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Review of Finite Element Vehicle Interior Acoustic Simulations Including Porous Materials

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Abstract: Recent trends in vehicle engineering require manufacturers to develop products with highly refined noise, vibration and harshness levels. The use of trim elements, which can be described as Poroelastic Materials (PEM), are key to achieve quiet interiors. Finite Element Methods (FEM) provide established solutions to simple acoustic problems. However, the inclusion of poroelastic materials, especially at higher frequencies, proves to be challenging. The goal of this paper was to summarize the state-of-the-art as well as to identify the challenges in the acoustic simulations involving FEM-PEM methods. This involves investigation of measurement and simulation campaigns both on industrial and fundamental academic research levels.

Keywords: finite element method, poroelastic materials, vehicle acoustics

1. Introduction

Continuous development in the automotive industry has led customers to expect quieter and more efficient vehicles for their daily commute. Megatrends, such as the need to reduce the environmental footprint of vehicles, urbanisation or the general increase in the need for mobility have shifted paradigms for vehicle producers. Buyers today prefer intelligent, efficient and comfortable cars instead of vehicles requiring the increased level of driver control or exposure to tiring noises, vibrations as well as harsh, intrusive effects in the cabin. As a result of the aforementioned effects, today's mainstream vehicles require engineering sophistication that was formerly typical for luxury cars only. This manifests itself in the need for accurate engineering methods to achieve the desired vehicle characteristics, rather than just the usage of fine materials. To satisfy customers, highly scientific methods have to be applied during the design process of cars. Today, the noise, vibration and harshness (NVH) level is one of the key vehicle properties customers care of. This does not necessarily mean quite vehicles in every case. While mainstream vehicle models are expected to be quieter and smoother, their performance oriented versions need to have a unique sound character to them, to provide an aural excitement beside driving enjoyment. Both of these development targets mean that achieving the desired acoustic characteristics has gained huge importance within the vehicle development process.

The advent of the electric car places even more focus on acoustics. Electric powertrains are on one hand quieter than internal combustion engines, while on the other hand will excite the vehicle chassis at much higher frequencies, than their traditional counterparts. This means that acoustic treatments are expected to perform well not only at low frequencies (where chassis noises "masked" earlier by the combustion powertrain will now be audible) as well as high frequencies, where the excitation from e-motors will dominate [1]. Alongside the improved acoustic performance, however, stricter efficiency goals will dictate continuous weight savings as well. Both of the above mentioned, conflicting requirements coupled with the accelerated development cycles necessitate accurate numerical methods to determine acoustic properties in the early phases of the design process.

The goal of the present paper is to provide a comprehensive overview in recent developments of simulating vehicle interior acoustics via FEM-PEM methods. The paper can aid researchers in understanding the state-of-the-art of vehicle interior acoustic simulations as well as the challenges of this field, thus leading to the identification of the gaps in knowledge.

2. Theoretical background of FEM-PEM simulations

2.1. Acoustic excitation modes in vehicle acoustics

Road vehicles are subjected to a wide range of mechanical and acoustic excitations.

Fundamentally, passengers hear noises through the pressure waves transmitted to their ear drums in the air in the cabin compartment. However, these pressure waves can be induced in two fundamentally different ways, which is the classical distinction between noise sources in a vehicle [2]:

- Structure-borne noise: Noise induced by the physical impact of an object on a structure and this signal travelling within the structure. For vehicles, this would correspond to time varying loads (i.e. vibrations) entering the vehicle chassis from the road and tyres, or elements of the powertrain. An example can be the vibratory load entering the chassis through the tyres, rims and suspension due to uneven road, or the vibrations entering the chassis from the engine vibrations. Typical frequency range is between 20-400 Hz

- Airborne noise: Noise induced by aerodynamic loads or the acoustic radiation of an object and entering the structure of interest via air. For vehicles, this corresponds to time-varying loads (i.e. vibrations) entering the vehicle chassis through the air around the vehicle. Examples are the vibration of the roof due to turbulent flow around it, the pressure waves radiated by the engine in the engine bay, or the tyre surfaces as the tyre is periodically deformed at the tyre-road interface Typical frequency range is 500 Hz and above.

These two noise categories are distinctively important in the literature review, since many software applications are restricted to the prediction of one or the other only. Fig. 1 illustrates the typical frequency ranges and sources of structure-borne and airborne noise for vehicles.

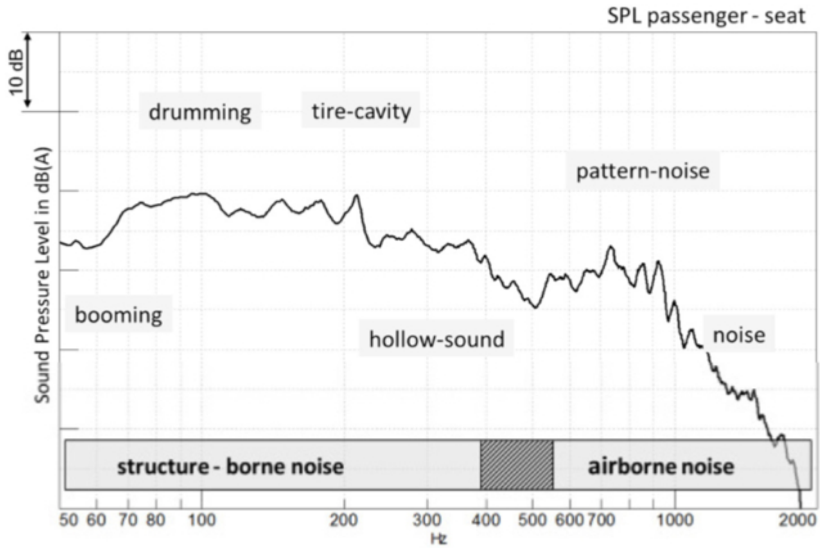

Figure 1.: Frequency range of structure-borne and airborne noise components [2]

Both noise sources lead to the vibration of the chassis elements (such as body panels, windshields, trim elements, etc.) and this vibrational energy transforms to pressure waves inside the passenger compartment, which then excite the eardrum of the passengers.

2.2. Finite Element Method

Finite Element Methods (FEM) can be used not only for stress and strength analysis, but also for predicting the acoustic behaviour of a structure. Structural problems are solved using a stiffness matrix as well as a displacement- and a load vector, while in acoustic dynamic analyses these are supplemented by mass/ and damping matrices, as well as acceleration and velocity vectors. Detailed description of structural FEM can be found in [3] and [4], while the acoustic coupling theory is expanded in [5] and [6].

For acoustic simulations, it is necessary to model the air surrounding the vibrating solid object. To determine the pressure field in air, coupled solutions are used where the stiffness matrix contains the coupling between the structure and air cavity. Song et al. [7] and Kim and colleagues [8] worked extensively on developing acoustic coupling formulations. For solving these equations, multiple solution schemes exist, with substructuring and modal analysis being the two most predominant ones in the industry.

2.3. Poroelastic Material (PEM) modelling

A recent addition to the Finite Element Method simulations has been the modeling of Poroelastic Materials (PEM). Poroelastic materials consist of a structural, porous matrix whose cavities can be filled with various fluids [9]. Figure 2. illustrates a sample cross-section of a poroelastic material. For automotive applications, poroelastic materials form for example the trim panels in the interior, carpets, seats and in general, i.e. most of the sound-absorption materials applied. Therefore, although their structure may differ, the fluid phase is always air. Interaction between these distinct phases complicates material behaviour in case of an excitation. As cavity diameters are on the microscale, computational capacity of current computers prevents the direct modelling of these structures, as cell numbers in a Finite Element solver would be excessive. To account for their effect, they can simply be modelled as non-structural masses added to the load-carrying structure [21, 22, 23, 24] or by applying the so-called Biot-theory for modelling PEM materials. Biot-theory describes the behaviour of PEM materials using about 13 macroscopic parameters (the number depends on the equation formulation) [10, 11, 12, 13], which are extracted from measurements. Since most of the Biot-parameters are both material- and frequency-dependent, their accurate measurement and application in numerical campaigns poses considerable challenges [14, 15, 16, 17]. However, the use of the Biot-theory in FEM simulations improves the comparison between experiments and numerical simulations greatly, especially when compared to the non-structural mass approach. FEM-PEM results match the experiments to higher frequencies, which justifies the extra computational cost imposed.

Figure 2.: cross-section of a generic poroelastic material [9]

The following sections will summarize selected works from the topic of FEM-PEM simulations, to highlight the state-of-the-art in this field and to point out certain gaps in the knowledge in this field.

3. Review of relevant literature

3.1. Poroelastic Materials as non-structural mass

Even though Biot`s theory was established in the middle of the 20th century [10, 11], his findings were not implemented into trim modelling until the 1980s [18, 19, 20]. It was only then, when the application of Biot's theory in vehiccle analysis gained importance with the increasing sophistication of vehicle design.

Despite the difficulty in applying the modelling equations, engineers still desired to approximate the acoustic effects of trims. The earliest approximation was done by adding non-structural mass elements on panels where trims would be installed, which – depending on the model – was sometimes extended with locally increased stiffness and damping values as well. One of the earliest examples of this technique was the work of Priebsch et al. [21]. This study examined noise levels emitted by an inline five-cylinder engine using finite element calculations. Figure 3. shows the engine-crankshaft system's FEM model.

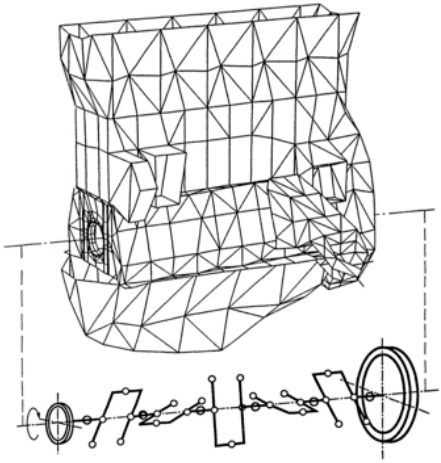

Figure 3.: finite element model of a 5-cylinder engine and crankshaft [21]

Poroelastic engine trims were modelled as extra mass without influencing the structural properties of the system. As trim elements were not the focus of this study, their effect was not quantified separately, however, the relatively good comparison between simualtion and measurements proved that the non-structural mass idea had merit in this particular application.

While trim elements are used on engines as well, their most prominent acoustic role is in the interior. Various damping mats as well as door cards, instrument panels and the seats themselves are used to attenuate disturbing noise from outside. Therefore, the application of non-structural mass models for interior poroelastic materials was a straightforward choice. An example for such a case is the work done by Sung and colleagues in 1999 [22]. As part of a series of investigations on trimmed bodies, the eigenmodes of a vehicle body were calculated, along with the frequency response. Besides their computational campaign, frequency response functions (FRFs) were also measured, shown on Figure 4. for four different locations.

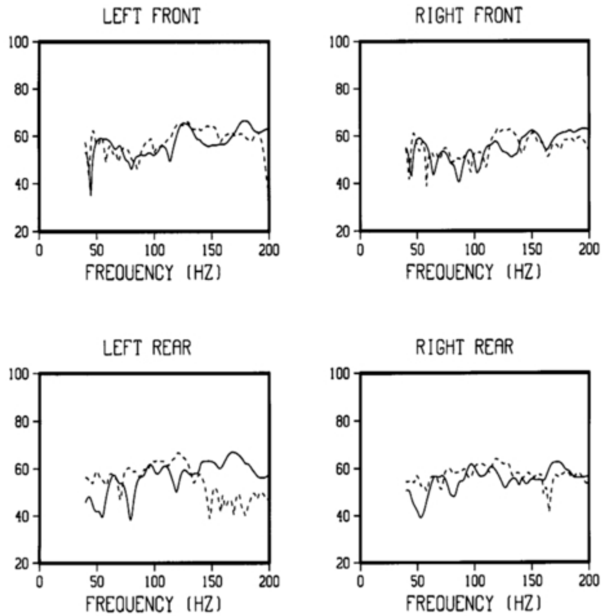

Figure 4.: Comparison of measured (dashed line) and calculated (solid line) SPL [22]

Using 24 reference points over the vehicle body and a single excitation input location, a spatially averaged FRF was compared with the calculated values (Fig. 4). As this was an early attempt at a trimmed body investigation, quite a number of simplifying assumptions were made during the simulation. Trim elements were represented only as non-structural masses, while the doors, hood and trunklid were reduced to point masses and replaced with rigid elements, joined to the bodywork through coupling factors. The relatively small amount of cells (35000 for the body model and 2500 for the acoustic cavity) also reduced accuracy, which manifested itself in the results. Trends were captured correctly, but – especially with increasing frequency – differences between measured and calculated results reached more than 5 dB both in FRF and sound pressure levels inside the cabin.

While this method has a lot of limitations in such a complex case, its quick computation time warrants its use in certain campaigns where short turn-around times are required. Subramanian et al. [23] performed a damping material optimization on a full-size truck cabin. Using a finite element solver, strain energy contours were calculated for treated body panels (Fig. 5). The contours were used to identify extreme energy points, where treatment was increased. By running this process in a feedback loop, the optimization was completed. Although the accuracy of the results was not perfect, the identified locations and treatment increases were proven useful later in the product development cycle. The overall goal of the optimization was achieved, as the production model received a damping treatment with reduced volume and mass.

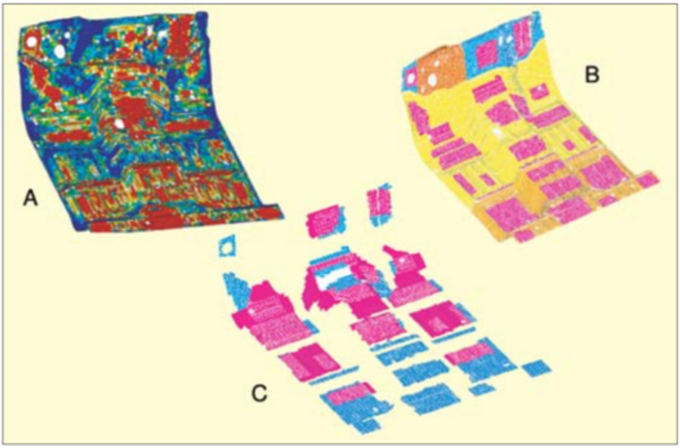

Figure 5.: Composite strain energy contour (A) and optimized damping layer treatment of a vehicle floor panel [23]

Despite being useful in predicting trends, this approximate method has very limited usage. As Nefske [24] also points out, its main advantage is the quick computational time and relatively good correlation with measurements below 50 Hz. Such limited frequency range means that it cannot be reliably used for interior noise assessments.

3.2. Biot-theory in Finite Element Methods

As computational capacities have grown exponentially, the more complex poroelastic formulations started to receive more popularity. One of the earliest to approach the Biot theory in Finite Element Methods was Panneton and Atalla [25]. Their 1995 paper describes a method where the Biot equations are formulated for a finite element solution approach, but simplified by mathematically separating the independent fluid and solid degrees of freedom and approximating the low-frequency material behaviour on the basis of air properties. Despite the mathematical efficiency, the method was only used in the 100-500 Hz region on a geometrically simple model seen on Fig. 6., and no practical application of this exact scheme was published. Göransson [26] also attempted a simplified FE solution of Biot`s equations by taking advantage of matrix symmetries between porous frame and pore fluids. The fully symmetric formulation, however, still requires five unknowns per node to solve for poroelastic effects, which is why its use is not shown in the paper above 160 Hz.

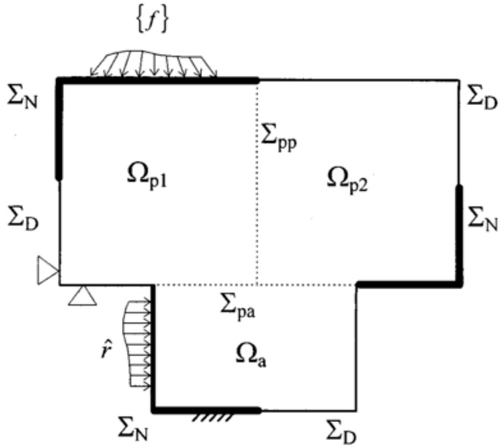

Figure 6.: Geometric setup of the problem tackled by Panneton and Atalla [25]

Deckers et al. [27] provides an excellent summary of the various formulations of the Biot-theory and their numerical applications into a solution scheme. As the goal of this paper was to review practical applications in vehicle industry, the work done by Deckers and colleagues will not be discussed in detail, however it is important as a milestone for summarizing the possible numerical ways to include poroelastic modelling. They also detail various approximation schemes, where either the PEM matrix is assumed to be fully rigid, or an equivalent fluid represents the material. To illustrate the complexity of Biot modelling, Deckers et al. lists seven different formulations that can be used in numerical solvers, with varying numbers of degrees of freedom. Transfer Matrix Method (TMM) is also described as a viable solution for higher frequency problems. Higher frequencies cause problems in FEM simulations due to the small element sizes necessitated by short wavelengths, a problem exacerbated by the presence of porous cavities which further impose wavelength restrictions. TMM alleviates this issue by describing wave propagation through a multi-layered medium. By assuming an infinite number of layers and keeping in mind that only two dilatational and one transversal wave can propagate through a poroelastic medium, the transfer matrix describes the wave field using a limited number of variables. However, the description of the exact coupling system to the boundaries and originally assuming a plane wave means that TMM`s limitations must be considered.

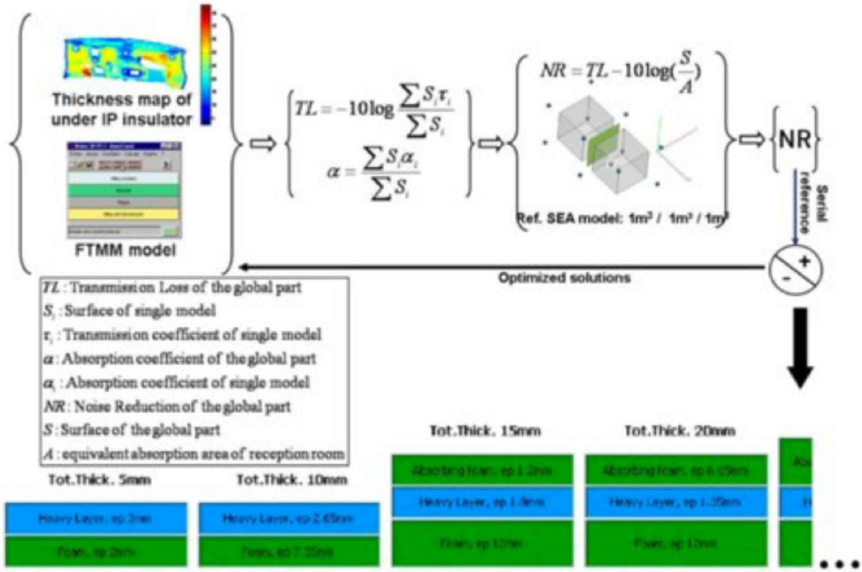

Figure 7.: Optimization loop involving TMM for an instrument panel [28]

Dejaeger et al. [28] used this method in an effective mass optimization study for an instrument panel insulator. Their goal was to retain or improve the acoustic properties of a firewall damping material while reducing its overall weight. Instead of performing detailed analysis of the entire firewall-insulator-instrument panel system in FEM for each iteration with exactly modelled poroelastic effects, the transfer matrix method was used to evaluate multiple concepts in a shorter timeframe, and only the best of those was validated using a complex FE simulation. The optimization loop developed in their paper is shown on Figure 7. Measurements confirmed the validity of the approach, with the optimized insulator providing improved transmission loss as well as lower weight than the baseline configuration. A similar optimization scheme was carried out by Rondeau et al. [29], however on a wider topic breadth: Biot parameter and transmission loss measurements were also carried out on a complex instrument panel model, while the numerical campaign involved FEM, boundary element method and poroelastic modelling as well. Correlation between measurement and simulation was overall good, especially in the above 1000 Hz frequency range, as shown on Fig. 8.

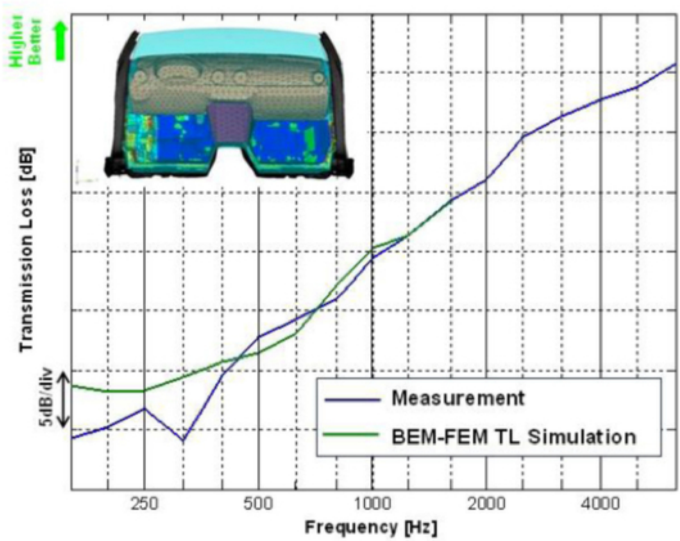

Figure 8.: Transmission loss of instrument panel insulator configuration – measurement and simulation [29]

While TMM is useful in the component optimization scheme, its limits are reached when dealing with multiple trim elements in a single acoustic system. For a more complex system, Biot equations or a model thereof is needed. In a 2015 publication, Pietrzyk [30] attempted to predict transmitted airborne noise into a vehicle passenger cabin. For the research, a body-in-blue configuration car model was used (no interior trim, doors and enclosures trimmed) with and without porous carpet material. By starting the investigation with a measurement campaign, baseline values were established. Calculations carried out with the porous carpet used the built-in Biot modeler of MSC Actran. Out of the 4 internal microphones, results for the chosen one show questionable accuracy, with differences reaching 10-20 dB in the worst locations. Frequency range was higher here, as expected for airborne noise transmission, however the maximum test frequency was only 400 Hz. Computational results followed measurement curves better in the 0-200 Hz region than above as Fig. 9. shows, although it should be noted that this model needs more refinement to obtain closer results. As mentioned before, the sheer number of material parameters influencing the Biot-model, as well as their frequency dependency can introduce large discrepancies into simulations, and possibly with more correctly set values, this calculation could also arrive at a better conclusion.

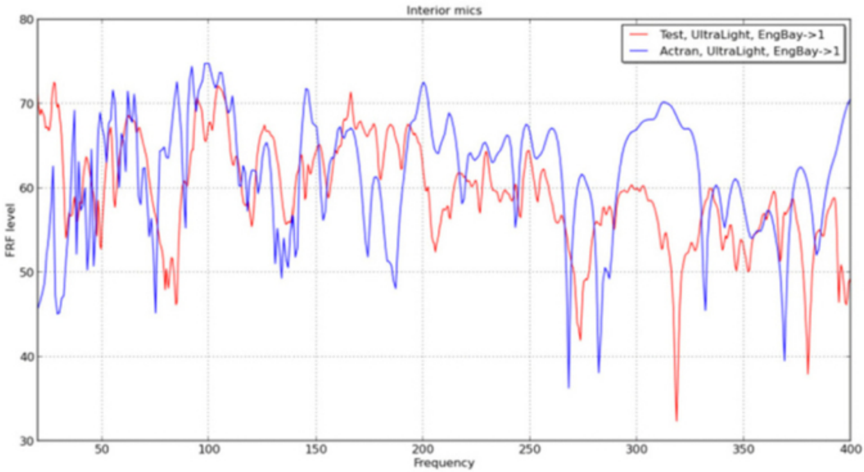

Figure 9.: Resulting FRFs at microphone 1 from measurement (red) and simulation (blue) [30]

The built-in capabilities of MSC Actran for poroelastic modelling were used by Guellec et al. [31] to perform trim element optimization on a passenger car. As the most recent published work on the FEM-PEM topic, it contains the latest simulation results obtained for a trimmed vehicle setup with floor insulator carpet. Two different calculation methods are presented and compared in their work. One involves MSC NASTRAN for a traditional modal solution, which includes calculation of the reduced impedance matrix in Actran. The other, however, takes place fully in Actran and starts with the computed cavity- and structural modes, with the ability to add frequency dependent damping – either globally, or node by node. As the amount of modes can increase rapidly, only a select number of them are stored for computational efficiency. During the carpet optimization, both solution strategies are used since changing embossments retains the originally calculated impedance matrix, while for changed trim material properties, the NLOPT sequence is used to obtain a new impedance matrix for the original modes. Using the NLOPT model enabled the mass optimization of the trim, and resulted in a 1.7 kg weight gain but 2.5 dB reduced power spectral density level over the whole frequency range up to 250 Hz (Fig. 10.). As future work it is recommended to investigate the high-frequency behavior of the optimized trim and the process itself as well.

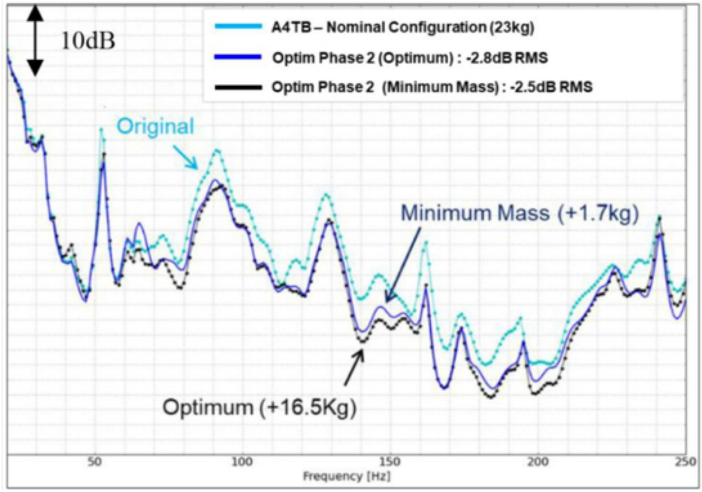

Figure 10.: Trim optimization results showing baseline (light blue), acoustically optimized (dark blue) and minimal mass (black) trim packages [31]

3.3. Hybrid modelling methods

High computational costs arising from the complexity of Biot`s theory have driven engineers to develop alternative methods to account for trim behaviour in acoustic models. Panel acoustic participation is a finite element-based method, as presented by Wang et al. [32]. This method calculates resultant pressures from node vibration, which are summed over each panel to account for their individual pressure contributions. Then, all surrounding panel induced pressures are summed to obtain the resultant pressure for the whole cavity. Thereby the panel participation is defined as the projection of the sound pressure of individual panels in the direction of the overall sound pressure. In their investigation, the panel participation method is used to identify large contributors to interior noise among the panels in contact with the interior cavity on a four seat family car with hammer excitations. Figure 11. shows panel divisions used in the study. Despite good agreement between measurement and simulation, the used frequency range is really narrow (20-50 Hz) and severely limits the applicability of this method.

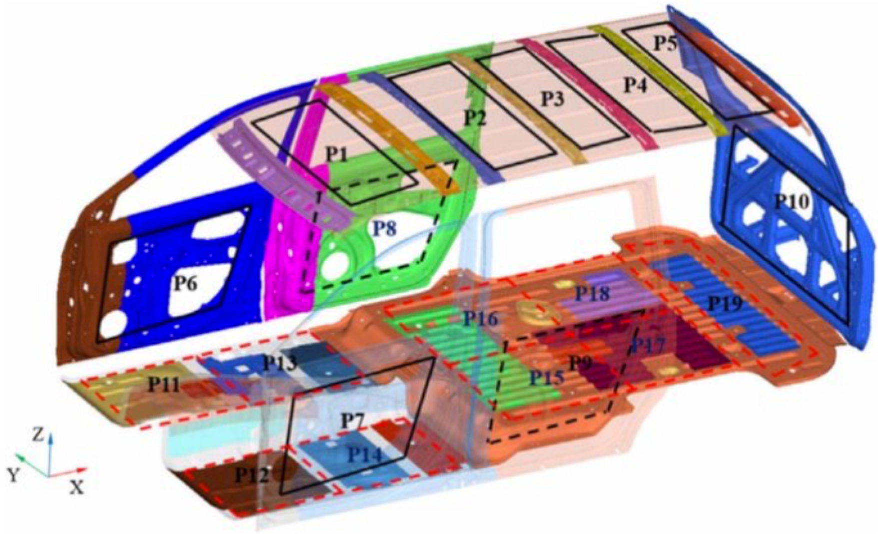

Figure 11.: Panel division in the investigated car for panel acoustic participation method study [32]

Duval and colleagues [33] also worked on an energy based hybrid method to obtain sound pressure levels for a trimmed body. The goal of the study was the implementation of a new sound package for a diesel hatchback car. In order to do so, acoustic power and transfer functions were measured in a fully lined interior with acoustic treatment and used to calculate sound pressure level at the driver`s ear level. The measurements resulted in an acoustic map of the interior, so the optimization could start with the most contributing elements. Figure 12. shows the acoustic power map of the interior panels investigated in the study. Even though the results were quite good, such an optimization relies on measurements on an actual prototype, so it cannot advise designers upstream in the design process.

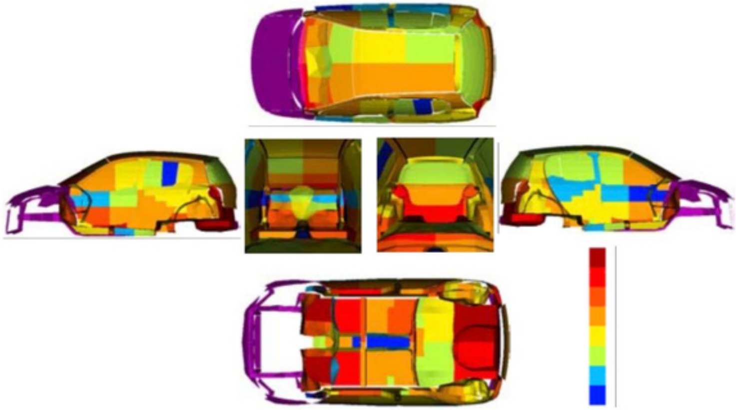

Figure 12.: Acoustic power map of interior trim panels in a small hatchback car [33]

Not focused on trim elements per se but on an entire design process from high-level targets to execution, Bergen et al. [34] presents a whole development strategy for vehicle manufacturing to deal with the challenge of designing an acoustic sound package. Their recommendation is to use panel participation method (or panel contribution analysis) and transfer path method to establish component goals based on overall design targets, and use more computationally costly procedures like FE models on smaller substructures, so they can be investigated in detail. However, the introduced example is only looking at the finite element results and does not mention the variety of errors that can be introduced into the process at the beginning with the establishment of component goals.

4. Conclusions

An overview of the most relevant literature on Finite Element Method simulations including trim elements made from Poroleastic Materials is given in this paper. Inherent complexities are introduced into an acoustic model once the representation of trim elements is required, since the complex micro-level interaction between fluid and solid phases need to be represented on a macroscopic level. Although Biot`s theory describes this behaviour, it is computationally expensive and introduces an unmanageably large amount of degrees of freedom to handle, especially at higher frequency. Improvements in computing power resulted in the broader application of these methods, and their accuracy is unquestionable below 250 Hz with correct parameters. Despite this, a distinctive lack of higher frequency trim simulations exists, with no articles citing objectively good correlation between measurement and simulation above the 250-300 Hz region.

Acknowledgements

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