Modeling the Vibro-Acoustic Effect of Trim on Full Vehicle and Component Level Analysis

D. Blanchet[1], M. Anciant[2], L. Mebarek[2]

[1] *ESI GmbH, Munich, Germany, Email:dbl@esigmbh.de*

[2] *ESI Group, Compiègne, France, Email: man@esi-group.com,lme@esi-group.com*

Abstract

In the automotive industry, the influence of poro-elastic components on acoustic comfort has been mostly investigated for airborne noise at mid- and high frequency ranges; however, due to the lack of adequate theoretical formulations, the influence of poro-elastic in numerical vibro-acoustic simulation at lower frequency range has often been ignored or simplified by the use of distributed spring/mass on the BIW structure and impedance on the acoustic medium. In the last few years, new theoretical developments contributed to overcome this limitation by providing an efficient FEM formulation for poro-elastic material modelling. This FEM approach, implemented in VTM (Vehicle Trim Modeller) software developed by ESI-Group, enables the computation of the coupled response of a fully trimmed vehicle by taking into account the BIW structure, the acoustic cavity and the poro-elastic components (carpet, dash insulator, headliner, seats…). This paper presents theoretical background and industrial application examples using these new developments.

Introduction

Over the last few years, a great deal of work concerning the modelling of trim in FE has been published. The development of a new (u,p) formulation for the representation of trim in FE has provided researchers and engineers the opportunity to investigate further the physics involved with full vehicle and component trim modelling. This paper summarizes the theoretical foundations of FE trim modelling as implemented in VTM. It also provides references to a wide range of industrial case studies from component TL to full trimmed vehicle simulation, as well as pillar fillar modelling and new trim concept design.

Trim FE modeling vs TMM

Duval and al. have shown in [1] that: " … the TMM (Transfer Matrix Method) was giving excellent results for flat samples with spatial windowing but was not at ease with curved shapes due to the decrease of Insertion Loss slopes of insulators with curvature [2][3]. First 3D investigation 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 [4].

Trim FE modelling vs Spring/Mass/Dashpot

It has been observed that the spring/mass/dashpot approach to represent the trim interaction with the structure is of limited value since it is not a predictive model, the properties of the spring/mass/dashpot having to be tuned to provide proper accuracy. The frequency domain where this approach is successful is well below 100 Hz. Trim FE modelling can represent the physics from 0 to 500 Hz for full vehicle and up to 700Hz for structureborne coupled vibro-acoustic response on a sub-assembly such as a floor and up to 1000 Hz for airborne transmission loss of sub-assemblies [1].

Hilbrunner and al. [5] have observed while studying pillar fillers that:"The efficiency of open cell or semi-closed foam are strongly due to the Biot parameters. That is why a simple mechanical model (without Biot parameters) is not accurate enough to represent the physical phenomena". Up to 20db variation between mass and Biot approach on the majority of the frequency bands between 100 to 2000 Hz for a simple tube and cavity setup was observed.

Industrial considerations

To be of value, a solution such as VTM has to be able to handle industrial class of problems. "Today (2005), the size of an FE model corresponding to a small car body in white (BIW) is ranging from 500,000 to 750,000 elements. When doors close the BIW, and equipments (such as the dash-board) are introduced the model size can reach 1,000,000 to 2,000,000 elements … It is now possible to realize calculations of vibration and acoustic frequency transfer responses up to 500 Hz for FE model sizes ranging from 500,000 to 1,000,000 elements…The proposed system approach facilitates data exchanges between carmakers and suppliers. By using incompatible meshes, supplier can totally ignore the FE model of the vehicle"[6].

Theoretical background

Porous material modelling

As described in references [7][8][9], propagation of elastic and acoustic harmonic waves, with an $e^{-i\omega t}$ time dependency, in porous elastic media is governed by the following system of modified Biot's equations:

$$\tilde{\rho}_s\, \omega^2 U + div\left(\sigma^s_{kl}(U) - \tilde{\alpha}\phi\, p\,\delta_{kl}\right) + \ldots \\ \ldots \tilde{\beta} grad(\phi p) = 0 \tag{1}$$

$$div\left(\frac{1}{\omega^2 \tilde{\rho}_f} grad(\phi p) - \tilde{\beta} U\right) + \frac{\phi p}{R} + \ldots \\ \ldots \tilde{\alpha}\, div(U) = 0 \tag{2}$$

Where:

The vector U represents the skeleton displacement, σ_{kl}^{s} the components of the stress tensor in the skeleton and p the acoustic pressure. ω is the angular frequency, Φ is the porosity and $\tilde{\rho}_{s,}\ \tilde{\rho}_{f}$ are respectively the skeleton and fluid equivalent mass densities, which are related to the real mass densities $\rho_{s,}\ \rho_{f}$ of the structure and fluid by:

$$\tilde{\rho}_s = (1-\phi)\rho_s + \phi\rho_f\left(1-\frac{\rho_f}{\rho_e}\right) \qquad (3)$$

$$\tilde{\rho}_f = \phi\rho_e \qquad (4)$$

Where ρ_e is the effective mass of the interstitial fluid given in reference [9]. Equation 5 represents the inertial coupling factor coefficient and equation 6 represents the stiffness coupling factor

$$\tilde{\beta} = \frac{\rho_f}{\rho_e} \quad , \quad \phi\tilde{\alpha} = 1-\frac{K_b}{K_s} \qquad (5,6)$$

The coefficient R represents the bulk modulus of the porous elastic media.

$$R = \frac{K_s\phi^2}{1-\phi-K_b/K_s+\phi K_s/K_e} \qquad (7)$$

Coefficients K_b , K_s represent respectively the bulk modulus of the skeleton with vacuum inside and of the material of the skeleton, and finally K_e represents the effective bulk modulus of the interstitial fluid as given in reference [9].

Biot parameters identification

At the beginning of the decade few engineers had the opportunity to work with Biot parameters. Then, only a handful of labs were able to provide these values based on extensive sophisticated measurements. Fortunately, todays situation has drastically changed. Biot parameters can be identified using an indirect method. Based on a simple impedance tube measurement, Biot parameters can be calculated using different optimization algorithms [10]. These identified Biot parameters are the intrinsic properties of the poro-elastic properties because only one set of parameter values is possible in the solution. To fully characterize foam type material extra properties such as the Young’s modulus and Damping of the foam structure is need. These are easily obtained from a quasi-static mechanical test.

Full vehicle modelling

As described in [6], the trim of a full vehicle analysis can be added to the classical structure/fluid coupled linear system as a trim impedance matrix $\tilde{Y}$.

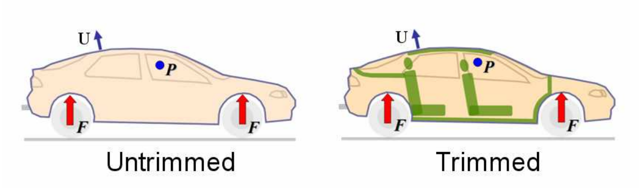

Figure 1: Untrimmed and trimmed configuration

The dynamic equation of the trimmed vehicle can be written in the following form:

$$\left[\begin{pmatrix} Z_s & C_{sc} \\ C_{sc}^t & A_c \end{pmatrix} + \begin{pmatrix} \tilde{Y}_{ss} & \tilde{Y}_{sc} \\ \tilde{Y}_{sc}^t & \tilde{Y}_{cc} \end{pmatrix}\right]\begin{bmatrix} U \\ P \end{bmatrix} = \begin{bmatrix} F \\ Q \end{bmatrix} \qquad (8)$$

Where Z_s is the mechanical impedance of the master-structure (car body in white), A_c is the acoustic admittance of the internal cavity. C_{sc} is the surface coupling operator between the untrimmed master-structure surfaces directly in contact with the internal acoustic cavity.

U is the displacement field vector of the master-structure, P the pressure field of the internal cavity; F the external force field applied to the master-structure, and Q represents internal acoustic sources. The matrix $\tilde{Y} = R'YR$ is the transferred impedance matrix of the porous component where R is the transfer operator relating the degrees of freedom of the porous component to the degrees of freedom of the master structure and of the internal cavity,

Linear system equation (8) is solved using structural and acoustic normal modes. This has the great advantage of keeping the trimmed linear system to be solve the same size as the initial BIW linear system.

Transmission Loss modelling

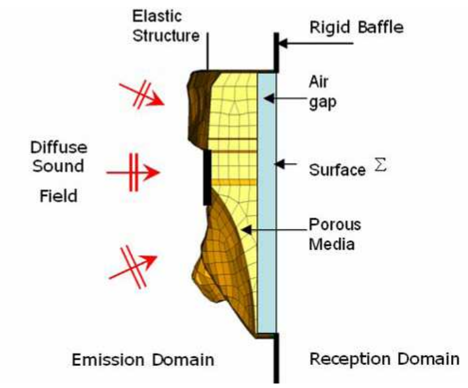

Figure 2: Diagram showing a transmission Loss setup for a trimmed dash panel

As described in [11] the implementation of the Transmission Loss (TL) computation is based on FEM-BEM method allowing calculation of the acoustic TL of a double wall trimmed component of arbitrary shape, involving in particular the use of poro-elastic materials (foams, fibers, etc..) between structural outer panels with or without air gaps (see Figure 2).

Theory used for TL modelling is based on displacement-pressure formulation (u,p) proposed by Atalla et al. [7][8]. This formulation has been extended to study vibro-acoustic problems involving interactions between a master structure, a fluid cavity and trim components made of poro-elastic material. This formulation has been implemented into VTM and details of this formulation are given in [12][13].

Complex industrial applications require the modelling of interactions between porous media and infinite fluids of the emission and reception rooms. The method proposed consists in adding an air gap between the porous media and the infinite fluid. On the coupling surface Σ separating the air gap and the external fluid (see Figure 2), two boundary conditions are defined: a) continuity of pressure : $p^g = p^e$ and b) continuity of normal displacement: $u_n^g = u_n^e$. Symbols g and e denote respectively the air gap and the external fluid. The weak formulation of the porous media including the air gap is given by:

$$\begin{aligned} &Z(U,V) + A(p,q) - \hat{C}(p,V) - \hat{C}(q,U) = \ldots \\ &\ldots \tilde{C}_s(T,V) + C_s\left(q,\phi\left(U_n^f - U_n\right)\right) \end{aligned} \quad (11)$$

Where, Z represents the mechanical impedance of the skeleton, A represents the admittance of the interstitial fluid and the air gap, $\hat{C}$ represents volume coupling terms between the skeleton and the interstitial fluid, $\tilde{C}_s$ represents surface coupling term between the porous media and the elastic structure, C_s represents surface coupling terms between the porous media, the air gap and the external fluid. The use of this formulation combined with integral representation of pressure in the emission and reception domains and the use of the boundary conditions defined above enables the creation of a linear matrix system that yields Transmission Loss values of the component. The full theoretical background can be found in [11][14].

Applications

Full vehicle analysis

Hamdi and al. have studied in [6] the damping effect of trim on cavity modes and panels vibrations: “Due to trim parts, FRF reaches 10dB attenuation even in the low frequency range, which illustrate the strong coupling of porous elastic components with car-body structure. Correct modelling of the interaction aspects is crucial for the vibro-acoustic analysis in low and medium frequency ranges.

Anciant and al. in [15] studied further correlation between test and simulation of a trimmed vehicle and investigated panel contribution: “For both [driver’s and passenger’s ear], a good agreement is observed between test and analysis pressure levels up to 400Hz. Resonance and global damping of acoustic responses is well reproduced by the simulation. This shows that VTM is able to correctly model the modification of the coupling between the structure and the cavity, due to the trim introduction in the coupled system …The good agreement between simulation and measurement results, demonstrate the capabilities of VTM software to predict accurately the coupled response of a full trimmed vehicle up to 500Hz, in a reasonable amount of time (≈ a night), by using a frequency distributed approach on a 4 multi-processor computation server”.

Efforts have been made to further reduce computation time and memory usage of full vehicle models by using partitioning technique of trim component. This method reduces memory usage but computation time is not significantly affected [16][17].

Transmission Loss

New trim construction design are created based on FE trim simulation such as in [18] by Monet-Descombey: ”First, new and classical soundproofing concepts are applied on the inner dash insulator and compared experimentally and numerically with a global performance criterion including automotive conditions … a finite element model of a fully trimmed car is used to validate the new concept. Thanks to this work, a three layered component, better performing and lighter than a classical two layers solution, has been validated by the two industrial partners…Both experimental mock-up and a numerical model allow the study of the influence of pass through trimming and the presence of the instrument panel. [VTM] is used to predict the vibro-acoustic response of the fluid-structure system including poroelastic parts and to test and analyze 2L- and 3L-trimming configurations”.

Zhang et al. have used FE modelling of the trim “…in order to predict efficiently the behaviour of the poroelastic parts (foam and felt) of the multilayer components. The numerical simulation not only reproduces the physical phenomena observed experimentally but also quantifies the influence of different factors. This comprehension leads to an optimal design to increase the acoustic performance with a light weight concept”[19].

Duval and al. have been working for quite some time on the modelling of trimmed flat and curved panels and made comparisons between meticulous tests and simulations method finally understanding the physics behind the peculiar TL values measured for trimmed curved panels: “We have been observing experimentally for years on real industrial cases, that the Insertion Loss slopes of “mass-spring” 3D shaped insulators, like foam-heavy layer systems, were much lower than the classical 12 dB / oct flat sample slopes. Indeed, a three dimensional coupling occurs between the curved steel structure and the heavy layer, through the foam or felt leading to 8 or 9 dB /oct Insertion Loss slopes typically. Unlike the Transfer Matrix Method which simulates exclusively the Insertion Loss of flat samples and cannot represent the 3D behavior of the insulator even combined with thickness cartographies, trim FEM simulation captures the 3D coupling phenomena correctly for both a simplified trimmed half cylinder case and a complete trimmed floor module in the low and middle frequency range. This Transmission Loss study shows that poroelastic FEM simulation addresses the physics properly in the low and middle frequency range for both a simplified trimmed curved and flat panel case, as well as a complete floor module with airborne excitation up to 1000 Hz (IL 9 dB/oct as measured); whereas the Finite Transfer Matrix Method, even with thickness 3D maps, is in difficulty not only in terms of slope after the respiration frequency (IL=12 dB/oct instead of 9 dB/oct) but also in terms of level. [20]

Expandable foam in pillars

Hilbrunner, Zhang, Wojtowicki studied in [5] the influence of a pillar filler in a simplified setup consisting of a concrete cavity and a pipe attached at the top. Different material were placed in the tube completely or partially filling the tube at the sample location. Also developed in this study is a new material model for closed cell foam: "… proposed a 5 parameters model with two different parts [21][22], see figure 13. …The 5 parameters are then adjusted to be used in the Biot model …VTM (FEM coupling formulation) is used to compute the vibro-acoustic simulation of the frame-cabin-sealing assembly up to 2000 Hz. The comparison between numerical and experimental results shows a good correlation for each pillar filler and validates the whole numerical model allowing further parametric simulations".

Conclusion

The FEM representation of the trim using Biot parameters is well appropriate at low to mid-frequency and in some cases critical in properly representing the physics of the interaction between a structure, a cavity and several trims.

References

[1] A. Duval, N. Kobayashi et al. "The FEM simulation of a dash and floor insulator cut out modules with structure-borne and airborne excitations" Euronoise 2008, Paris.

[2] A. Duval and al. "Faurecia vehicle acoustic synthesis method: A hybrid approach to simulate interior noise of fully trimmed vehicles", Congrès SIA Confort automobile et ferroviaire 2004, Le Mans.

[3] A. Duval and al. "Vehicle acoustic synthesis method 2nd generation: an effective hybrid simulation tool to implement acoustic lightweight strategies" Journées SFA/Renault/SNCF, Guyancourt, 2005

[4] A. Duval, L. Dejaeger and J. Baratier. "Structureborne and airborne Insertion Loss simulation of trimmed curved and flat panels using an (u,p) poroelastic FEM formulation" ESI Group VA User Conf. 2007, Cologne.

[5] O. Hilbrunner, C. Zhang, J-L. Wojtoviky, "Modeling the vibro-acoustic behavior of a simplified car frame and cabin assembly by predict the overall efficiency of pillar filler parts", Automobile Comfort Conference, November 15-16 2006, Le Mans, France

[6] Hamdi M.A., Zhang C, Mebarek L., Anciant M. and Mahieux B. "Analysis of vibro-acoustic performances of a fully trimmed vehicle using an innovative sub-system solving approach facilitating the cooperation between carmakers and sound package suppliers", Eurodyn 2005.

[7] N. Atalla, R. Panneton and P. Debergue, " A mixed pressure-displacement formulation for poroelastic materials " J.A.S.A., 104(3), 1444-1452, (1998).

[8] N. Atalla, M.A Hamdi, R. Panneton,"Enhanced weak formulation for mixed (U,p) poroelastic equations" J.A.S.A., 109(6), (2001)

[9] J.F Allard: "Propagation of sound in porous media, Modelling of sound absorption", Charpman & Hall,North Way Andover, Hampshire, SP105BE, England , (1993).

[10] Panneton R., Atalla Y., Blanchet D., Bloor M., Validation of the Inverse Method of Acoustic Material Characterization, SAE2003, 2003-01-1584

[11] A. Omrani, L. Mebarek and M.A. Hamdi, "Transmission Loss Modeling of Trimmed vehicle Components", ISMA, (2006)

[12] M.A. Hamdi, L. Mebarek, A. Omrani, N. Atalla, M. Fortez, G. G. G. Crignon, S. Lullier, "An efficient Finite Element Formulation for the analysis of Acoustic and Elastic Waves propagation in Sound Packages". SAE 2001, 01 NVC -165.

[13] C. Zhang, M.A. Hamdi, L. Mebarek & B. Mahieux, « Influence of porous elastic components on structure and air borne noise in low and medium frequency ranges », Reiter Automotive Conference June5th-7th (2005)

[14] A. Omrani, D. Blanchet and Hideki Niwa, "Theoretical Foundation for the Modeling of Transmission Loss for Trimmed Panels". Presented at 2007 JSAE Congress.

[15] M. Anciant, L. Mebarek, C. Zhang, J. Monet-Descombey, "Full trimmed vehicle simulation by using RAYON-VTM", . Presented at 2007 JSAE Congress.

[16] J. Monet-Descombey,C. Zhang, M.A.Hamdi, "A partition finite element method allowing the calculation of the vibro-acoustic response of fully trimmed vehicles in medium frequency range", presented at Internoise 2008, Shanghai, China

[17] J. Monet-Descombey, O. Thuong, C. Zhang, M.A.Hamdi, "Presentation of an efficient method for solving large coupled vibro-acoustic systems including porous elastic components", presented at ISMA2008

[18] J. Monet-Descombey, C. Zhang, F. Lhuillier, A. Duval, « A new light weight concept to enhance both absorption and insulation in passenger cars : A numerical and experimental joint study », SIA 2008, Le Mans, France

[19] C. Zhang, O. Thuong, F. Lhuillier, A. Duval, "Numerical and Experimental Study of Light Weight Concept under Automobile Applications Conditions", 5th international Styrian NVH Congress (2008)

[20] A. Duval, L. Dejaeger, J Baratier, J-F.Rondeau, "Structureborne and airborne Insertion Loss simulation of trimmed curved and flat panels using Rayon-VTM-TL: implications for the 3D design of insulators " Automobile and Railroad Comfort – 2008, Le Mans

[21] R. Panneton & J.-L. Wojtowicki: "Improving the efficiency of sealing parts for hollow body network", SAE conference, 2005-01-2279.

[22] C. Zhang, O. Hillbrunner & J.-L. Wojtowicki: "Modeling the vibro-acoustic behavior of a simplified car frame and cabin assembly to predict the overall efficiency of pillar-filler parts", ESI Group EuroPAM 2005 (Potsdam, Germany).