimized acoustic model. The models were developed using tetra elements with an average grid spacing of approximately 50 mm that results in 25,000-30,000 grids and degrees-of-freedom. The traditional acoustic model in Fig. 3a represents the passenger and trunk compartment cavities with the front and rear seat volumes modeled using “heavy air” properties identified from test-measured lowest cavity mode frequency [5]. The improved acoustic model in Fig. 3b includes traditional acoustic models of the passenger and trunk compartments, as well as acoustic models of the front and rear doors, and “heavy air” models of the front and rear seats, parcel shelf, door, instrument panel and HVAC volumes, and trunk wheel well volume. The optimized acoustic model in Fig. 3c was developed from the improved acoustic model to more accurately represent the IP panel and parcel shelf panel by modifying their “heavy air” properties.

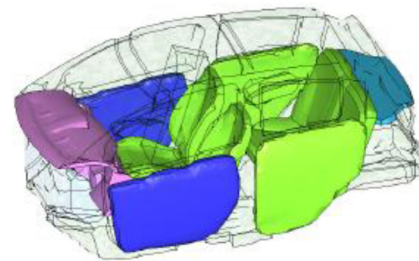
In the traditional acoustic model in Fig. 3a, the seats are modeled using the “heavy air” traditional value of 50 times the density of air [5], while the IP Volume, IP Panel, Parcel Shelf, and Wheel Well volumes are assumed to have standard air properties, and there are no door models. In the improved acoustic model in Fig. 3b, the door volumes are added and assumed to have standard air properties, while the IP volume and wheel well volume are assumed to have “heavy-air” properties of 10 times the air density. In the optimized acoustic model in Fig. 3c, the acoustic properties of the improved acoustic model were modified to obtain the best correlation as described below. In this case, the IP panel and parcel shelf are represented with a best-fit density value, and the IP volume and wheel well were assumed to have standard air properties. In addition, the loudspeaker device volume was represented as a rigid wall box to account for the large volume it occupies and account for its influence on the effective acoustic boundary length. Any increased acoustic boundary length will decrease natural frequency of the particular affected mode [7].



(a). Traditional Acoustic Model Configuration



(b). Improved Acoustic Model Configuration



(c). Optimized Acoustic Model Configuration

Figure 3. Passenger compartment acoustic finite element model configurations.