Truck Interior Noise Prediction by FEM and BEM

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ABSTRACT

A Computer Aided Engineering (CAE) method is developed for noise prediction in a truck cab interior. The finite element (FE) and the boundary element method (BEM) are used to characterize the acoustic field of a truck cab interior in terms of its natural frequencies and mode shapes. Structural vibration responses of the cab are computed for excitations at the cab mounts in a frequency range from 50 to 250 Hz. Then interior noise levels at the driver's right ear location are computed using the boundary element method for such excitations at the cab mounts.

INTRODUCTION

Globally, customer appreciation and demand for quieter products has driven noise control engineers to develop efficient and quieter products in a relatively short time. In the automotive industry noise has become an important attribute because of the competitive market and increasing customer awareness. To have a viable vehicle design in a short cycle time, computer-aided-engineering (CAE) methods are being used extensively. This enables the analyst to perform numerous design iterations and study their effects using a computer without fabricating hardware. In the past, CAE techniques such as the finite element method (FEM) have been used for durability (stress/fatigue) analysis and limited noise, vibration and harshness (NVH) analysis. However, in recent years NVH related analysis has become relatively easy and quicker with the emergence of new CAE methodologies like the boundary element method (BEM) for acoustical analysis.

Use of the BEM has made it possible to predict the interior noise levels in cavities [1,2]. However, the methodology has not been widely adopted like the FEM, which is used for structural analysis. In this paper a study is made to understand the physics and then use the BEM along with FEM to predict interior noise in the passenger compartment area due to structure-borne excitations.

Usually structure-borne noise in automobiles is predominant in frequencies below 250 Hz since diesel engines typically used in trucks have predominant structural excitation levels up to a frequency of 250 Hz [3]. In the BEM model the mesh should be fine enough to capture the modes of interest on the structural model, as well as the wavelength of the fluid in the acoustic medium. The size of the boundary elements is determined by the geometry of the structure being modeled and the structural or acoustical wavelength, whichever is smaller. For linear elements, at least four elements per wavelength are required [4]. For dynamic structural analysis, the present FE model of the truck cab with all the major components like the door, the steering column, the seat and the instrument panel has a total of about 150,000 grid points. The super-element method in MSC/NASTRAN is chosen to determine the dynamic response of the cab. MSC/NASTRAN version 68.0 is used for all computations. The FE model of the body-in-prime (BIP) cab with the major structural components is shown in Fig. 1. This size of a FE model, with six degrees of freedom at the majority of the grid points is quite a computational challenge, the Cray C-90 supercomputer is used for all the finite element and the boundary element computations.

CAVITY ACOUSTIC MODES AND NOISE PREDICTION

The acoustic cavity modes of the cab interior are computed both by the FE and the BE methods to compare and check the validity of both methods. In the FE method a normal mode analysis (Solution 103 in MSC/NASTRAN) of the cab air-mass model is performed. In the BE method a forced frequency response analysis is performed on the BE model of the cab.

Cavity Acoustic Modes by FEM

For the normal mode analysis of the cab interior air mass, the air mass is modeled as a fluid using bulk properties of speed of sound in air and the density of air. The air mass in the interior

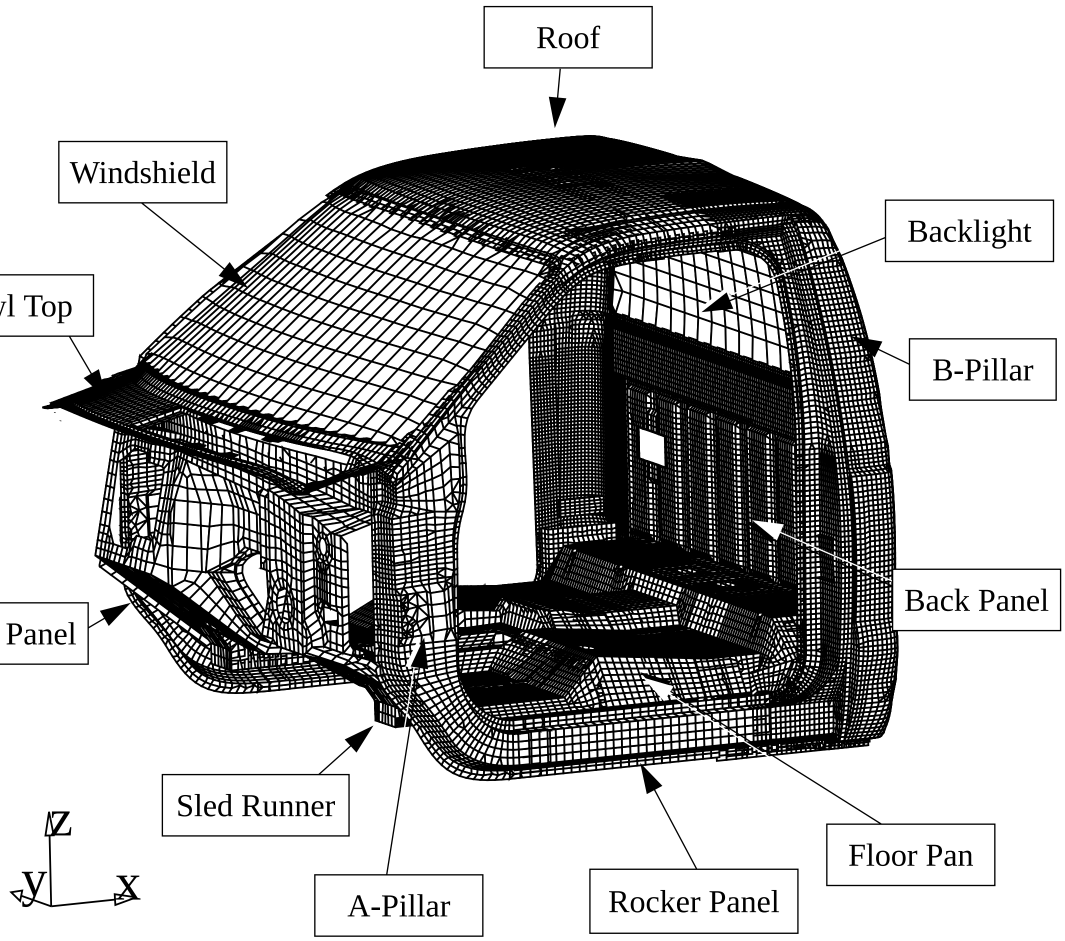

Figure 1. A MSC/NASTRAN FE model of Body-In-Prime cab.

of the cab is modeled with tetrahedral, pentahedral and hexahedral solid elements in MSC/NASTRAN [5]. The model has in total about 4000 solid elements. The seats present in the cab are also included in the model. Ten acoustic modes of the air-mass are obtained in frequencies up to 250 Hz.

Cavity Acoustic Modes by BEM

In order to compare the natural frequencies obtained by the FE method with those obtained using the BE method, the following is done. A forced frequency response analysis is performed using BEM. From the modeshape plots obtained by normal mode analysis using the FEM, a side panel in the right B-pillar of the cab is observed to be an antinode point for almost all the modes. The corresponding structural location in the BE model is excited with a harmonic velocity of 0.1 m/s. Then, due to this excitation at the antinode point, the interior SPL at the driver's right ear (DRE) location is computed using the commercial BEM code COMET/Acoustics. A frequency

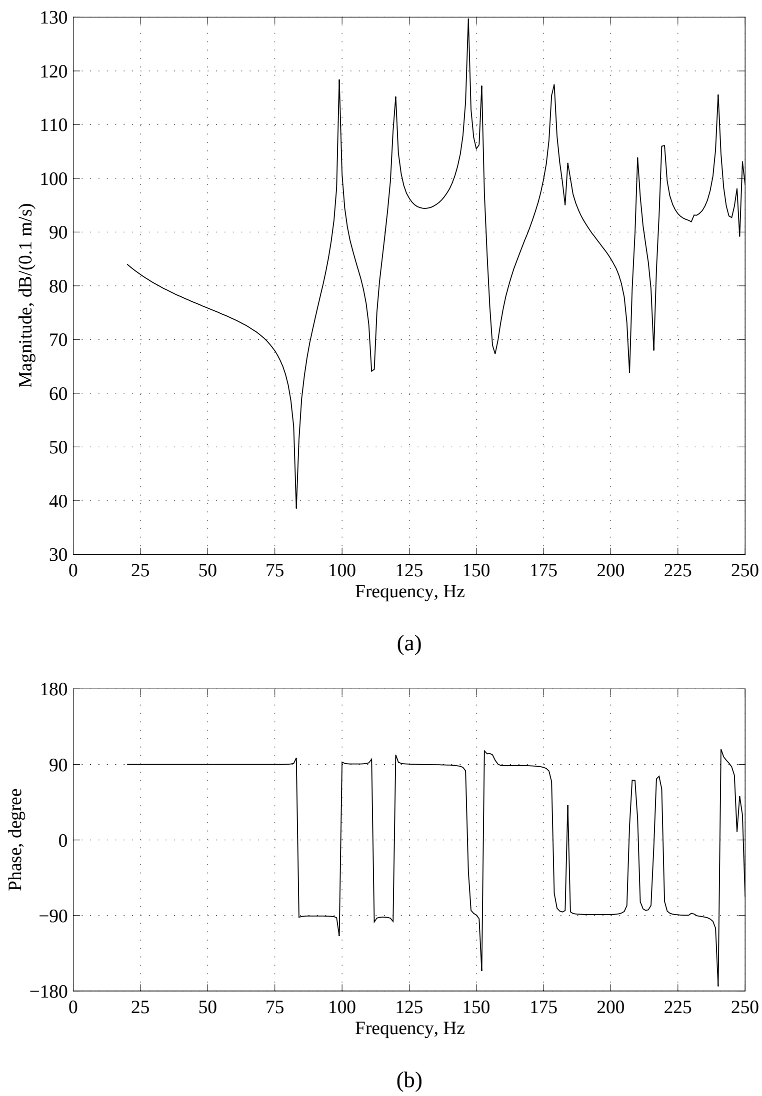

Figure 2. The acoustic frequency response of the cab interior at the driver's right ear, (a) Magnitude, (b) Phase.

sweep is done from 20.0 Hz to 250.0 Hz in steps of 1.0 Hz. The frequency response at the driver's right ear location is shown in Fig. 2. By picking the peaks in the frequency response curve in Fig. 2 the natural frequencies of the cab acoustic cavity are determined. A comparison between the FEM and the BEM results are shown in Table 1. The seats in the BE model are modeled to be rigid, though they can be modeled as porous with sound absorbing properties. It has been found that by modeling the seats as porous, the cab acoustic modes do not change significantly, hence are not reported here. The BE model has around 2000 elements in the boundary. Excellent agreement of the natural frequencies obtained by the FEM and the BE method is seen in Table 1. The mode orders are also indicated in Table 1, the first three axial modes are at 97.8 Hz, 120.6 Hz and 146.9 Hz.

TABLE 1 Comparison of natural frequencies calculated by FEM and BEM analysis.

Mode No.	Mode Order (x,y,z)	FEM (Hz)	BEM (Hz)
1	(0,1,0)	97.8	99.0
2	(1,0,0)	120.6	120.0
3	(0,0,1)	146.9	147.0
4	(1,1,0)	152.0	152.0
5	(0,1,1)	178.8	179.0
6	(0,2,0)	182.3	184.0
7	(1,0,1)	208.2	210.0
8	(2,0,0)	217.0	219.0
9	(2,1,0)	226.5	240.0
10	(0,2,1)	240.1	247.0

Sensitivity studies on the Acoustical BE model

A sensitivity study is performed to check the linearity of the BE model for the cab. The amplitude of the excitation at the same anti-node point on the right B-pillar is increased from 0.0001 m/s to 1 m/s by factors of 10. The SPL at the driver's right ear for the frequencies of 147 Hz, 179 Hz and 247 Hz are shown in Fig. 3. It is seen that with an increase in the amplitude of the velocity by a factor of 10, the SPL in the cab increases by 20 dB. This is in agreement with the theory of linear acoustics where pressure is proportional to velocity [6]. Hence, the current model is suitable for further acoustical analysis and SPL predictions in the cab interior.

STRUCTURE-BORNE NOISE ANALYSIS

In the past it has been determined experimentally by researchers that in automobiles the structure-borne noise path is predominant in frequencies below 250 Hz and the air-borne noise path is predominant at frequencies above 400 Hz [1]. In the present study the primary focus is on the structure-borne noise contribution in the cab interior.

For structure-borne analysis the frequency range of interest is determined by the major sources of input energy. Road excitations and powertrain excitations are some of the major sources of input energy. Road noise is typically predominant at frequencies less than 25 Hz, and it is usually random in nature and decreases rapidly with increasing frequency. Powertrain idle and its harmonics have a strong excitation of frequencies below 250 Hz. As determined in the previous section, there are ten acoustic modes of the cavity below 250 Hz. Thus, due to the various excitations present below 250 Hz, a very strong acoustic resonance can occur in the cab interior. The objective of this study is to determine the acoustical response (SPL) at these frequencies

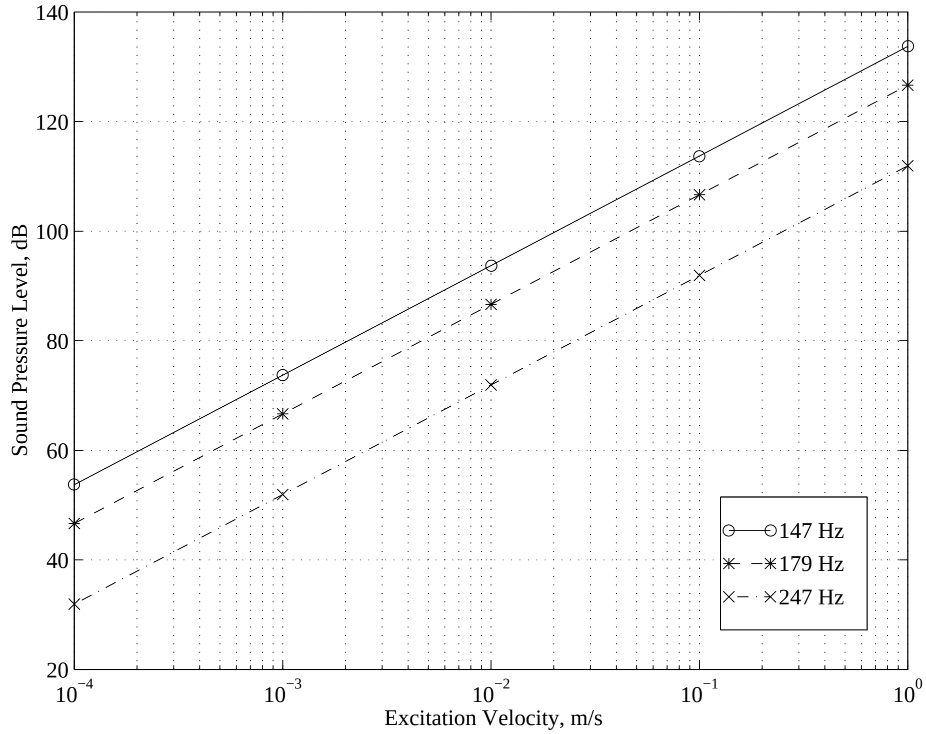

Figure 3. Sensitivity study of SPL variation with velocity change.

in the cab interior.

To determine the acoustical response, one must first determine the velocity response for the frequency range of 50-250 Hz on the structure of the cab by the finite element method and then compute the interior SPL by the boundary element method. The excitations for the velocity response are the accelerations at the cab side of the cab mounts. This velocity response is translated into a boundary condition in the BE model for the prediction of interior SPL [4].

Structural Frequency Response Analysis

The primary path of structural energy into the cab is through its four cab mounts. These mounts are used to provide vibration isolation for the cab by reducing the transmission of energy into the cab. In a vehicle like a pickup truck, the cab is on one side of the mount and the frame on the other. The vibration isolation characteristics of the mount can be determined by knowing the vibration levels on both sides of the mount as a function of frequency. In the present study, in order to provide structural excitation to the structural model, an acceleration of 0.01 g is input at the cab side of the cab mount in the vertical direction, at the four mount locations with a phase difference of 90^o between them. From past experiences with measuring the acceleration levels at the cab mount location on the cab side for similar medium duty trucks, a value of 0.01 g is a representative value. In the present simulation there are no inputs in the longitudinal and fore-aft directions since those levels are orders of magnitude less than the vertical level. However, if needed, the present FE model of the cab can easily be excited in the longitudinal and fore-aft direction as well. The acceleration levels at the cab mount of the cab FE structural model will be the forcing function. A frequency sweep is done with these constant acceleration levels of 0.01 g in the range from 50 Hz to 250 Hz. The velocity response is then obtained on the entire structure of the cab as a function of frequency.

Since the velocity on the entire structure is requested and the model has about 150,000 grid points, the velocity is requested in a binary results file. Since super-element method is used the following DMAP sequence is used:

```
COMPILE SUPER3 SOUIN = MSCSOU
ALTER 94
OUTPUT2 // -1 / 33 // OMAXR $ write file label
ALTER 238
OUTPUT2 // -9 / 33 // OMAXR $ write end-of-file marker
COMPILE SEDRCVR SOUIN = MSCSOU
ALTER 338
OUTPUT2 OUGV1 // 0 / 33 // 4098 $ write OUGV1 data block
END ALTER
```

The above DMAP alter writes a file label, the OUGV1 data block, and an end-of-file marker to I/O file unit 33. In addition to one of the above DMAP alters, velocity output must be requested using the VELOCITY case control statement [7]:

```
VELOCITY (SORT1, PLOT, REAL) = ...
```

Interior Acoustics Prediction due to Structural Excitations

The structural velocity obtained from the FE model frequency response analysis is interpolated into the boundary grid points of the BE model. This is done through a translator program provided by COMET/Acoustics which is able to read the binary results file containing velocities [4]. Then an interior acoustics computation is done by using the direct BE method to determine the SPL at the driver's right ear (DRE). Fig. 4 shows the SPL levels at DRE due to cab mount excitations. The cab in this case has no sound packaging treatment. It is seen from Fig. 4 that when the cab mount acceleration levels are constant over the entire frequency range of excitation of 50 Hz to 250 Hz, the peaks of the interior SPL spectrum occur at the frequencies of 80.0 Hz, 120.0 Hz, 145.0 Hz, 210.0 Hz and 240.0 Hz. From Fig. 2 it is seen that these peaks at the frequencies of 120.0 Hz, 145.0 Hz, 210.0 Hz and 240.0 Hz are the acoustic cavity modes of the cab interior.

The cab mounts play an important role in the structure-borne noise in a cab, since they constitute the major path through which energy comes into the cab interior. An increase in the acceleration levels by a factor of 10 times at the cab mounts will have a corresponding increase in the structural velocities by 10 times. From Fig. 3 it is seen that a 10 times increase in the structural velocity increases the interior SPL by 20 dB. So, good isolation at the mounts will be beneficial to reduce structure-borne noise.

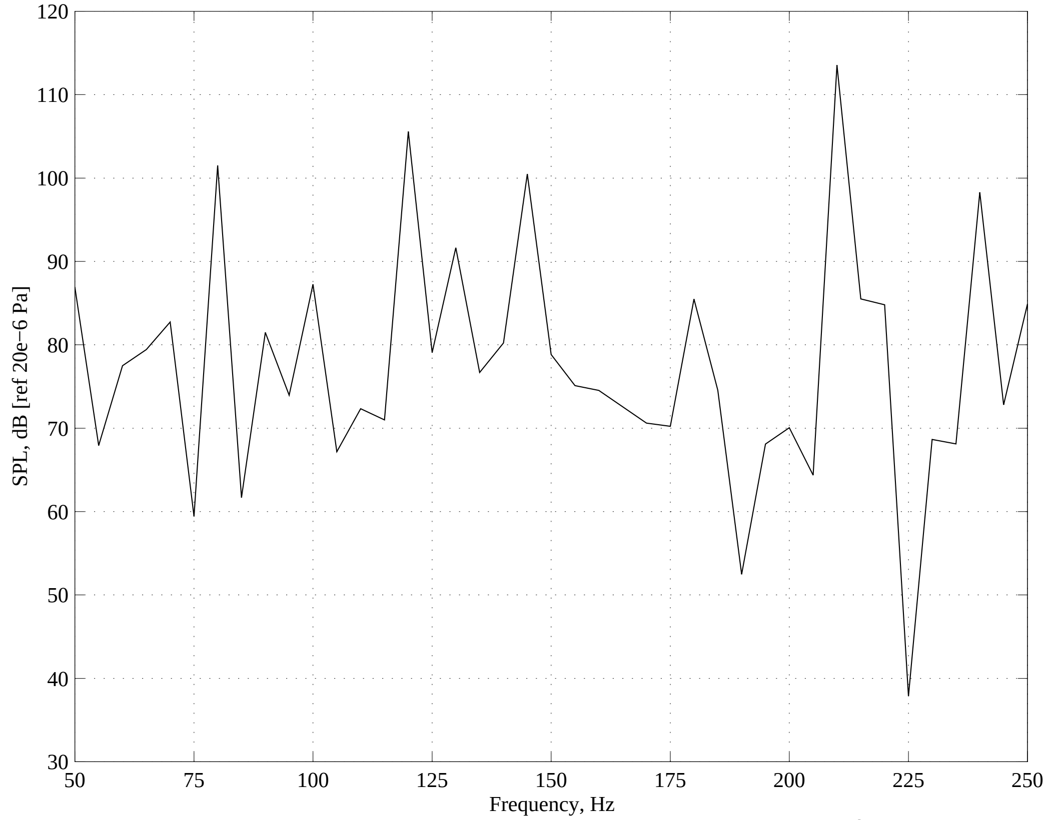

Figure 4. SPL at DRE due to structural excitation at cab mounts of untreated cab.

CONCLUSIONS

A successful CAE methodology has been developed to predict the interior noise in a truck cab due to structure-borne excitations by using FEM and BEM. Cab mounts play a very important role in noise control, since they are the major paths through which structure-borne energy enters the cab and in turn produces noise. A reduction of acceleration level across the cab mounts by ten times produces a 20 dB change in the interior SPL. Measured acceleration levels at the cab mounts can be used to excite the structural model and then predict interior noise by the above methodology.

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