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Transmission Loss trim FEM simulation of lightweight automotive dashboard insulators with consideration of the instrument panel

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In previous studies, good correlation results between Transmission Loss measurements and finite elements simulations on dashboard or floor insulators were obtained. Regarding dashboard insulators, those works were done without considering the Instrument Panel. The weight reduction is a strong trend in vehicles design and impacts obviously the noise treatments. In order to ensure an equivalent noise reduction performance, a way to proceed is, for example, to reduce the heavy layer weight and to compensate the loss of insulation by adding an absorption layer. When neglecting absorption and masking effects linked to the presence of the Instrument Panel, the overall insulation performances of the front vehicle unit are not properly estimated then. In this paper, a numerical study dealing with Transmission Loss simulation of a dashboard insulator with consideration of the Instrument Panel, including absorbing felts, was carried out in order to predict the insulation performances of a complete front vehicle unit. In this study, weight localization on dashboard insulators was optimized in order to improve noise reduction performances for airborne and structure-borne noise excitation in the middle frequency range. Finally a ranking, between all studied solutions, from pure absorbing systems to highly insulating noise treatments is presented.

1 Introduction

Due to European regulations reducing CO_2 emissions from passenger cars, the OEMs have to decrease vehicle weight. This weight reduction has also an important impact on the NVH package overall mass. But in spite of insulators weight loss, the acoustic comfort in passengers compartment must be maintained.

A way to take up this challenge is to use multilayered insulators, such as insulating foam / heavy layer / absorbing foam (called Light Weight Concept or LWC on some figures below) for example, in order to reduce the weight (until 25%) while maintaining the noise reduction performance in passenger compartment. The principle of this solution is to decrease the heavy layer weight and to compensate the loss of insulation by adding an absorption complex [1].

The vehicle global NVH target is defined by the OEM. To meet this target, modules objectives are chosen. These acoustic requirements are often expressed by an insulation performance curve, the Transmission Loss (TL), and an absorption performance curve, the absorption coefficient.

TL is measured with substantial facilities: the coupled reverberant rooms. The implementation of such measurements is expensive and requires a long preparation time: the module is cut out from a car body, mounted between the two rooms and mocked up carefully in order to focus on the studied area (leakage or flanking path have to be avoided).

Coupling reverberant rooms measurements on real dash insulator part were performed many times and presented for example in [2]. Two configurations were tested: insulators with and without passthroughs and Instrument Panel (IP). The multilayered insulator (LWC) is about 30% lighter than the standard insulator (mass-spring system: insulating foam / heavy layer): this explains the lower TL performance between 500 and 2500 Hz (Figure 1). But by setting up passthroughs and IP, the TL of this multilayered insulator becomes better upper than 500 Hz. The lighter weight combined with this good TL result confirms the multilayered insulator as an attractive solution. However, because of the heavy layer weight loss and the low absorption compensation in low frequency range, the TL performance can not be as effective as the standard insulator performance in the frequency range between 200 and 400 Hz. This fact can be considered as a problem and will be addressed later in this paper.

In order to minimize the amount of measurements, simulation tools are nowadays deployed to design and evaluate insulators performances.

Several simulation methods are available for predicting the acoustic performances of layered insulators containing sound absorbing materials. Most of them implement the Biot-Allard theory to describe porous material [3]. The simplest one is an analytic method using the Finite Transfer Matrix Method (FTMM). Measurements campaigns, in which a flat case was mounted in between the coupled reverberant rooms, have validated the TL simulation for flat samples: correlations are excellent. But for real case, such as dash module, which includes curved panels, this is not the case anymore, the slope of the measured TL curve is not well reproduced [4].

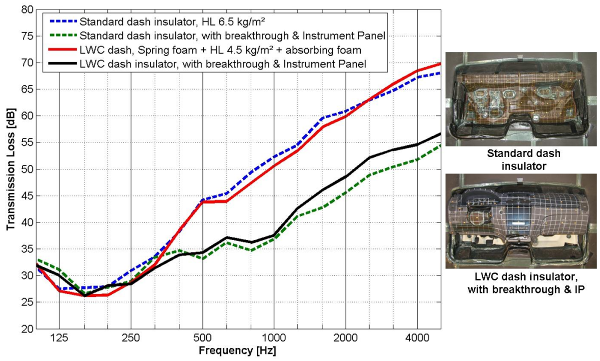


Figure 1: Dash insulator measurements.

Finite Elements (FE) modeling of coupled fluid-structure problems integrating poroelastic elements provides good results in low and middle frequency range. Previous studies have allowed to build our know-how in term of generation of numerical models and to gain confidence in results, towards good correlations with measurements [4] [5].

As an example in [6], TL measurements of formed parts were compared to TL simulations. Special care was taken to build assembly and model, in order to obtain good correlation. The TL results (Figure 2) show very good correlation up to 1250 Hz, with a slight overestimation for the trimmed dash upper 800 Hz (beginning of convergence issue due to mesh size).

Trim FE simulation is thus a relevant method to evaluate insulation performance of real formed parts: standard insulator, but also multilayered insulators, combining insulation and absorption.

As shown by measurement results (Figure 1), the insulation performance of multilayered insulators can only be properly estimated with the IP combination where absorption effects occur. Therefore the IP must also be integrated in FE simulation model, in order to evaluate the overall insulation performances of the front vehicle area.

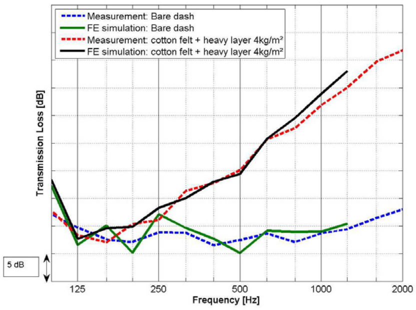


Figure 2: Dash insulator Transmission Loss correlation.

2 Consideration of the Instrument Panel

According to measurements carried out for years in coupled reverberant rooms, mounting a classical IP on vehicle front unit improves the TL of about 3 to 4 dB on the global frequency range (Figure 3). The same masking effect is expected on FE simulation results by modeling the IP.

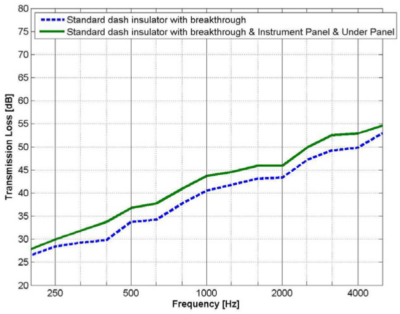


Figure 3: TL measurements by mounting the IP.

2.1 Instrument Panel modeling

In order to integrate the IP into our models, it has been considered as a part of the FE trim model. The IP will be simplified and modeled as an elastic solid component. Only plastic surfaces, then considered as a kind of screen, the cavity between IP and insulator and sub-cavities in the IP will be represented.

The IP surfaces mesh is defined from the available CAD data. The secondary cavities are formed by the electronic box, the HVAC and the glove box. The studied IP has some absorbing parts on the back side. These absorbing felts are also modeled with poroelastic elements and corresponding Biot parameters (see Figure 4).

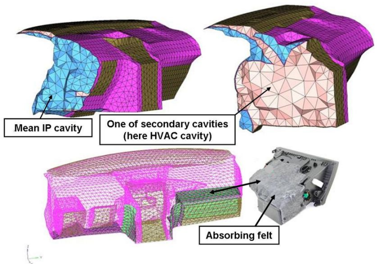


Figure 4: Instrument Panel Definition.

In order to evaluate the masking effect of the IP, and check the 3 to 4 dB TL improvement, models were built with and without IP. For avoiding too long calculation, these simulations were carried out with a simplified structure, which has the advantage to have a reduced modal basis, and on a reduced frequency range. This structure comes from the extrusion of the lower skin of the dash insulator model. The perspicacity of this method is shown in reference [4]. Figure 5 represents the TL model with mounted IP on a standard insulator.

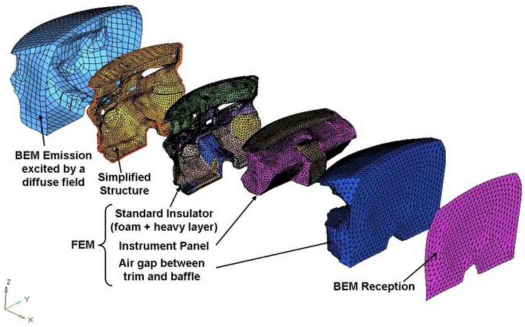


Figure 5: TL model including IP in Trim FEM.

The aperture towards the passenger compartment under the IP has an area of 0.07 m^2 (Figure 6). This area is the only airborne path, because the IP is airtight, but radiates noise. In order to avoid unrealistic structure-borne coupling with IP, it is completely mechanically decoupled from insulator: no contact with the insulator upper surface. The structure-borne excitation is shut down and the IP is only airborne excited through the under-IP cavity (Figure 6). An improvement from 1.5 to 6 dB is observed compared to the model without IP (Figure 7). This model seems to ensure an improvement on a large frequency range.

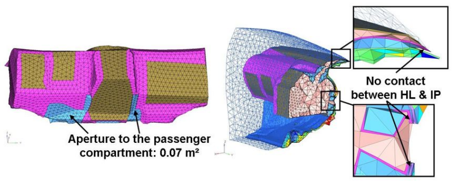


Figure 6: IP on standard insulator.

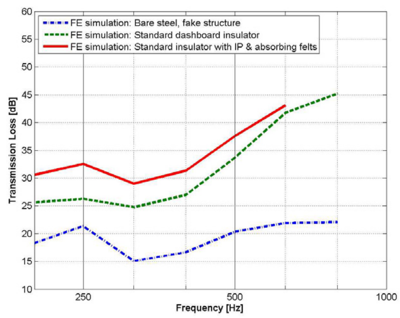


Figure 7: TL simulation results with & without IP.

2.2 IP on multilayered dash insulator

The previous results were obtained with a standard insulator. A multilayered insulator was also designed and modeled respecting the overall dimensions of the serial insulator part (Figure 8).

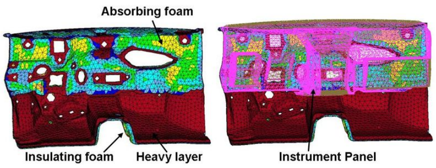


Figure 8: multilayered insulator model w/o & with IP.

TL simulations with simplified structure were likewise done with and without IP. Results are compared to standard insulator results on the Figure 9. The TL performance of the multilayered insulator, either with or without IP, are lower than the TL performance of the standard insulator. Indeed, compared to the standard insulator, the heavy layer weight was reduced by 40% on the global surface, including on the lower area without absorption compensation. The main comment regarding these results is the significant TL improvement from 315 Hz of the multilayered insulator by mounting the IP. For example, putting the IP on multilayered insulator improve the TL results by more than 6 dB at 500 Hz, while putting the IP on standard insulator brings only 4 dB at 500 Hz. The simplified model of the IP is also appropriate since it reproduces the observed effects by measurement. It will be now used with a vehicle front unit model.

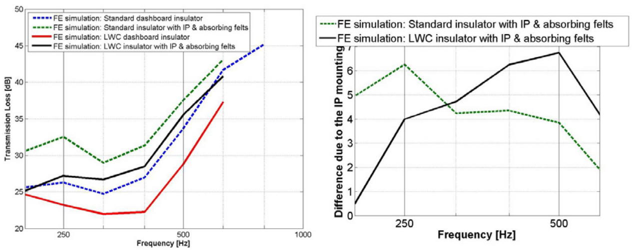


Figure 9: Standard & multilayered insulators TL results.

3 Simulations with vehicle front unit

From this chapter, all TL models use a real vehicle front unit, instead of the simplified structure previously. By strictly respecting the overall dimensions of the serial standard insulator, different noise treatment concepts described in [7] and illustrated on Figure 10, were modeled with the IP . These several concepts were designed only on the dash area covered by the IP. On the lower surface connected directly with the reception room, the insulator remains the same for all cases. So the insulator weight reduction takes place only on the surface behind the IP and the insulation performance out of it is equivalent here.

The TL results of the four insulators with IP are showed on Figure 11. The multilayered insulator (LWC) with 2D felt (single thickness) absorber and the hybrid insulator (injected foam on felt) have nearly the same TL, although the multilayered insulator is slightly better from 630 Hz up to 2 dB. But these two solutions have lower insulation performances compared to the standard insulator and the multilayered insulator with 3D foam absorber (about 4 to 5 dB from 500 Hz). Indeed the disadvantage of these two concepts is induced by

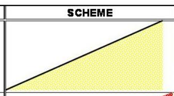
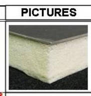
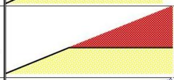
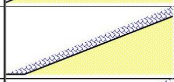
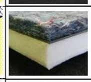
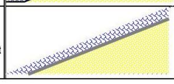
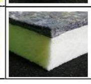
DESIGNATION	SCHEME	PICTURES	WEIGHT behind IP	GLOBAL WEIGHT
Mass/Spring Foam/HL			Reference	Reference
Light Weight Concept Foam/HL/Absorb. Foam			- 34 %	- 14 %
Light Weight Concept Foam/HL/Felt 2D			- 21 %	- 9 %
Hybrid Injected Foam/Felt			- 68 %	- 28 %

Figure 10: Modeled cases with front unit and IP.

low thicknesses area, where insulating foam can not be injected, due to the single thickness of the upper absorbing felt: the heavy layer (for the multilayered insulator with 2D felt absorber) or the upper felt (for the hybrid one) are directly in contact with the bare steel. The insulation performance of these two insulators is strongly linked to the part geometry.

The multilayered insulator with 3D foam absorber has equivalent insulation performance compared to the standard insulator from 500 Hz, despite the lower heavy layer weight (34% weight reduction behind IP compared to standard insulator). The 3D foam absorption layer under the IP compensates the loss of insulation due to heavy layer weight reduction. However under 500 Hz, the absorption, which is especially effective in the high frequency range, is not able to compensate the insulation loss.

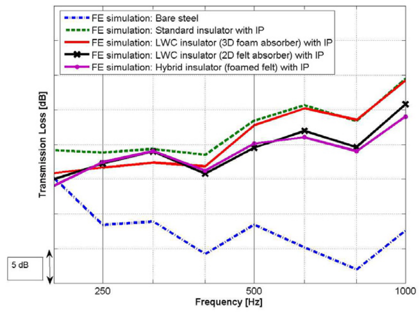


Figure 11: TL simulation results with front unit and IP.

This lack of performance could be considered as a problem, regarding the sound pressure level in passenger compartment, which is significant in the middle frequency range. New developments on the multilayered insulator, based on heavy layer localization, are presented in the next chapters in order to improve the insulation performance under 500 Hz.

4 Multilayered insulator optimization regarding TL performance

4.1 Optimization preset using FTMM

In the previously presented multilayered insulators, the heavy layer weight was considered as uniform. In order to improve the TL in the middle frequency range, the idea is to localize the heavy layer weight according to the thicknesses of the part [8]. An optimization loop (Figure 12), based on FTMM, is used to optimize the weight localization following the multilayered insulator design rule of Figure 10.

The FTMM simulation results are recomposed to obtain the global TL and absorption performances of the part. To take into account both insulation and absorption, the Noise Reduction (NR) is calculated.

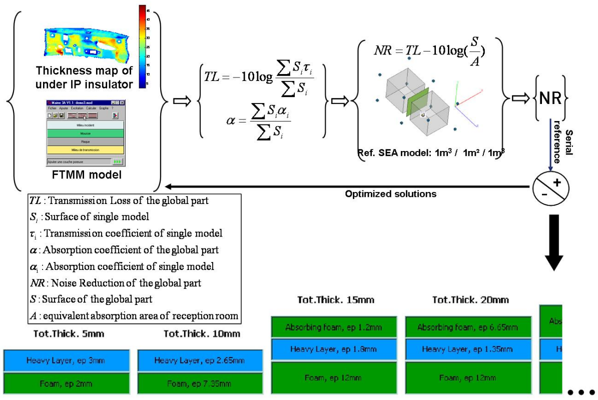


Figure 12: Optimization loop with FTMM simulations.

The obtained results do not reproduce measurements well, as a flat sample assumption is taken, despite the real part curvature [6]. Nevertheless, this method, considering the Noise Reduction in a reduced reception room, is relevant to do a ranking between different insulators: pure insulating ones, pure absorbing ones or those combining both properties.

Several optimization loops have yielded the red curve on Figure 13, result of the optimized location of the heavy layer weight. Compared to the multilayered insulator with uniform heavy layer, the multilayered insulator with localized heavy layer shows better insulation performance in the middle frequency range between 250 and 400 Hz.

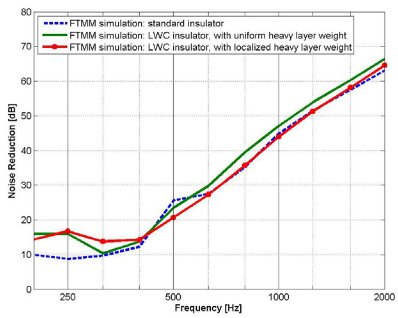


Figure 13: Weight optimization results with FTMM.

FTMM simulations allowed to quickly optimize the heavy layer weight localization. This localization will be transferred into the FE model in order to validate these results in a real 3D model (Figure 14).

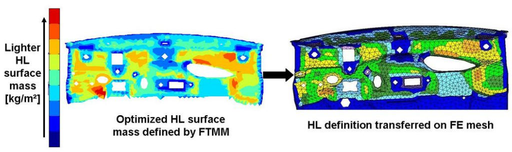


Figure 14: Heavy Layer localization.

4.2 FE simulation validation

A FE model of a multilayered insulator is built with the defined heavy layer weight localization. The weight of this insulator is 19% lighter as the reference standard insulator, this is a further 5% weight reduction compared to the multilayered insulator with uniform heavy layer.

The TL performance of the multilayered insulator with localized heavy layer is in line and even beyond expectations (Figure 15). Indeed, as predicted by FTMM simulation (Figure 13), this insulator has better insulation performance (up to 5 dB) between 250 and 400 Hz than the multilayered insulator with uniform heavy layer. In this middle frequency range, the performance is even better up to 3 dB compared to the standard insulator. From 500 Hz, the three insulators with IP have an equivalent TL.

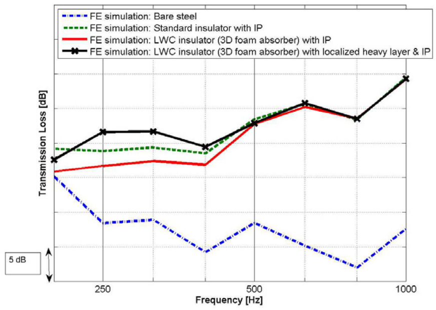


Figure 15: TL results of insulators with IP.

This good result give confidence on the capacity of multilayered insulator with localized heavy layer to have equivalent insulation performance than standard insulator already from the low frequency range (250 and 500 Hz). How is this performance with a structure-borne excitation?

4.3 Structure-borne excitation

The insulators FE models are coupled to the complete passenger compartment cavity. In order to obtain relevant Sound Pressure Levels (SPL), microphones were located at the driver and passengers head. A unit force is applied on the left engine mounting in order to have a structure-borne excitation on the front dash unit (Figure 16).

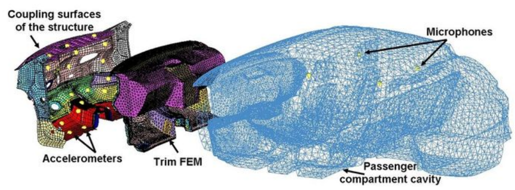


Figure 16: Vibro-Acoustic model.

Figure 17 shows the averaged SPL in passenger compartment. Basically, almost the same effects are observed as for TL simulation. The performance of the multilayered insulator with uniform heavy layer is lower under 400 Hz compared to the standard insulator: more level induces more noise in passenger compartment. Then except by 800 Hz, where a peak is observed for the standard insulator, the performances are equivalent. Furthermore, localizing heavy layer weight on the multilayered insulator allows to match the standard insulator performance. So heavy layer localization defined with airborne excitation is also valid for structure-borne excitation (others excitation nodes were also tested).

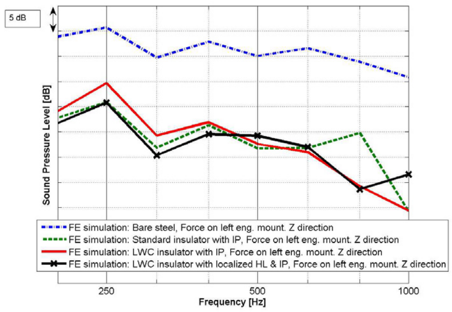


Figure 17: Average SPL results in passenger compartment.

5 Multilayered insulator optimization regarding sound radiation

On a second vehicle FE model, a heavy layer weight localization was done according to the dash insulator sound radiation into the passenger compartment. The localization process is described on Figure 18. From a standard insulator, with uniform heavy layer and without IP, sound intensity cartographies have been computed for the interested third octave frequency bands and selected excitation nodes. From these cartographies, surfaces have been defined, where heavy layer weight localization has to be done in order to minimize the sound radiation. This localization is then transferred on the multilayered insulator FE mesh.

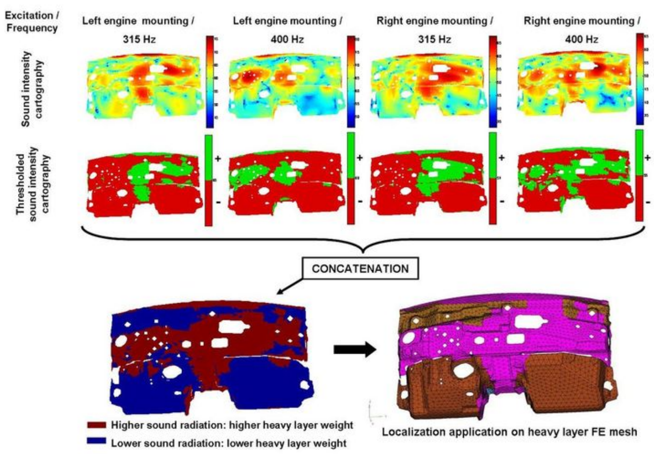


Figure 18: Heavy layer localization process.

On Figure 19, FE simulation results with IP are represented for a standard insulator and a multilayered insulator with the defined heavy layer weight localization. The multilayered insulator is here about 10% lighter as the standard insulator, but the insulation performance is equivalent and even better between 100 and 200 Hz and between 500 and 800 Hz.

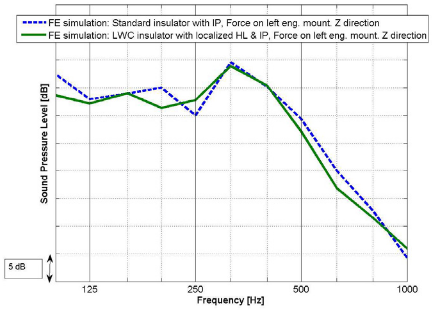


Figure 19: Average SPL results in passenger compartment.

6 Conclusion

In order to take into account both insulating and absorbing properties of multilayered insulators, the instrument panel was integrated, for the first time, into the FE models.

The multilayered insulator with localized heavy layer weight answers to the OEM request to have a light weight insulator, which has as good acoustic performance, even in the middle frequency range, as a standard mass-spring insulator for both airborne and structure-borne excitation.

The simulation performed in this article was done without consideration of noise radiation through vehicle passthroughs. Depending on the significance of leaks caused by this passthroughs, others insulator concepts could be highlighted. As a perspective, simulations including a leakage percentage in the insulator FE models will be carried out next.

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