

**Experimental and Numerical Prediction of Structure Borne
Noise in a Heavy Duty Truck Cabin and a Novel Regularization
Methodology for FRF Based Substructuring**

by

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and found that it is complete and satisfactory in all aspects,
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ABSTRACT

Automotive manufacturers invest a lot of money and time to enhance the vibro-acoustic performance of their products. In a complex dynamic system such as a heavy duty truck cabin, the vibro-acoustic enhancement and redesign effort may be very difficult and time-consuming without prior knowledge about the noise contributors.

This study proposes a design methodology that employs high fidelity computational models together with experimental methods to predict the vibro-acoustic performance of a heavy duty truck cabin. The analysis tools and the methodology presented provide a systematic and quantitative way to investigate the vibro-acoustic performance of such systems and identify the critical components of the system that degrade the performance.

The noise inside the vehicle cabin originates from various sources and travels through many pathways. The first step of sound quality refinement is to find the pathways and corresponding operational internal forces. For that reason, acceleration responses and frequency response functions (FRFs) are measured on a prototype truck to determine the excitation forces while engine is running in operational conditions. Once these internal forces are identified using the experimental force identification (FI) technique, they can be utilized to predict the total sound pressure level inside the cabin and also perform the panel acoustic contribution analysis (PACA) to determine the most problematic panel of the cabin. An uncoupled vibro-acoustic finite element model (FEM) is used and validated with experimental measurements to predict the sound pressure level inside the cabin. When the most noise radiating panel is identified, it can be redesigned to improve the sound pressure level inside the cabin.

Second half of this thesis focuses on response prediction with FRF-based Substructuring (FBS) technique where the FRFs of the truck cabin obtained from a numerical model is coupled with the experimentally measured FRFs of the truck chassis. The influence of each substructure modification, on the chassis and/or cabin, can be easily

assessed by reformulating the FBS equations. However, the coupling procedure of the substructures involves an inverse problem which requires regularization techniques since the matrix being inverted can be ill-conditioned. The inverted FRF matrix is called the Kernel matrix. Accuracy of the predictions made with this technique is highly dependent on the inversion of the Kernel matrix and in general, singular value decomposition is used for the inversion procedure. In this thesis, a novel methodology is adapted for lowering the condition number of the Kernel matrix by selective omission of the cross-coupling terms such that the matrix inversion procedure can be improved. Response predictions obtained by the proposed regularization method are compared with the experimentally measured responses and also with those obtained through Moore-Penrose pseudo-inverse technique with Singular Value Decomposition (SVD) to verify the accuracy of the developed approach. Finally, sensitivity analysis of the joint stiffnesses connecting the substructures is performed to determine their effect on vibro-acoustic performance of the structure. It is shown that the regularization method plays a significant role on the outputs of the sensitivity analysis.

All the studies presented in this thesis, demonstrate that the methodologies developed in this thesis accurately predict the end-to-end vibro-acoustic performance of the heavy duty truck cabin and also they can be utilized effectively to identify and redesign the components that degrade the performance.

Keywords: vibro-acoustics, FEM, frequency response function, noise transfer function, force identification, cross-coupling effect, panel acoustic contribution analysis, FRF-based substructuring

ÖZET

Otomotiv üreticileri ürünlerinin vibro-akustik performanslarını iyileştirmek için birçok para ve zaman yatırımı yapmaktadır. Örnek olarak bir ağır kamyon kabini, gürültü ve titreşim performans geliştirme çabaları gürültü kaynakları ve yolları hakkında ön bilgiye sahip olmadan, deneme yanılma yöntemleri ile ciddi zaman kaybına neden olabilir.

Bu çalışma, bir ağır kamyon kabini vibro-akustik performansını tahmin edebilmek için deneysel yöntemler ile birlikte yüksek gerçeklik oranlı hesaplama modelleri kullanan bir tasarım metodolojisi önermektedir. Önerilen analiz araçları ve sunulan metodoloji, bu sistemlerin vibro-akustik performansını araştırmak ve sistem performansını düşüren kritik bileşenlerini tanımlamak için sistematik bir yol sağlamaktadır.

Araç kabini içindeki gürültü, çeşitli kaynaklardan birçok aktarım yolları üzerinden sürücüye ulaşmaktadır. Gürültü kalitesinin iyileştirilebilmesi, yol şartlarında motordan gelen tahrik kuvvetlerinin tespiti ile başlar. Bu çalışma kapsamında tahrik kuvvetleri deneysel verileri kullanarak FRF matrisi tersini alma işlemi ile hesaplanır. Bu hesaplanan tahrik kuvvetleri, akuple olmayan vibro-akustik sonlu elemanlar modeli (FEM/FEM) kullanılarak kabin içindeki toplam ses basıncı seviyesini tahmin edilmesinde ve deneysel ölçümler ile karşılaştırılmasında kullanılır. Kabin içinde oluşan gürültü seviyesine artı veya eksi yönde katkıda bulunan en sorunlu panel/panelleri tespit etmek için simülasyon ortamında Panel akustik katkı analizi (PACA) metodu kullanılır. Bu panel/paneller tespit edildikten sonra kabin panellerinin pasif kontrol sistemleri ile yeniden tasarlanarak gürültü performansının 5dB'e kadar iyileştirilebileceği gösterilmiştir.

Bu tezin ikinci bölümünde, FRF temelli alt sistem (FBS) analizi kullanılarak sistem tepkisinin tahmini için kamyon şasesinin deneysel ölçülmüş FRF'leri ile kabinin simülasyon modeli üzerinden ölçülmüş FRF'lerinin akuple edilmesi incelenmiştir. Bu metot sayesinde herhangi bir alt sistem değişikliğinin genel sistem üzerine etkisi kolayca ilgili alt sistem FRF'lerindeki değişiklik ile FBS denklemlerinin yeniden formüle edilmesi

ile değerlendirilebilir. Fakat, alt sistemleri birleştirme işlemi aslında bir tersine problem olduğu için ortaya kötü koşullanmış FRF matrisleri oluşmaktadır ve bu matrisler düzenlileştirme tekniklerinin müdahalesini gerektirirler. Tersine alınan FRF matrisi Kernel matrisi olarak bilinmektedir. Bu teknikle yapılan sistem tepki tahminlerin doğruluğu tersine alınan Kernel matrisinin kalitesine bağlıdır ve genel olarak, tekil değer ayrışım (singular value decomposition) tekniği kullanılarak tersine alınan matrisler düzenlileştirilebilir. Bu çalışmada özgün bir yöntem geliştirilerek, matris tersini alma işlem kalitesinin önemli ölçüde artırılması sağlanmıştır. Bu yeni yöntem, diğer düzenlileştirme yöntemlerine göre farklı bir bakış açısıyla soruna yaklaşmaktadır. Yeni yöntem, Kernel matrisinde bulunan diyagonal olmayan bütün elemanların, matrisin kondisyon sayısını azaltmaya yönelik, seçici bir şekilde göz ardı ederek çalışmaktadır. Son olarak, alt sistemleri bağlayan burçların genel sistemin gürültüsü ve direksiyon titreşimi azaltmaya yönelik hassasiyetleri tespit edilmiştir. Bu çalışmada Kernel matrisi için kullanılan tersini alma tekniklerinin, hassasiyet analizi sonuçları üzerinde yadsınamayacak bir etkiye sahip olduğu gösterilmiştir.

Bu tezde sunulan tüm çalışmalar, geliştirilen yöntemlerin bir ağır kamyon kabinin vibro-akustik performansının yüksek doğruluk oranı ile tahmin edilmesinde ve ayrıca akustik performansı düşüren bileşenleri tanımlamak ve yeniden tasarlanmasında etkin bir şekilde kullanılabileceğini göstermektedir.

Anahtar kelimeler: vibro-akustik, FEM, frekans tepki fonksiyonu, gürültü transfer, fonksiyonu, kuvvet tahmini, çapraz akuple etkisi, panel akustik katkı analizi, FRF temelli alt sistem analizi

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CHAPTER 1

INTRODUCTION

1.1 Introduction

The transportation industry is forced to produce more comfortable and quieter vehicles because of high competition in the market. Comfort, safety and performance are the three major customer's selection criteria that make significant differences in the sales performance of heavy duty trucks. Due to these requirements, noise, vibration and harshness (NVH) exposure of the passengers has been an important topic for vehicle manufacturers in the last decade (e.g. [1-10]).

Many sources may cause an increase in the sound pressure level affecting comfort inside the passenger cabin. These sources are classified as structure-borne and airborne [8]. In this study, the structure-borne noise, which is mainly caused by the vehicle' engine located on the chassis, is investigated. The excitation coming from the engine passes through many pathways and causes the cabin panels to vibrate, and further, leads to an undesirable booming noise, which occurs in the low frequency range (below 100 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 the cabin interacts with the air inside.

During the design phase of heavy duty trucks, addressing the noise and vibration problems is very difficult, time consuming, and costly due to the complexity of the problem. In those cases, computer-based models and simulations become an integral part of the development process to test and verify the system level vibro-acoustic performance.

In this thesis, we propose a design methodology that employs high fidelity computational models together with experimental methods to predict the vibro-acoustic performance of a heavy duty truck cabin. The analysis tools and the methodologies

presented in this thesis provide a systematic and quantitative way to investigate the end-to-end vibro-acoustic performance of such systems, perform panel contribution analysis, and identify the critical components of the system that degrade the noise and vibration performance. Although some of the standard commercial tools and well established techniques are utilized in this study, the integration of the results of such analysis tools and experimental findings is unique and the methods that we propose provides a lot of insight to the design engineers.

The following sections present some of the engineering methods currently being used to solve noise and vibration problems used in the literature and provide an insight about the tools and methodologies that are utilized in this thesis.

1.1.1 Vibro-Acoustic Analysis

The powertrain systems, tire/road interactions and airflow over the vehicle are the fundamental excitation sources for the acoustic domain inside the vehicle [8]. They are categorized in two groups: Structure-borne and airborne noise problems. A schematic representation of the problem is depicted in Figure 1.1

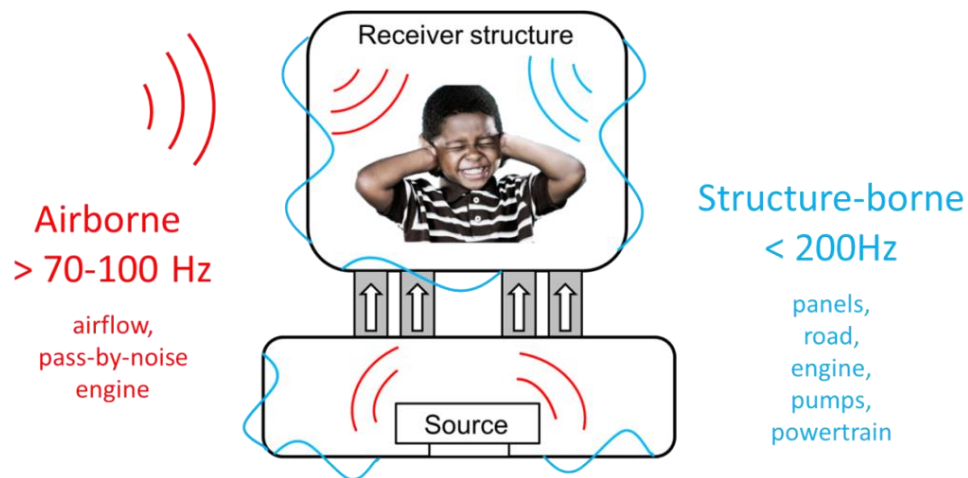


Figure 1.1: Example of structure-borne and airborne noise problems in an enclosed environment (e.g. a passenger cabin of heavy duty truck).

The first one is the structure-borne noise mainly caused by the engine excitation or road disturbances that are transmitted through the tires and the suspension system. Second, the air-borne noise mainly caused by the airflow over the vehicle body. In this thesis, only the structure-borne noise caused by the engine vibrations is investigated.

One of the most fundamental papers about the structure-borne noise was presented by Dowell [10]. He studied the structure-borne noise analytically in terms of acoustic/structural coupling. He gave a general theory of noise transmission analysis and developed an efficient numerical method for computing acoustic natural modes and interior noise levels due to panel vibrations of the passenger cabin.

On the other hand, many researchers studied the accuracy and the limitations of Finite Element (FE) and Boundary Element (BE) methods and reported on this topic so far [11-25]. Unlike BEM, the FEM is a numerical technique for solving models in differential form. For a given design, the FEM requires the entire design, including the surrounding region, to be modeled with finite elements. Therefore, the basic difference between these two techniques is the fact that BEM only needs to solve the unknowns on the boundaries, whereas FEM solves for a chosen region of space and requires a boundary condition bounding that region. Unlike FEM, which must use a 3D finite element mesh, BEM uses only 2D elements on the surfaces which are the material interfaces or assigned boundary conditions. Since only elements on interfaces are involved in the solution procedure, problem modifications are also easy.

Both techniques have advantages and restrictions. In this thesis, only FEM/BEM approach will be used for the structural and the acoustic domain.

1.1.2 Force Identification

In automotive applications, the primary noise inside the passenger cabin generally is a function of the vehicle's engine rpm. The excitation coming from the engine causes the panels of the passenger cabin to vibrate, and further, causes an increase in the noise level.

The increase in the sound pressure level generally leads to an undesirable booming noise, which usually occurs in the low frequency range (below 200 Hz) inside the passenger cabin [26]. In order to reduce the interior noise, it is critical to understand the dynamics of the vehicle, and more importantly, how the forces are transmitted to the passenger cabin. In vibro-acoustic systems, vibration energy is transmitted through a number of discrete connection points between source and receiver structure [27,28]. The goal is to identify those forces by using forward and inverse-calculation techniques as presented in the below literature review which is influenced by [28].

The measurement of the physical forces at the contact points is sometimes very difficult, and can only be measured by disassembling the structure such as mount stiffness method. An inverse method, so called matrix inversion method, is generally preferred to calculate the pseudo-forces acting on the transmission paths of the system [29]. Many researchers [29-31] have compared pseudo-forces method with direct measurement methods for practical applications. They have concluded that the accuracy of pseudo-force approach is suitable for source ranking and transfer path analysis.

Within the scope of this study, the internal forces at the (1) cabin mounts, (2) engine mounts are calculated using the pseudo forces technique. Cabin mount forces are used to predict the SPL and perform PACA to determine the most problematic panels which will be described in Chapter 4. Engine mount forces are used to predict the SPL and the vibration level on the steering wheel by performing FRF-based substructuring technique which will be described in Chapter 5 as shown in Figure 1.2.

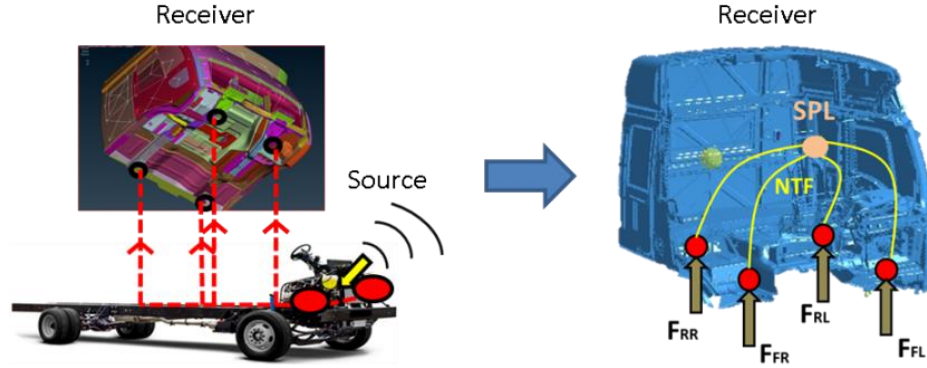


Figure 1.2: The calculated pseudo forces: (1) engine mounts, (2) cabin mounts.

1.1.3 Panel Acoustic Contribution Analysis

The Panel Acoustic Contribution Analysis (PACA) is utilized in this thesis to identify the most contributing panels to the sound pressure level inside the truck cabin [31,32]. Theoretically, PACA can be performed by identification of the partial contributions of each panel that surrounds the inner surface of a passenger cabin and sort their contributions towards the SPL at each reference microphone location. The panels are then categorized as positive (increases SPL), negative (decreases SPL) or neutral (no effect) according to their contribution to the total SPL.

PACA utilizes the Acoustic Transfer Vector (ATV) method initially introduced by [9]. 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 position of the driver's ear [9,62]. The pressure is calculated at the user defined locations using the following [10].

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

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 ATV 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,

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

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).

The experimental method uses sensors called “particle velocity probes” for measuring source strength as well as transfer functions (with a reciprocal measurement) [32]. Traditionally, depending on the panel size, the method requires many probes to be attached to interior panels (generally above 50), hence non-stationary conditions can be measured as well. Typically the measurement of panels is performed in multiple measurements and because of the low discretization resolution of the panel surfaces, the method can only be used at mid-low frequency range [32].

1.1.4 FRF-Based Substructuring

During the development phase of a new vehicle, there are many possible prototypes of a truck due to combinations of different cabins, engines, axle configurations. A practical method that predicts the vibro-acoustic response of the possible configurations of the vehicle is always needed to reach an optimal design that governs the whole configurations. The frequency response function-based substructuring (FBS) technique is a practical and robust method for computing the vibro-acoustic response of complex structures with moderately high-modal density characteristics [33-36]. Like any other method, previous

implementations of the FBS method, so called impedance coupling [37,38], state that the main causes of FBS errors are due to the inconsistencies of measured data (disturbance, frequency shifts), the inability to measure correctly all DOFs, and ill-conditioned inversion of Kernel matrices.

Recent developments of data acquisition and pre/post-processing techniques show that the errors related to the inconsistency of the measured data can be improved up to a certain degree. Based on Jetmundsen method [35], various research studies [39,40] proposed numerical techniques to filter the measured FRFs and thereby to increase the accuracy of the results of the assembled systems in experimental substructuring.

The influence of neglecting the rotational DOFs in FBS calculations is studied by many researchers [41-43]. In practice, the measurement of these rotations and torques is very difficult to accomplish. In that condition, a large number of elements in the Kernel matrix are omitted, and depending on the substructure connection flexibility, this omission may result in inaccuracies in the calculations [43].

Besides these improvements, many researches developed new techniques for conditioning the Kernel matrix in order to improve the inversion process. Generally, the singular value decomposition (SVD) is performed for the Kernel matrix to avoid the numerical ill-conditioning of the coupling calculations [44-46]. Leuridan et. al focused on pre-processing (“smoothing”) the measured substructure FRFs based on SVD analysis for each frequency band, before performing the inversion of Kernel matrix [47]. The Kernel matrix is then reconstructed using these smoothened FRFs. The predictions with their proposed smoothing algorithms can lead to some improvements in the predictions but cannot eliminate the influence of error completely [48,49]. However, incorrect use of singular value elimination of Kernel matrices may lead to elimination of strong FRF contributions from the system substructures such that the coupled response could be under-predicted. Bregant et. al states that elimination of the strong contributing FRFs have not

been studied so far, which could result significant errors in the coupling calculations [40]. The author states that some indicators must be developed in order to assure that the eliminated singular values do not contain significant structural information. Considering these needs, in this thesis, we utilize a new approach for lowering the condition number of the Kernel matrix by selective omission of the insignificant cross-coupling terms such that the matrix inversion procedure can be improved.

1.1.5 Design Modifications Based on Vibro-Acoustic Analysis

Design modifications for vibro-acoustic performance can be implemented by a variety of ways in automotive industry. Applications may include but not limited to redesigning sound sources and noise barriers, implementing sound absorbers, panel stiffeners, and buffer zones as discussed by Marburg et. al [18-22]. Alternatively, changing the stiffness properties of the components or bushings along the transmission pathways is an effective solution [50-55].

Within the scope of this thesis, the implementation of the panel stiffeners based on panel acoustic contributions and stiffness change of the bushing that connect the cabin and the chassis will be presented in the following chapters of this thesis.

1.1.6 Limitations on the Problem

The methodologies applied and developed in this study is valid for linear, time invariant parameter systems, where principles of superposition and reciprocity are valid. These are fundamental assumptions of linear acoustics theory, which is exploited, in practice, for most of the practical noise emission problems with success [56-58].

On the other hand, commercially available acoustical absorption materials, which have heat dissipation characteristics in accordance with either porous or fibrous type heat dissipation mechanisms, are very little effective at the low frequency ranges. So they are not considered as a solution alternative in the improvement of the SPL inside the cabin.

1.2 Contribution of this Thesis

The overall proposed design methodology in this thesis employs high fidelity computational models together with experimental methods to predict the vibro-acoustic performance of the cabin. The analysis tools and the methodology presented provide a systematic and quantitative way to investigate the vibro-acoustic performance of such systems and identify the critical components of the system that degrade the performance. This thesis utilizes standard well established techniques and aims to improve the accuracy of the methods with the new approaches. Therefore, the integration of the results of such analysis tools and experimental findings are unique. The major contributions of the thesis can be listed as follows:

- Integrated the high fidelity models with experimental measurements to predict the vibro-acoustic performance of a heavy duty truck cabin.
- Developed a systematic and quantitative approach to investigate the end-to-end vibro-acoustic performance of the heavy duty truck cabin using PACA method and identified the critical components that contribute to the SPL. This approach can be generalized to any complex vibro-acoustic structure.
- Adapted a new technique for regularization of the Kernel matrix by selective omission of the cross-coupling terms such that the matrix inversion procedure can be improved in the FRF based substructuring method.
- Demonstrated the effectiveness of the proposed regularization approach by a case study where the FRFs of a truck cabin obtained from a numerical model are coupled with the experimentally measured FRFs of the truck's chassis using the FBS approach.
- Compared the proposed regularization method results with the experimentally measured responses and also with those obtained through Moore-Penrose pseudo-inverse technique with SVD to verify the accuracy of the developed approach.

1.3 Outline

The present chapter covered general approaches in NVH problems with a brief literature review related to the vibro-acoustic problems, force identification and FRF-based substructuring fields.

Chapter 2 presents the details for the vibro-acoustic model of the heavy duty truck cabin which includes a FE model for the structural and acoustic domains. The vibration and SPL predictions are then compared with the experimental measurements obtained in an anechoic chamber to validate the vibro-acoustic model.

Chapter 3 presents the details of the identification of the operational loads exerted on the cabin and engine mounts. The details of the force identification procedure, and the experimental conditions and set-up are also included in Chapter 3.

Chapter 4 presents the PACA method and the results that demonstrate the major panels that contribute to the SPL inside the passenger cabin. This chapter also presents some case studies for improving the design of the panels to reduce interior noise based on the results of the PACA.

Chapter 5 presents a new technique for lowering the condition number of the Kernel matrix by selective omission of the cross-coupling terms such that the matrix inversion procedure in FRF based substructuring method can be improved. The effectiveness of the proposed approach is demonstrated by a case study where the FRFs of a truck cabin obtained from a numerical model are coupled with the experimentally measured FRFs of the truck's chassis using the FBS approach. Response predictions obtained by the proposed regularization method are compared with the experimentally measured responses and also with those obtained through Moore-Penrose pseudo-inverse technique with SVD to verify the accuracy of the developed approach.

Finally in Chapter 6, the main results, conclusions and future work recommendations are summarized.

CHAPTER 2

VIBRO-ACOUSTIC SIMULATION MODEL AND VALIDATION WITH EXPERIMENTS

2.1 Introduction

In this section, the details of the vibro-acoustic model of the heavy duty truck cabin are presented. Finite element method is used for modeling the structural and acoustical domains to perform the vibro-acoustic analysis. The results predicted with the vibro-acoustic model are compared with the experimental measurements obtained in a semi-anechoic room. The validation procedure consist of the following steps: (1) experimental setup building, (2) instrumentation, (3) experimental transfer function measurements, (4) simulation transfer function measurements, and (5) comparison of the simulation and experimental data.

2.2 Vibro-Acoustic Analysis

When analyzing the systems that represent the vibro-acoustic system, the FE model contains two different element types, namely structural and fluid (air) elements. These elements completely or partially have an influence on each other. Acoustic pressure along the fluid-structure interface may create loading on the structure which influences the structural vibrations. Conversely, the structural vibrations may affect the acoustic pressure field. The strength of this vibro-acoustic coupling interaction is largely dependent on the geometry of the structure, the fluid domain, material properties and the frequency of interest. Depending on the strength of the mutual interaction, vibro-acoustic systems may be classified into coupled and uncoupled systems [9].

In the coupled vibro-acoustic analysis approach, the mutual vibro-acoustic coupling interaction between the structural and fluid components is no longer negligible and the system can be defined as a coupled system. However, the uncoupled vibro-acoustic analysis does not consider the mutual coupling interaction between the structural and the fluid components. Only one way of interaction is considered and structural components are regarded as independent force excitations for the fluid components. In this study, the mutual effects between the truck cabin and enclosed cavity are not considered since the air has very small effect on the structural vibrations of the cabin.

The processes of two methods are shown in the Figure 2.1. Coupled method starts with developing the finite element model of the structure. After the modal analysis is performed, the structural mesh is imported to construct the surrogate and the cavity model. A surrogate model is required to represent the structural model since the mesh size of the cavity model and the structural model does not match and a representative model is needed to import the modal analysis results. The cavity model represents the fluid properties of the air inside the passenger cabin. After the force disturbances are applied at the engine mounts, the coupled analysis is performed to predict the SPL inside the cabin.

Similar to the coupled analysis, the uncoupled analysis starts with the construction of the structural finite element model of the vehicle. Forced response analysis is performed on the structure under the effect of engine disturbances. The velocity outputs of the analysis are used as the boundary conditions on the cavity model. The advantage of the uncoupled analysis is that it allows separate analysis of the structural and acoustic model which reduces the time required for the computations.

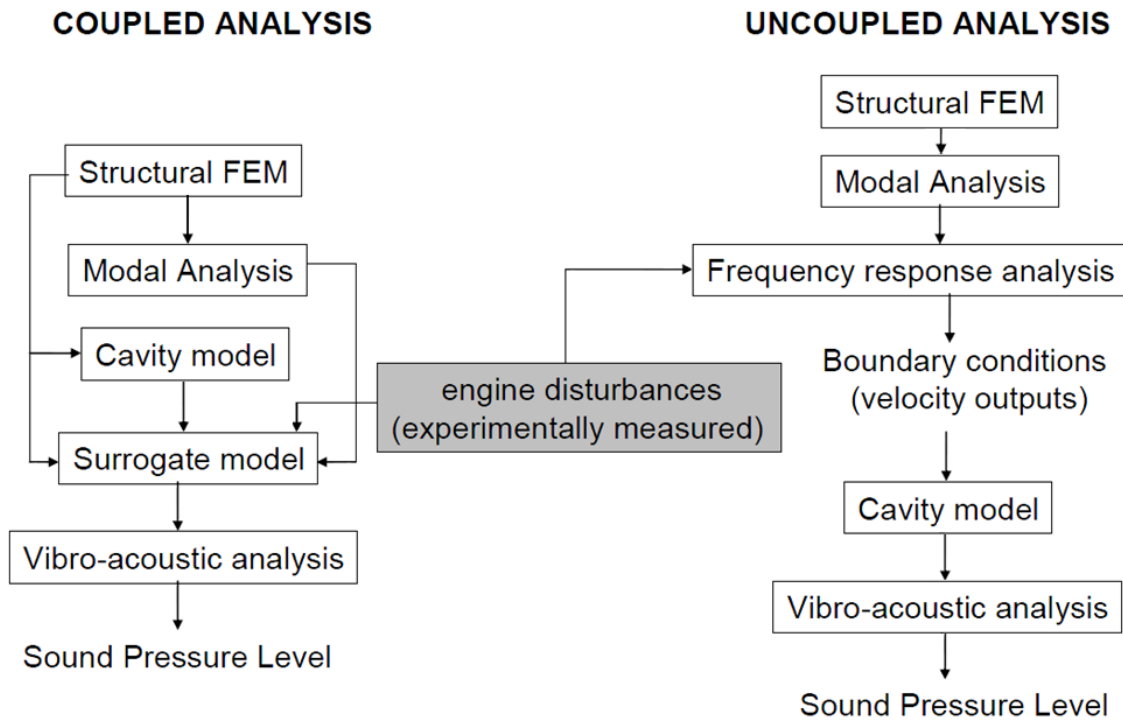


Figure 2.1: Flow chart shows the coupled and uncoupled analysis

2.2.1 Structural Finite Element Model

The first step is to create the structural FEM of the heavy duty truck cabin. The model captures the structural characteristics of the vehicle and it is built together with the engineers from Interior Quietness team of Ford Otosan R&D Center. The analysis model only includes the heavy duty truck cabin which is separated from its chassis. It includes shell (tria, quad), solid (tetra, hexa), mass and spring element types and the sheet metals are modeled by using shell elements with different thicknesses. Lumped masses are used to represent the welding of the mounts to the main chassis structure. The structural FE model of the cabin is illustrated in Figure 2.2.

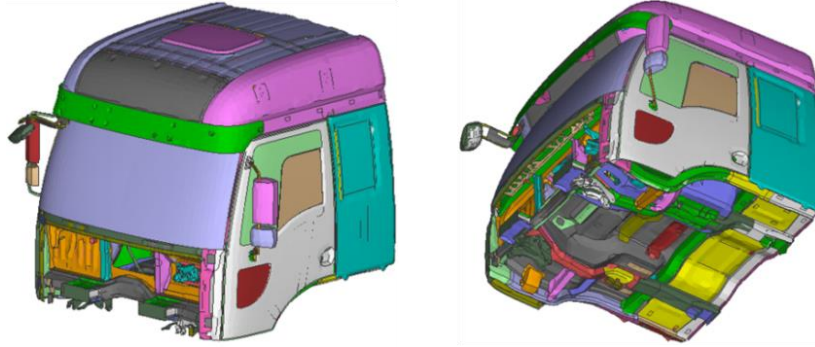


Figure 2.2: Structural FE model of truck cabin illustrating the main panel regions with different colors.

2.2.2 Cavity (Acoustic) Finite Element Model

In order to perform vibro-acoustic analysis of the passenger cabin, the acoustic cavity surrounding the passenger cabin needs to be defined. The interior volume should be meshed such that the vibration from the structure can be transferred to the cavity across the outer envelope of the cavity mesh. Later, these transferred vibrations are used to calculate the interior noise at the driver or passenger ear. In this study, a FEM is built that captures acoustical properties of the air inside the cabin. The volume mesh is created directly from the structural finite element model in LMS Virtual Lab (Sysnoise) [60] as illustrated in Figure 2.3.

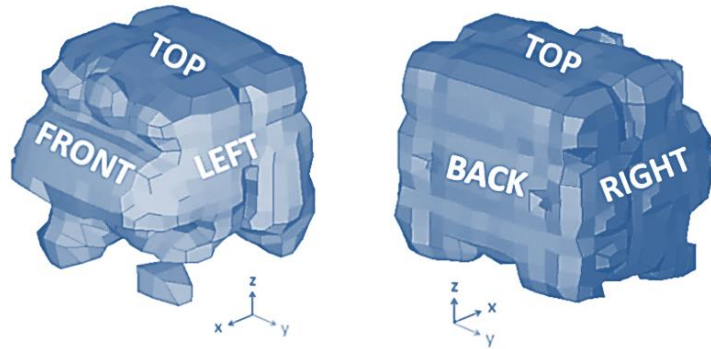


Figure 2.3: Cavity FE model illustrating the main panel regions with different colors

There are few things to pay attention while creating the cavity mesh. In order to create the cavity mesh, first, the holes in the geometry which are bigger than a single element size are detected and enclosed the by the algorithm. The meshing algorithm in LMS Virtual Lab uses HEXA element type. Moreover, the cavity can be partitioned in to many regions and then merged again by the user, which gives great control flexibility to the user on the cavity model [9]. Once the topology of the cavity mesh is defined, the next step is mapping it with the structural mesh.

2.2.3 Mesh Mapping

The structural and cavity meshes are generally modeled incompatible in terms of mesh size. This is due to the different mesh size requirements of the structural and acoustic solvers. Generally, the structural mesh size is determined to obtain good convergence; whereas the acoustic mesh size is determined by the wavelength of the fluid and the frequency range that the user is interested [9]. The appropriate acoustic element size should be determined in terms of maximum frequency range and the wavelength of the air by the following formula (Eq. 3);

$$\lambda = \frac{v}{f} \quad (3)$$

where:

λ : Wavelength (maximum allowed mesh size)

f : Maximum frequency range

v : Speed of sound waves in air (at room temperature and atmospheric pressure)

The acoustic element mesh size has to be smaller than the wavelength that can be calculated by the Eq.3 formula. By using this formula, the wavelengths of the human audible sound frequencies between 20 Hz – 20 kHz can easily be calculated. This formula yields 17 m and 17 mm respectively.

In order to compensate the element size differences between structural and acoustic mesh, a mesh mapping is required to link/match these incompatible meshes. The mapping is performed by creating the 3D outer envelope of the main cavity and linking the nodes of the structural mesh that to the node of the acoustic mesh. In this particular study, the nodes that are closer than 50mm are linked to each other. This is found to sufficient to obtain a good mesh coupling between the structural and acoustic mesh. Once the mesh mapping process is finished, the surface vibrations can be transmitted to the cavity from the mapped nodes on the boundaries. The next step is to combine the structural and cavity mesh.

2.2.4 Vibro-Acoustic Model

The cavity mesh was linked to the structural model with a mesh mapping such that the velocities calculated at the structural nodes are used for the acoustical analysis. For this model, a global 3% structural and 7% acoustical damping were used. The structural model of the truck cabin has over 1 million nodes and over 6 million degree of freedom. Almost 800 structural and 12 acoustic uncoupled modes were found between 0-200 Hz range separately which shows the complexity of the system. As seen in Figure 2.4, the FE model of the cabin was divided into groups of elements known as panels including the back panel, roof, front panel, left-door, right-door, seat and the remaining elements of the cabin model.

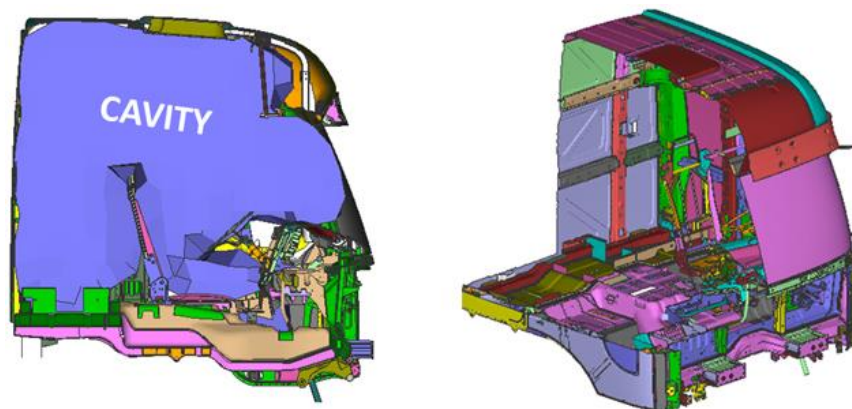


Figure 2.4: Vibro-acoustic model of the uncoupled (a) cavity and receiver (b) cabin

2.3 Validation Studies

In this section, the validation process of a FE model which has acceptably similar dynamic behavior to that of an experimental test setup is discussed. The dynamic behavior of the structure is usually represented by the experimental data that are obtained from tests undertaken on an actual test structure. There are several forms of experimental data that can be used in the process of model validation and the most widely used are: frequency response functions (FRFs) and modal data (eigenvalues and eigenvectors).

In this chapter, the experimental procedure for model validation with FRFs will be discussed. The applications during the test planning and measurements are explained for the purpose of ensuring that experimental data from a modal test can fulfil the requirements.

2.3.1 Description of the Experimental Setup

In order to investigate the acoustic characteristics of the system model more accurately, an anechoic test chamber located in Automotive Technologies Research Center (OTAM) in Istanbul Technical University was used (<http://www.otam.com.tr/nvh.html>) to validation experiments. In the scope of bench testing experiments, the reactions of the cabin to the excitations coming from the cab mounts were analyzed by using the data collected from the structural accelerometers and acoustic microphones. The frequency response functions (FRF) and noise transfer functions (NTF), which are obtained from experiments, were compared with the simulation FRF and NTF measurements to show the accuracy level of the FE model. Due to the confidentiality agreements between Koç University and Ford Otosan, some of the pictures from the experiments were blurred.

2.3.2 Instrumentation and Vehicle Preparation

Experimental studies were conducted in a semi-anechoic chamber in OTAM where the truck cab was placed on a chassis-like frame support. The system was excited from the 4 suspension points by an impact hammer in 3 directions (X,Y,Z) and vibration responses of the main cabin panels (left, right, top, rear, and front) were measured using accelerometers attached to the panels. In order to transmit the impact hammer power to the cabin mounts in all principal directions, special apparatus were mounted to the measurement locations. These mounted apparatus also ensured repeatability of the impact tests. In Figure 3.1, test bench setup in semi-acoustic room and the corresponding FE model of the cabin is presented.

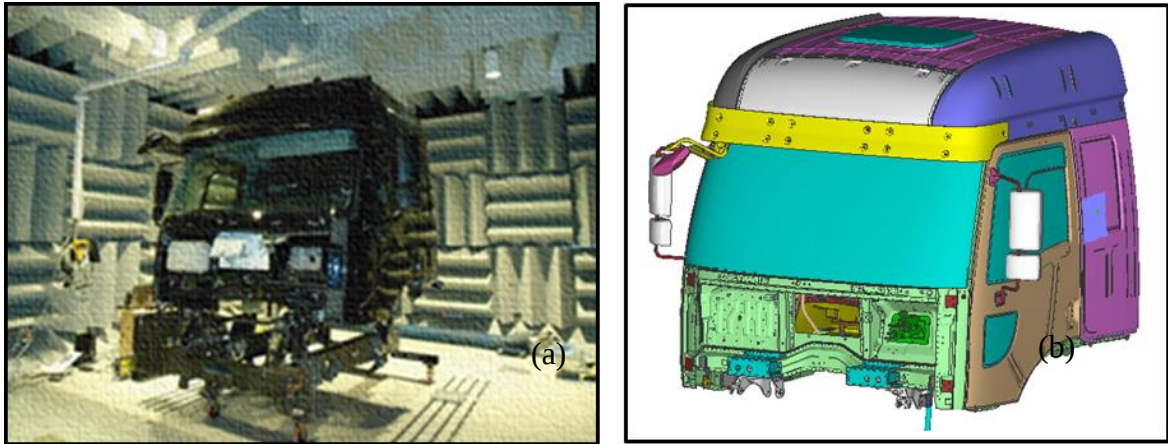


Figure 3.1: (a) Test setup in semi-acoustic room and (b) FE model of the heavy duty truck cabin

Multi-channel data acquisition system was used to collect the acceleration and microphone measurements. The accelerometers used in this experiment were piezo-electric based sensors.

On each main cabin panel, there were 25 different vibration measurement points. The actual measurement locations on the cabin panels are shown in Figure 3.1 explicitly. As seen in Figure 3.1, the semi-anechoic room was not big enough to fit the whole truck.

Therefore, the cabin was placed on top of a frame-like structure to represent the actual truck chassis.

2.3.3 Experiment: Transfer Function Measurements

FRF measurements were performed using an impact hammer. Firstly, the frequency response functions from the cabin suspension to the microphones and steering wheel were measured. The frequency response functions between the cabin mounts themselves and those from the cabin mounts to the target response locations were measured afterwards. All the FRF and NTF measurements points were shown in Figure 3.2



Figure 3.2: Accelerometer and microphone instrumentation on the cabin for FRF and NTF measurements

The structural FRF measurements were collected from 125 points normal to the panel surfaces. Impacts are made at each individual path location for each of the three orthogonal directions. The impact hammer tip and the data recording settings for the data acquisition system are selected to obtain the best measurements for a frequency range of up to 200 Hz at a resolution of 0.5 Hz. Sampling rate of the measurement set to 1024 Hz to avoid signal overlap problems. During the experiments total of 1500 structural acceleration data and 48 acoustic microphone responses were collected and stored in corresponding data files.

2.3.4 Simulation: Transfer Function Measurements

In order to obtain correlation between the model and experiments, a quite detailed model of the actual truck cabin was used as described in the previous chapter. The modal analysis results were first obtained by Nastran, then LMS virtual Lab software was used to obtain frequency response and noise transfer functions of the FE vibro-acoustic model.

The structural modal analysis of the truck cabin model was performed within 0-200 Hz frequency band. Since an impact hammer was used in the experiments, the same analogy was applied in the simulations. The forces were applied at the suspension points of the finite element model (Figure 3.3) in 3 principal directions (XYZ) and the structural accelerations were measured from the panel surfaces of the cabin which was determined by experiment measurements.

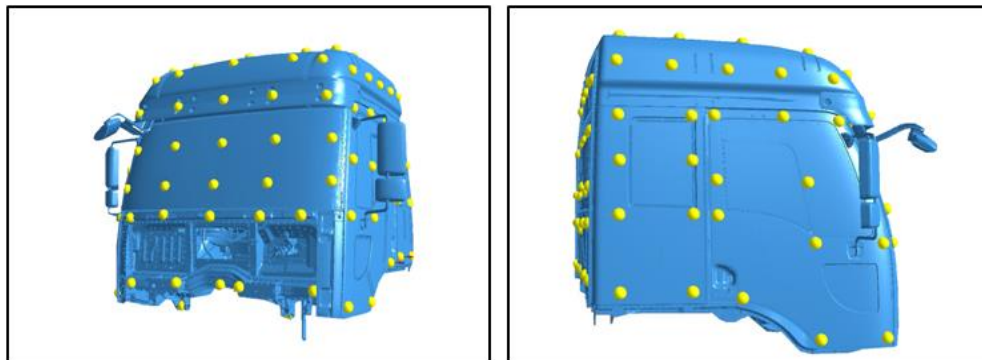


Figure 3.3: Model validation experiments of vibro-acoustic heavy duty truck cabin in a semi-anechoic room

For the calculation of NTFs, the temperature of the air inside the cabin was assumed as 25 degrees and the material properties are obtained accordingly. The NTFs were determined by using the vibro-acoustic FE model where the cabin excitation forces are applied at the cabin suspension connection points and the sound pressure level is measured at the passenger ear locations. Structural and acoustic damping coefficients were selected as 3% and 7% respectively in order to obtain consistency with the simulation and experimental results. In Figure 3.3, the acceleration measurement locations on FE model are depicted.

2.3.5 Comparison of Simulation and Experimental Results

In this section, the comparison of the acoustic and structural measurement results obtained both from simulation and test bench experiments are presented. Since the number of the acceleration measurements from panel surfaces is high, some sample measurements are selected for illustration (125 sensor x 4 mounts x 3 directions = 1500 FRF).

In Figure 3.4, the acoustical and structural transfer functions are analyzed. The figure involves the comparison results for 6 of the selected sensor points. In the figures, the red dotted lines represent the experimental measurement results, whereas the blue straight lines represent the simulation predictions. The upper three graphs include NTF and lower FRF comparison. Some unmatched peaks are due to the inconsistencies of the measured data (disturbance, frequency shifts, nodal points) or the incorrect modelling parameters (modal damping values, linearity).

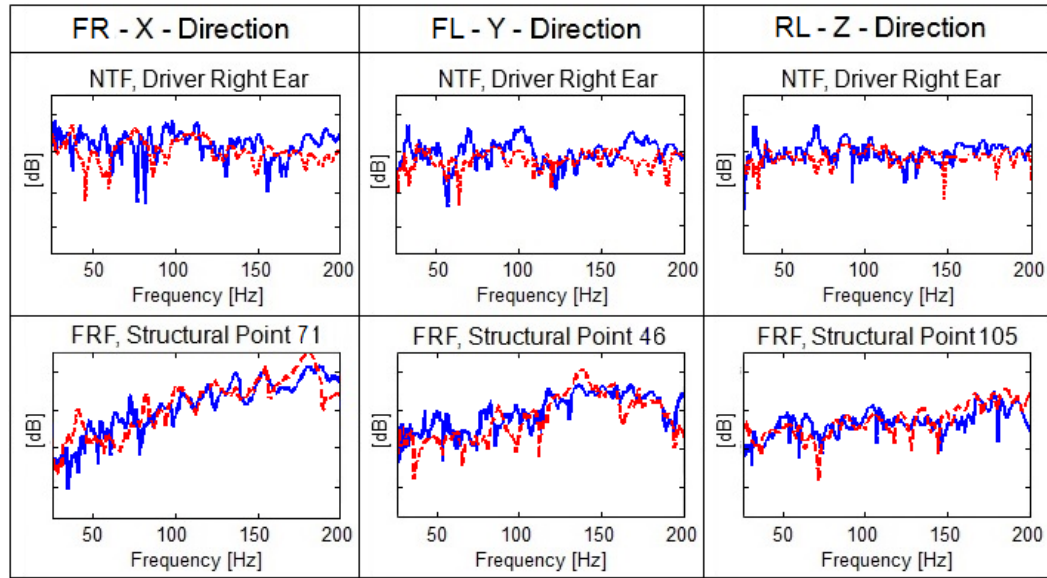


Figure 3.4: Structural and acoustical transfer functions of the passenger cabin when excited from the cabin mounts (in dB scale). Solid lines: simulation. Dashed lines: experiment. The y-axis of the figures is not displayed due to non-disclosure agreement between Koç University and Ford Otosan.

When the numerical results obtained with the FEM are compared with the experimental measurements, there was a good match at some frequencies as well as, some discrepancies at certain frequencies. The other possible causes of these differences could be due to the instrumentation constraints. For instance, the impact hammer excitations introduced errors since the measurement of the accelerations and forces at a same point is quite impossible. In every hit, the energy transmitted to the structure varies and introduces error to the results. As a result, the errors introduced either during the modeling stage (coefficients, connections, linearity, etc.) or experimental measurement stage (impact hammer issues, coherence, etc.) effect the prediction accuracy mostly. As stated by Stan et. al, this mismatch between the modeling and the experimental measurements is still an ongoing research topic concerning the biggest automotive manufacturers [25].

CHAPTER 3

FORCE IDENTIFICATION

3.1 Introduction

The noise inside the vehicle cabin originates from various sources and travels through many pathways. The first step of sound quality refinement is to find the pathways and corresponding operational internal forces. For that reason, acceleration responses and frequency response functions (FRFs) are measured on a prototype truck to determine the excitation forces while engine is running in operational conditions. Once these internal forces are identified using the experimental force identification (FI) technique, they can be utilized to predict the total sound pressure level inside the cabin. In this study, the internal forces at the (1) cabin mounts, (2) engine mounts are calculated using the FI technique.

Figure 3.1, illustrates the engine mount and cabin mount forces by red circles for a typical heavy duty truck and the placement.

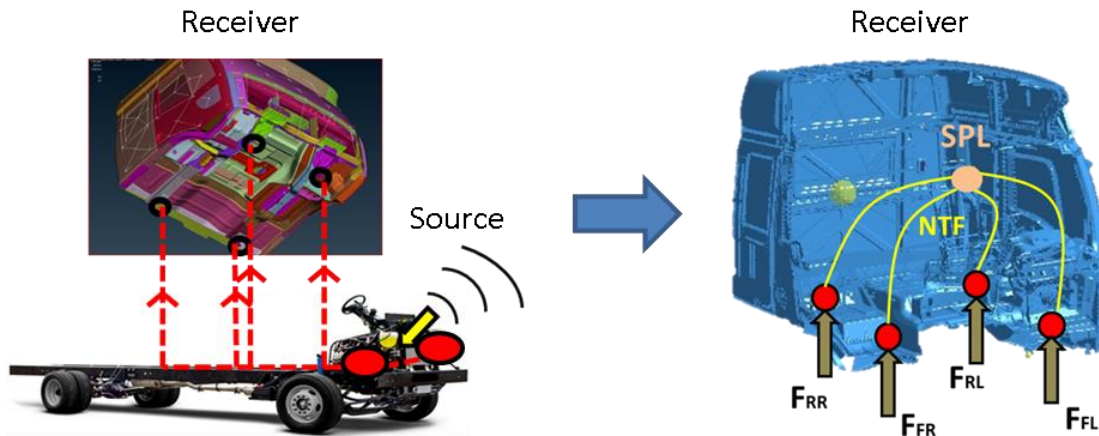


Figure 3.1: Definition of source and receiver structures for a typical heavy duty truck [11]

Cabin mount forces are used to predict the SPL and perform PACA to determine the most problematic panels which will be described in Chapter 4. Engine mount forces are used to predict the SPL and the vibration level on the steering wheel by performing FRF-based substructuring technique which will be described in Chapter 5. For more details the reader is referred to [25,61].

3.2 Transfer Path Analysis and Force Identification

Identifying the operational internal forces acting on the transmission paths between the active and passive structures is the first step required for the TPA. When the relative displacement between the active and passive sides is very small, the matrix inversion method must be used for identification of the internal forces. This method requires direct or reciprocal measurement of transfer functions between the paths and operational response measurements at the connections that define each individual path. To obtain the operational internal forces, an accelerance matrix composed of these transfer functions is inverted at each frequency and then multiplied with the measured operational responses at the path interface as shown in Eq. (4) [6].

$$\{f(\omega)\} = [H(\omega)]^{-1}\{\ddot{x}(\omega)\} \quad (4)$$

$f(\omega)$ – forces acting on the paths

$H(\omega)$ – accelerance matrix

$\ddot{x}(\omega)$ – operational responses at path interface locations

The accelerance matrix can be square or rectangular. Forces can be correctly identified when the number of response points to be measured is at least equal to the number of forces that have to be determined. Equation (4) then becomes:

$$\begin{bmatrix} f_1 \\ \vdots \\ f_N \end{bmatrix} = \begin{bmatrix} \frac{\ddot{x}_{11}}{F_1} & \frac{\ddot{x}_{11}}{F_2} & \dots & \frac{\ddot{x}_{1n}}{F_n} \\ \vdots & \ddots & & \vdots \\ \frac{\ddot{x}_{m1}}{F_1} & \dots & \dots & \frac{\ddot{x}_{mn}}{F_n} \end{bmatrix}^{-1} \begin{bmatrix} \ddot{x}_1 \\ \vdots \\ \ddot{x}_m \end{bmatrix} \quad (5)$$

where:

f_i : the internal operational forces at path i

$\frac{\ddot{x}_{ij}}{F_i}$: the FRF between the acceleration at point j and the applied force at transfer path i

$\ddot{x}_i$: the operational acceleration at interface point i

m : number of responses measured around the path interface locations

n : number of paths

Using more response point measurements than the number of transfer paths leads to an over-determined system of equations [6]. In this case, the solution is obtained by using a pseudo-inverse technique, the most widely known type is called the ‘‘Moore-Penrose’’. Inaccurate predictions caused by measurement errors can be reduced when an over-determined system is used. Especially at higher frequencies where modal overlap is significant, over-determination gives significant reductions in condition numbers and thus improvement of force and response predictions [66].

3.3 Description of the Experimental Setup

Before the vehicle preparation for the experiments, the significant transfer paths must be identified such that the measurements can be performed at those locations. In this study, main goal of the FI process is to determine the force inputs generated by the engine and to understand the contribution of each of these forces to the overall response at various locations. Therefore, the schematic representation of transmission paths on a system composed of source and receiver structure is shown in Figure 3.2.

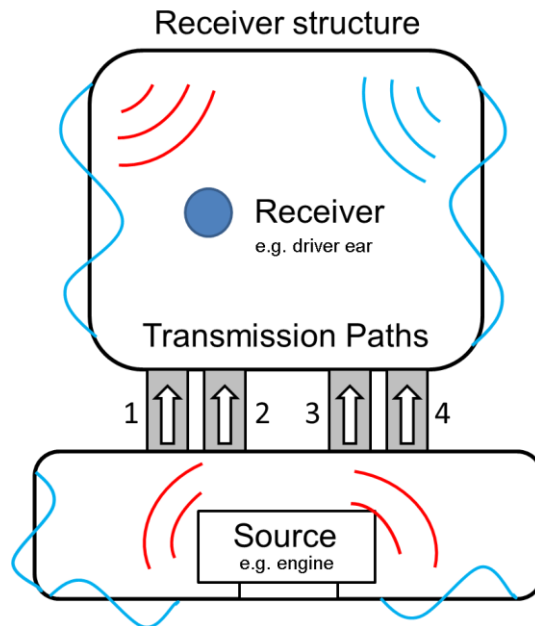


Figure 3.2: TPA - One source, two transmission paths and one receiver.

In order to simplify the analysis, transfer paths coming from the exhaust system and other secondary paths are neglected. The transfer path locations include four engine mount locations and four cabin mount locations. Each transfer path is assumed to transmit force in three orthogonal directions.

3.4 Instrumentation and Vehicle Preparation

Tri-axial accelerometers are attached to the truck frame at the location of the engine mounts at the chassis side, at the both sides of all the four cabin suspension connections and on the steering wheel. Microphones are attached to the seats, close to the position of the driver and passenger left and right ears. As an example, Figures 3.3 and 3.4 show the positioning of the accelerometers on the rear cabin suspension connections and the microphone locations.

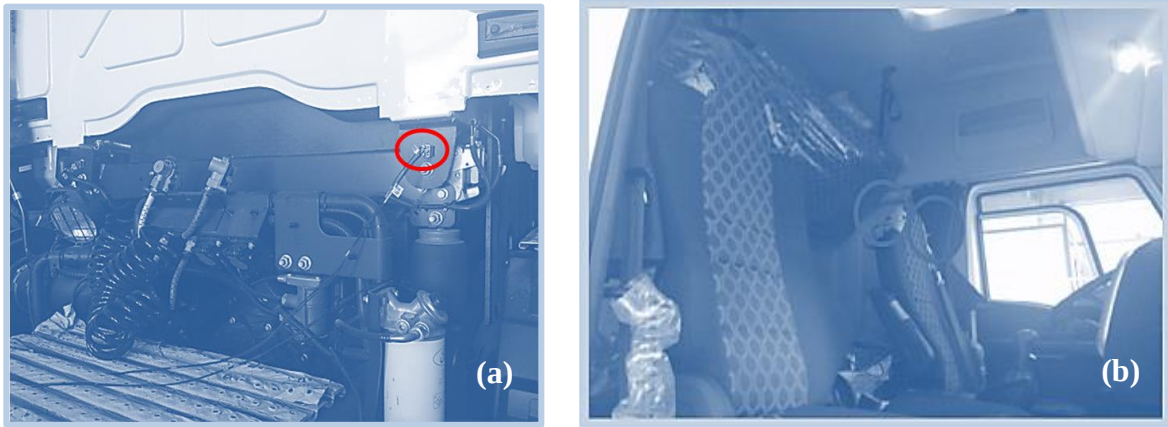


Figure 3.3: (a) Positioning of the tri-axial accelerometer on the rear right cabin connection points and (b) microphones on the driver and passenger seats (right)

3.5 Frequency Response Measurements

All the necessary FRF measurements to fill the accelerance matrix were taken inside an anechoic enclosed area when the truck was in a stationary condition. The FRFs are measured by using an impact hammer. First, the accelerance matrix FRFs between the cabin mount themselves are measured in order to identify the cabin mount forces. Afterwards, the accelerance matrix FRFs between the engine mounts themselves are measured in order to calculate the engine mount forces. Impacts were hit at each individual path location for each of the three orthogonal directions. The tip of the impact hammer and data acquisition settings for the data acquisition system is selected to obtain the best measurements for a frequency band up to 200 Hz at a resolution of 0.5 Hz.

3.6 Operational Data measurement

The operational accelerations and sound pressure levels need to be recorded while the truck is driving on a test track in the 10th gear during a wide-open-throttle (WOT) engine run-up. The recorded mount accelerations are used for the determination of the operational internal

forces. Later, the measured SPL will be used to validate the accuracy of the predicted results in Chapter 4 and 5.

For the current setup, the recorded data includes the tri-axial accelerations from each of the engine mounts and cabin suspension connections, and also the recorded outputs from the microphones and the accelerations on the steering wheel.

The experiments are performed for a wide-open-throttle (WOT) engine run-up operating conditions on a smooth track path as illustrated in Figure 3.4.

The measurements are recorded in time domain over a period of approximately 10 seconds after which the time domain data is converted in frequency domain by taking FFT.

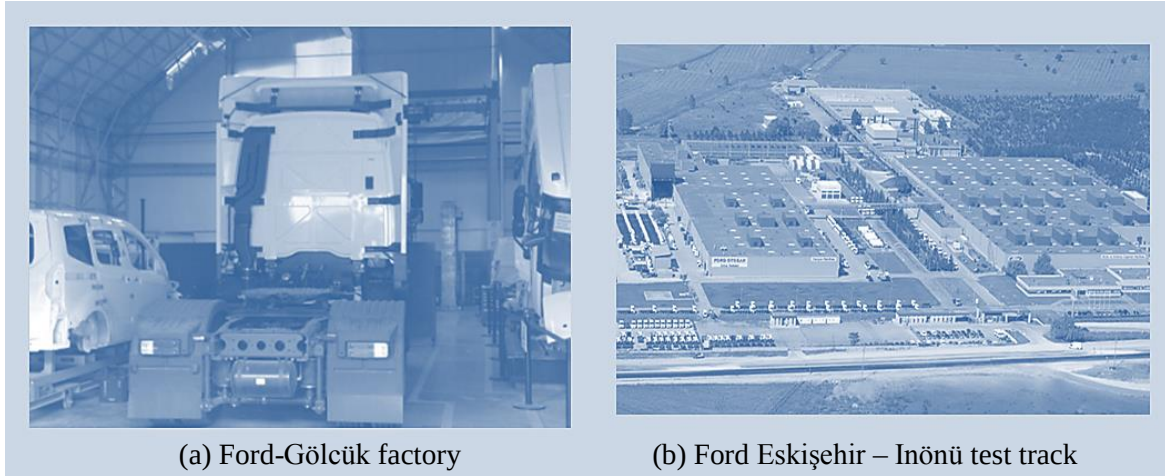


Figure 3.4: Locations where the operational measurements were performed

3.7 Results of Pseudo Force Identification: Cabin and Engine Mounts

The calculated cabin and engine mount forces for 10th gear WOT run-up condition on smooth test track are illustrated in Figure 3.5. It should be noted that, depending on the type of road condition, forces acting on the cabin and engine mounts may differ. As discussed before, the identified cabin mount forces are used in Chapter 4 and the engine mount forces are used in Chapter 5. The accuracy of these forces is crucial for the accuracy of the following simulation results.

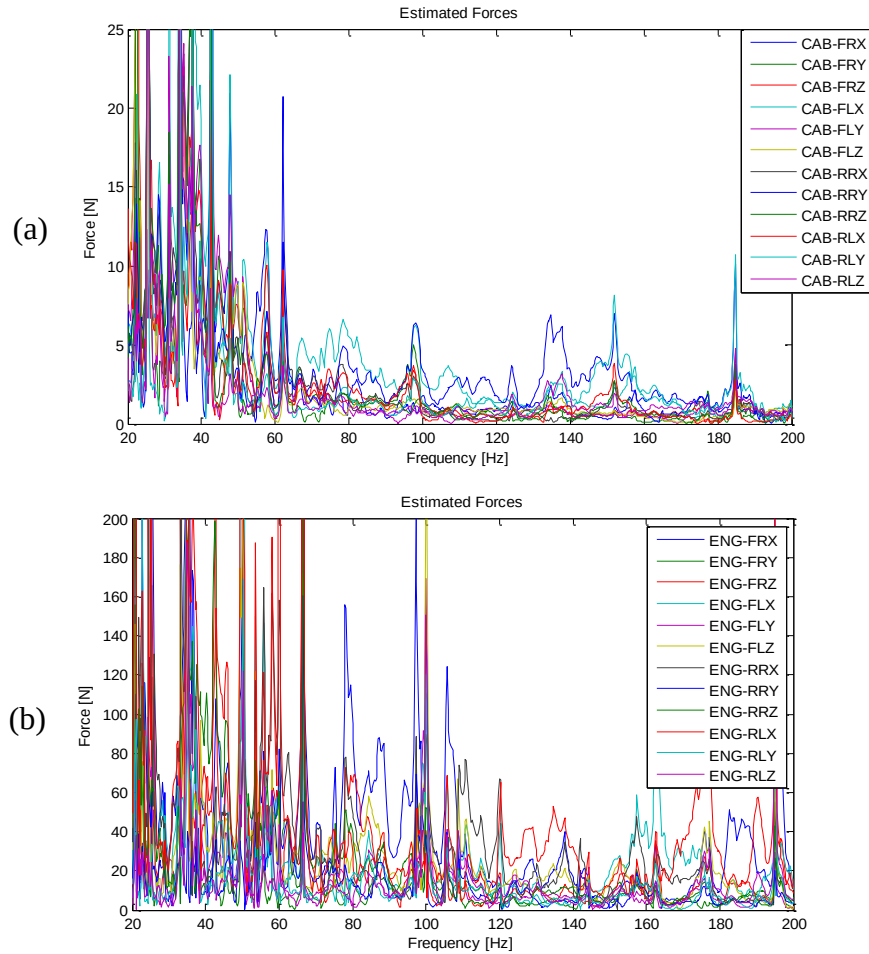


Figure 4.5: Calculated pseudo forces: (a) cabin mounts and (b) engine mounts during 10th gear WOT running condition

Looking at the spectral information of the forces, in overall, the forces acting on engine side are up to 6-7 times higher than those predicted at cabin mounts during operational conditions. The influence of the ill-conditioning of the FRF accelerance matrix played a significant role where unexpected peaks in the force have appeared. This specific phenomenon for this setup is studied by Stan et. al in [25]. Therefore, the regularization of the force estimation is studied in [59], but the applied methods are not explained within the scope of this thesis.

CHAPTER 4

PANEL ACOUSTIC CONTRIBUTION ANALYSIS

4.1 Introduction

This chapter deals with Panel Acoustic Contribution Analysis (PACA) of the experimentally validated FE cabin model. This chapter consists of the following sections; (1) determination of low frequency acoustic mode shapes, (2) theory and calculation steps of Acoustic Transfer Vectors (ATV), (3) determination of contributing body panels by PACA, and finally (4) evaluation of proposed design improvements of critical panels. Eventually, the investigations enable the redesign of most problematic body panels that degrade the vibro-acoustic character of the system. The following section mainly focuses the explanation of the steps involved in PACA.

4.2 Panel Acoustic Contribution Analysis (PACA)

The origin of the problem can be identified by investigating the modal characteristics of the cabin structure and acoustic cavity separately. Information on whether the identified problem is structural or acoustic transfer dominated can be reached from the PACA study since the panel vibrations and corresponding ATVs of the vibrating points can be investigated separately. The flow diagram of this approach was illustrated in the following section.

4.2.1 PACA Process Flow Diagram

Figure 4.1 illustrates the steps involved in PACA process. The PACA process starts with the calculation of the structural modes and frequencies. Then, Forced Response Vibration Analysis is performed using the cabin mount forces (as calculated in Chapter 3.7) to calculate the nodal velocities on the structure. Then, the acoustic analysis is performed by creating the cavity mesh using the LMS Virtual Lab. The panel definitions are created on the cavity mesh (see Figure 2.3 for the panels). Modes of the acoustic cavity are calculated in SYSNOISE software [9]. Acoustic Transfer Vectors (ATVs) are calculated by using the modes of the cavity. The coupling of the structural model with the acoustic cavity is handled by the mesh mapping technique. Finally, ATV response analysis is performed to predict the contribution of each panel at the driver and passenger ears. Then, results of the PACA are evaluated to determine the most contributing panel. Red colored regions represent the highest contributions to the sound pressure level at the driver's ear.

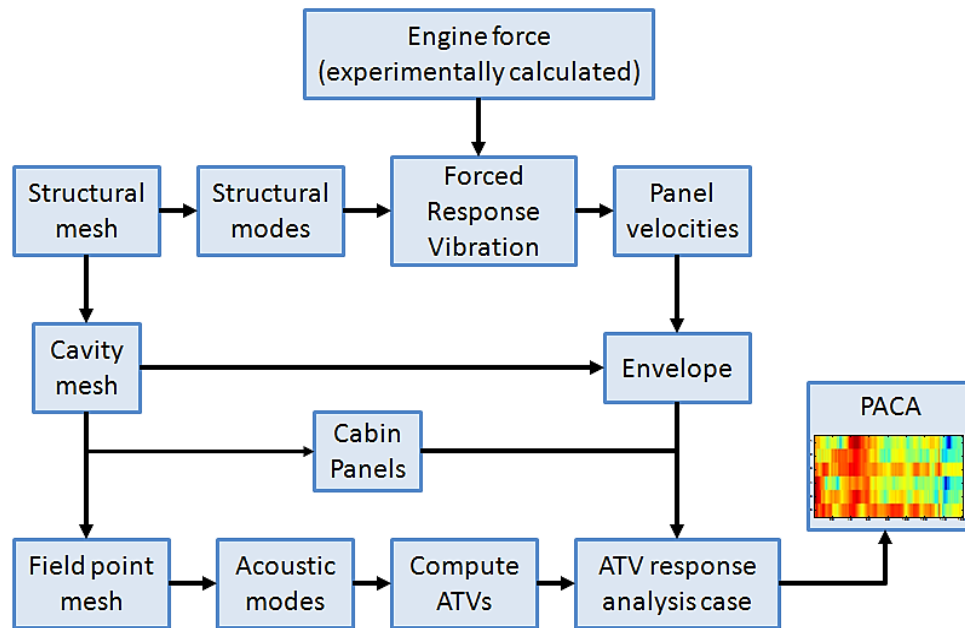


Figure 4.1: Flowchart showing the processes involved in PACA

The processes involved in PACA are visually illustrated in Figure 4.2. The method starts with structural forced response analysis. Low frequency acoustic mode shapes are determined after the structural analysis. ATVs are calculated after the acoustic analysis which is independent from the surface vibrations calculated by the forced response. After the combination of the surface vibrations with the corresponding ATVs, the SPL inside the cabin is predicted. This prediction consists of panel contributions.

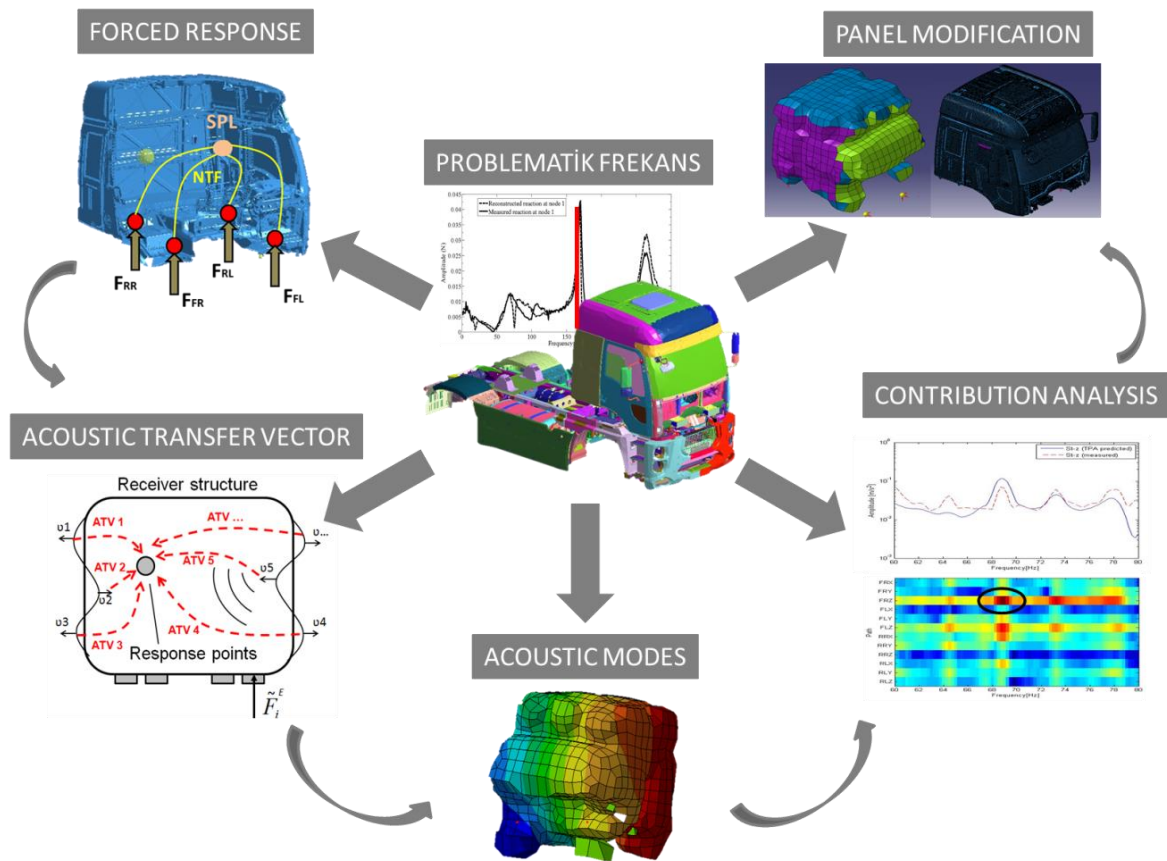


Figure 4.2: The visual illustration of the processes involved in Panel Acoustic Contribution Analysis

4.3 Determination of Low Frequency Acoustic Cavity Modes

First, the resonance frequencies and corresponding mode shapes of the acoustic cavity are determined using the FE model. The acoustic cavity modes are extracted between 0 – 200 Hz frequency range in LMS Virtual Lab (Sysnoise) Figure 4.3 demonstrates the acoustic cavity 12 modes and the corresponding mode shapes of the cabin structure.

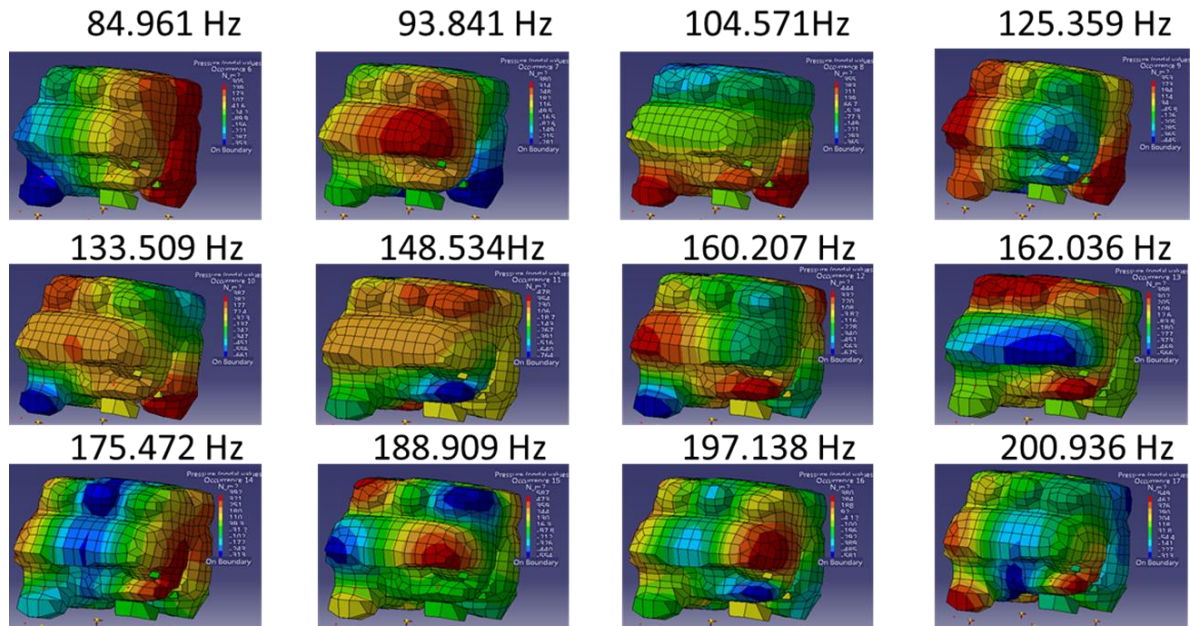


Figure 4.3: Mode shapes of the acoustic cavity inside the truck cabin and frequencies

These cavity resonances are prone to generate a booming noise when they are coupled with the structural vibrations. On the other hand, between certain frequency ranges where the acoustic transmission is low, high levels of structural vibrations may not cause noise problems in the cabin. The determination of so-called problematic frequency range is very crucial; therefore, engineers have to avoid matching of the cavity resonances with the structural modes when designing their systems.

4.4 Theory of Acoustic Transfer Vector (ATV) Analysis

The ATV method is based on Acoustic Transfer Vectors (ATVs) that combine the input of the structural velocity of the radiating surface and the sound pressure level at the desired target output point [9, 62]. The sound pressure is calculated at the user defined response points using the following equation [10].

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

where $p(\omega)$ is the sound pressure level at the driver's ear position, $v_n(\omega)$ represents the nodal velocities of the panels surrounding the cavity, and $ATV(\omega)$ shows the acoustic transfer vector between the input (i.e. point velocities) and the output (i.e. the driver or passenger 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 are used for panel contribution analysis, only taking into account the normal velocity boundary conditions on the specified radiating surface. The partial contribution of each panel to the total sound pressure level is written as follows;

$$p_c(\omega) = \sum_{e=1}^{N_p} \{ATV^e(\omega)\}^T \{v_n^e(\omega)\} \quad (2)$$

where the superscript c denotes the contribution of an element (i.e. panel). This way, the contribution of groups of elements, corresponding to distinct panels of the structure is calculated. This process is called Panel Contribution Analysis (PACA), and is used to identify the contribution of the panels to the sound pressure level. ATVs in Equations [4]

and [5] are dependent only on the geometry characteristics of the acoustic domain, the frequency, and location of the target output points. They are independent of the loading conditions, implying that they are especially well-suited for design studies that require experimentation with multiple configurations [63]. Figure 4.4 shows the input-output locations of the ATVs.

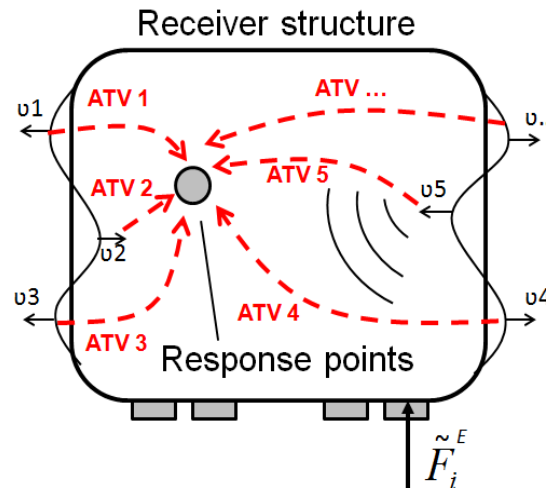


Figure 4.4: Schematic overview of the point velocities and point ATVs of the panel surfaces involved in the SPL calculations caused by cabin mount forces. Adapted from [64]

The classical FE and BE 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, ATV approach becomes more efficient in acoustic problems. In this study, LMS Virtual Lab was used for the ATV Response Analysis tool to calculate the sound pressure level as a function of frequency. First the acoustic transfer function is obtained from the radiating surface to the specific output field point for the predetermined frequency range. Then, structural velocities (indirectly loading) are introduced to the model as the boundary conditions.

4.5 Operational Panel Acoustic Contribution Results

After implementing the required steps in PACA process, the last step is to calculate the operational SPL and corresponding panel contributions. For this purpose, the experimentally calculated operational loads at the cabin mounts coming from the road conditions are required. The operational forces are imported to LMS Virtual Lab where those forces are applied to the cabin mounts of the FE model. The noise levels at the driver and passenger ears inside the cabin are calculated in terms of the partial panel acoustic contributions. To perform PACA, acoustic transfer vectors (ATVs) are calculated. ATVs are the linear operators which give the transfer ratio between the velocity of the vibrating surface and the sound pressure level at the microphone location. For the simplicity of the analysis, the cabin panels are separated into 6 groups as described in previous chapters; (left, right, front, top, back, floor).

Figure 4.5 shows the sound pressure level occurred at the passenger ear under the excitations exerted from the road conditions with a high gear at wide open throttle (WOT) operation condition. The trends of microphone measurements and simulation sound pressure levels match very well in the overall frequency range.

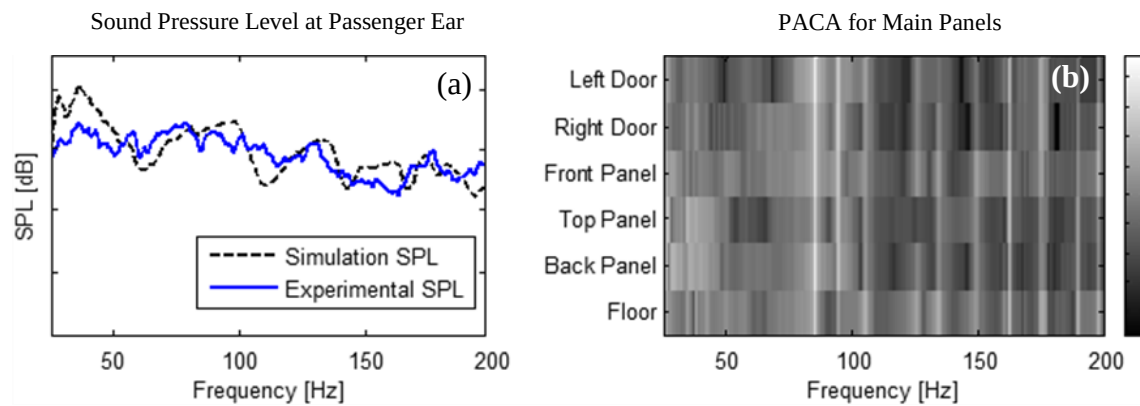


Figure 4.5: (a) Measured noise levels at the passenger ear and FE simulation result: during high gear WOT during road conditions between 20-200 Hz. (b) PACA results of the main panels forming the truck cabin (left, right, front, top, back, and floor)

As illustrated in Figure 4.5, the driver ear SPL has peaky frequency regions around 35-40 Hz and 70-95 Hz. In the PACA results, bright colored regions represent high contribution, and dark colored regions represent low contribution to the overall sound pressure level. Since the modal density of the truck cab is very high (around 800 structural modes between 0-200Hz band), it is hard to identify the problematic panel regions just by looking at the structural modes. PACA results in Figure 4.5 shows that most of the frequency bands in bright color, except 35-40 Hz, are dominated by the acoustic resonances of the cavity. The most critical acoustic mode for the high sound pressure levels at the driver's right ear is found to be the first acoustic mode at 83 Hz. The panel contributions at 83 Hz are not distinguishable by naked eye; therefore, a closer investigation of the contribution values is necessary.

4.5.1 Implementation of Sub-Panel Idea for Main Panels

In order to investigate the partial panel contributions in detail, the main cabin panels are separated into smaller sub-panels in the later sections. Depending on the size of the main panel, each panel is also divided into 5-by-5 or 3-by-3 sub-panels. The detailed analysis was performed between 35-95 Hz frequency band with 5 Hz bands. The size and the orientation of these sub-panels have to be decided carefully by the simulation engineers so that the resultant discretized regions belong to a meaningful region in continuous domain. Figure 4.6 illustrates the sub-panels and the SPL contributions for each frequency band separately.

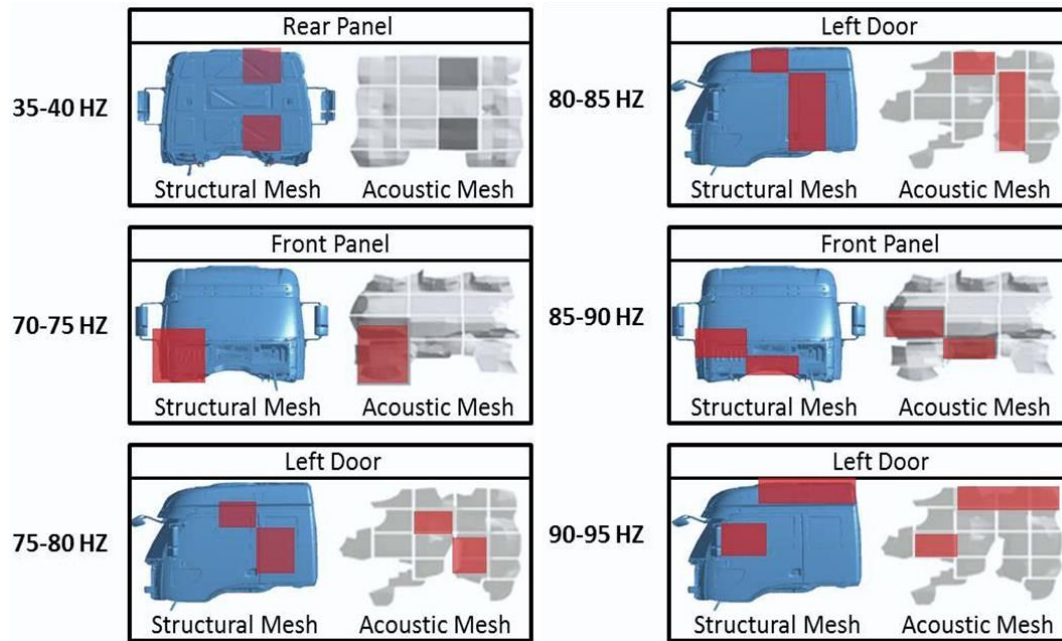


Figure 4.6: Most contributing cabin sub-panels to the SPL between 35-95 Hz frequency band with 5 Hz bands.

In Figure 4.6, the hotspots of the cabin sub-panels can be extracted clearly from the figures. The Figure explicitly shows that sub-panels around the left door have the highest contributions between 75-80 Hz and 80-85 Hz frequency bands that are defined as critical. On the other hand, the most critical contributing panels between 70-75 Hz and 85-90 Hz was identified as the glove box region. The overall stiffness of the glove box helps to transmit the vibration to the cavity; therefore, the booming noise effect can easily be occurred.

The origin of the problem can be identified by investigating the modal characteristics of the cabin structure and acoustic cavity separately. Information on whether the identified problem is structural or acoustic transfer dominated can be reached from the operational PACA study since the panel vibrations and corresponding ATVs of the vibrating points can be investigated separately.

4.6 Design Improvement Studies Based on Vibro-Acoustic Simulations

In the previous section, the vibro-acoustic analysis is performed to predict the SPL and then PACA is performed. After identifying the most problematic panels with PACA, further design studies can be done on the panels to improve the SPL inside the cabin.

By looking at SPL levels in Figure 4.5, the most critical frequency band is determined as 70-90 Hz range which corresponds to 3rd order 1400-1800 rpm engine rotation. For this specific range, the hotspots of the 70-90 Hz range are presented in Figure 4.7 in order to identify the specific sections of the problematic panels.

The hotspots shown in Figure 4.7, which are calculated with PACA, are very stiff compared to the other plate-like components in the cabin (i.e., mirrors, flat plates, etc.). Therefore, in terms of structural vibrations, these regions should not be considered as the root of the vibro-acoustic problem. But the introduced effects of the acoustic cavity resonances make these regions important in terms of noise contributions.

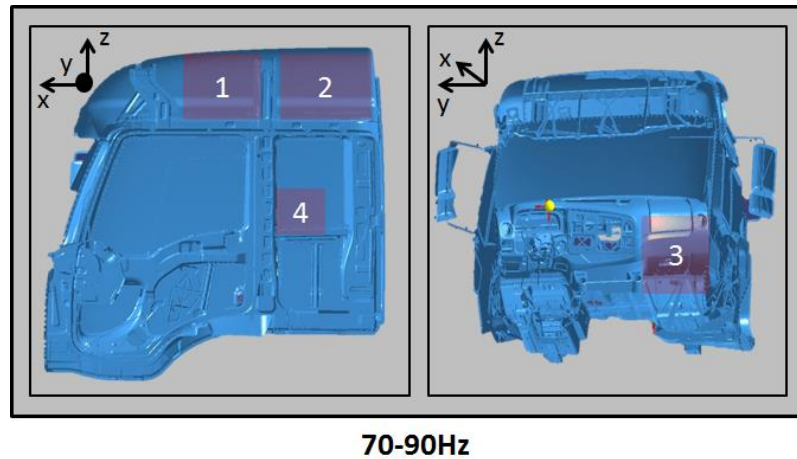


Figure 4.7: Four critical sub-panel locations between 70-90Hz identified by conducting PACA

The root of the ‘booming’ noise problem is the acoustic mode shapes. When we analyze the mode shapes of the air inside the cabin, first mode shape of the cavity looks

more critical than the other modes. 1st acoustic cavity mode is at 83 Hz. This mode shape has also the same critical locations as identified with PACA. Deformation of the air is high around the corner regions. The first acoustic mode shape can be seen in Figure 4.8.

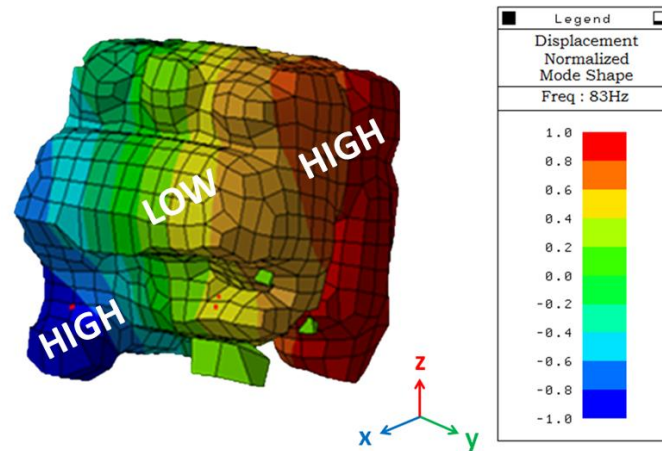


Figure 4.8: Deformation shape of the 1st acoustic cavity resonance at 83 Hz

Therefore, the structural modifications should be decided such that, the contributions from the resonance effects of the cavity to the SPL are minimized. Next section summarizes couple of design studies possible design modifications.

4.6.1 Design Modification of Sub-Panels

The sound pressure level inside the truck cabin was expected to be reduced through the help of the structural and acoustical modifications that was explained in this chapter. These modifications were chosen to be simple methods in order to understand the benefits of the PACA study. The empiric modifications are listed as follows;

- Making the young modulus of problematic panels thousand times stiffer
- problematic regions more stiff than default case by defining new material,
- Stiffening the problematic sub-panel region by using rigid body elements such as RBE2,

In order to illustrate the sensitivity of SPL and its dependence on the panel modification in a wide aspect, each modification region was shown on the detailed simulation model as illustrated in Figure 4.9 and 4.10.

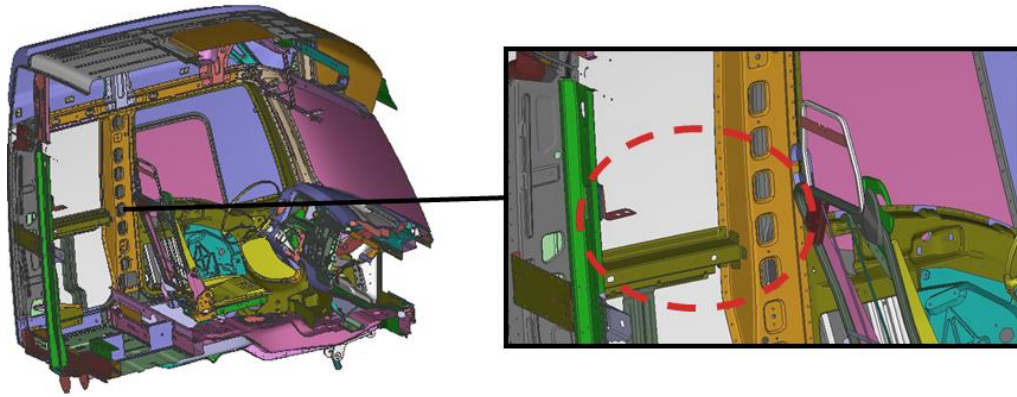


Figure 4.9: Critical region identified by PACA located behind the left door (B-collar) later will be modified via passive stiffening methods.

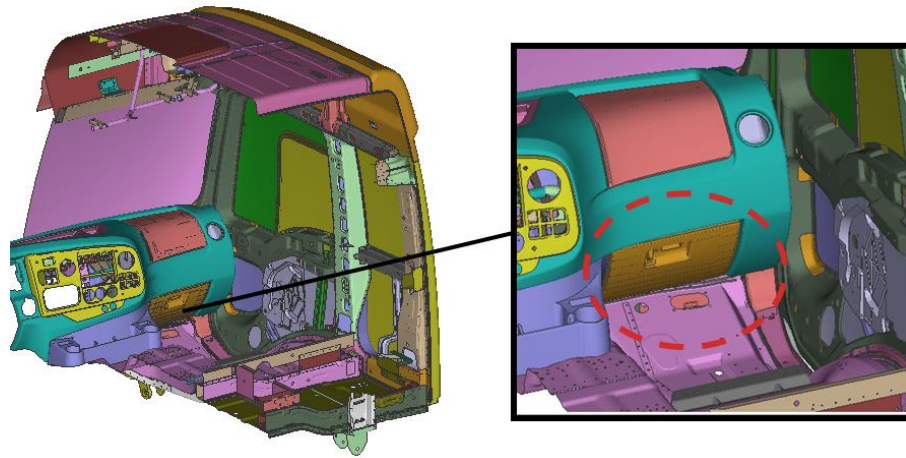


Figure 4.10: Critical region identified by PACA under the glove box located around front panel which will be modified via passive stiffening methods.

Finally, the SPLs of the corresponding design modifications inside the truck cabin are calculated and compared for each of the above mentioned cases. This yields to 7 design

configurations. All the curves are added into Figure 4.11 to show the overall influence of the studied variations and to show the robustness of the achieved results. The solid blue line in Figure 4.11 represents the default SPL and the others are for the modified versions.

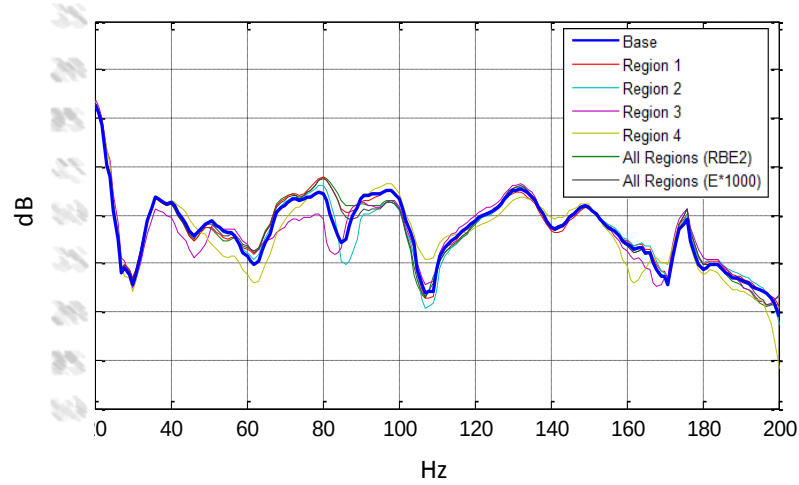


Figure 4.11: Vibro-Acoustic SPLs of the different structural modifications of problematic cabin sun-panels based on the PACA results. Blue line represents the default SPL. Other colors are for different sets of modifications.

Figure 4.12 illustrates robustness of the modifications between 60-110 Hz frequency band in detail for the overall influence of the studied variations.

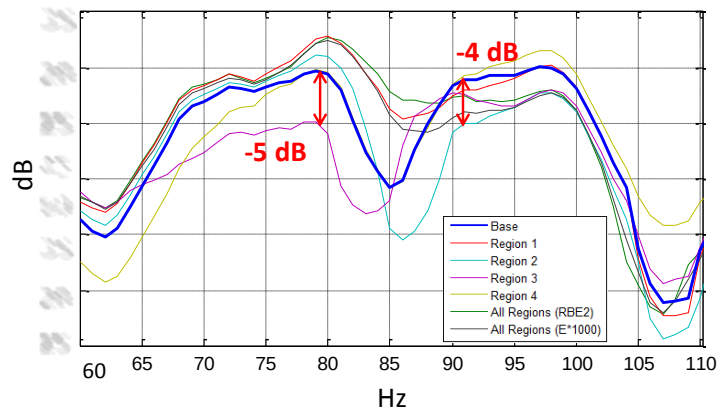


Figure 4.12: A closer look to comparison of SPL between 60-110 Hz band after the design modifications

It is clear that the effect of small changes of the sub-panels is very significant for the critical frequencies. The reduction of response level compared to the original configurations is on the orders of ($\sim 4\text{-}5\text{dB}$) magnitude. However, all the amplitudes between these frequencies could be reduced. There are some amplifications at certain frequencies but they are not at the highest amplitude regions.

At the end of this study, the topic becomes a multi-disciplinary optimization problem that can be solved by optimization techniques which is out of the scope of this paper. Within the scope of this study, the region #3, which corresponds to the outer surface of glove box, is selected among 7 structural modifications that give the minimum response levels for desired frequency bands.

CHAPTER 5

A NOVEL APPROACH FOR FRF-BASED SUBSTRUCTURING

5.1 Introduction

In this chapter, the classical FBS theory will be explained and verified on a case study which includes a truck chassis and a cabin. However, when using FBS method, it is important to take into account the error due to modal truncation in the Kernel matrix. This chapter focuses on developing new techniques for lowering the condition number of the Kernel matrix by selective omission of the cross-coupling terms such that the quality of the matrix inversion procedure is improved. The new novel approach deals with the problem from a different perspective compared to other regularization methods. The effectiveness of the proposed method reducing the influence of pre-existing errors in the Kernel matrix is investigated theoretically and computationally. In addition to comparing the predicted results with the measured values, the predicted results will be also compared with those obtained through the Moore-Penrose pseudo-inverse technique with singular value rejection threshold to verify the validity of the selective omission technique.

Finally, in order to identify the potential of vibro-acoustic characteristics optimization of the rubber bushings connecting the truck chassis and cabin, a sensitivity analysis is performed where the mounting stiffness of the bushings are altered within the design constraints. When searching for the best bushing configuration among all combinations of parameter study, it is shown that the regularization method plays a significant role on the outputs of the sensitivity analysis. So, it is concluded that this effect must be also considered during the design process of the system components.

5.2 Basic Theory

The FBS technique predicts the dynamic response of a coupled system on the idea of free-interface FRFs of the uncoupled substructures and, optionally, with corresponding coupling joint stiffness's. For every substructure the available degrees of freedom (DOFs) are named into two sets, they are (1) the coupling DOFs and (2) the internal substructure DOFs. These internal DOFs correspond with the input and output DOFs of the substructures are illustrated in Figure 5.1. "S" superscript is used to represent system consist of substructure A rigidly coupled to substructure B at the connection DOFs "c"

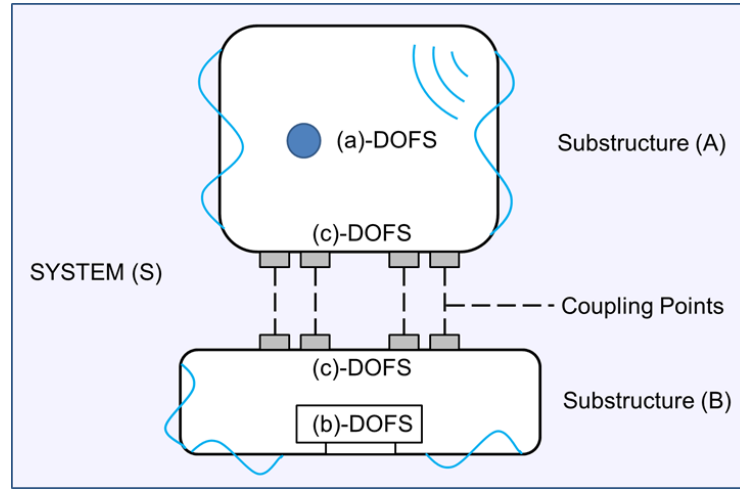


Figure 5.1: System (S), rigidly combining the two uncoupled substructures A and B to each other. The number of "c" coordinates is the same on substructure (A) and (B). The figure is adapted from [65].

The equation of motion for each uncoupled free-free substructures A and B is then [35],

$$\begin{Bmatrix} x_a^A \\ x_c^A \end{Bmatrix} = \begin{bmatrix} H_{aa}^A & H_{ac}^A \\ H_{ca}^A & H_{cc}^A \end{bmatrix} \begin{Bmatrix} F_a^A \\ F_c^A \end{Bmatrix} \quad (6)$$

$$\begin{Bmatrix} x_b^B \\ x_c^B \end{Bmatrix} = \begin{bmatrix} H_{bb}^B & H_{bc}^B \\ H_{cb}^B & H_{cc}^B \end{bmatrix} \begin{Bmatrix} F_b^B \\ F_c^B \end{Bmatrix} \quad (7)$$

H_{vw}^U is the FRF matrix of substructure U between the DOF groups with subscripts V and W,

a , the subscript for the internal DOFs of A,

b , the subscript for the internal DOFs of B,

c , the subscript for the coupling DOFs of A and B,

$\{x_v^U\}$, the V -set DOFs of substructure U,

$\{F_v^U\}$, the forces on the V -set DOFs of substructure U.

The main formula for the FRFs of a rigidly coupled two substructure system S, assembled by the substructures A and B as derived in [35], is then

$$\begin{bmatrix} H_{ab}^S \end{bmatrix} = \begin{bmatrix} H_{ac}^A \end{bmatrix} \begin{bmatrix} H_{cc}^A + H_{cc}^B \end{bmatrix}^{-1} \begin{bmatrix} H_{cb}^B \end{bmatrix} \quad (8)$$

This formula shows that only one inversion has to be calculated, namely the inversion of the sum of the driving point FRF matrices of both substructures which is also called Kernel matrix. The accuracy of the coupled response predictions is highly dependent on this inversion. To improve this inversion numerically singular value decomposition (SVD) can be used [46]. The condition number of this SVD is a measure for the quality of the inversion. The condition number is calculated as the ratio between the largest and the smallest non-null singular values of the FRF matrices.

5.2.1 Procedure for the Response Prediction

The procedure for response prediction consists of three major steps, (1) calculation of the required FRFs, (2) coupling of the uncoupled FRFs and (3) prediction of the response by the multiplication of FBS functions with the operational forces.

5.2.2 Calculation of the FRFs

The first step in the FBS procedure is the calculation of the required FRFs. The FRFs have to be determined for the desired input and output points of the substructures and driving point FRFs for the coupling points. Moreover the driving point FRFs in the coupling points are essential for the FBS procedure. The ways to determine the FRFs are discussed [33]. The available methods are (1) direct calculation of FRFs using FE data, (2) FRF synthesis using natural frequencies of FE data, (3) experimental measurements of FRFs from test structure.

If the substructure FRFs are experimental, the effects of higher modes already exist in the measured FRFs. This is a significant advantage of the experimental FBS compared to other available methods. If, however, the substructure FRFs are from finite element (FE) analysis, they may suffer from the effects of truncation since a limited number of low frequency modes are calculated and synthesized to generate the FRFs. Increasing the number of modes does improve the FRF accuracy but may never know how many are enough. Direct calculation of FRFs using FE model can be adopted to obtain the “correct” FRF data if the size of FE model is small. However, for many engineering applications, the FE models are too large for this approach to be practically applicable.

5.2.3 Coupling of the FRFs

In the case of a two substructure coupling via flexible joints, the Kernel matrix $[H_{A_{ss}} + H_{B_{ss}}]^{-1}$ in equation (7) has to be replaced $[H_{A_{ss}} + H_{B_{ss}} + [K_S]^{-1}]^{-1}$, with $[K_S]^{-1}$ being the matrix that represents the connection stiffness between the coupling DOFs of both substructures. Note that $[K_S]$ has frequency-dependent elements which required additional experimental measurements using the connection. The frequency range and resolution of the substructure FRFs determine the frequency range and resolution of the coupled FRFs.

5.2.4 Response Prediction

The superposition principle states that the total response at a receiver can be expressed as the sum of its individual path contributions. The response R_i^S (vibration or SPL) in global A due to operational forces, $F_j(\omega)$, at input location B can be written as the multiplication of the vibro-acoustic transfer function vector $[H_{ab}^S]$, with the operational forces. Then, total response (vibration or sound pressure level) at a specified location is obtained as Eq.9:

$$R_i^S(\omega) = [H_{ac}^A] [H_{cc}^A + H_{cc}^B]^{-1} [H_{cb}^B] F_j(\omega) \quad (9)$$

The vibro-acoustic transfer functions, $[H_{ab}^S]$, which are calculated by Eq.8, may be obtained from experimental model on the individual subsystems with free-free conditions at the coupling points, or can also come from finite element (FE) analysis, which allows a hybrid prediction of the response at the receiver.

In the following sections, the response predictions obtained through Eq. 9 will be compared with the measured response to demonstrate the validity of the developed approach.

5.2.5 Moore-Penrose Pseudo-Inversion with Singular Value Rejection

As it can be observed, the important part of the Eq. 8 is the inversion of the Kernel matrix. Therefore, if this inversion is ill-conditioned, then problems will be encountered during the direct inversion. Thus, alternatively, pseudo-inversion by means of SVD with singular value rejection is the most common method used for the regularization of the Kernel matrix inversion and it can be used to overcome this problem. The calculation of singular value decomposition of Kernel matrix is given in Eq.10 and the pseudo inverse of Kernel matrix with singular value decomposition is given in Eq.10.

$$[H_{A_{ss}} + H_{B_{ss}} + [K_S]^{-1}] = U \cdot S \cdot V^H \quad (10)$$

where S is a diagonal matrix containing the singular values of. In case Kernel matrix is square matrix, U and V are square unitary matrices. H represents the Hermitian transpose of matrix V . The pseudo inverse of Kernel matrix then becomes:

$$[H_{A_{ss}} + H_{B_{ss}} + [K_S]^{-1}]^+ = V \cdot S^{-1} \cdot U^H \quad (11)$$

where “+” indicates the pseudo-inverse. The inverse S^{-1} consists of the same diagonal terms of S^{-1} . Small singular values S have a large effect over the inverse of the Kernel matrix. A certain number of the lower singular values can be forced to zero by replacing the S^{-1} terms with 0 in order to avoid the ill-conditioning.

When the basic Kernel matrix is decomposed into 3 sub matrices and reconstructed again after the singular value rejection, all the elements of the Kernel matrix are modified. In other words, all the information regarding the original system's FRFs change. The problem with SVD method is that the optimum regularization parameter depends on both the condition number and the measurement errors, making the selection of this parameter a difficult process. For further information, the reader is referred to Reference [66,67].

5.2.6 The Cross-Coupling Omission Method

Generally, most of the matrix regularization methods, including SVD technique, aim to replace the current problem with a problem whose solution is less sensitive to perturbations [35]. In any regularization scheme, there is a regularization parameter which is a quantity that can be adjusted in order to change the degree of regularization of the solution. For some values of this parameter the solution can be very sensitive to noise in the measured data set [68]. By selective omission of the cross-couplings, the proposed cross-coupling omission method changes the initial matrix slightly and makes it better conditioned and as a result more accurate results are obtained after the inversion process.

For a complex system like the heavy duty truck studied in this study, the ill-conditioning state of the Kernel matrix is tied to the presence of cross-couplings that make the inversion process complex. The newly proposed method of selective cross-coupling omission [59], intends to change the initial Kernel matrix in a different way than the SVD with singular value rejection. It uses the condition number as a running parameter to regularize the initial Kernel matrix by selective omission of the cross-couplings at each frequency and as a result more accurate predictions are obtained after the inversion process. It will be shown that with the new approach, a better conditioning in terms of singular values can be obtained. In other words, most of the information regarding the original Kernel matrix FRFs stays unchanged [59].

In order to regularize the Kernel matrix, first matrix elements that belongs to the low contributing paths are identified and then selective omission of the cross coupling terms are performed. The algorithm can be described as follows:

- 1) Compute the condition number of Kernel matrix
- 2) Replace $\text{Kernel}_{ij} = 0$ and $\text{Kernel}_{ji} = 0$; where $i \neq j$ and $i, j = 1, 2, \dots, n$ – for the pair i, j where the reduction in condition number would be maximal;
- 3) Repeat step 2) until a desired condition number of the matrix is reached or further decrease in condition number is impossible

For a better understanding of the algorithm, the flow diagram of a sequential identification and omission of the cross-couplings in the Kernel matrix is described in Figure 5.2:

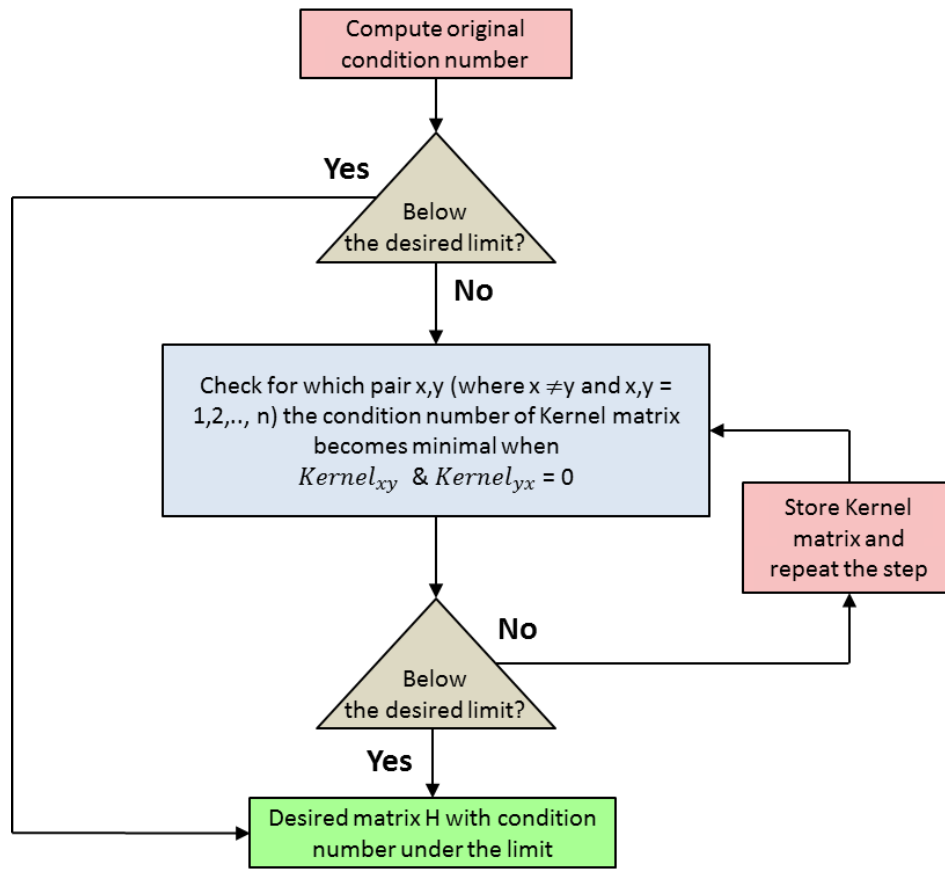


Figure 5.2: The new method flow diagram: Selective cross-coupling omission algorithm procedure to reduce the condition number of the Kernel matrix.

In practice, one must be careful when targeting a very low threshold (e.g. <10) for the condition number since the algorithm may remove too many of the cross-couplings such that, eventually, lead to an under-prediction of the target response. The amount of change in condition number for each removed cross-coupling will be different for every system because high condition number of the Kernel matrix can be due to the sum of two small contributions from substructures A and B, but also due to the sum of two large contributions with opposite sign as stated in [67]. The procedure aims to keep the system of FRFs as much as possible such that the initial problem remains “unchanged”. Reaching a

desired condition number without removing many cross-coupling pairs means that the ill-conditioning problem is solved while keeping the most of the information provided by the system [59]. The implementation of the methodology on a heavy duty truck model is presented in the following sections.

5.3 Case Study: Coupling of the Truck Chassis with the Cabin

In this section, the proposed methodology is demonstrated with a case study which involves the main substructures of a heavy duty truck. The truck cabin is represented by an experimentally validated numerical model whereas the chassis FRFs are measured experimentally. After FBS is performed using the provided FRFs, the results are compared with different regularization methods to validate the accuracy of the proposed method.

5.3.1 Description of the Coupled Model

The assembled system, cabin and chassis, is connected to each other with different front and rear rubber bushings. The substructures are connected from 4 different locations. The front and rear sides have different geometrical features that distinguish the dynamic stiffness properties. The axial direction of front bushings is aligned with the y-direction in the global coordinate system. Differently, the axial direction of rear bushings is aligned with the x-direction in the global coordinate system. A schematic representation of the hybrid system assembly of the substructures can be seen in Figure 5.3.

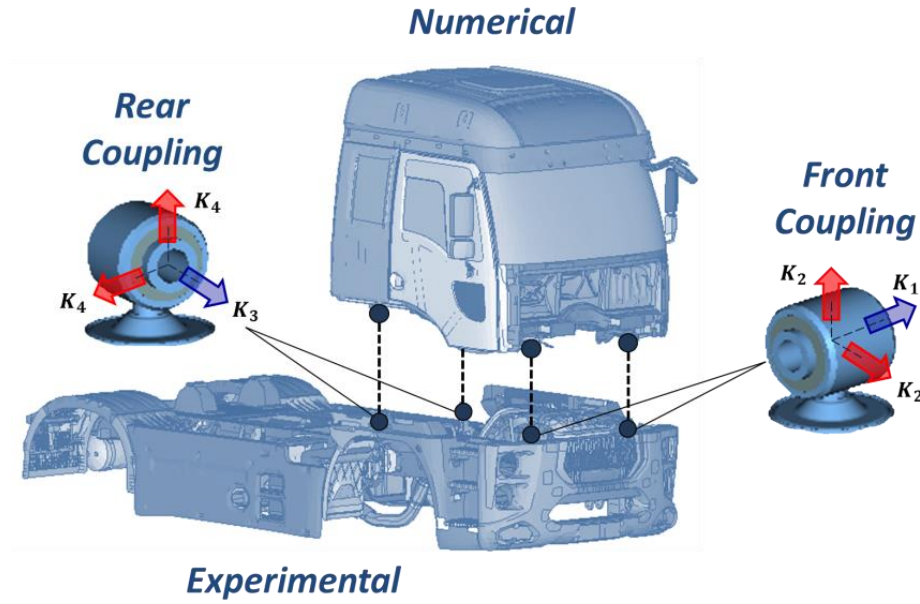


Figure 5.3. 3D representation of the assembled system, the cabin and chassis, and the geometry of the front and rear coupling bushing consisting of a rubber tube with shore hardness 60 between inner and outer sleeves (K1 represents the front side radial and K2 front axial. K3 represents the rear side radial and K4 rear axial).

The magnitudes of the dynamic stiffness of each rubber bushings are obtained through testing results performed by the supplier as illustrated in Figure 5.3. It should be noted that there is an order of magnitude difference in the dynamic stiffness amplitudes of radial and axial directions. Later, the obtained dynamic stiffness information of the bushings was implemented by the FBS coupling equation through Kernel matrix in Eq.(8).

There are two types of bushing used for front and rear side of the cabin. The orientation of the axial and radial stiffness's of the front and rear bushings and the arrangement of the diagonal stiffness matrix is described in Figure 5.4. The Kernel matrix has the following format; $[H_{A_{ss}} + H_{B_{ss}} + [K_S]^{-1}]^{-1}$.

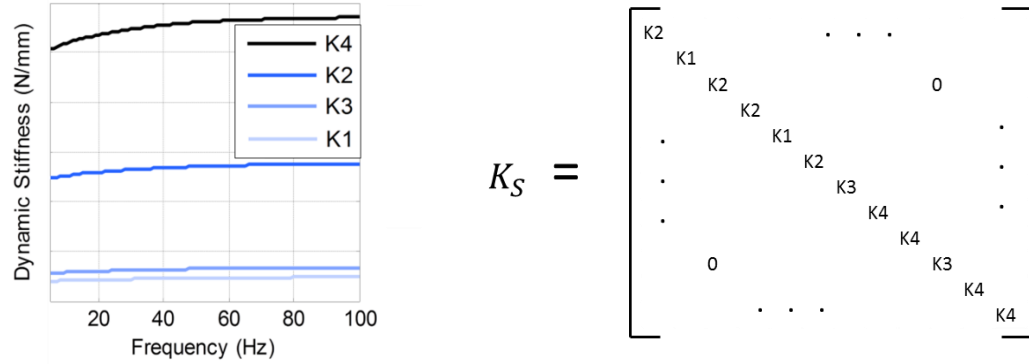


Figure 5.4. The dynamic stiffness components of the rubber bushing structures obtained from the supplier (K1 represents the front side radial and K2 axial. K3 represents the rear side radial and K4 axial). The y-axis of the figure is not displayed due to non-disclosure agreement between Koç University and Ford Company.

The actual formulation of K_S often depends on the type of connections used in the system [48]. In Figure 5.5, a schematic representation of the FRF measurements on the free-free substructures along with the diagonal coupling stiffness matrix, K_S , is depicted. The figure only shows one input-one output FRF of the coupled system. Note that, in the total system of equations, there are 4 input points on the chassis, 1 structural response and 1 acoustic response points on the cabin side.

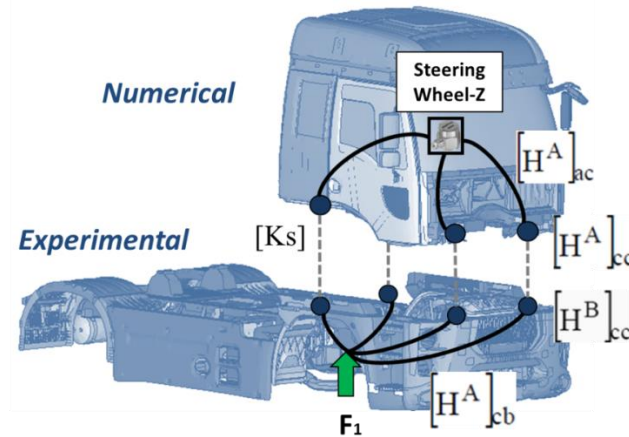


Figure 5.5. 3D representation of the substructures and the coupled system, where the cabin and chassis are connected through flexible bushings. Each of the FRF matrices which are used in the FBS formulation is depicted inside the figure.

5.3.2 Comparison of Predicted Response for Different Inversion Methods

In the literature, there is not any determined threshold values for considering the matrix well or ill-conditioned. Many researchers determine their own thresholds according to their experience on the system. The condition number of the original problem which is a 12x12 Kernel matrix as a function of frequency is shown in Figure 5.6.

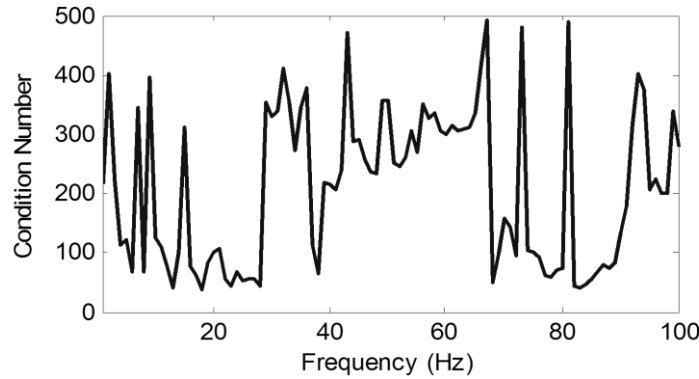


Figure 5.6: Original condition number of the Kernel matrix

In this section, the measured responses are compared with the response predictions (Refer to Eq. 9) obtained with the three inversion methods, direct-inversion, pseudo-inverse with singular value rejection and the cross-coupling omission method, by considering different threshold values. The effects of singular value rejection thresholds for 2%, 3% and 5% of the highest singular value are investigated in order to obtain the maximum 50, 30 and 20 condition numbers over the 0-100 Hz frequency band. Left column of Figure 6.7 depicts the vibration response at the steering wheel-Z and the right column depicts the sound pressure level at the driver left ear. The resonance predictions are good regarding to the coincidence of natural frequency peaks; however, there is sometimes a difference in amplitude values between the measured and predicted for both responses.

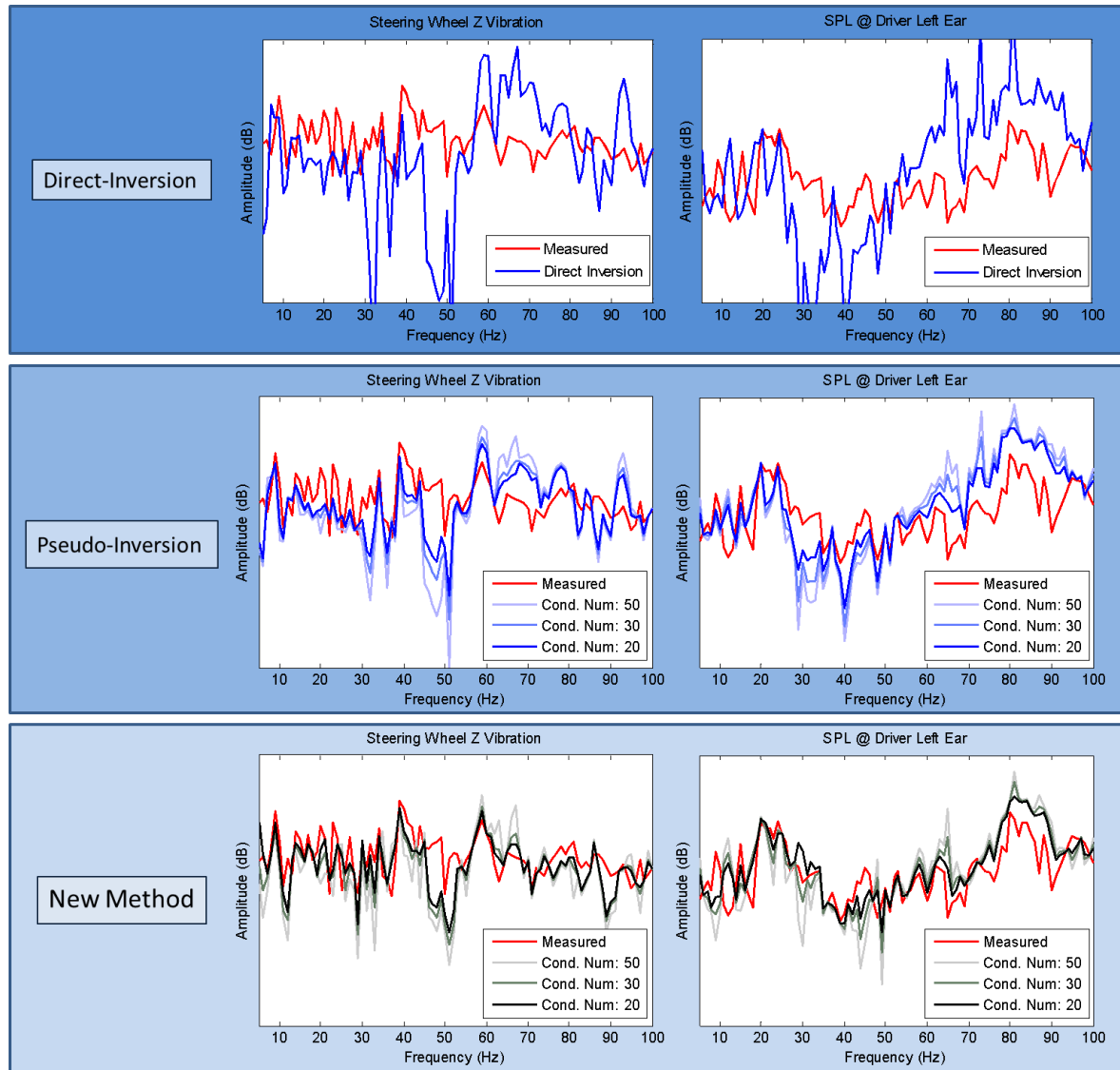


Figure 5.7: Demonstration of the predicted responses: Steering wheel Z vibration (left), (b) Sound pressure level at driver left ear (right). a) Comparison of the results obtained by direct inversion and measured ones, b) Comparison of the results obtained by pseudo-inverse (when condition number is limited to 50, 30, and 20) and measured ones, c) Comparison of the results obtained by the new method (when condition number is limited to 50, 30, and 20 and measured ones).

As it can be observed from Figure 5.7, the match between the measured and predicted results improved significantly as we gradually switch from direct inversion to SVD and then to from SVD to cross-coupling omission method especially at the frequency regions corresponding to higher condition numbers (see Figure 5.6). It can be observed that the regions where the condition number of the initial Kernel matrix was lower than the condition number limit remained unchanged for both regularization methods. The steering wheel vibration response results at 60-70 Hz and the sound pressure level results at 20-30 Hz significantly improved with the proposed method. Although the predictions look very similar when both methods are compared in the other frequency regions, a gradual increase in the prediction accuracy is observed when the condition number is decreased from 50 down to 20. As discussed before, some unmatched prediction discrepancies are due to the inconsistencies of measured data (disturbance, frequency shifts), the inability to measure correctly all DOFs, ill-conditioned inversion of matrices and wrong force estimations. Since the forces, $F_j(\omega)$ (Refer to Eq. 9.), are calculated by matrix inversion procedure [25], it is highly probable that they introduce error to the predictions.

In order to quantify the prediction accuracy of both methods in specific frequency bands, an RMS metric is adapted from Stan et al [59]. The frequency band of 0-100 Hz is divided into 10 equal sub-intervals (every 10 Hz). For each of the subintervals, the ratio between the root mean square (RMS) values of predicted and measured in Figure 5.7 is calculated according to Eq. 12:

$$\frac{\text{RMS}_{\text{pred}}}{\text{RMS}_{\text{meas}}} = \frac{\sqrt{\frac{1}{\omega_{\text{max}} - \omega_{\text{min}}} \int_{\omega_{\text{min}}}^{\omega_{\text{max}}} |\text{R}_{\text{pred}}(\omega)|^2 d\omega}}{\sqrt{\frac{1}{\omega_{\text{max}} - \omega_{\text{min}}} \int_{\omega_{\text{min}}}^{\omega_{\text{max}}} |\text{R}_{\text{meas}}(\omega)|^2 d\omega}} \quad (12)$$

The results are tabulated in Tables 1 in order to compare both methods. Results in the Table 1a and 1b show the ratios between the predicted and measured responses of the

steering wheel acceleration in z direction and SPL at the driver's ear using the pseudo inverse and direct inversion methods, respectively. The first column gives the information regarding the condition number limit and the type of method used. The other columns show the ratio between the actual and predicted root-mean square (RMS) values for each frequency band obtained with the pseudo-inverse and newly-proposed method for the 50, 30 and 20 condition number limits. As it could be also observed from Figure 5.7, the cross-coupling omission method returns better predictions when the condition number limit is lowered. This trend can be easily observed in Table 1. The highlighted regions in Tables 1 show which method had the best prediction in the specified interval (the ratio closest to 1). As it can be observed from the Table, in the majority of the results, the new method gives a better match with the measured response. For the remaining regions, it was not that different from the results obtained by the SVD approach.

(a) Steering Wheel Z Vibration										
Frequency Condition Number	1-10 Hz	10-20 Hz	20-30 Hz	30-40 Hz	40-50 Hz	50-60 Hz	60-70 Hz	70-80 Hz	80-90 Hz	90-100 Hz
CN50-Pinv	0.87	0.91	0.90	0.84	0.73	0.99	1.18	1.12	0.92	1.08
CN50-New	0.87	0.92	0.92	0.95	0.82	0.93	1.09	0.97	0.89	0.97
CN30-Pinv	0.90	0.94	0.93	0.88	0.80	0.98	1.13	1.10	0.93	1.06
CN30-New	0.92	0.93	0.96	0.97	0.86	0.94	1.05	0.97	0.92	0.98
CN20-Pinv	0.93	0.95	0.94	0.90	0.84	0.98	1.10	1.09	0.93	1.05
CN20-New	0.97	0.95	0.98	0.98	0.88	0.95	1.03	0.97	0.93	0.98

(b) SPL @ Driver Left Ear										
Frequency Condition Number	1-10 Hz	10-20 Hz	20-30 Hz	30-40 Hz	40-50 Hz	50-60 Hz	60-70 Hz	70-80 Hz	80-90 Hz	90-100 Hz
CN50-Pinv	0.95	1.07	0.83	0.76	0.84	1.08	1.25	1.22	1.21	1.10
CN50-New	0.88	1.07	0.91	0.94	0.85	1.05	1.15	1.09	1.17	1.01
CN30-Pinv	0.96	1.06	0.85	0.80	0.87	1.06	1.18	1.21	1.19	1.08
CN30-New	0.91	1.06	0.96	0.97	0.90	1.03	1.11	1.08	1.15	1.01
CN20-Pinv	0.97	1.05	0.88	0.86	0.91	1.04	1.14	1.17	1.17	1.05
CN20-New	0.93	1.06	1.02	0.99	0.94	1.02	1.06	1.08	1.13	1.00

Table 1: The ratio between the predicted and measured RMS using Eq.12 when predicting (a) the steering wheel-Z vibration and (b) sound pressure level at driver left ear with the SVD and new method for each 10 Hz bands.

In this section, two different criteria that are directly connected to each other are investigated: the number of removed cross-couplings pairs and the condition number reached. The Figure 5.8 is included such that one can have an idea of how many cross-

couplings pairs are removed to reach a certain condition number with the new method. It can be observed that the number of removed cross couplings terms does not exceed 40 even when the condition number is decreased from 50 to 20. Note that on the right hand side of Figure 5.8, both approaches yield a different condition number for each frequency of interest.

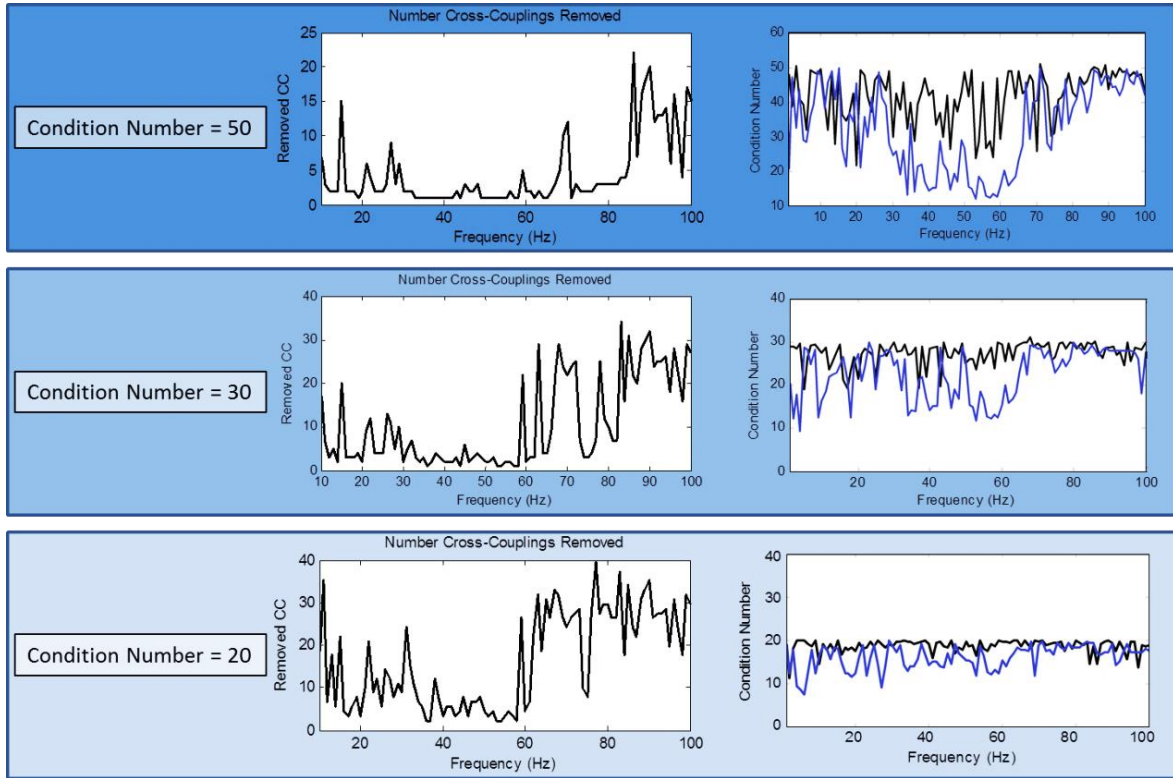


Figure 5.8: (Left) the number of cross-couplings removed to reach the desired condition number. (Right) the corresponding condition number obtained with the pseudo-inverse technique with singular value rejection threshold (blue) and also with the cross-coupling omission method (black) when condition number is limited to 50, 30, and 20.

5.4 Sensitivity of the Predicted Response to Connection Stiffness

In this section, a sensitivity analysis of joint stiffnesses connecting the substructures of the truck is performed to determine their effect on vibration and acoustic performance of the structure. In the literature, many researchers [50-55] studied the effect of dynamic stiffness of coupling mounts by parametric studies on substructure levels. The dynamic stiffness of the rubber mounts at lower frequencies is smaller than its stiffness at higher frequencies. Although they show good performance at low frequencies, they provide relatively poor isolation at higher frequencies [48]. The newly proposed method is used to investigate the effect of bushing stiffness on the steering wheel vibration and SPL level performance of the heavy duty truck. The sensitivity analysis identifies the most critical design parameters to improve the performance. Determining sensitivity of design parameters can provide useful information when the system does not meet the specified requirements. For systems with many design parameters, sensitivity information can identify which parameters in the system are the most significant. These parameters might be the focus of redesign efforts that attempt to improve the performance. In order to compute the sensitivities the following expression must be evaluated:

5.4.1 Sensitivity Analysis of Bushing Stiffness

The first step of the sensitivity analysis is identifying the critical frequencies of interest where the highest peaks occurs in the steering wheel vibration and SPL response. Looking at the results in Figure 5.7, two frequency bands (56-63 Hz for steering wheel, 77-83 Hz for the SPL) are selected as the most critical frequency bands that need to be improved. The design parameters are selected as the dynamic stiffness of the cabin connection bushings which have three orthogonal components, 1-axial and 2-radial directions. The stiffness magnitudes of the radial and axial direction of the bushings are considered independent on each other. Changing the connection location of the mounts is not considered during this study, since it would require the re-measurement of all the FRFs.

The default values of the initial stiffness values (K_1, K_2, K_3, K_4) (Refer to Figure 5.4) were altered simultaneously between $\pm 20\%$ from the initial values with 10% increments for frequency bands. This yields to 625 possible combinations and the range of dynamic stiffness for the sensitivity analysis is shown in Figure 5.9.

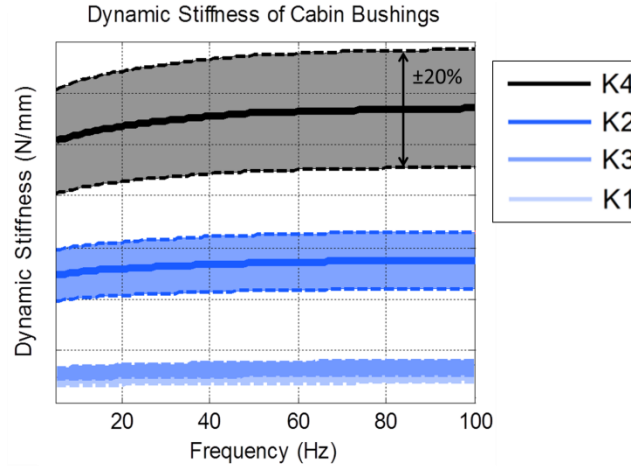


Figure 5.9. The selected range of dynamic stiffness ($\pm 20\%$ for each magnitude) for the sensitivity analysis of the cabin mounts. The y-axis of the figure is not displayed due to non-disclosure agreement between Koç University and Ford Otosan.

All of the corresponding response curves are added into Figure 5.10 to show the overall influence of the studied variations. The solid black lines in Figure 5.10 are included to visualize the original configurations of the bushings. Note that, in all of the calculation, the condition number of the Kernel matrix is limited to 20 with the new method.

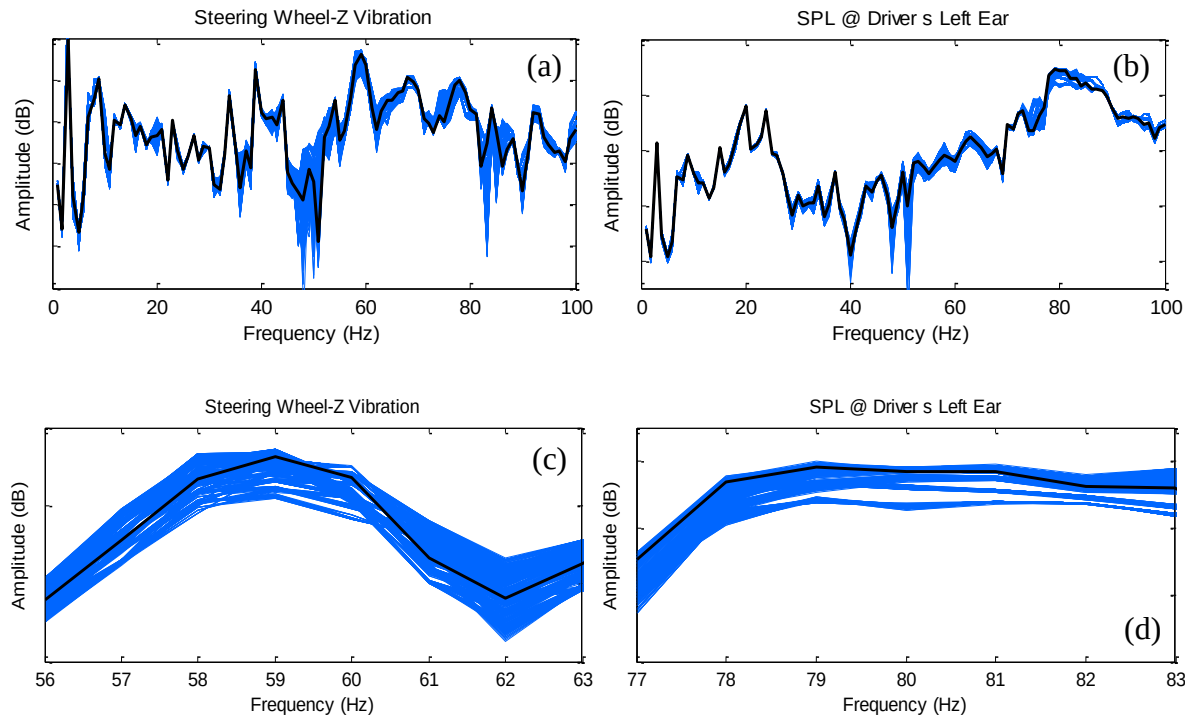


Figure 5.10. (a) Steering wheel–Z vibration with all different possible bushing parameters, (c) most critical frequencies 56–63 Hz. (b) SPL at driver’s left ear with all different possible bushing parameters, (d) most critical frequencies 77–83 Hz. Original (black), with all possible configurations of bushings (blue)

It is clear that the effect of slight changes in the bushing stiffness is very significant for the critical frequencies (56–63 Hz for the steering wheel, 77–83 Hz for the SPL). For the steering wheel vibration response, a reduction of 5 dB can be achieved in the 56–63 Hz range at the highest peak by altering the stiffness values. Also it can be observed in Figure 5.10(b) that, significant improvements can be achieved in the frequency band of interest for the SPL response when the bushing stiffnesses are altered.

Among 625 possible configurations, the configuration of the bushings that give the minimum response levels for desired frequency bands are selected for the steering wheel vibration and SPL responses. In Figure 5.11(a), the response levels that have the minimum

average amplitudes between 56-63 Hz the steering wheel vibration is given with red line. In Figure 5.11(b), the same approach is applied between 77-83 Hz for steering wheel vibration is given with solid red line.

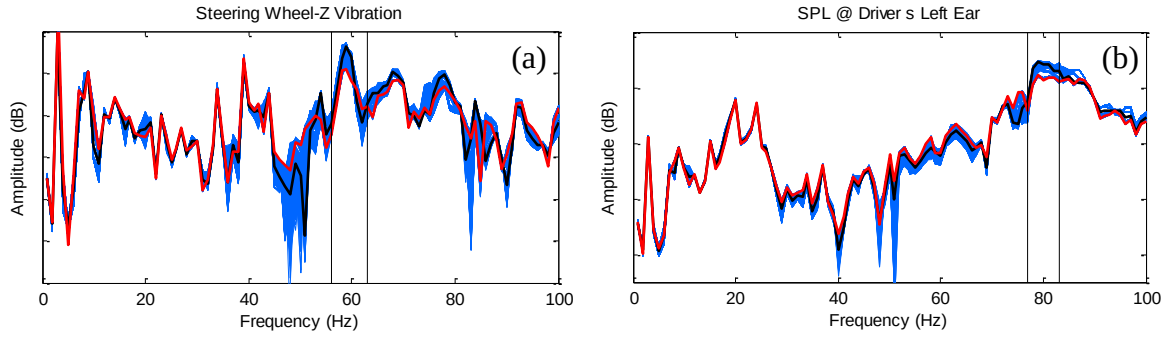


Figure 5.11. (a) Steering wheel-Z vibration with different bushing parameters targeted for 56-63 Hz (b) SPL at driver's left ear with different bushing parameters targeted for 77-83 Hz. Original (black), with 625 bushing configurations (blue), minimum response average between target frequency bands (red).

The potential of reduction in the vibration or acoustic response levels is dependent on the frequency of interest. Each targeted frequency band leads to a different bushing configuration and response level characteristics. However, there is always a trade-off between the original and optimum design for certain frequencies. Depending on the system characteristics, each different bushing combination gives lower or higher responses for certain frequencies. At the end, the topic becomes a multi-disciplinary optimization problem that can be solved by optimization techniques which is out of the scope of this paper.

Table 2 shows the bushing stiffness configuration that gives the minimum vibration response for steering wheel-Z between 56-63 Hz, as illustrated in Figure 5.11(a). Table 3 shows the bushing stiffness configuration that gives the minimum for booming SPL at driver's left ear between 77-83 Hz, as illustrated in Figure 5.11(b).

Table 2. The optimal configuration (380th) of bushing stiffness for steering wheel-Z between 56-63 Hz

Configuration	K1	K2	K3	K4	Change in Vibration	Change in SPL
<u>380th / 625</u>	+10%	-20%	-20%	+20%	- 4.9 dB	- 0.7 dB

Table 3. The optimal configuration (422th) of bushing stiffness for booming SPL ear between 77-83 Hz

Configuration	K1	K2	K3	K4	Change in Vibration	Change in SPL
<u>422th / 625</u>	+10%	-10%	+20%	-10%	- 1.3 dB	- 4.2 dB

The potential for reducing the transmitted vibro-acoustic energy of the cabin bushings is highly dependent on the type of excitation direction, frequency and interested response point. Table 2 and 3 clearly shows that, for the two target responses, the minimum level of response of the targeted frequencies can be obtained by having the same amount of change in the axial stiffness of front bushing, but different for the other directions of the bushings.

5.4.2 Effect of Kernel Matrix Regularization on Response Optimalization

Previous section illustrates the different potentials of each combination of the bushing for reducing the vibro-acoustic response at different responses. However, it is found out that the type of regularization method used for Kernel matrix influences the response optimalization result. The best stiffness values with respect to the new inversion method for two target responses were demonstrated in Table 2 and 3. In this section, the FBS analysis is performed by using three previously mentioned regularization methods. The affected selection for best bushing combination performed for two different response points and frequency bands using different regularization techniques is demonstrated in Table 4 and 5. Each obtained number is a representation of the corresponding bushing combination which is between $\pm 20\%$ ranges of initial stiffness values with $\%10$ increments.

Each resultant configuration number points a different bushing stiffness for K1, K2, K3 and K4. The values in the table are shown as percent change with respect to the initial values.

Table 4. The optimal bushing stiffness configuration for steering wheel-Z vibration between 56-63 Hz calculated by three methods.

Steering Wheel Z	Configuration	K1	K2	K3	K4
Direct	<u>415th</u>	+10%	-10%	0	+20%
Pseudo Inversion	<u>424th</u>	+10%	-10%	+20%	+10%
New Method	<u>380th</u>	+10%	-20%	-20%	+20%

Table 5. The optimal bushing stiffness configuration for booming SPL at driver's left ear between 77-83 Hz calculated by three methods.

SPL @ Driver Ear	Configuration	K1	K2	K3	K4
Direct	<u>394th</u>	+10%	-20%	+10%	+10%
Pseudo Inversion	<u>380th</u>	+10%	-20%	-20%	+20%
New Method	<u>422th</u>	+10%	-10%	+20%	-10%

Table 4 and 5 lists the optimum stiffness configurations obtained by the three inversion methods for steering wheel vibration and SPL responses. The optimum stiffness value for K1 remains the same for all the methods. However, K2, K3 and K4 values change depending on the regularization method used in the inversion of the Kernel matrix. These results demonstrate the significance of the inversion process especially when the joint stiffnesses are designed for improving vibration and/or noise performance of the structure.

CHAPTER 6

DISCUSSION & CONCLUSION

In a complex dynamic system such as a heavy duty truck cabin, the vibro-acoustic enhancement and redesign effort may be very difficult and time-consuming. This study focuses on the development of new methods to predict and improve the structural borne noise inside the passenger cabin of a heavy duty truck by incorporating a combined usage of simulation and experimental methodologies. The analysis tools and the methodology presented provide a systematic and quantitative way to investigate the vibro-acoustic performance of such systems and identify the critical components of the system that degrade the performance with vibro-acoustic requirements.

Essentially, five major disciplines have been considered: implementing a vibro-acoustic FE model, conducting experiments for model validation, experimental identification of the cabin and engine mount forces, identifying the most problematic cabin panels, and developing a novel methodology for solving ill-conditioned matrix problems encountered during FRF-based substructuring technique.

In Chapter 1, the vibro-acoustic problem was introduced and a review of the methodologies used throughout the thesis was given. The review topics included vibro-acoustic modelling, force identification and FRF-based substructuring.

In Chapter 2, the details of the vibro-acoustic model were presented. The interactions between the structural model, cavity mesh and surrogate mesh were explained. The vibro-acoustic simulation model was validated by comparing the predicted results with the experimental measurements obtained in a semi-anechoic room. The vibrational and acoustic transfer functions (FRFs and NTFs) were compared with the measured ones in all

principal directions. It was observed that the FRF predictions of the model were matching well with the experimental results in 0-200 Hz frequency band.

In Chapter 3, the operational forces were calculated at the cabin mounts and engine mounts using the matrix inversion method. The cabin mount forces were applied to the cabin model to perform PACA to identify the most contributing panels in FE model in Chapter 4. The engine mount forces were used with the FRF-based substructuring analysis to predict the SPL and steering wheel vibration experiments in Chapter 5.

In Chapter 4, the panel acoustic contribution analysis (PACA) was performed for identifying the highest contributing panels to the SPL inside the cabin. The methodology benefits from ATV approach, which links the input of the structural velocity of the radiating surface and the sound pressure level at the position of the driver's ear. The detailed sub-panels analysis showed that the critical regions on the panels can be easily modified by passive methods to improve the sound pressure level on the order of 4-5 dB in the cabin.

In Chapter 5, a new iterative method was proposed to enhance the ill-conditioning state of the Kernel matrix for FRF based substructuring. The proposed methodology was demonstrated for the coupling of the truck cabin numerical FRFs with the chassis experimentally measured FRFs using the FRF-based substructuring analysis. It was shown that, the newly proposed inverse FBS approach resulted in better response predictions when compared with Pseudo inversion with singular value rejection threshold. In the second part of Chapter 5, sensitivity analysis of the joint stiffnesses connecting the substructures was performed to determine their effect on vibration and acoustic performance of the structure. It was shown that the regularization method played a significant role on the outputs of the sensitivity analysis. So, it was concluded that this effect must be also considered during the design process of the system components.

The preliminary results show that the developed methodologies can provide a lot of insight for the vehicle design engineers to improve the noise level inside the passenger cabin of heavy duty trucks and reduce the cost of vehicle development in the early stage. The developed methodologies are demonstrated for a heavy duty truck case study however they are all applicable to any complex system.

Naturally, like every scientific work, this study raises new questions for further research. The cross-coupling omission method for improving the inversion procedure of Kernel matrix can also contribute to future studies regarding the values of the condition number at which a matrix becomes ill-conditioned. For future studies, new and improved algorithms for selecting the omission of cross-couplings can be developed such that the duration of the inversion process is optimized. Another suggestion is an alternative method by omitting the diagonal terms in the Kernel matrix together with the cross-coupling terms that degrade condition number. Preliminary findings show that the omission of diagonal terms helps to decrease the condition number with less iterations. Finally, the effects of the following topics to the real-life SPL and vibration levels on prototype vehicle should be investigated: (1) the proposed design modifications, (2) the imperfections in the production of modifications, (3) the errors in actual hardware installation of modifications.

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