SZÉCHENYI ISTVÁN UNIVERSITY

DOCTORAL SCHOOL

OF MULTIDISCIPLINARY ENGINEERING SCIENCES

Numerical vehicle acoustic simulation development using SEA-based methods

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Doctoral dissertation

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1. INTRODUCTION

1.1. Motivation

Noise, Vibration and Harshness (NVH) research and development has become increasingly important in the vehicle industry recently. NVH deals with predicting and controlling vibroacoustics phenomenon, which can cause passenger discomfort in the form of vibrations and noise or fatigue of the car components. Therefore, the automotive sector invests lot of effort in predicting and controlling the undesired effects of vibroacoustics phenomenon. There are various trends for vibration- and noise control. One of the simplest methods is to change the mass, stiffness or damping of a component participating in the vibration or noise transfer, since these can alter the response of a system to an excitation. However, by for example adding mass, one contradicts an important aspect of modern vehicle development: the need to reduce the weight of the car due to the tightening emission standards. Another challenge nowadays is related to the NVH of electric vehicles, since the excitation due to an electric motor occurs at much higher frequencies, than for combustion engines. Also, noise sources, which have been "masked" before by the internal combustion engine, might now become important. Such noise sources include the gear noise, road noise or wind induced noise, which itself is characterized by high frequency random excitation due to the turbulent boundary layer around the vehicle. Furthermore, the heavy battery package of electric vehicles calls for the usage of new lightweight materials, such as lightweight alloys or carbon fibre composites. For such complex lightweight structures, one has to analyze the effect of high frequency excitations from structural, fatigue, failure as well as noise point of view. For this reason, electric vehicles require full car simulations, what represent a quite significant computational challenge [1].

From the frequency point of view, it is common in vehicle NVH to distinguish between low-frequency (up to 400 Hz), mid-frequency (400-1000 Hz) and high-frequency (above 1000 Hz) noise. There are quite matured methods established for predicting the low-frequency or high-frequency noise, however, there is no common approach to predict the mid-frequency range [2]. Several researchers attempted to investigate and solve this grey area in the vehicle industry, and a range of new simulation methods were developed, such as wave-based methods, wave finite element methods, energy distribution analysis methods or hybrid Finite Element-Statistical Energy Analysis methods [2].

The present work aims to use hybrid Finite Element-Statistical Energy Analysis (Hybrid FE-SEA) method to investigate the challenges of predicting mid-frequency and high-frequency vibroacoustic phenomena for vehicles. Although the main goal of the research is to enable the virtual development of vehicles, various simplified vehicle-like structures have been investigated, in order to build up a simulation know-how step-by-step from the basics. This Chapter will first describe the peculiarities of vehicle structures, followed by reviewing the state-of the-art of SEA and Hybrid FE-SEA simulations, from which the research gaps in these areas will be identified.

1.2. Specifics of vehicle structures

Vehicle chassis structures must provide comfort and safety to the passengers and must enable the desired handling qualities of the vehicle. Therefore, the vehicle chassis must fulfil multiple criteria from mass, stiffness, impact crash, strength, durability and fatigue, manufacturability as well as cost point of view. One important aspect of passenger comfort is the vibroacoustics characteristic of the vehicle. From this point of view, the goal is to create such vehicle structure, which minimizes the sound pressure level at the ear position of the passengers as well as the vibration levels at the contact points with the passengers, i.e. at the steering wheel, the floor, and the seats. In order to meet these complex challenges, modern chassis design involves the combination of strong frame elements and sheet panels as well as multiple materials, as shown in Fig. 1.

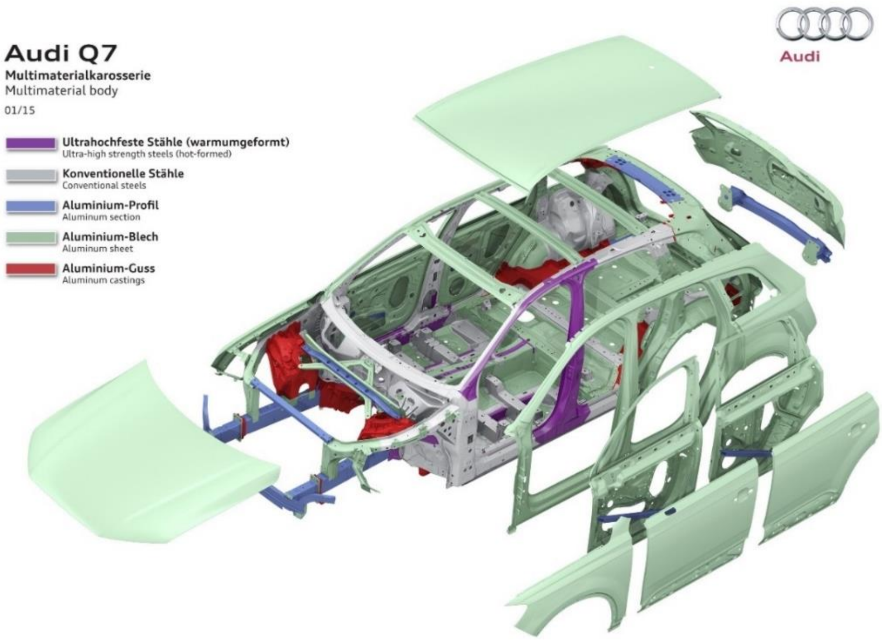

Figure 1: Illustration of the vehicle structure of the frame and panels as well as of the range of materials used in a modern chassis [3].

As a consequence of the material diversity, various binding techniques are utilized during the manufacturing process. As Fig. 2 shows, modern vehicle binding techniques include spot welding, line welding, bolting, riveting, gluing, as well as their combinations. As will be seen later, the modelling techniques for capturing the effect of junctions will appear crucial to obtain credible and accurate vibroacoustics simulation results.

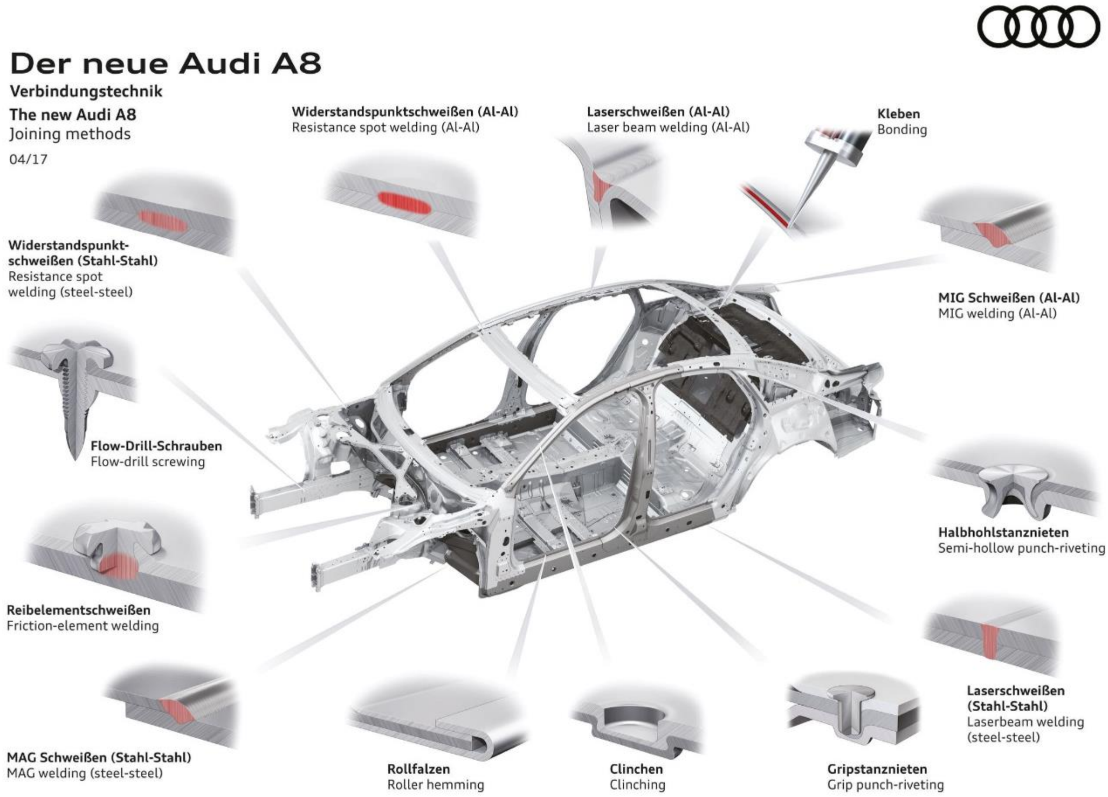

Figure 2: Various binding techniques on a passenger car [4].

In general, full vehicle structures can be subdivided into the following main elements (or “levels”):

- Stiff frames (e.g.: A-pillar, B-pillar, C-pillar, stiffeners, etc.)
- Sheet plates (e.g.: sheet metals and glasses)
- Joints (e.g.: welding, gluing, riveting, bolting, and their combinations)
- Trim materials (e.g.: foams, carpets, dashboard, seats, etc.)
- Air cavities (e.g.: air ducts, air cavity of the passenger compartment)

These elements will be used in this work to determine the various levels of validation models for the simulations. A range of simplified test apparatuses were designed to enable the detailed understanding and validation of the key parameters of simulations. The test equipment was a simplified equivalent of their full vehicle applications.

1.3. Vibroacoustics phenomenon in vehicles

As it was mentioned above, in vehicle NHV development, the main goal is to reduce the SPL level at the ear positions of the passengers as well as the vibration at the contact points between the driver and the vehicle, i.e. the steering wheel, floor, seats, etc. During a vibroacoustics measurement or simulation campaign of a vehicle, the response points (where noise or vibration data are evaluated) are therefore defined by these positions and target criteria are typically prescribed at these response points. Note though, that for methodology development and/or validation purposes, other response point locations can be considered as well.

In order to be able to control the NVH characteristics of a vehicle structure, the so called "transfer paths" between the excitation sources and the response points need to be known. Several excitations can occur on vehicles, such as the ones from engine vibrations, gearbox vibrations, the contact with the road, the turbulent boundary layer around the chassis etc., and their paths to the passenger ears and body contact points can also be diverse. From physics point of view, the transfer path can be either of "air-borne noise" or "structure-borne noise" type. While the previous one means a vibration or noise signal transmitting through air, the latter one represents a vibration signal transmitted through a solid body. Both of them occur in vehicles and therefore their accurate capturing is important for vehicle NVH simulations. From structural point of view, transfer paths between an excitation and response point can develop through several paths, i.e. a chain of chassis elements, as illustrated in Fig. 3.

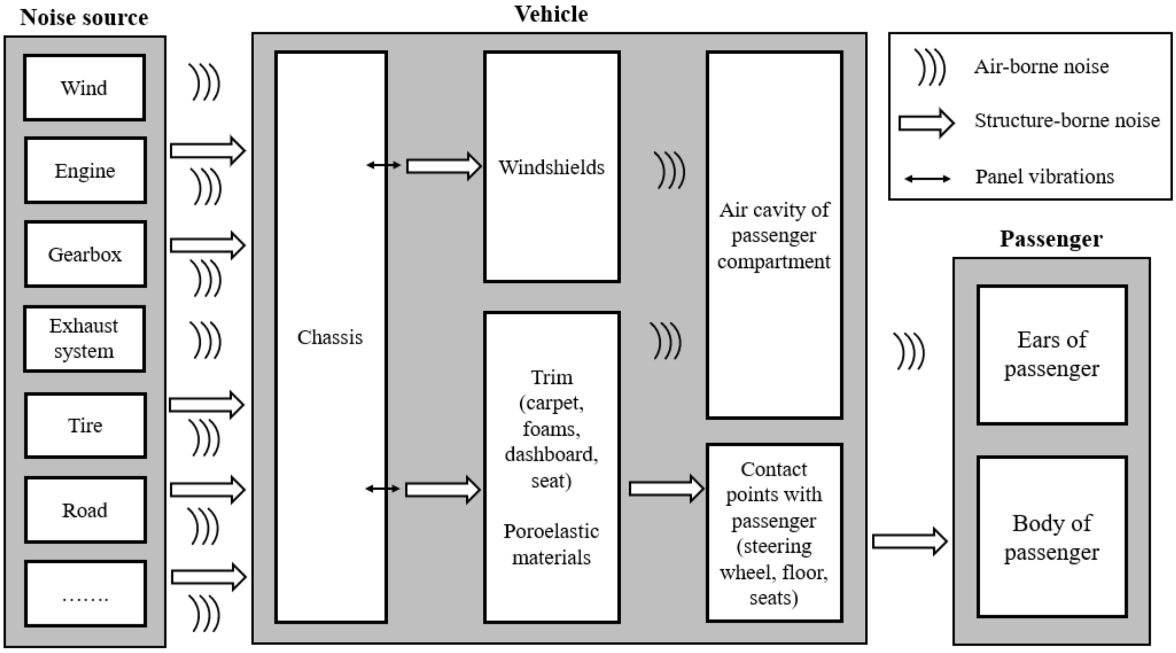

Figure 3: Typical transfer paths of a vehicle.

Consider for example the transfer path from the engine to the ear of the driver. The engine excites its mounting points in the chassis as well as the air in the engine bay. The vibrations from the mounting points are transmitted in the form of structure-borne noise all over the chassis, but also as airborne-noise in the engine compartment towards the vehicle firewall. The sum of vibrations generated in the chassis transfers into the chassis panels, such as the roof, windscreen and side windows, in the form of structure-borne noise. The chassis elements act just as membranes of a loudspeaker and transfer the signal – via airborne noise - further to the passenger ears. In parallel, the structure-borne noise propagating in the vehicle structure will transform into body vibrations of the passenger at the contact points between the passenger and the vehicle.

Finally, an important aspect of vehicle NVH analysis is the determination of the resonance frequencies and the corresponding modes shapes of the vehicle chassis. A trimmed body configuration 10.000+ of modes can be occurred up to 1000 Hz. If an excitation signal coincides with any of these, the chassis will tend to amplify the input signal and thus deteriorate the NVH characteristic of the vehicle. For example, if the chassis' first bending mode's natural frequency is close to the engine idle frequency, the body will resonate and make idling noisy. The chassis natural frequencies are a function of the mass, damping and stiffness characteristics of the chassis, thus it would be logical to modify these for improving vehicle vibroacoustics phenomenon. However, the very same parameters also define other important characteristics of the vehicle, such as the handling qualities, dynamic performance, collision safety, fuel consumption, etc, so the vibroacustics-based needs cannot be viewed separately when altering the chassis characteristics.

1.4. Principles of the simulation methods

There are several simulation methods that deal with vibroacoustic problems, such as, Boundary Element Method, Finite Element Method, Ray-tracing Method, Statistical Energy Analysis, etc. The current research focuses on pure SEA and hybrid FE-SEA methods, and for this reason the purpose of this section is to acquaint the reader with the general concepts of these methods before reviewing the literature.

At lower frequencies, the so-called "deterministic methods" (such as Finite Element Method - FEM) are capable to solve the vibroacoustic problems accurately. These methods require a detailed model of the vehicle, where the element size is dictated by the wavelength of the highest frequency to be captured, typically, about 6-8 elements need to cover one wavelength.

This on one hand necessitates high computing capacity, - thus limiting the applicability of FEM methods to higher frequencies, - while on the other hand requires a very detailed geometry of the vehicle, which is often not available at the early phases of the development process, when the geometry is far from the final one. In addition, at higher frequencies FEM is very sensitive to any deviation in the material characteristics (inhomogeneity, wall thickness changes, manufacturing inaccuracies, etc.), which can cause significant differences even in the nominally identical structures, as Fig. 4 illustrates.

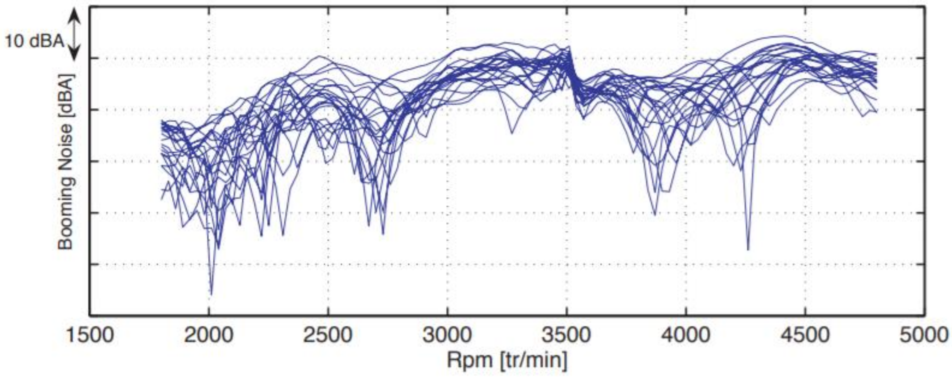

Figure 4: Vibroacoustic measurements of 20 nominally identical vehicles [5].

As a consequence, FEM is more suitable for lower frequencies, where computational costs are bearable and modal density is low. On the other hand, Statistical Energy Analysis (SEA) subdivides the vehicle model not into small elements, but rather large panels, and thus appears to be one of the most efficient methods for higher frequencies, where the modal density is sufficiently high and thus the local behavior of the structure dominates, instead of the global one (as at low frequencies) (Fig. 5).

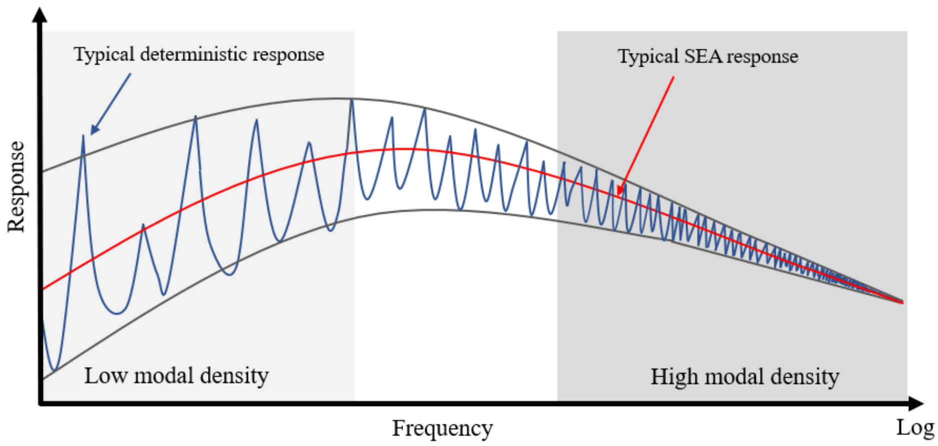

Figure 5: Deterministic methods cannot solve dense modal frequencies, while SEA predicts the ensemble average.

Since the general aim of this thesis is to develop simulation methodologies for the mid-frequency (400-1000 Hz) and high-frequency (over 1000 Hz) range, the research herein will focus on pure SEA and hybrid FEM-SEA methods. For this reason, the purpose of the next section is to acquaint the reader with the general concepts of these methods before reviewing the literature and identifying the research gaps.

1.4.1. Statistical Energy Analysis

Statistical Energy Analysis (SEA) is a statistical method, in which results are averaged spatially as well as throughout the frequency band, therefore the method neglects most of the details of the model. The fundamental idea of SEA is to divide the geometry into subsystems, on which the governing equations are solved. Consider a single subsystem, e.g. a flat plate. The injected power equals the power loss, as the following equation shows:

$$P_{in} = \omega\, E\, \eta, \qquad \textbf{(1)}$$

where P_{in} is the injected power, ω is the angular frequency, E is the stored vibrational energy of the subsystem, η is the Damping Loss Factor (DLF).

In the case of a coupled structure, e.g. where two subsystems are connected through a junction, the power flow can be represented as shown in Fig. 6.

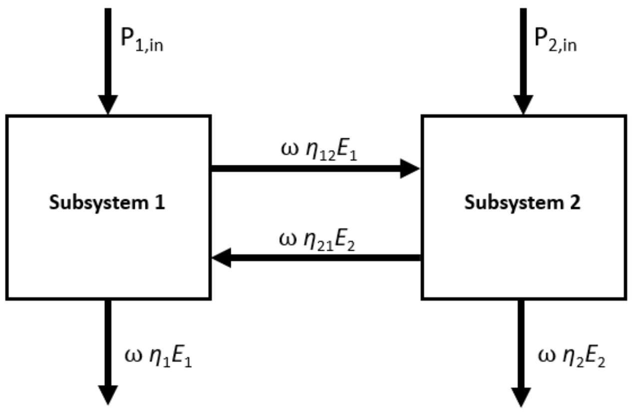

Figure 6: Energy balance between two subsystems.

The energy balance equations of the two subsystems can then be expressed as:

$$\begin{pmatrix} P_1 & 0 \\ 0 & P_2 \end{pmatrix} = \omega \begin{pmatrix} \eta_1 + \eta_{12} & -\eta_{21} \\ -\eta_{12} & \eta_2 + \eta_{21} \end{pmatrix} \begin{pmatrix} E_{11} & E_{12} \\ E_{21} & E_{22} \end{pmatrix}. \qquad \textbf{(2)}$$

The left side of the equation is the matrix of the input power, while the right side contains the frequency, the loss matrix, and the energy matrix, respectively. The loss matrix contains the damping loss factors (η_1, η_2) and coupling loss factors (η_{12}, η_{21}) of the structure. The damping loss factor (DLF) is proportional to the subsystem damping and occurs due to the internal friction of the subsystem material, acoustic radiation loss of subsystem vibration to the environment and the energy loss caused by the boundary connection damping of the subsystem. The coupling loss factor (CLF) defines the energy loss at a junction between two subsystems. For the case of a vehicle, the meaning of DLF and CLF can be and interpreted as shown in Fig. 7.

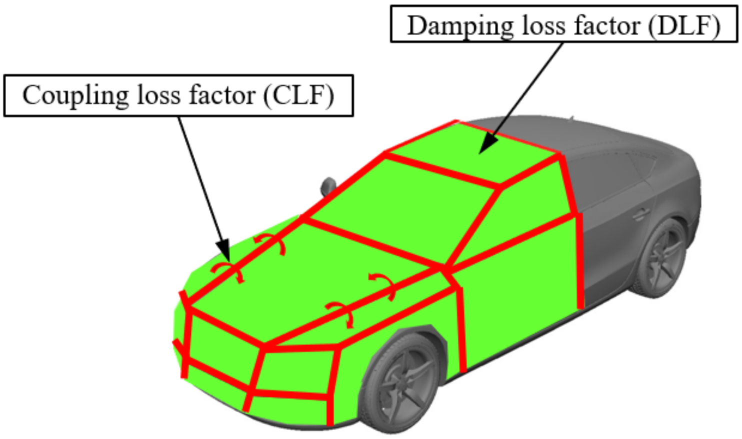

Figure 7: The front part of the vehicle is divided by SEA subsystems, the back part is the real geometry. The damping loss factor is equivalent to the damping of the subsystem while the coupling loss factor is equivalent to the energy loss between the subsystems.

The two oscillators shown in Fig. 6 are in essence an analogy to a heat transfer problem of two identical thermally conducting parts. When thermal energy is injected into the right subsystem, then one part dissipates to the environment, while the other part transfers heat energy into the left subsystem via the coupling. The modal density corresponds to the thermal capacity, the damping to radiation loss, the conductivity to the coupling loss factor and the flow of heat is similar to the flow of vibrational energy. The energy equilibrium between subsystem as well as the theory of coupling mechanisms will be introduced in more detail later, in Chapter 4.1.4.

The Loss Matrix can be obtained through the Power Injection Method. In this method, the first subsystem is excited with a certain (known or measured) injected power, while all other subsystems (response) energies are measured. Then, the next subsystem is excited, and all other subsystem's energies are measured and so on. As the Eq. 1 and Eq. 2 show the loss factors have huge importance in SEA, because they directly influence the results. Consequently, the accurate estimation of these parameters is crucial. There are three different ways to obtain DLFs and CLFs as the Fig. 8 illustrates.

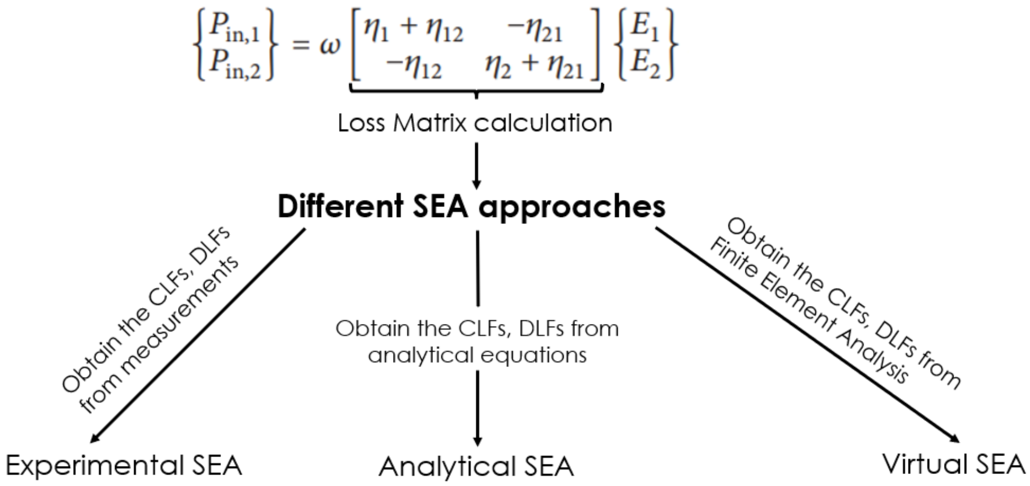

Figure 8: The 3 different approaches of SEA.

The current research will employ mainly the experimental SEA and partly the analytical SEA. Virtual SEA methods are relatively new and only a handful of publications exist in this field, with no industrial usage so far [6].

Fig. 9 illustrates the importance of using the proper damping values, since both the averaging process in the SEA simulations, as well as the different philosophies in evaluating results from FEM and SEA methods can have great influence on them. Fig. 9 shows two frequency response functions (FRF): one from an FEM simulation and the other from experimental results. Between 200-600 Hz, the FEM results are compared to the experiment without averaging, while above 600 Hz, the same FEM results are employed but by averaged in the $1/3^{rd}$ octave band. Recall, that the FEM results show the exact location of the peaks as well as the magnitude of each peak, hence this method is called “deterministic” method. On the other hand, the SEA results show only the $1/3^{rd}$ octave averaged results, i.e. the average magnitude of the numerous peaks in each $1/3^{rd}$ octave frequency band. Thus, this “hides” the data about the individual peaks, and one cannot know anymore, where they would lie. However, it gives a good idea about their average magnitude over a certain (in this case $1/3^{rd}$ octave) frequency band. Therefore, the SEA method gives information about the statistical distribution of the peaks, instead of their individual Details, hence this is termed as a “statistical” method, in contrast to FEM, which is termed as “deterministic” method.

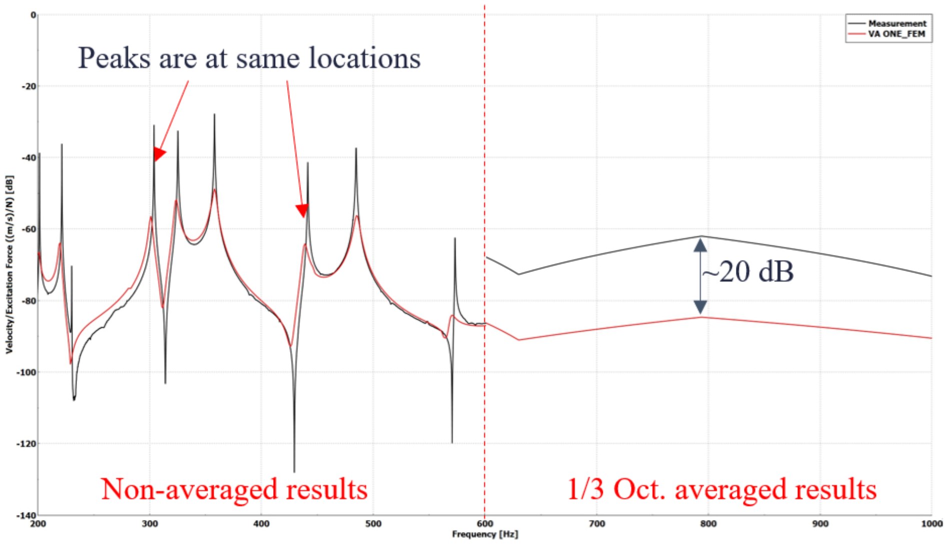

Figure 9: Comparison of FEM and experiment curves. From 200 to 600 Hz the FEM results without averaging while over 600 Hz averaged in 1/3rd octave band. As a consequence of the averaging the proper amplitudes so, damping values have significant impact to the simulation results in SEA.

Now, for the FEM analysis, the main aspect of the results is the location of the peaks i.e. the frequency differences between measurement and simulation results for the same peaks. Notice that the location of the peaks matches well, but the amplitudes do not. This is due to the errors in the damping values. However, one would be able to judge, that the FEM results match quite well the experiment, except for the amplitude. In contrast, the SEA results show much more pronounced difference from the experiment, and this is because the results are mostly influenced by the damping values due to the frequency band averaging process. Note that the two set of simulations results (FEM and SEA) are equally inaccurate with respect to experiment, yet this is much more pronounced visually in the SEA results. On the other hand, we have no information about the number or the peaks in SEA.

Although SEA in general works better at higher frequencies than FEM, the damping values can have massive effect on the results and wrong input values can literally ruin the results, as it was seen above. Consequently, entering the proper damping value in SEA is crucial for accurate simulations.

1.4.2. Hybrid FE-SEA method

SEA is ideal for panel-like structures that have a large modal density [2], [7-9]. As a rule of thumb, 3 modes/third octave band are required as a minimum for an SEA subsystem [10]. This gives a limitation to the applicability of SEA simulation methods for vehicles, where smaller subsystems or stiffer parts often cannot fulfill this requirement. Consequently, these smaller or stiffer parts would be perfectly modelled by FEM, while the larger parts (that fulfill the SEA requirement on modal density) as an SEA subsystem. This is the main benefit of the hybrid FEM-SEA method in comparison to either pure SEA or pure FEM, since it perfectly combines the advantages of the two simulation methods. For simplified geometries, as shown in Fig. 10, the subdivision between the FEM- and SEA like subsystems is quite obvious, because it is easy to distinguish between the stiff (FEM) and panel (SEA) parts.

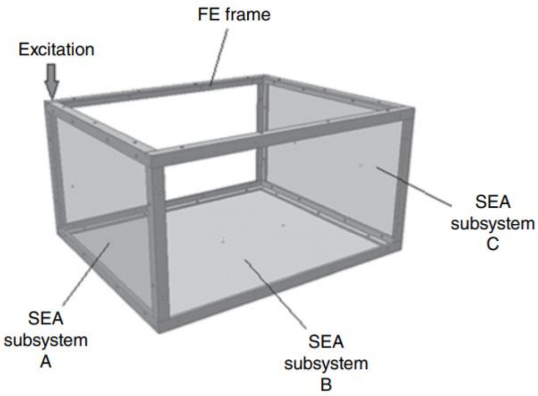

Figure 10: Ideal hybrid FE-SEA geometry with stiff parts for FEM and large panels for SEA simulations [10].

However, in vehicles the subdivision is not as straightforward as in the example above, since there are many components, which are not suitable neither for SEA nor FEM modelling. This is the real problem of the mid-frequency gap (400-1000 Hz), where some parts have too many modes to predict them accurately via FEM but they do not have enough modes to assign them as SEA subsystems [2]. These parts are then typically modelled based on the experience of the user, which can lead to accuracy problem at the end of the analysis. Fig. 11 illustrates an example of the division of the full vehicle

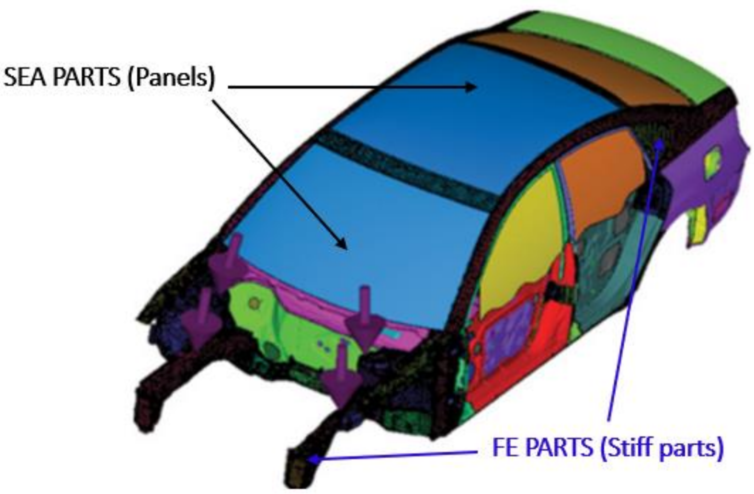

Figure 11: General philosophy of the FE and SEA division of a vehicle [11].

This Chapter highlighted the fundamental philosophy and key problems of vehicle vibroacoustic simulations in the mid-frequency and high-frequency range. In the next Chapter, we will review the state-of-the-art in these two field in detail, with the aim of identifying the concrete research gaps to be addressed in this thesis.

2. TEST CASES

In this section, the test apparatuses used for the research are introduced. The research is conducted on a set of test cases, which start from very simple „academic” test cases and gradually increase in complexity. For this, a set of test cases of various complexity were defined. The major requirements for these were that:

- they should range from very simple to complex structures,
- they should feature characteristics (i.e. materials, thickness, joints, etc.) typical of vehicle chassis structures.

With these requirements in mind, set of Test Apparatuses have been developed by the research group (see Tab. 2 and Fig. 41).

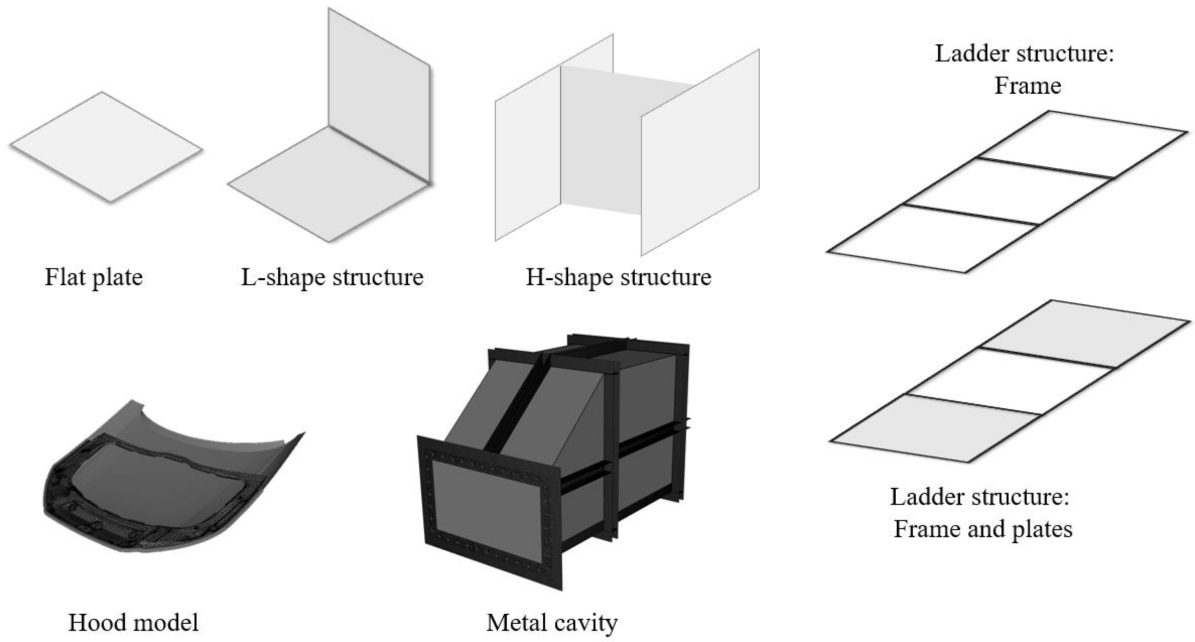

Figure 12: Test apparatuses.

The main purpose of using test apparatuses is to validate the simulation results with experiments. A major challenge was to create such structures, which can be used for validation with the SEA simulations as well. This requires such modal density, so that minimum 3 modes are contained in one-third octave. In order to satisfy this criterion, all test apparatuses were simulated via FEM and SEA software to make sure that the required modal density is achieved. All test apparatuses have been designed in-house and custom manufactured for the MTA-SZE Lendület Vehicle Acoustics Research Group. This author was personally responsible for designing test Apparatuses level 1, 2 and 3 from Tab. 2.

The design of the two cavities was especially challenging. Here, the design requirement was to have minimum 3 fluid modes in each octave above 400 Hz in order to fulfill the assumptions made for SEA simulations, with which validation was also targeted. Both cavities were modeled and simulated in FEM to make sure that the right combination of size and wall thickness is applied. The criteria for choosing the cavity size were to have the minimum total number of modes for the 400-1000 Hz range, but with a modal density of minimum 3 modes/third octave at 400 Hz. At the top of the complexity, a “rigid” (made of concrete) as well as a “soft” (made of steel) cavity has been created to be able to analyze the differences between the finite and infinite volumes [92].

Table 1: Details of the test apparatuses.

Test case name	Level	Validation methods	Simulation methods	Goal of the study
Flat plate	1	Laser doppler vibrometer, Shaker testing, Impact testing	SEA	Comparison of the measurement techniques. Determination of the damping parameters. Investigation of the influencing factors of the damping loss factor during the measurement process.
Flat plate + bitumen layer	1b	Impact testing	SEA	Comparison of the different damping treatment layups and they implementation into the simulation software.
L-shape structures	2	Impact testing	SEA	Comparison of the different binding techniques (19 variants), riveting, bolting, glueing, point welding, line welding and one reference structure without any junction, only bended. Comparison of the two different connection angles: 60° and 90°. Comparsion of the measured CLFs to the analytical values.
H-shape plate	3	Impact testing	-	Highlight the difficulties of the experimental PIM.
Ladder structure: Frame	4a	Impact testing	FEM	Comparison of the different simulation methods.
Ladder structure: Frame and plates	4b	Impact testing	FEM, Hybrid FE-SEA	
Ladder structure: Frame, plates and foams	4c	Impact testing	FEM, Hybrid FE-SEA	
Hood model	5	-	FEM, SEA, Hybrid FE-SEA	Comparison of the different simulation methods.
Metal cavity	6	Shaker testing	FEM, Hybrid FE-SEA	Investigation of 5 different hybrid subdivisions. Comparison of the calculation times as well as the velocity results at the response locations.

Every additional detail of the apparatuses such as sizes, material properties, etc. will be introduced directly in their sections.

3. THESES

3.1. Thesis No. 1

I formulated a novel procedure for the accurate determination of the damping for SEA panels. The procedure was deduced from a systematic approach and appears to provide relatively accurate Damping Loss Factor values in every frequency band, when compared to other method. I investigated the effects of the boundary conditions on the results, such as those of the coherence, the driving force or the type of the excitation. In addition, the method of data processing was also examined with log-normal distribution as well as through arithmetical average. The approach was validated on a rather sensitive, lightly damped flat plate, and the acquired data were implemented in SEA simulations, yielding minimal deviation (less than 1.5 dB) compared to the experiment results [I], [J].

3.1.1. Thesis No. 1.b

In SEA simulations the Noise Control Treatments are considered as a coverage on the surface and there is no information about the spatial distribution of them.

I proved that the different distributions of the damping layers can be taken into account during the SEA simulations if the panel damping is obtained during the measurements [G].

3.2. Thesis No. 2

Different bindings yield different CLF values, that can significantly influence the simulation results. In addition, in the numerical method only a few types of junctions exist and for this reason the reliability of the analytical formulas for CLFs is a key factor from accuracy point of view for complex cases.

In relation to this, I proved that [F]:

- The connection angle has an impact on the CLF values and this might require modification to the analytical CLF formulas to include this effect.
- The type of the adhesive has an impact on the CLF values.
- At combined junctions, such as simultaneous riveting and gluing, the CLF is not a summation of the individual binding CLFs, but rather a combination of them, namely: $CLF_{rivet} + CLF_{glue} \neq CLF_{rivet+glue}$

- Analytical equations can be used with high precision for common cases, if the modelling technique is reliable.

3.3. Thesis No. 3

For vehicle NVH characteristics, the main goal is to reduce the vibration of the panels that radiate noise into the passenger compartment. Traditionally, the vibrational energy of the panel is controlled locally on the panel by adding viscoelastic damping material to it, which in turn adds mass to the system.

I proved that the adhesive binding technique can more effectively reduce the vibrational response of a panel, than a viscoelastic damping layer applied directly on the receiver plate. While this leads to reduced vibration on the plate, the stiffness of the system remains unchanged, while the mass of the system could be reduced in comparison to a classical solution of adding viscoelastic damping layer to the plate [E], [B].

3.4. Thesis No. 4

According to the theory of Hybrid FE-SEA, the larger panels should be modeled as SEA subsystems, while the smaller ones and stiff components as FE parts. In complex structures, the determination of the borderline between the FE parts and SEA subsystems is not straightforward, since many components could fit both the FE or SEA methodology.

In relation to this, I proved that [A]:

- The divison of the hybrid model has an impact on the results, namely the proportion of the SEA subsystems related to the FE parts.
- The logic of which parts shall be represented by SEA is driven by the minimum modes in band requirement.
- The divison of the structure massively influences the computational time.

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