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Trim FEM simulation of a dash and floor insulator cut out modules with structureborne and airborne excitations

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During a vehicle development, measurements on cut out modules in large coupled reverberant rooms are often carried out in the middle and high frequency range in order to optimize the insulation performance of the trims (Transmission Loss). Using optimal controlled mounting conditions, we have been able to extend the frequency range to the low frequencies in order to validate trim FEM models of a dash and floor insulator modules with structureborne and airborne excitations.
Both coupled response with movable concrete cavities (structureborne excitation) and Transmission Loss with the coupled reverberant rooms (airborne excitation) have been measured and simulated for various types of insulators on the same setup, without any change on the mounting conditions. An additional movable absorbing environment in the large reception room has been deployed in order to carry out laser vibrometer (skeleton velocity) and p-u probes (particle velocity and intensity) measurements on the surface of the trims.
By incorporating the maximal treatment mock-ups of the cut out modules as additional trims in the models, we have obtained good correlation results between measurements and simulations for both bare and trimmed configurations for a dash and floor insulators with structureborne and airborne excitations.

1 Introduction

Acoustic comfort is a key issue for the car makers and often a way to differentiate between brands. Modeling accurately porous materials is therefore of the utmost importance in the transportation industry not only in order to simulate the acoustic performance but also in order to develop new noise treatment technologies, especially towards weight reduction.

The Biot-Allard porous media theory is now extensively used in the automotive industry, particularly by trim suppliers, through its Transfer Matrix Method implementation (T.M.M), generally combined with Quick SEA approaches [1, 2]. We have shown that the T.M.M was giving excellent results for flat samples with spatial windowing but was not at ease with curved shapes, due to the decrease of the Insertion Loss slopes of insulators with curvature [3, 4]. First 3D investigations on a simplified trimmed half-cylinder structure using poroelastic trim FEM modeling in the low and middle frequency gave promising results and proved that this approach was necessary to catch the three dimensional coupling effects between structures and trims [5].

The development of the numerically efficient mixed (u,p) formulation for poroelastic material by Atalla et al. with only 4 degrees of freedom per node (instead of 6 for the former (u,U) formulation) and its finite element implementations on the first commercial softwares gave a real start to large scale industrial applications [6, 7]. Except for the case of simplified setups like the RTC III, most of the industrial applications with partially trimmed or almost fully trimmed vehicles were carried out in the low frequency range up to $400\,Hz$ typically [8, 9, 10, 11, 12].

The first aim of this study was to extend the trim FEM simulation from the low frequency range to the middle frequency up to $700\,Hz$ for structureborne coupled vibro-acoustic responses and up to $1000\,Hz$ for airborne Transmission Loss, by working on dash and floor insulator cut-out modules. The reduced size of the problem enables to go upper in frequency, while maintaining reasonable computational times. These cut out modules are classically mounted, during a NVH development, in large coupled reverberant rooms for trim insulation performance optimization and pass-through analysis in the middle and high frequency range. The challenge has been to design optimal controlled mounting conditions in order to be able to simulate correctly the modal dynamical behavior of the dash and floor cut out structures. The second aim of this study was to validate separately the trim FEM models of a dash insulator and a floor insulator for both induced damping effects on the structure (acceleration) and radiation efficiency in the cavity (sound pressure levels), which has been made possible by the separated measurement setups and by incorporating maximal treatment mock-ups of the cut out modules as additional trims in the models.

2 Mesh criterion

The problem of convergence of poroelastic finite elements is always a delicate issue, especially for 3D behaviors [13]. The most common mesh criterion for poroelastic finite elements, considered as necessary, is to define 6 linear elements per wavelength, the wavelength being the shortest one of the three Biot waves propagating in the considered porous material (two compressional waves and one shear wave) [1]. In fact, the first compressional wave propagates mainly in the solid phase and is classically called the solid-borne wave (k_1 wave number), the second compressional wave propagates mainly in the fluid phase and is classically called the fluid-borne wave (k_2 wave number), the third single shear wave propagates mainly in the solid phase (k_3 wave number).

Therefore, one has to compute the three Biot waves for each trim and look for the shortest one at the upper frequency ($700\,Hz$ here):

$$\lambda_i = Re(\frac{2\pi}{k_i}) \;\; for\ i = 1, \ldots, 3 \qquad (1)$$

The figures 1 and 2 show the three Biot wavelengths for the dash and floor trims (plus the equivalent fluid one) and illustrate different behaviors for the two trims. Indeed, a strict application of the $\lambda/6$ mesh criterion at $700\,Hz$ leads for the dash trim to an extremely small $1,5\,mm$ linear element length and for the floor trim to a small $5,4\,mm$ linear element length. Due to RAM memory limitations, we have been obliged to define an average $7,5\,mm$ element length in the thickness, which is the most important one for automotive applications (it goes up to $20\,mm$ for the lateral dimensions for the

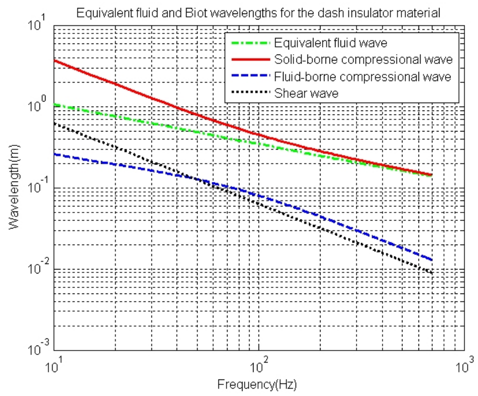

Figure 1: Biot wavelengths for the dash trim.

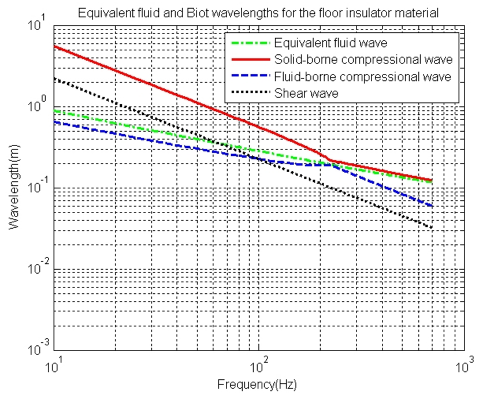

Figure 2: Biot wavelengths for the floor trim.

dash and up to 30 mm for the floor), but with compulsory quadratic elements for the dash trim (one quadratic element being equivalent to three linear ones as a rule of thumb) and linear elements for the floor. This means that the $\lambda/6$ mesh criterion is respected up to 450 Hz and that only a $\lambda/4$ mesh criterion is respected up to 700 Hz.

3 Dash trim FEM model

3.1 Bare dash panel configuration

The well controlled boundary conditions of the cut out dash panel have been obtained by continuously welding a 3 mm thick plate on the one hand to the dash structure and on the other hand to a 100 $mm \times 100\,mm$ metal frame filled with concrete and fixed to the vertical separation wall of the coupled reverberant rooms (a movable absorbing environment was used for the structureborne configurations cf. figure 3). For the complex shaped cut reinforcement beams and for the replacement of the windshield, the 3 mm thick plate has been cut and deformed following the exact shapes and welded continuously in order to ensure a pure clamped boundary conditions. These additional metal sheets have been 3D modeled and integrated in the FEM model.
An experimental modal analysis has been carried with a scanning laser vibrometer in order to check if the modal dynamical behavior was correctly simulated by the bare cut out dash FEM model, as well as identify the experimental modal damping of the structure. The correlation

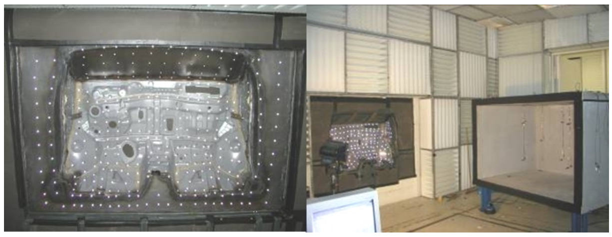

Figure 3: Bare dash panel experimental setup.

between measurement and simulation of the first modes up to 200 Hz is good, as illustrated figure 4.

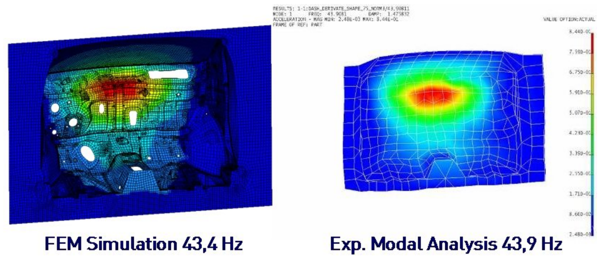

Figure 4: Modal analysis: first mode correlation.

The Bare dash panel structure is excited in Z direction by a shaker positioned at an engine mount location. The point to point correlations between measurements and simulations for the structural Velocity / Force Frequency Response Functions (FRF) are good up to 400 Hz, as illustrated figure 5.

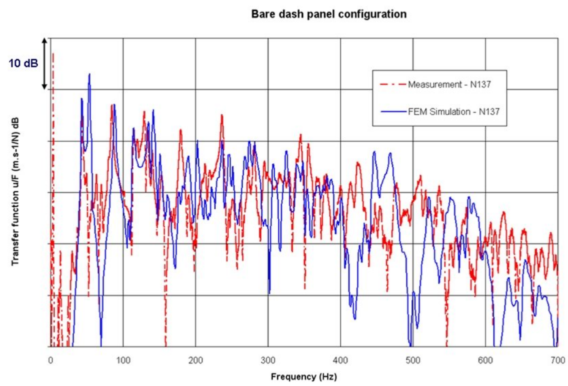

Figure 5: Structural u/F FRF correlation.

A movable concrete cavity is positioned in front of the dash cut out module fitting exactly with the supporting frame of the dash panel with 8 microphones positioned in two rows with non-coïncident positions in order to catch different modal behaviors in the cavity. The point to point correlations between measurements and simulations for the vibro-acoustic Pressure / Force FRF are good for all microphones up to 600 Hz, as illustrated figure 6.

3.2 Trimmed dash panel configuration

Two kinds of dash insulators have been modeled, an absorption type, sometimes called bi-permeable concept, with a 1250 g/m^2 compressed felt layer superposed on

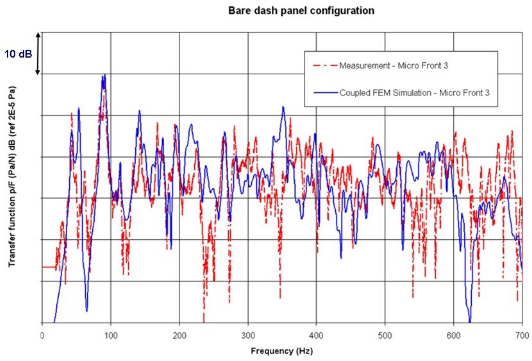

Figure 6: Vibro-acoustic p/F FRF correlation.

a soft spring felt layer and an insulation type with an $3,9\,kg/m^2$ heavy layer superposed on a felt spring layer. The Biot-Allard porous material parameters are the input data of the trim FEM model and have to be characterized carefully: σ airflow resistivity, ϕ porosity, α_∞ tortuosity, Λ and Λ' viscous and thermal characteristic lengths, ρ density, E young modulus, η damping loss factor and ν poisson ratio. All these parameters have to be characterized for each thickness, especially for thermoformed felt materials, where the Biot-Allard parameters change drastically with density.

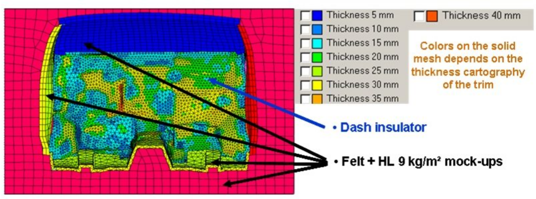

Figure 7: Dash trim FEM model with mock-ups.

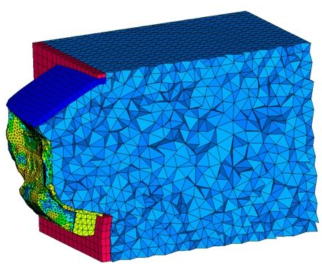

Figure 8: Complete Dash trim FEM model with cavity.

An important characterization effort have been carried out here, by measuring more than 40 samples per insulator type, in order to get Biot parameters every $5\,mm$. Using a thickness cartography of the insulators and the resulting $5\,mm$ thickness distribution, the quadratic tetrahedral elements have been automatically given the corresponding Biot material card (cf. figure 7). In some areas, we have identified air gaps between the trim and the structure. Some tests with the coupling distances have shown that the best modeling technique was to integrate these air gaps directly in the trim complex domain.
Despite the use of incompatible meshes for the structure, the trim and the cavity (cf. figure 8), the size of the problem remains big: 46662 nodes meaning 186648 dofs for the insulation type dash and 76033 nodes meaning 304132 dofs for the absorption type dash. For the trimmed dash panel configuration, four accelerometers have been positioned on the structure in the emission side in order to measure the damping induced by the trim. Additional particle velocity measurements have been carried out at the surface of the trims using p-u probes (for the absorption type trims) and compared to a scanning laser vibrometer (only in the case of insulation type trims). The fully trimmed dash vibro-acoustic point to point FRF results presented figure 9 and figure 10 are quite good up to $600\,Hz$ for both acceleration on the structure and pressure in the cavity for the bi-permeable insulator. This is the same microphone position plotted here figure 10 as for the bare configuration figure 6 and one can clearly see the actual noise reduction and smoothing of the resonance peaks induced by the dash insulator.

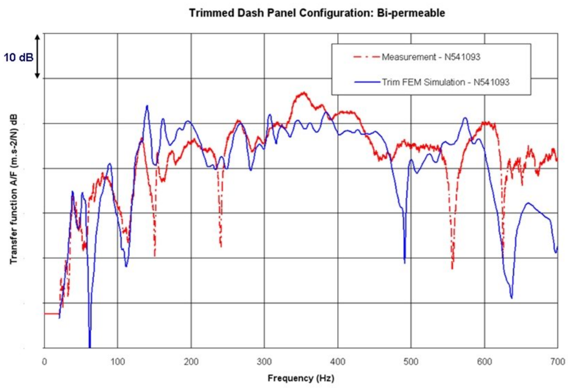

Figure 9: Dash trim FEM A/F FRF correlation.

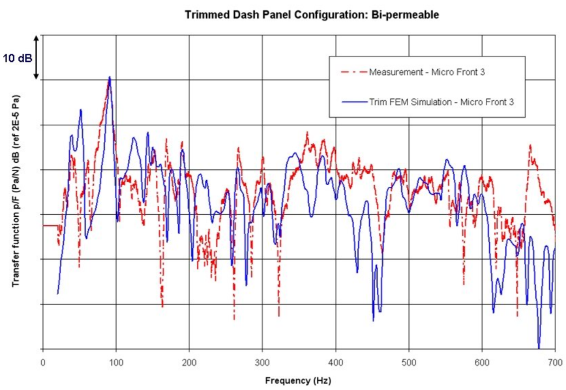

Figure 10: Dash trim FEM p/F FRF correlation.

4 Floor trim FEM model

4.1 Bare floor panel configuration

The same kind of boundary and mounting conditions as for the dash cut out module has been applied for the floor module, except that it has been fixed in the horizontal separation wall of the coupled reverberant rooms (cf. figure 11).

Figure 11: Bare floor panel experimental setup.

The Bare floor panel structure is excited in Z direction by a shaker positioned at a front reinforcement beam location. The point to point correlations between measurements and simulations for the vibro-acoustic Pressure / Force FRF are very good for all microphones up to 700 Hz, as illustrated figure 12.

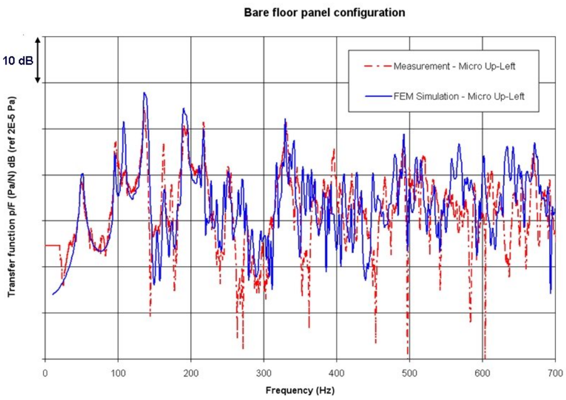

Figure 12: Vibro-acoustic p/F FRF correlation.

4.2 Trimmed floor panel configuration

Two kinds of floor insulators have been modeled, an absorption type, sometimes called bi-permeable concept, with an airflow resistive densified binappe carpet superposed on a soft spring felt layer and foam spacers, and an insulation type, with an 2 kg/m^2 heavy layer behind the binappe carpet superposed on a felt spring layer and foam spacers (cf. figure 13).

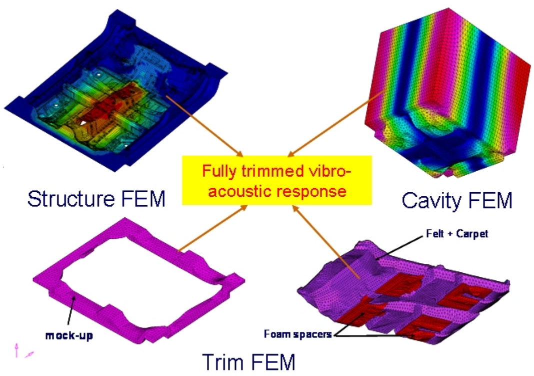

Figure 13: Complete floor trim FEM model with cavity

The point to point vibro-acoustic correlation results between measurements and simulations presented figure

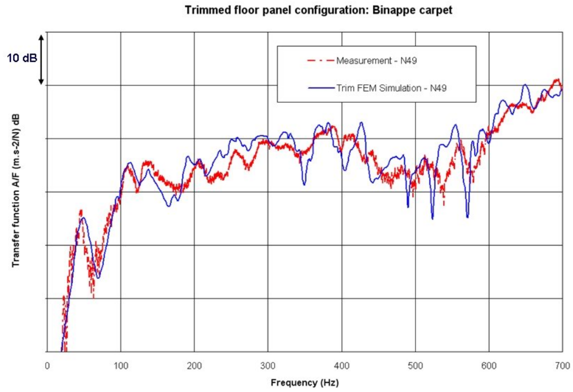

Figure 14: Floor trim FEM A/F FRF correlation.

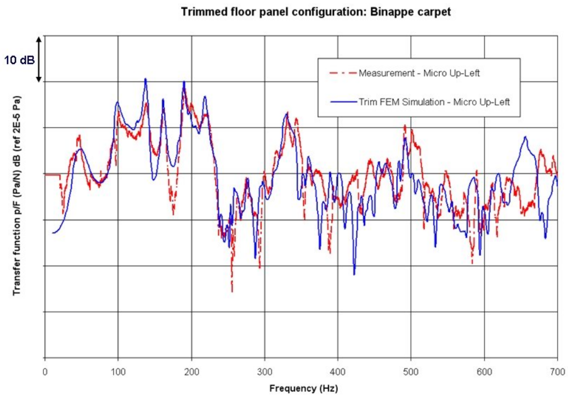

Figure 15: Floor trim FEM p/F FRF correlation.

14, figure 15 and figure 16 for the trimmed floor are very good for both absorbing and insulating material concepts configurations up to 700 Hz. In fact, the correlation is a little bit better for the absorption concept upper than 350 Hz for the pressure, because we should have modeled very thin decoupling air gaps between the felt and the heavy layer (like recent systematic investigations on flat samples have demonstrated). These correlations are the best results of this study and show the good prediction of the acoustic transmission (radiation efficiency) through the trims computed by trim FEM simulation, as well as the good prediction of the damping induced by the trims to the structure. It proves as well the importance of good coverage and clear contact between the trims and the structure (situation which is better here than for the dash).

4.3 Transmission Loss of the trimmed floor module

A Transmission Loss model have been built using the same floor trim FEM model, by adding an emission and reception BEM mesh instead of a cavity, excited by a diffuse field represented by 17 random incident planes waves (cf. figure 17 and the theory in reference [14]).
The Transmission Loss correlation results presented figure 18 are quite satisfactory up to 1000 Hz with the right insulation slopes, which is the real issue here.

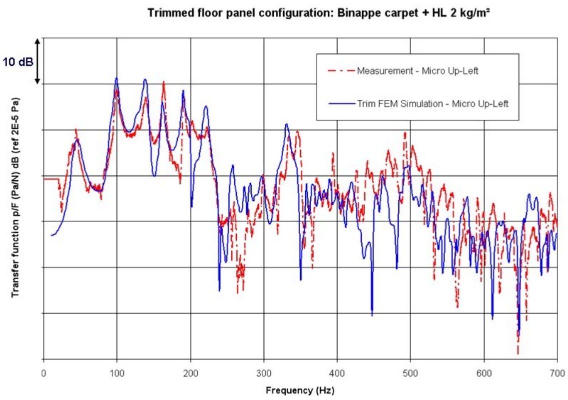

Figure 16: Floor trim FEM p/F FRF correlation.

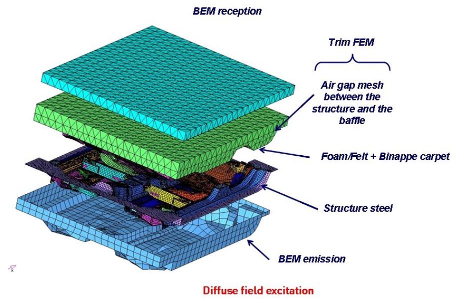

Figure 17: Floor trim FEM Transmission Loss model

5 Conclusion

The correlation results between measurements and simulations for both bare and trimmed configurations are quite good for the industrial cases of a dash insulator and floor carpet insulator cut out modules for both structureborne and airborne excitations in the low and middle frequency range up to 700 Hz. In fact, this quality remains for absorption or insulation acoustic material concepts and proves that the complex three dimensional coupling phenomena are well reproduced here.

The original mounting conditions of the cut out modules have proved to be innovative here, as well as the modeling and integration of the mock-ups as additional trims, in order to validate separately the trims NVH performance. This procedure could thus become a new standard for fully trimmed vehicle developments as an intermediate step, not only for the middle and high frequency range, but also for the low frequency range.

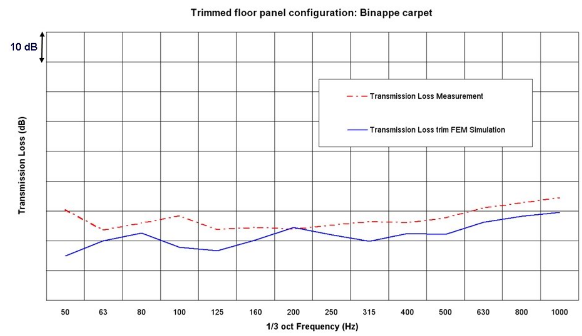

Figure 18: Trim FEM Transmission Loss correlation

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