Proceedings of 20th International Congress on Acoustics, ICA 2010

23-27 August 2010, Sydney, Australia

Vibro-Acoustic Analysis of a Vehicle Integrated with Design of Experiments Methodology using Three Performance Criteria

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PACS: 43.40.-R: Structural acoustics and vibration

ABSTRACT

The interior noise inside the passenger cabin of automobiles can be classified as structure-borne or airborne. In this study, we investigate the structure-borne noise, which is mainly caused by the vibrating panels enclosing the vehicle. Excitation coming from the engine causes the panels to vibrate at their resonance frequencies. These vibrating panels cause a change in the sound pressure level within the passenger cabin, and consequently generating an undesirable booming noise. It is critical to understand the dynamics of the vehicle, and more importantly, how it interacts with the air inside the cabin. Two methodologies were used by coupling them to predict the sound pressure level inside the passenger cabin of a commercial vehicle. The Finite Element Method (FEM) was used for the structural analysis of the vehicle, and the Boundary Element Method (BEM) was integrated with the results obtained from FEM for the acoustic analysis of the cabin. The adopted FEM-BEM approach can be utilized to predict the sound pressure level inside the passenger cabin, and also to determine the contribution of each radiating panel to the interior noise level. The design parameters of the most influential radiating panels (i.e., thickness) can then be investigated to reduce the interior noise based on the three performance metrics. The performance metrics selected for this study are “Percentage over 80dBA”, “Max Amplitude”, and “Idealized Performance Error”.

Design of experiments (DOE) technique was employed to understand the relationship between the design parameters and the performance metrics. The components that have the highest contribution to the sound pressure levels inside the cabin are identified. For each run, the vibro-acoustic analysis of the system is performed, the sound pressure levels are calculated as a function of engine speed and then the performance metrics are calculated. The highest contributors (design parameters) to each performance metric are identified and regression models are built. These regression models can be used in future studies to employ optimization runs to find the optimum configuration of the panel thicknesses to improve the sound pressure level inside the cabin.

INTRODUCTION

The reduction of vehicle interior noise has become one of the most important issues related to driving conveniences. Vehicle interior noise is usually quantified by sound pressure level (a.k.a., sound level), which is a logarithmic measure of the sound pressure of a sound relative to a reference value. Sound pressure level is measured in decibels (dB). The commonly used reference sound pressure in air is p_{ref} = 20 µPa, which is usually considered to be the threshold of human hearing. However, for automotive applications, frequency weighted, dB(A) measure is used to approximate the human ear's response to sound using the A-weighted scale [1].

Sound pressure level varies as a function of the speed of the vehicle’s engine. There are many sources that may cause the sound pressure level to increase inside the passenger cabin. These sources can be classified as structure-borne and airborne [2]. In this study, we investigate the structure-borne noise, which is mainly caused by the vibrating panels enclosing the vehicle. The excitation coming from the engine may cause these panels to vibrate, and consequently, cause an increase in the sound pressure level. The increase in the sound pressure level generally corresponds to an undesirable booming noise, which is usually felt in the low frequency range of 50-200 Hz inside the passenger cabin. In order to reduce the interior noise, it is critical to understand the dynamics of the vehicle, and more importantly, how it interacts with the air inside the cabin. The objective of the present study is to demonstrate a methodology that can be used to identify the key contributors to structure-borne vibration induced interior noise in a commercial vehicle. A generalized methodology was developed for defining the relationship among the panel design parameters and vibro-acoustic performance of the vehicle using three performance criteria.

For the vibro-acoustic study, mainly two simulation methods, which are Finite Elements Method (FEM) and Boundary Elements Method (BEM) are used. Many researchers have studied and reported on the accuracy and limitations of both of these methods. For example, structural vibro-acoustic analysis of vehicles, in which both FEM and BEM have been used, was reported in the studies [3, 4]. In these papers, the vibro-acoustic response of vehicle using either a strongly coupled structure-fluid interaction method or an uncoupled structure-fluid interaction have been investigated and discussed in great detail. Suzuki et al. [3] performed BEM to overcome the noise problems inside a vehicle cabin. They studied the effect of absorbent materials adhered to vibrating surfaces to prevent air leakage from the cabin walls. In Pal and Hagiwara's study [4], FEM was performed for the coupled structure-fluid problems. They analyzed the correlation between vibration of cabin walls as well as the sound pressure level at the position of the passenger ear. Liu et al. [5] created a vibro-acoustic model to predict the noise inside the tracked vehicles. They determined the interaction forces in the vehicle by using the ADAMs software. In their study, both FEM and BEM models were used for the vibro-acoustic analysis.

In this study, we adopt a combined use of FEM and BEM methodologies in order to predict the sound pressure level inside the passenger cabin of a commercial vehicle. We use FEM for the structural analysis and BEM for the acoustic analysis. The adopted FEM-BEM approach takes advantage of the Acoustic Transfer Vectors (ATV) to calculate the sound pressure levels at the predefined locations as a function of engine speed. ATVs are transfer functions that link the structural vibrations of the radiating surfaces and the sound pressure levels at the desired output field points. The contribution of each radiating panel to the interior noise is calculated using Panel Acoustic Contribution Analysis (PACA) [6]. PACA takes advantage of the ATVs and enables the user to identify the most critical radiating panels. This information can then be utilized to reduce the cabin noise by focusing on these critical panels.

Although the results of the PACA contains a lot of useful information, it does not provide any guidance to the designer regarding how these panels should be modified so that the sound pressure levels can be reduced. The PACA results are only based on the current design and they do not provide any information regarding how the performance changes if any or all of these panel design variables are changed within a certain range. The vehicle contains many structural panels and consequently, there are many design variables that should be looked at when a redesign effort is considered. Especially, when the coupling between the structure and cavity and the interactions between the panels are considered, the reduction of sound pressure level forms a highly non-linear optimization problem, and hence is still considered to be a complicated task even for a simple vibro-acoustic problem.

In this study, we employ techniques from the fields of Design of Experiments (DOE) to understand the relationship between the studied design parameters and sound pressure level. The PACA results are chosen as the basis for the DOE study. The thicknesses of the same panels used in the PACA are used in a DOE study to identify the important ones. Following the DOE analysis, response surfaces are built to be used for future optimization studies The following section gives the details of the vibro-acoustic model that was used in this study.

VIBRO-ACOUSTIC MODEL AND SOUND PRESSURE LEVEL ESTIMATION

Structural Finite Element Model

Figure 1(a) shows the structural model used for the vibro-acoustic analysis of the vehicle. In this model, SHELL181 and MASS21 were used as element types. After determining the element types and identifying material properties, modal analysis was performed in the frequency range of 0-160 Hz. The analysis model only includes the hull system enclosing the cavities of the vehicle and the engine mounts are simply connected to the main body by welding. The main structure of the vehicle was modeled by using shell elements with different thicknesses. MASS 21 element type was used to represent the welding of the engine mounts to the main structure.

The forced response analysis for obtaining the velocity boundary conditions was performed in the same frequency range (0-160 Hz) with an interval of 1 Hz. The disturbances, measured experimentally at the engine mounts, were used as the excitation source for the structure. A global 1% structural damping was used. From the analysis, the velocities were obtained at every finite element node and then used as velocity boundary conditions in the acoustic analysis.

The Cavity (Acoustic) Boundary Element Model

In order to analyze the interior noise, the interior (volume) should be meshed such that the vibrations coming from the structure can be transferred across the outer envelope of the cavity mesh. Cavity mesh (volume mesh) was created directly from the structural finite element model in LMS Virtual Lab, as shown in Figure 1(b). The mid-panel (bulkhead) separates the cavity into two divisions: the passenger cabin and the baggage compartment. The SPL is measured in the passenger cabin at the driver's ear location.

The cavity mesh is much coarser when compared with the structural mesh. A process called *mesh mapping* is performed to link the structural model nodes with the acoustical model nodes such that the velocities calculated at the structural nodes can be used for the acoustical analysis.

Engine Disturbance Model

Engine forces are considered to be transferred to the vehicle panels at the mount application points (see Figure 1(a)). There are three engine mount locations which are called left, right and transmission engine mounts. Each of them has three directions, (x, y, z), where the amplitudes may vary. The amplitude variation of the force data was obtained experimentally at the mount locations while the engine is running at different speeds (i.e., RPMs). This study has focused on the causes of the mid-speed boom that occurs around 2800-3700 RPM with increments of 20 RPM.

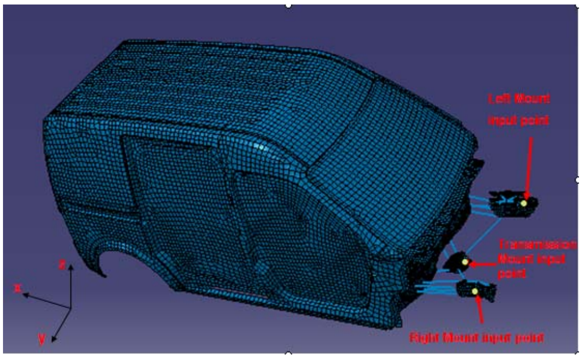

Figure 1(a): Structural model showing the engine mounts as the disturbance input locations

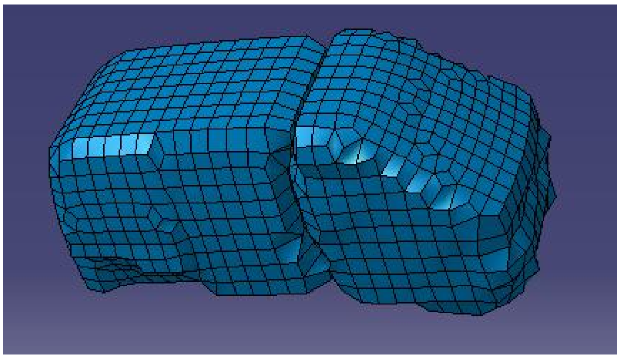

Figure 1(b): Acoustical (cavity) model

Predicting the Sound Pressure Level using the ATV Response Analysis

The LMS Virtual Lab uses the ATV Response Analysis tool to calculate the sound pressure level as a function of engine speed. ATV method is based on Acoustic Transfer Vectors (ATVs) that link the input of the structural velocity of the radiating surface and the sound pressure level at the desired output field point [6, 7]. The pressure is calculated at the user defined locations using the following [6]

$$p(\omega) = \{ATV(\omega)\}^T \{v_n(\omega)\} \tag{1}$$

where $p(\omega)$ is the sound pressure level at the position of driver's ear, $v(\omega)$ represents the nodal velocities of the panels surrounding the cavity, and $ATV(\omega)$ shows the acoustic transfer vector between the input (the nodal velocities) and the output (the position of driver's ear).

The ATVs from the radiating surface to specified field points are evaluated in the first step across the frequency range of interest at fixed frequency intervals. In the second step, the acoustic response in the field points is calculated for all loading conditions by combining the ATV with the normal structural velocity boundary condition vector at the frequency range defined by the user.

This technique has the important advantage that the frequency dependent ATVs can also be used for panel contribution analysis, only taking into account the normal velocity boundary conditions on part of the radiating surface,

$$p_e(\omega) = \{ATV(\omega)^e\}^T \{v_n(\omega)^e\} \tag{2}$$

where the superscript *e* denotes the contribution of an element (i.e.,panel), *e*. This way, the contribution of groups of elements, corresponding to distinct panels of the structure, can be calculated. This process, as mentioned before, is called Panel Contribution Analysis (PACA), and can be used to identify the panels that contribute to the sound pressure level the most. ATVs in Equations (1) and (2) are dependent only on the geometry characteristics of the acoustic domain, the frequency, and location of the output field points. They are independent of the loading conditions, implying that they are especially well-suited for design studies that require experimentation with multiple configurations. In contrast, classical FEM and BEM approaches use the structural vibrations directly to define the boundary conditions for the acoustic radiation problem. The drawback of the traditional approach is that the acoustic response must be calculated by solving the system equations for each loading condition. Since solving the equations for each load at each frequency is quite time consuming, predicting the sound pressure levels using ATVs becomes more efficient in acoustic problems.

The ATV approach is particularly well-suited for the design studies presented in the following section. Since the acoustical model does not change, ATVs are calculated only once and then multiplied by the normal velocities calculated for each new design configuration to predict the sound pressure level. The structure changes slightly due to the modifications in the panel thicknesses, and hence, the normal velocities must be recalculated for each new design configuration. This approach reduces the computation time significantly since many different design configurations need to be investigated to perform the factor DOE analysis presented in the next section.

Figure 2 shows the steps followed in PACA. The process started with the calculation of the structural modes and frequencies. Then, Modal Based Force Response Analysis was performed using the engine disturbances to calculate the nodal velocities on the structure. Cavity mesh was created using the LMS Virtual Lab. Panels were created on the cavity mesh (see Figure 3 for the names of the panels). Table 1 lists the names of the panels and corresponding thicknesses used for the baseline configuration. Data transfer analysis was then performed to transfer the velocities calculated in the Modal Based Force Response Analysis. Finally, ATV response analysis was performed to predict the contribution of each panel at the field point mesh symbolizing the head of the driver (see Figure 4).

Figure 4 shows the results of PACA. The x-axis shows the engine speed in RPMs. The y-axis is divided into segments to show the contribution of each panel. Red colored parts represent high contribution, in the order of more than 90 dB(A)s, and yellow colored parts represent low contribution, less than 60 dB(A)s, to the overall sound pressure level. According to PACA results, all panels except back floor and doors have contribution at different engine speeds, changing from 900 to 4500 RPM.

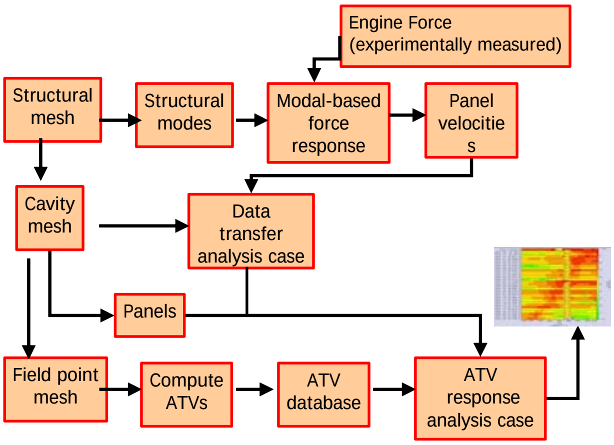

Figure 2: PACA Process Flow Diagram

We complement the results of PACA by employing a DOE study to determine the impact of various panel thicknesses on the sound pressure level. Observing from PACA results that front floor, front door, bulkhead, window, front panel and roof panels were all significant contributors to the sound pressure level, we decided to choose the thicknesses of these panels as our design factors. We narrowed our study to the engine speed range of 2800 to 3700 RPM, since the highest values of sound pressure level occur in this range of RPMs. In this study, DOE and RSM techniques are employed together to find the relationship between the panel thicknesses and the three performance metrics. The DOE and RSM methods are described in the following sections.

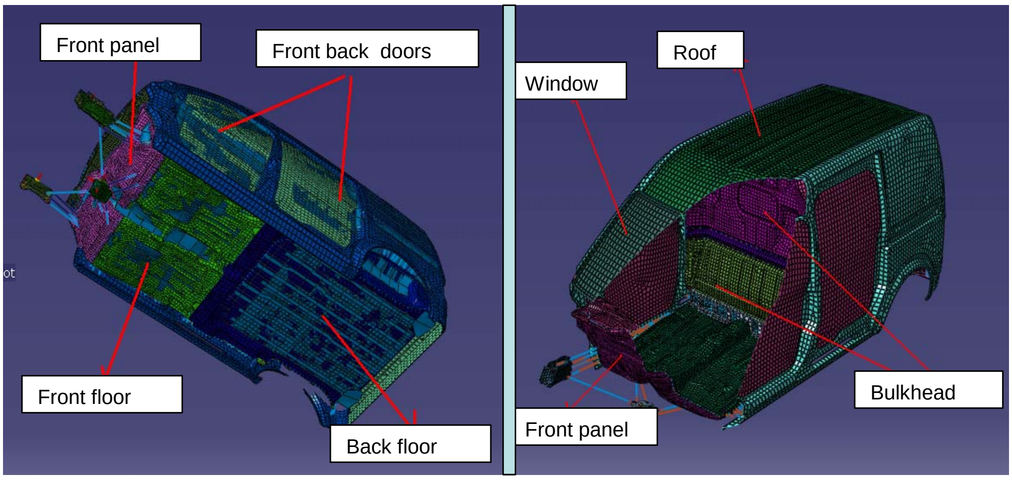

Figure 3: Panels of the vehicle model

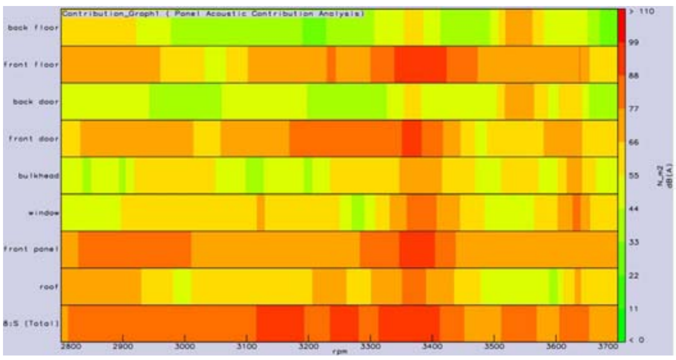

Figure 4: PACA results for the baseline configuration

DESIGN OF EXPERIMENTS (DOE) AND RESPONSE SURFACE METHOD (RSM)

Background on DOE & RSM

DOE is a statistical methodology that is primarily concerned with the development of an effective experimentation plan. The fundamental ideas of DOE were introduced by R.A. Fisher in the 1920s to address agricultural experimentation problems [8]. DOE methods have since evolved into a comprehensive body of theory and application tools, and are being widely used in natural sciences, industrial, and engineering applications due to their effectiveness [9]. Fundamentally, DOE aims to determine the appropriate set of experiments that are sufficient to attain the desired level of information by varying the main factors of interest over an operating range in a structured manner using statistical tools.

The fundamental goal of RSM is to obtain an approximate functional relationship between the input variable(s) and the output objective function(s) to construct a model over the entire domain of interest. A common method to obtain this model is to employ regression, which relates controllable variables to responses. The regression equations provide information about the properties of the system from which the data (generally obtained through a set of designed experiments) is taken, and can be used to improve/optimize processes through appropriate deterministic optimization procedures. Myers et al., [10] review the progress of RSM in the general areas of experimental design and analysis, and indicate how advances in other fields of applied sciences have affected its role. For further references on RSM methodology and applications, the reader is referred to [11]. The increased use and availability of computational models to assess performance of different product designs has allowed the use of RSM for computer experiments (rather than physical experiments for which RSM has been initially developed). The use of RSM presents two fundamental advantages over other optimization schemes (e.g., random search schemes such as genetic algorithms). First, RSM yields a functional relationship between the factors (i.e., various components and parameters of the product design) and response variables (i.e., some performance metric for the design) of interest over the search space. This provides a better understanding of the system behavior, and complements the product designer's expertise with the system. The step-by-step nature of the technique also allows interaction of the designer with the optimization scheme. Secondly, such functional relationships (obtained through regression analysis, in general) allows for much faster search of the design space compared to random search schemes. This is particularly important for computer models that require significant amount of computational time to run.

In the domain of acoustic analysis, the following studies are noteworthy, and are relevant to our work. Liang et al. [12] utilized DOE (in particular a three-level fractional factorial design) to optimize three response variables with respect to three factors (two thicknesses and one material). The authors obtained (through regression analysis) a response surface to analyze the effect of design parameters on the sound radiation from a vibrating panel with point force excitation. The authors approximated the structure-born noise problem by a series of second-degree polynomials, and considered three objectives of mean quadratic velocity, sound radiation power and system loss factor. The paper presents a simple case study to demonstrate the effectiveness of the methodology. The study presented in [13] is similar; the authors used Central Composite Design (instead of a three-level fractional factorial design) in this paper. In an earlier study by Kamci et al. [14], a screening study was performed for the thicknesses of seven panels surrounding the cabin to identify the panels that have the highest contribution to sound pressure level. Fractional factorial design was selected for the DOE analysis and the most significant panels were determined. Marburg and his co-workers [15], [16], and [17] published series of papers on the investigation of vibro-acoustic interior noise of the vehicles and the optimization of several structure-acoustic systems.

Two Level Full Factorial Experimentation

In this study, we employed techniques from the fields of DOE to understand the relationship between the studied design parameters and sound pressure level. The PACA results were chosen as the basis for the DOE study. A Two Level Full Factorial Experimentation was employed to find the significant design parameters. Two level full factorial experimentation is a common experimental design with all input factors set at two levels each. These levels are called `high' and `low' or `+1' and `-1', respectively. A design with all possible high/low combinations of all the input factors is called a two-level full factorial design. The number of the experiments of a two level-full factorial design is 2^k where k is the number of factors [9].

In order to determine the panel thicknesses that are the highest contributors to the structure-borne interior noise, a study was performed using DOE. For this purpose, the first seven factors listed in Table 1 (Factors A to G, excluding Factor H) were considered. Back door thickness (i.e., Factor H) was not taken into account due to the fact that sound pressure level is measured in the passenger cabin only, and back doors are known to contribute minimally to the total sound pressure level (see Figure 4). For each factor, two-levels of settings were used (a high level and a low level). For example, front panel thickness (Factor F) was set at two levels of 0.72 mm (low level) and 0.88 mm (high level). The high and low levels for each factor were determined by increasing and decreasing the baseline values of the factors by 10%, respectively.

A full-factorial experiment was employed which required 2^7 (=128) runs. An additional run was added to test the factors at their baseline values (e.g., Factor F was set to 0.8 mm for this run), which was also used to check for curvature in the RSM analysis.

Table 1: Panel thicknesses considered in DOE study (low & high values are also included)

Factors	Panel Name	Baseline Thickness (mm)	Low (mm)	High (mm)
A	Back floor	0.9	0.81	0.99
B	Front floor	0.8	0.72	0.88
C	Front & Back doors	1.0	0.90	1.10
D	Bulkhead	0.8	0.72	0.88
E	Window	5.0	4.5	5.5
F	Front panel	0.8	0.72	0.88
G	Roof	0.7	0.63	0.77

Three Response Variables are selected in DOE study as our performance metrics (objective functions). The vibro-acoustic analysis were employed 129 times (number of experiments required for the full factorial design) for each metric and the results were used for the regression analysis to obtain the response surface. The explanation of each metric is as follows:

"Percentage over 80dBA" Metric

The response variable was selected as the fraction of RPMs that are greater 80 dB(A) within the engine speed range of 2800 to 3700 RPMs (this range corresponds to 46 distinct engine speeds due to 20 RPM increment used in **evaluating** sound pressure level for a given engine speed). This metric is preferred when more than one SPL values are over the threshold value. In this study, 80 dB(A) is selected as the threshold but this value can be altered based on the desired noise levels defined by the automobile manufacturers. After the DOE is completed, there are 129 experiments including the centroid point (the configuration with the baseline values of the design variables). The percentage value obtained for the baseline configuration is 67.39%. The percentage values of the full-factorial experiment vary between 17.39%-69.56%. A lower percentage value shows that with the right design parameter configuration, the performance can be improved.

"Max Amplitude" Metric

In some cases, the highest amplitude of SPL can be the focus of the redesign efforts. In those cases, there is only one dominant peak that stands out in the SPL performance and the rest of the SPL values are in the acceptable range. In this metric, the response variable was selected as the max SPL amplitude (dBA) within the engine speed range of 2800 to 3700 RPMs. The max amplitude obtained for the baseline configuration is 93.97 dBA. The max amplitude values of the full-factorial experiment vary between 89.73-118.42 dBA. A lower dBA value shows that with the right design parameter configuration, the performance can be improved.

"Idealized Performance Error" Metric

Usually, automobile manufactures have a desired performance curve for the SPL as the engine speed increases. This curve is usually an idealistic curve and usually hard to achieve. However, the NVH designers would like to approach this curve as much as possible. A linear relationship between the SPL and the engine speed was defined as the desired performance curve. In this metric, the response variable was selected as the least mean square error (mse) between the idealized curve and the SPL predictions for a given design. The mse for the baseline configuration is 173.12 dBA. The mse values of the full-factorial experiment vary between 62.69-419.45 dBA. A lower dBA value shows that with the right design parameter configuration, the performance can be improved.

Response Surface Modeling

The response surface model is usually assumed as a second-order polynomial, which can be written for n_v design variables below [9]

$$y^{(p)} = c_o + \sum_i c_i x_i + \sum_{1 \le i \le j \le n_v} c_{ij} x_i x_j \qquad p = 1, \cdots, n_s \tag{3}$$

Where $y^{(p)}$ is the dependent variable of the response surface model, c_o, c_i, c_{ij} are the regression coefficients, x_i is the design variable, and n_s is the number of observations. The basis functions for the regression model of Equation 3 lead to an over-determined matrix problem and the regression coefficients are obtained to minimize the total statistical error.

The results presented with the bar charts in Figures 5, 6 and 7 show the contribution of each design variable to the performance metric under consideration. When the results of the three metrics are compared, it is observed that among the seven panels considered, front panel is the most significant for all of the metrics considered. The front doors are also significant for the "Max Peak" and "Idealized Performance Error" metrics. The two way interactions are also taken into account in the regression model. Interesting observations can be made from the contribution plot. There are cases when the significance of a single design parameter in the overall response may increase when the two way interactions are considered (eg. "Max Amplitude metric": roof only contribution vs roof and front panel interaction contribution).

When the regression parameters are considered, the adjusted R^2 of the "Percentage over 80dBA" metric is 0.977. That means 97.7 % of the variability in the response variable can be explained with the regression model. The adjusted R^2 of the "Max Peak" and "Idealized Performance Error" metrics are 0.713 and 0.983, respectively. Based on these adjusted R^2 values, the regression models are acceptable and they can be utilized for future optimization studies.

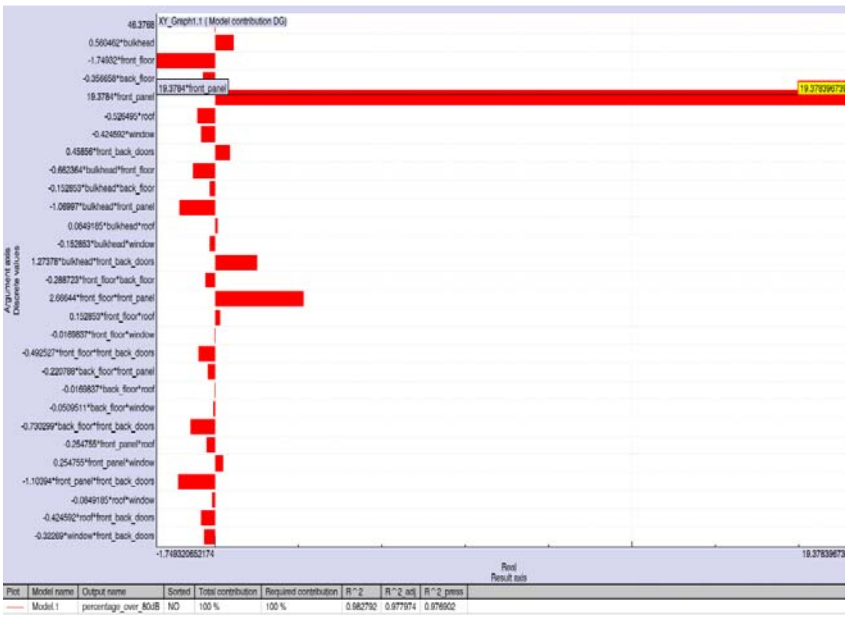

Figure 5. Contribution chart of the design variables for the "*Percentage over 80 dBA*" metric

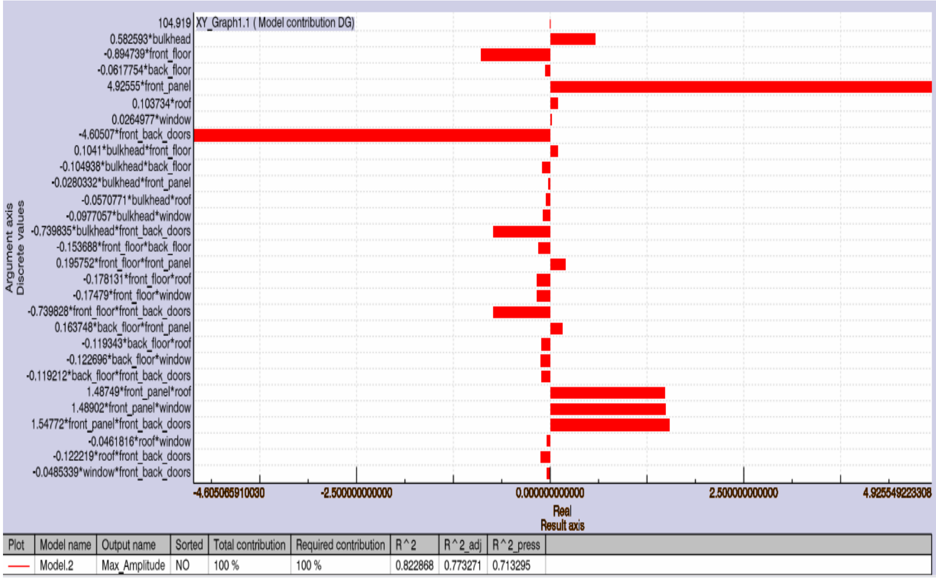

Figure 6. Contribution chart of the design variables for the "*Max Amplitude*" metric

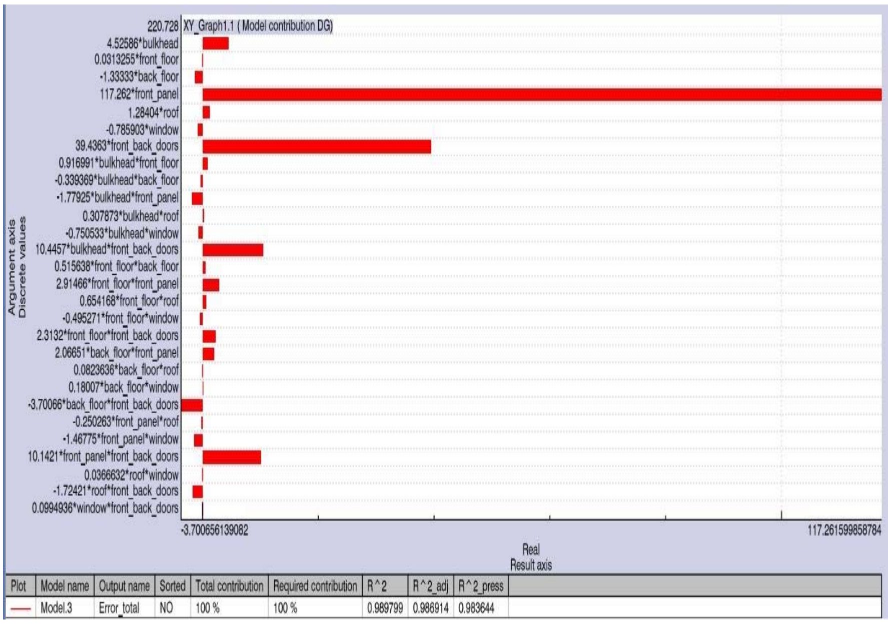

Figure 7. Contribution chart of the design variables for the "*Idealized Performance Error*" metric.

CONCLUSIONS

In this study, we built a vibro-acoustic model of a commercial vehicle using a FEM-BEM approach. The model was utilized to predict the sound pressure level inside the passenger cabin, and also to determine the contribution of each radiating panel to the interior noise level. A DOE study was performed to identify the most influential radiating panels (i.e., thickness) based on three performance metrics. Based on the DOE study, RSM of the three performance metrics were built and they will be utilized to perform the optimization runs to improve the interior sound pressure levels by finding the optimum configurations for the panel thicknesses. Our results show that the methodology developed in this study can be effectively used for improving the design of the panels to reduce interior noise when the vibro-acoustic response is chosen as the performance criteria.

ACKNOWLEDGEMENTS

This work is supported by Scientific and Technological Research Council of Turkey (TUBITAK). Authors also acknowledge in-kind support from Ford Otosan, Turkey.

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