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A structural modification methodology adapted to a vibro-acoustic model to improve the interior noise

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ABSTRACT

The structural-borne noise inside the passenger cabin of automobiles, which is mainly caused by the vibrating panels enclosing the vehicle, dominates the low frequency noise inside the cabin. Excitations coming from the engine cause the panels to vibrate mainly at their resonance frequencies. These vibrating panels cause a change in the sound pressure level (SPL) within the cabin, and consequently, an undesirable booming noise. The SPL can be predicted using a vibro-acoustic model which includes the Finite Element Model (FEM) for the structural analysis of the passenger cabin, and the Boundary Element Model (BEM) for the acoustic analysis of the cavity inside the cabin. The structural design of the panels can be modified (eg: adding a stiffener to the panel, adding or removing mass, etc) to improve the SPL. The modal analysis of the structural model has to be repeated after every modification before the reanalysis of the vibro-acoustic model. However, such changes require considerable computational time especially when the structural models are very complex.

In anticipation of these needs, we adapted a methodology that utilizes the frequency response functions (FRFs) of the original model for the reanalysis of the structure that is subjected to structural modification. We used a rectangular box with a very flexible mid-panel to demonstrate the developed methodology. In the presented method, the receptances of the original structure are used together with the dynamic stiffness of the components that are added or removed from the original structure to perform the structural modification and to calculate the receptance matrices of the modified system. Then, the receptance matrices are used to calculate the mobility matrices of the modified structure. The mobility matrices are then used to calculate the velocities at the structural nodes under the effect of the excitation forces. The calculated velocities are used as boundary conditions for the acoustic analysis to calculate the nodal pressures at the predefined locations.

Keywords: vibro-acoustic analysis, structural modification, vehicle interior noise

1. INTRODUCTION

The reduction of vehicle interior noise has become one of the most important issues related to driving conveniences. There are many sources that may cause high peaks at the sound pressure levels inside the passenger cabin. These sources can be classified as structure-borne and airborne [1]. The structure-borne noise 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 (SPL). The increase in the sound pressure level usually generates an undesirable booming noise which is generally seen 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 couples with the air inside the cabin.

In this study, we adopted a combined usage of FEM/BEM methodologies in order to predict the sound pressure level inside a rectangular box that is separated by a flexible mid-panel. Note that, a similar approach can be also used to predict the SPL inside the passenger cabin in the transport vehicles where the passenger and luggage compartments are also separated with a panel. However, the rectangular box used in this study is constructed only for the demonstration of the developed methodology. The dimensions and the material properties do not represent a vehicle model. Finite Element Method (FEM) is used for the structural analysis whereas Boundary Element Method (BEM) is used for the acoustic analysis. The adopted FEM-BEM approach takes advantage of the Acoustic Transfer Vectors (ATV's) to calculate the sound pressure levels at the predefined locations. 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 radiating panels can be redesigned such that the sound pressure levels can be reduced. The most common methods to reduce the sound radiation are to modify mass and stiffness properties of the panels and apply damping treatments. The modal analysis of the structural model has to be repeated after every modification before the reanalysis of the vibro-acoustic model. However, such changes require considerable computational time especially when the structural models have high modal densities. In this paper, we are going to adapt a methodology that utilizes the frequency response functions (FRFs) of the original model for the reanalysis of the structure that is subjected to structural modification. The methodology takes advantage of the formulation initially proposed by Özgüven [2] for predicting the dynamic response of a structure with structural modifications from the response of the original structure itself and dynamic stiffness matrix of the modifying structure. In the presented method, the receptances of the original structure are used with the dynamic stiffness of the components that are added or removed from the original structure to perform the structural modification. These matrices are used to calculate the mobility matrices of the modified structure. Although a matrix inversion is involved in the presented method, the order of the matrix to be inverted is much smaller than the order of the main structure. The mobility matrices are then used to calculate the velocities at the structural nodes under the effect of the excitation forces. The calculated velocities are used as boundary conditions for the acoustic analysis to calculate the nodal pressures at the predefined locations. The original and modified structure SPL results are then compared to see the effect of the structural modification on the interior noise levels.

Structural modification is about determining the effects of mass, stiffness and/or damping changes on the dynamic response of the mechanical system. According to the Snyder's [3] and Avitabile's surveys [4], structural modification falls into two categories: the direct structural modification problem and the inverse structural modification problem. When dealing with the direct structural modification problems, the dynamic response of the system is calculated after the modification takes place. On the other hand, in the case of the inverse structural modification problems; the modifications needed to achieve the required dynamic characteristics (eg: desired values for natural frequencies and mode shapes) are determined. This paper focuses on the direct structural modification problem and predicts the performance of the vibro-acoustic system before and after the modification takes place. The direct structural modification problem can be classified into four sub-categories: (1) lumped modification without additional degrees of freedom (DOF), (2) lumped modification with additional DOFs, (3) distributed modification without additional DOFs, and (4) distributed modification with additional DOFs. The problem of lumped modification, with or without additional DOFs, related to both the prediction and optimization aspects has been solved at the end of 1980s. However for the distributed modification case, with or without additional DOFs, most of the advancements are more recent. D'Ambrogio and Sestieri [5-7] proposed a new modeling approach to combine the measured frequency response functions (FRFs) of the original structure and the distributed modifications to predict the dynamic response of the modified structure. They have applied reduction and expansion techniques to overcome the difficulties raised by the mismatch of the DOFs of the original structure and the modifying structure due to the fact that the dynamics of the original structure is represented by only translational DOFs whereas structural information of modifying structure contains both rotational and translational DOFs. In a later work, by Hang, et al. [8], they focused on the distributed structural dynamics modification with additional DOF by using the original relationship developed by Özgüven [2] and modeling method of the distributed modification developed by D'Ambrogio and Sestieri [6]. In this paper, we implement the approach developed by Canbaloglu and Özgüven [9] for predicting the dynamic response of the structure with distributed modifications from the response of the original structure itself and dynamic stiffness matrix of the modifying structure. In this approach, the frequency response function of the original structure can be obtained either experimentally from modal testing or theoretically by using finite element method (FEM), and the modifying structure is modeled in such a way that consistent DOF are present at the connection nodes. The method proposed is validated by different case studies in [9].

This paper summarizes our initial efforts in implementing the structural modification methodology to a vibro-acoustic problem. Utilizing the proposed approach, it is much easier to predict the acoustic performance of the vibro-acoustic system, which reduces computational time considerably when several design alternatives are to be tested. The following section summarizes the vibro-acoustic model and analysis. Then, the details of the structural modification technique are presented in Section 3. The comparison of the acoustic performance of the vibro-acoustic system before and after the structural modification is given in Section 4.

2. VIBRO-ACOUSTIC MODEL AND ANALYSIS

2.1 Structural Finite Element Model

Solid elements are used to mesh the rectangular box structural model (See Figure 1). After assigning the element types and material properties, modal analysis is performed using ANSYS Software [10]. The analysis model includes a rigid rectangular box separated by a flexible mid-panel (eg: modulus of elasticity of the rectangular box is much higher than the modulus elasticity of the mid-panel). The flexible mid-panel divides the acoustic cavity into two parts. The flexible panel is excited at the center point with excitations ranging from 30-190 Hz. A global 1 % structural damping is introduced. The velocities are obtained at the flexible panel nodes using the structural modification technique presented in Section 3. Then, these normal velocities are used as boundary conditions in the vibro-acoustic analysis.

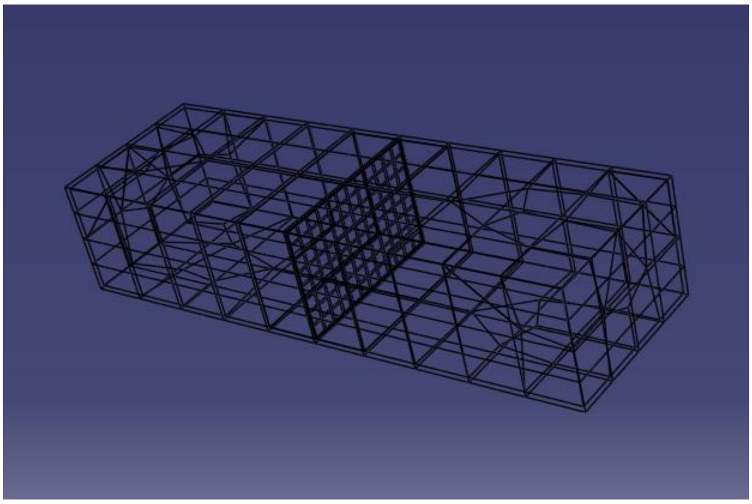

Figure 1: Structural finite element model

2.2. The Cavity (Acoustic) Boundary Element Model

In order to analyze the interior noise, the acoustic cavity of the rectangular box must be defined and meshed. In order to analyze the interior noise, the interior (volume) should be meshed such that the vibrations caused by the excitations can be transferred to the cavity across the outer envelope of the cavity mesh.

A cavity mesh (volume mesh) was created directly from the structural finite element model in LMS Virtual Lab (SYSNOISE) [11]. The cavity mesh can be seen in Figure 2, which includes the two divisions of the acoustic cavity separated by the mid-panel. After the cavity mesh is generated, a skin meshing algorithm is used to build the "*skin mesh*" of the cavity such that the BEM can be used for the sound pressure calculations. As it can be seen from Figures 1 and 2, the structural and cavity meshes are incompatible in terms of the mesh size. The velocity outputs calculated at the structural nodes should be linked to the nodes at the "*skin mesh*" such that they can be used as the boundary conditions for the acoustic analysis. Mesh mapping algorithm is utilized in LMS Virtual Lab (SYSNOISE) such that the structural and cavity meshes are compatible for transferring the velocity data.

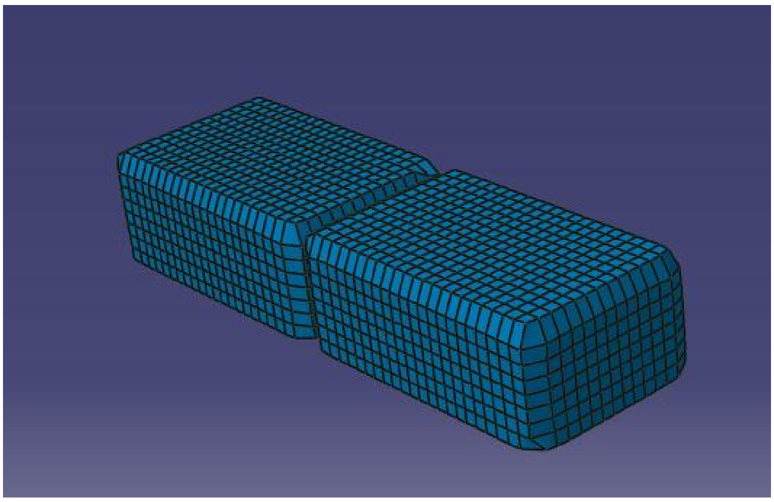

Figure 2: Cavity model

2.3. Vibro-Acoustic Analysis using the Acoustic Transfer Vectors

The acoustic transfer vector (ATV) is an array of transfer functions between the surface normal velocities and the pressure at the field point [12, 13]. The ATVs from the radiating surface to specified field points are evaluated in the first step of the vibro-acoustic analysis 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 any frequency within the range. This ATV-response calculation is a vector-vector product, given as,

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

and involves negligible computation time. The $p(\omega)$ is the sound pressure level at the predefined position, $v(\omega)$ represents the normal velocities calculated at the boundaries surrounding the cavity and $ATV(\omega)$ shows the acoustic transfer vector between the input and the output.

3. STRUCTURAL MODIFICATION

3.1. Modeling Approach for Distributed Modifications

In a structural modification problem the additional dynamic stiffness matrix due to structural modification is given by

$$[\Delta D] = [D] - [D_0] \tag{2}$$

where $[D]$ and $[D_0]$ are the dynamic stiffness matrices of the modified and original structures, respectively. For lumped modifications $[\Delta D]$ corresponds directly to dynamic stiffness matrix of the modifying structure. However for distributed modifications, it has to be calculated by using Equation 2, which may not correspond to the dynamic stiffness matrix of the modifying structure. In order to apply Equation 2, the dynamic stiffness matrices of the original and modified structures should be available. This requires the computation of the FE models for original and modified structures. However, if such FE models were available, the advantage of using structural modification method would be limited. A different approach for the application of the structural modification technique for distributed modifications is proposed by Canbaloglu and Özgüven [9]. If distributed modification is applied to an original structure in such a way that additional DOF is introduced, then it is not necessary to use Equation 2 in order to calculate the additional dynamic stiffness matrix due to structural modification, as the problem will be a structural coupling problem. In that case, the additional dynamic stiffness matrix due to structural modification will be equal to the dynamic stiffness matrix of the modifying structure which can directly be used in the structural modification method. The following section summarizes the structural modification technique used in this study.

3.2. Structural Modification Method with Additional DOF

In this section, the formulation given by Özgüven [2] is summarized. FRF matrix of a modified system can be partitioned as; DOFs which correspond to original structure only (superscript a), DOFs at connection points (superscript b), and DOFs that belong to modifying structure only (superscript c). Then the following equations can be written for original and modifying structures.

$$[\alpha_0]^{-1} = \begin{bmatrix} \alpha_0^{aa} & \alpha_0^{ab} \\ \alpha_0^{ba} & \alpha_0^{bb} \end{bmatrix}^{-1} = [K_0] - \omega^2 [M_0] + i[H_0] \tag{3}$$

$$[\alpha]^{-1} = \begin{bmatrix} \alpha^{aa} & \alpha^{ab} & \alpha^{ac} \\ \alpha^{ba} & \alpha^{bb} & \alpha^{bc} \\ \alpha^{ca} & \alpha^{cb} & \alpha^{cc} \end{bmatrix}^{-1} = \begin{bmatrix} \begin{bmatrix} \alpha_0^{aa} & \alpha_0^{ab} \\ \alpha_0^{ba} & \alpha_0^{bb} \end{bmatrix}^{-1} & 0 \\ & 0 \\ 0 \quad\quad 0 & 0 \end{bmatrix} + \begin{bmatrix} 0 \quad 0 & 0 \\ 0 & \\ 0 & \begin{bmatrix} D_{\text{mod}}^{bb} & D_{\text{mod}}^{bc} \\ D_{\text{mod}}^{cb} & D_{\text{mod}}^{cc} \end{bmatrix} \end{bmatrix} \tag{4}$$

where $[\alpha_0]$ and $[\alpha]$ represent the receptance matrices of the original and modified structures, respectively. After some matrix manipulations, the receptance submatrices of the modified system can be obtained as

$$\begin{bmatrix} \alpha^{ba} \\ \alpha^{ca} \end{bmatrix} = \left[\begin{bmatrix} I & 0 \\ 0 & 0 \end{bmatrix} \begin{bmatrix} \alpha_0^{bb} & 0 \\ 0 & I \end{bmatrix} . [D_{\text{mod}}] \right]^{-1} \begin{bmatrix} \alpha_0^{ba} \\ 0 \end{bmatrix} \tag{5}$$

$$\begin{bmatrix} \alpha^{bb} & \alpha^{bc} \\ \alpha^{cb} & \alpha^{cc} \end{bmatrix} = \left[\begin{bmatrix} I & 0 \\ 0 & 0 \end{bmatrix} \begin{bmatrix} \alpha_0^{bb} & 0 \\ 0 & I \end{bmatrix} . [D_{\text{mod}}] \right]^{-1} \begin{bmatrix} \alpha_0^{bb} & 0 \\ 0 & I \end{bmatrix} \tag{6}$$

$$[\alpha^{aa}] = [\alpha_0^{aa}] - [\alpha_0^{ab} \mid 0] [D_{\text{mod}}] \begin{bmatrix} \alpha^{ba} \\ \alpha^{ca} \end{bmatrix} \tag{7}$$

$$[\alpha^{ab} | \alpha^{ac}] = [\alpha_0^{ab} | \; 0] - \left[[I] - [D_{\text{mod}}] \begin{bmatrix} \alpha^{bb} & \alpha^{bc} \\ \alpha^{cb} & \alpha^{cc} \end{bmatrix} \right] \tag{8}$$

It should be noted that the order of the matrix to be inverted is equal to the DOF of the modifying structure, which is usually much less than then the total DOF of the structure.

3.3. Case Study-Vibro-Acoustic Analysis using Structural Modification Technique

This study summarizes our preliminary results to demonstrate the implementation of the structural modification technique to a vibro-acoustic problem. The sound pressure level at the predefined location is investigated before and after adding a stiffener (eg: distributed modification) to the mid-panel. The location and shape of the stiffener added to the mid-panel is shown in Figure 3.

The receptance matrices of the original structure can be obtained by modal analysis using the FEM of the original structure. Then, the Equations 5 to 8 are used to calculate the receptance matrices of the modified structure. When the receptance functions at the excitation point are compared for the modified and the original structure, we observe a frequency shift at the 1st and the 2nd resonances after the modification takes place (See Figure 4). The resonance peak at 62 Hz of the original structure is shifted to 80 Hz, whereas the second resonance at 152 Hz is shifted to 167 Hz.

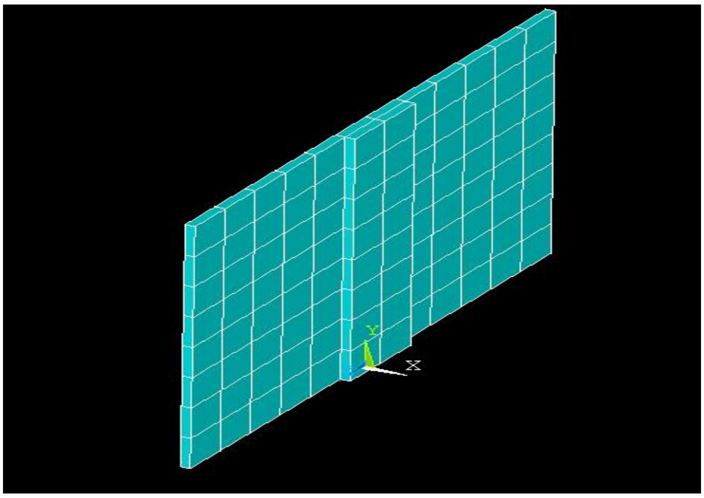

Figure 3: FEM of the mid-panel with the structural modification

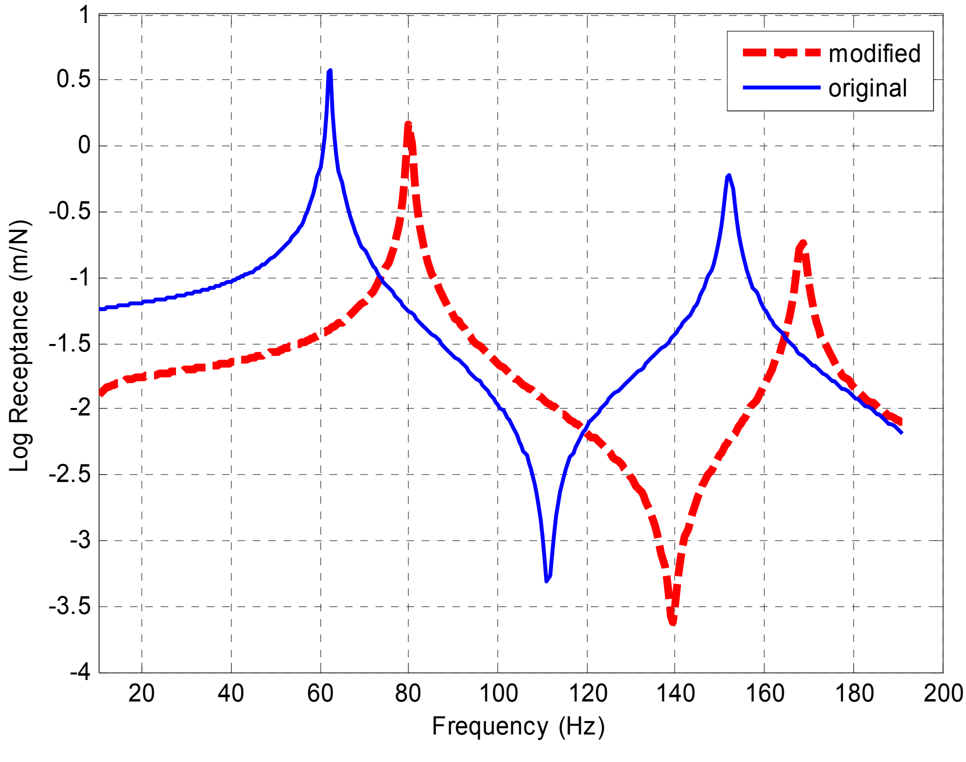

Figure 4: Comparison of the receptances of the original and modified structure

The receptance matrices are then used to calculate the mobilities of the mid-panel nodes. Finally, by multiplying the mobilities with the excitation forces at different frequencies, the velocities normal to the mid-panel are calculated (Note that the velocities of the rectangular box is not taken into consideration since these panels are much stiffer than the mid-panel). The normal velocities are then imported to LMS Virtual Lab (SYSNOISE) to perform the vibro-acoustic analysis using the ATV method described in Section 2.3. The SPL results are compared for the modified and the original structure and demonstrated in Figure 5. As it can be seen from the figure, the structural modification improves the SPL in the overall frequency range

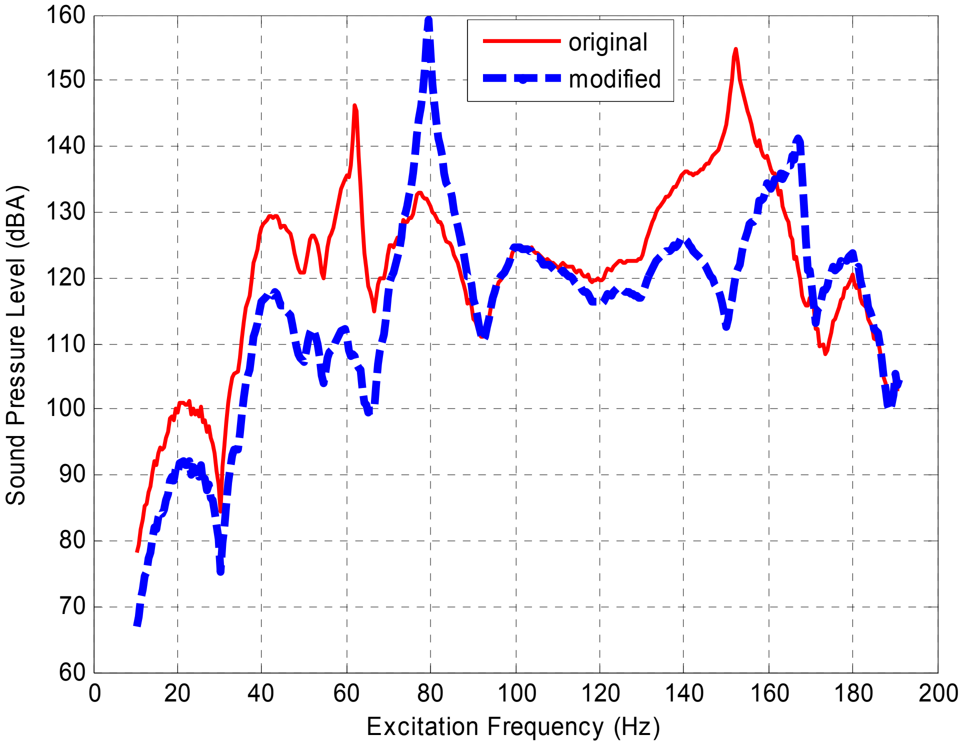

Figure 5: Comparison of the SPL before and after the structural modification

except at 80 Hz. The effect of the shifted 1st and 2nd resonances of the mid-panel can be clearly observed. The high peak at 80 Hz should be further investigated to reduce the effect of the shifted 1st resonance of the mid-panel. Because of the non-linear and complex nature of vibro-acoustic problems, in these types of modification efforts, it is very difficult to foresee the effect of these changes and also to decide how much each design variable should be altered to optimize the performance.

4. DISCUSSION AND CONCLUSION

In this paper, we successfully adapted a structural modification methodology to a vibro-acoustic problem to investigate the effects of the structural modification to the acoustic performance of the system. The structural modification technique utilizes the frequency response functions (FRFs) of the original model for the reanalysis of the structure that is subjected to structural modification. In the presented method, the receptances of the original structure are used with the dynamic stiffness of the distributed component that is added to the original structure to perform the structural modification. The modification technique presented in this paper proves itself to be very efficient when structural modifications are considered to improve the sound pressure levels in vibro-acoustic systems. Further studies are required to determine where and how these modifications should be applied to the original structure so that the vibro-acoustic performance can be optimized. Implementation of the same methodology to more complex systems such as vehicle models will be addressed in future studies.

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