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Theoretical Foundation for the Modeling of Transmission Loss for Trimmed Panels

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In the automotive industry, the influence of poro-elastic components on acoustic comfort has been mostly investigated for air borne noise at mid and high frequency ranges. However, due to the lack of adequate theoretical formulations, the influence of poro-elastic material in numerical vibro-acoustic simulation at lower frequency range has often been ignored or roughly approximated by the addition of distributed spring/mass on the BIW structure. This paper presents the theoretical formulation for the calculation of the acoustic Transmission Loss (TL) of simple or double wall trimmed panels. The poro-elastic materials are described by a FEM model describing geometry and the intrinsic properties of the trim: the BIOT parameters. This method provides an efficient way of building predictive models of a trimmed structure. The resulting Transmission Loss module is implemented in ESI-GROUP software solutions. Numerical results are compared with other software tools available on the market and with experimental data.

Key Words: Trim, Sound Package, FEM, BEM, TL,Transmission Loss, Biot,

1．INTRODUCTION

Increasingly car manufacturers use numerical simulation methods such as Finite and Boundary Element Methods (FEM and BEM) for the prediction of the vibro-acoustic performance of car components. The main objective of such vibro-acoustic studies are the design of effective trim for the reduction of internal noise for improved passenger comfort. To reach this objective, car manufacturers want to optimize the vibro-acoustic performance of a vehicle at an early stage of the design cycle therefore reducing long and costly tests on real prototypes. For example, transmission of sound by structures is of great interest for the transportation industry. The part of the structure (the specimen) to be tested is mounted on a hard wall commonly called baffle separating a reverberant room (source) and an anechoic room (receiver) and excited by acoustic sources placed in the reverberant room. The Transmission Loss (TL) is defined by the logarithmic ratio between the incident acoustic power and the transmitted acoustic power to the receiver room and is used as an indicator of the capability of the component to reduce transmitted noise. This corresponds to the standard test well known by acousticians for the qualification of windows or other structures.

To respond to the demand of car manufacturers and suppliers, ESI-Group developed a TL Module resulting from the recent industrial software development done in cooperation with RENAULT [4]. The latter development called VTM is capable of handling vibro-acoustic simulation of a fully trimmed vehicle in a highly automated process. These capabilities have been recently extended to allow the modeling of transmission problems of sandwiched panels composed by different layers of poro-elastic materials and air gaps.

The numerical method implemented within the TL Module to calculate the Transmission Loss (TL) of trimmed structural panels make use of 2 existing module at ESI-GROUP:

1. PEM module which uses a Finite Element Method to solve the modified Biot's equations and to calculate mechanical impedance of trim
2. BEM module which uses a Boundary Element Method (BEM) to calculate the acoustic loading induced by the incident acoustic excitation (plane

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waves, diffuse field), the radiation impedance of the panel structure and the acoustic power radiated by the structure in the receiving media.

2 . THEORETICAL BACKGROUND

The problem of interest deals with the prediction of transmission loss of a multi-layer sandwiched panel schematized in figure 1 and placed in a rigid infinite baffle. The panel is submitted to an acoustic diffuse field composed by a set of uncorrelated plane waves with equal repartition in the source room or to a mechanical force. The acoustic and mechanical excitations are considered harmonic with time dependency of $e^{-i\omega t}$.

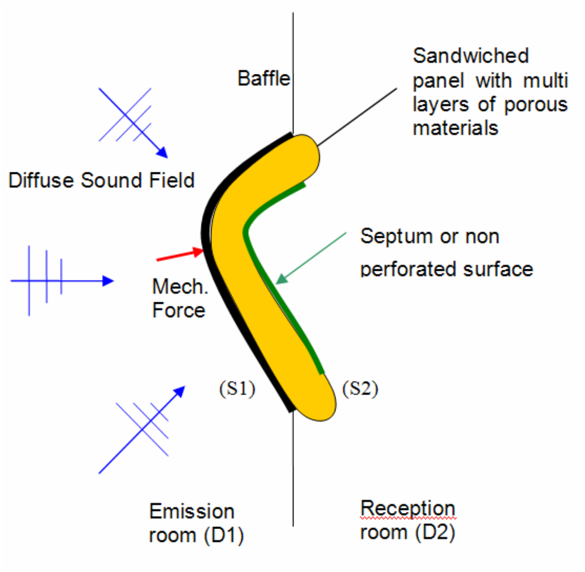

Figure 1: Sandwich panel with multiple layers of porous materials

2.1 Surface impedance matrix of a porous component

As detailed in [1], a dedicated mixed BEM-FEM formulation has been developed for the calculation of TL of multilayered structural panels. As shown in [1], the condensation of the internal degree of freedom (DOF) of the porous media leads to a surface impedance matrix added on the structure:

$$[Y]=[Z_{ss}]-[Z_{si} \quad C_s]\begin{bmatrix} Z_{ii} & C_i \\ C_i^t & A \end{bmatrix}^{-1}\begin{bmatrix} Z_{si}^t \\ C_s^t \end{bmatrix} \qquad (1)$$

2.2 Equation of the complete coupled system

The equation of the coupled fluid structure system is given by the following system [1]:

$$\begin{bmatrix} [Z_s+Y+A_1+A_2] & \left[\frac{1}{2}C_1-B_1\right] & \left[\frac{1}{2}C_2-B_2\right] \\ \left[\frac{1}{2}C_1^t-B_1^t\right] & [D_1] & 0 \\ \left[\frac{1}{2}C_2^t-B_2^t\right] & 0 & [D_2] \end{bmatrix}\begin{Bmatrix} W \\ P_1 \\ P_2 \end{Bmatrix}=\begin{Bmatrix} F-P_\infty \\ U_\infty^a \\ 0 \end{Bmatrix} \qquad (2)$$

Where,

- Z_s is the impedance matrix of the structure.
- Y is the surface impedance matrix of the porous media
- A_1, A_2 are pseudo added mass fluid matrices respectively of the domain D_1 and D_2
- C_1 and C_2 are the fluid structure FEM coupling matrices of the fluid domains D_1 and D_2
- B_1 and B_2 are the fluid structure BEM coupling matrices for the domain D_1 and D_2
- D_1 and D_2 are the admittance fluid matrices of the domain D_1 and D_2.

Finally the elimination of the fluid pressure DOF in the equation (2) leads to the following equation in terms of the structure DOF's only.

$$\left[Z_s+Y+Z_1^{rad}+Z_2^{rad}\right]W=F+F_{b1} \qquad (3)$$

Where Z^{rad}_i represents the radiation impedance matrix of the fluid domain number (i), given by

$$\left[Z_i^{rad}\right]=[A_i]-\left[\frac{1}{2}C_i-B_i\right][D_i]^{-1}\left[\frac{1}{2}C_i^T-B_i^T\right] \qquad (4)$$

and where F_{bl} represents the blocked acoustic load commonly called blocked pressure due to the plane waves exiting in the domain D1

$$F_{b1}=\left[\frac{1}{2}C_i-B_i\right][D_i]^{-1}U_a \qquad (5)$$

Equation (3) can be solved using normal mode projection technique, which have the great advantage of reducing the size of the final coupled system governing the dynamic behavior of the sandwiched panel. In this paper results are obtained by using the mode shapes of the body-in-white (BIW).

2.3. Radiated energies

Using the second Green's formula and taking into account the boundary conditions, the acoustic energy radiated in the reception domain, is given by the following formula:

$$E^{(2)}{}_{ray} = \frac{1}{2} \mathrm{Im} \int_{S_\infty^+} p^* u dS \qquad (6)$$

where S_∞^+ is an infinite hemisphere located in the right side of the baffle, is also given by the formula:

$$E^{(2)}{}_{ray} = \frac{1}{2} \mathrm{Im} \int_{S_2} p_2 w_n{}^* dS_2 \qquad (7)$$

2.4. Transmission Loss (TL)

Transmission Loss (TL) is the logarithmic ratio between the incident acoustic energy induced by the incident waves on the structure S1 in the source domain D1, and the acoustic energy radiated or transmitted by the structure S2 in the receiver domain D2. For an incident plane wave of amplitude P_i this leads to the formula,

$$TL(dB) = 10 \log(\frac{E_i^{(1)}}{E_{ray}^{(2)}}) \qquad (8)$$

where E_i is the incident energy per period corresponding to the incident plane wave. In that case the acoustic excitation is a diffuse field modeled by a set of plane waves statistically distributed in the emission space. The wave directions are equally probable and are defined by wave longitude and latitude. The diffuse sound field input energy is the sum of the input energy of each plane wave weighted by a function β_i,

$$E_i = \frac{S(\omega) S_a}{2 \omega \rho c} \sum_{i=1}^{Npw} \beta_i \qquad (9)$$

where S_a is a reference area, ω is the circular frequency, ρ the fluid mass density, c is the speed of sound and N_{pw} is the number of plane waves representing the diffuse sound field.

3. DESCRIPTION OF VALIDATION CASES

To validate the developed formulation for transmission loss (TL) calculation, two test samples of structural panels has been benchmarked. The first one is a flat double wall panel made of aluminum, with dimensions of 1,57m by 1,105m. The plate on the source side is 2mm thick and the plate at the receiver side is 1,6mm thick. The plates are 2cm apart and a foam is placed between them (Fig. 2 -left).

The plates of the second panel are separated by 5cm, with a superposition of two layers made of 2cm thick felt and an air gap of 3cm (Fig. 2 -right).

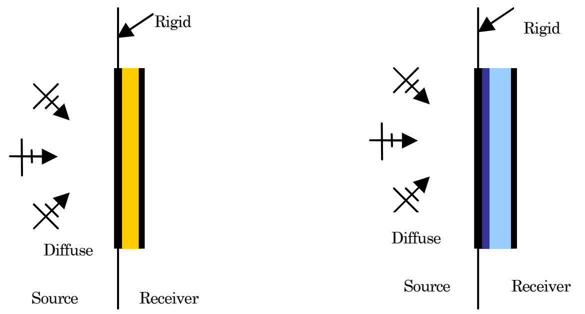

Figure 2: Plate-Foam-Plate (left) & Plate-Felt-Gap-Plate (right)

For these two examples the calculated TL is compared to results obtained by NOVA FEM (a flat sample FE prediction software) and experimental results.

To describe the trim, a FEM mesh is build which represents the actual geometry of the trim layer by layer. When poro-elastic materials are used, Biot parameters are associated to the 3D elements by the use of MAT cards commonly used in Nastran. Figure 3 shows a detailed view of the felt and air gap FEM mesh.

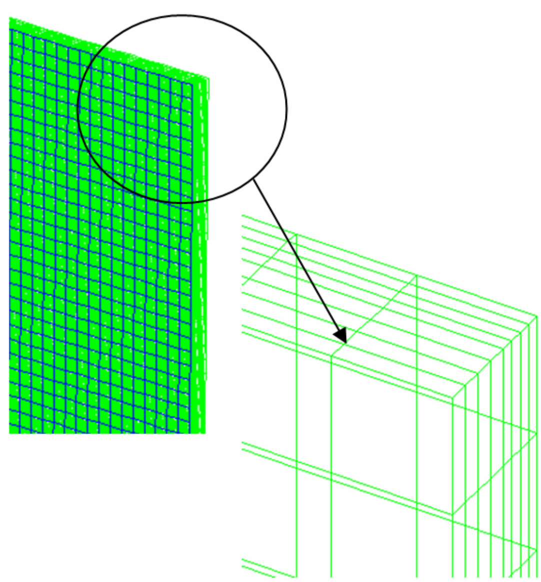

Figure 3: FEM mesh of Trim (Felt and Gap)

Table 1 shows the physical properties of poro-elastic media used in the validation test cases. Biot parameters can be derived from simple normal incidence impedance tube measurements and the use of commercial softwares such as FoamX. This makes the process of getting Biot parameters relatively easy, cheap and fast compared to individual testing of each parameters

Table 1: Physical property of porous media for test cases 1 & 2

	Foam	Felt
Fluid phase density	1.213 kg/m3	1.213 kg/m3
Fluid phase sound speed	342.2 m/s	342.2 m/s
Porosity	0.97	0.984
Resistivity	22098.	1564.
Tortuosity	2.2	1.01
Viscous characteristic length	3.9E-5	2.18E-4
Thermal characteristic length	2.75E-4	3.18E-4
Solid phase density	24.48 kg/m3	7.3 kg/m^3
Young Modulus	91500.N/m^2	0
Poisson Ratio	0.4756	0
Structural damping	0.11	0

4. VALIDATION RESULTS

The results for both test cases are presented in figure 4 and 5. Results predicted by the TL Module correlate well with other commercial software and also with test data.

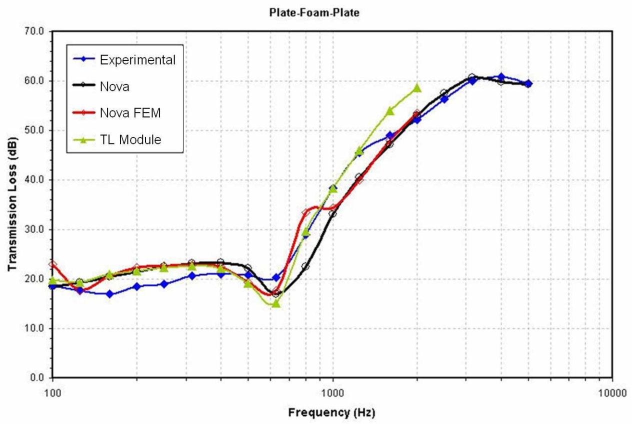

Figure 4: TL with TL Module (green), NOVA FEM (red) and experimental results (blue)

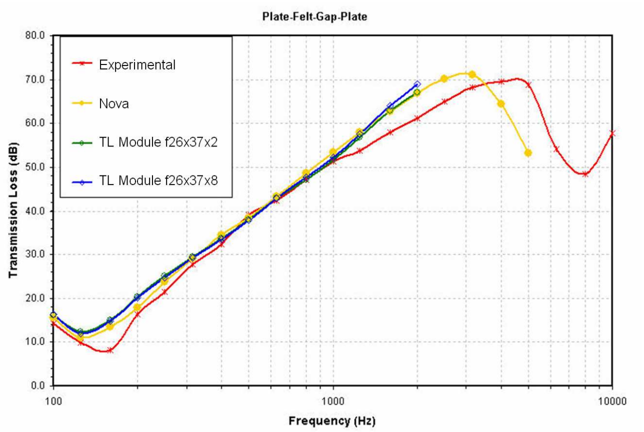

Figure 5: TL with TL Module (blue), NOVA FEM (yellow) and experimental data (Red)

Further investigations are underway to assess accuracy of the newly developed TL module for more complex and industrial structures such as the dash of a car.

5. CONCLUSION

This new TL module opens the way to the modelling of trimmed component in an efficient automated manner. The algorithms developed in VTM are reused in the automatic model setup and coupling between the different vibro-acoustic domains of the TL Module. Furthermore the computation of the trim impedance matrix allows the user to add trim to its current fluid/structure computation without adding DOFs. Finally, the combination of VTM and TL module offers the capability to design trim both on the system level and on the component level of the vehicle.

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REFERENCES

[1] A. Omrani, L. Mebarek and M.A. Hamdi, *"Transmission Loss Modeling of Trimmed vehicle Components"*. ISMA 2006, Sept. 18-20, 2006, Catholic University of Leuven-Belgium.

[2] Noureddine Atalla, Raymond Panneton and Patricia Debergue, "*A mixed displacement-pressure formulation for poroelastic materials*". 1998 JASA [S0001-4966(98)06008-1].

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

[4] 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.