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STUDIES REGARDING THE EFFECT OF THE SOUND ABSORBING MATERIALS ON THE SOUND PRESSURE LEVEL INSIDE THE SIMPLIFIED PASSENGER CABIN

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Abstract: *During a vehicle development, the acoustic comfort represents an important field of research for the car makers and often a way to differentiate between brands. The acoustic comfort depends on the multilayer materials from inside the passenger compartment. The acoustical performance of a material is evaluated by means of the sound absorption coefficient and the sound transmission loss. In the first part of this study, three acoustic treatments from inside Logan model are presented. In the second part, the Finite Element Method (FEM) was used for a vibro-acoustic analysis of a simplified passenger compartment. In order to simulate the sound pressure level (SPL) at the hearing point of the driver while the firewall panel is excited, the modal frequency response analysis has been performed with Radioss solver (a module of HyperWorks software package). To study the effect of the sound absorbing material on acoustic comfort, the sound pressure level inside the simplified passenger cabin with and without the acoustic treatment was investigated and analyzed by using Computer Aided Engineering (CAE).*

Key words: *poro-elastic materials, impedance tube, finite element analysis, fluid-structure interaction, acoustic analysis*

1. INTRODUCTION

The interior vehicle sound package can be divided into 8 sets (or composite structures) as follows: vehicle floor carpet, headliner, passenger seats, trunk cover, pillars, control board, doors and firewall. Each set represents a collection of the passenger cabin interior trim components being a multi-layered or sandwich construction that can contain the following materials: polyurethane foam, fiber, felt, porous film, fabric, plastic, metal sheet each of them serves a specific purpose.

The acoustical performance of these materials is evaluated by means of two important acoustic parameters: the sound absorption coefficient and the sound transmission loss, both depend on the frequency of the sound waves. At low frequencies, an isothermal process takes place, according to which the porous material absorbs sound by energy loss caused by heat exchange. The absorption represents the capacity of a material to dissipate the present noise in the passenger compartment. The insulation quantifies the capacity of a material to prevent the noise penetration in the passenger compartment.

The reduction of sound pressure level inside the passenger compartment is an actual problem of interest for automakers. There is a relation that provides the calculus for the global sound pressure reduction (ΔL) of a passenger cabin [2] as follows:

$$\Delta L = 10\lg\left(\frac{A}{A_0}\right) \tag{1}$$

where A represents the equivalent absorption area of the passenger cabin after the acoustic treatment and A_0 represents the equivalent

absorption area of the passenger cabin without the acoustic treatment.
The calculus relation for the equivalent absorption area (with / without attached sound absorbing materials) of the passenger cabin is:

$$A = \sum_{i=1}^{n} \alpha_i S_i \qquad (2)$$

where S_i is the area of the surface number i, that represents the surface of a panel and α_i is the absorption coefficient of the surface S_i.

In the automotive industry can be mentioned three types of trim part construction placed inside the passenger compartment of a car (see Figure 1). The three types are as follows [9]:

- *Insulative type*: consisting of a steel panel on which is superimposed a porous material and on top a heavy impervious layer;
- *Absorptive type*: consisting of a steel panel on which is superimposed a porous material and on top a compressed porous material;
- *Insulative and absorptive type*: which has the same structure as the absorptive type but between the compressed porous material and the porous material there is a heavy impervious layer.

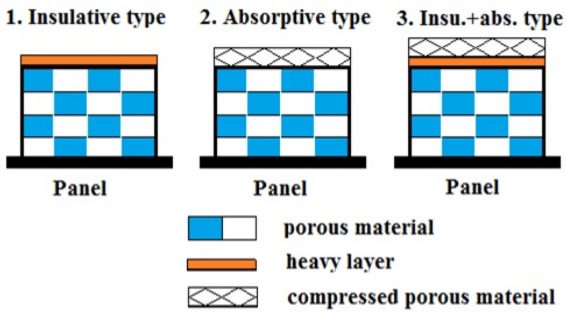


Fig. 1. Types of constructions for multi-layered trim constructions

According to Biot's theory the porous materials are divided into two phases: a solid phase known as the skeleton and a fluid phase that is normal air in pores, [6] (Figure 2). There are different types of porous materials: fiber, fabric, foam and vegetable binders, used as passive noise – control materials. It is very important to study the acoustical performance of these materials in order to know their proper location inside the passenger compartment.

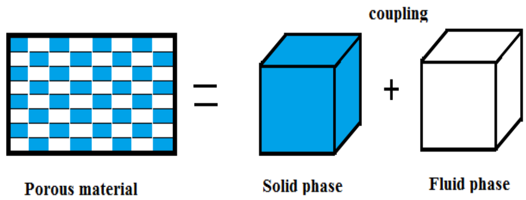


Fig. 2. Composition of porous material

A porous material is defined by eleven parameters. Two parameters define the fluid phase: the fluid density and the speed of sound. Five parameters describe the coupling between the fluid phase and the skeleton: porosity, flow resistivity, tortuosity, viscous and thermal characteristic length. The solid phase is defined through four parameters: solid phase density, Young modulus, Poisson coefficient and frame structural damping factor. The detailed description of the meaning of these parameters and measurement techniques are outlined in [1], [16], [12].

The main purposes of this study are as follow: presentation a list of Logan trim parts; determination of the absorption coefficient by using the impedance tube; modeling a coupled acoustic-structure system, that represents a simplified passenger compartment, by means of finite element method; performing an acoustic analysis to simulate the sound pressure level at the driver's ear location for the fluid-structure FE model without acoustic treatment; modeling with finite elements and attaching the sound absorbing material to the floor panel; comparison the sound pressure level at the driver's ear position for the FE model with/without acoustic treatment.

2. DETERMINATION OF THE ABSORPTION COEFFIECIENT AND NORMAL SURFACE IMPEDANCE BY USING IMPEDANCE TUBE

2.1 The Logan acoustic treatments

Normally, to reduce the sound pressure level inside the passenger compartment, a vehicle requires a good balance of the insulation and absorption provided by the acoustical trim parts.

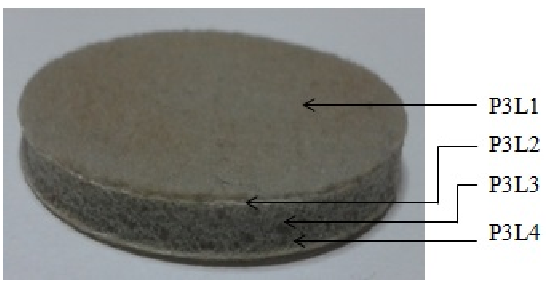
Table 1.

The Logan trim parts

Samples	*Nomenclature*	*Structure*	*Layer label*	*Layer Thickness*	*Estimated total thickness*
1	floor carpet	nonwoven polyester fabric	P1L1	2 *mm*	13 *mm*
		polypropylene extruded with latex	P1L2	1 *mm*	
		cotton felt bonded with epoxy – PES resin	P1L3	10 *mm*	
2	luggage carpet	nonwoven polyester fabric	P2L1	3.5 *mm*	5 *mm*
		polypropylene extruded with latex	P2L2	1.5 *mm*	
3	headliner	nonwoven polyester fabric	P3L1	1 *mm*	7.5 *mm*
		Fibreglass with polyurethane adhesive	P3L2	0.5 *mm*	
		polyurethane foam	P3L3	5 *mm*	
		polyester fabric	P3L4	1	

a)

b)



c)

Fig. 3. The acoustic treatments: a)floor carpet, b) luggage carpet, c) headliner

The acoustic treatment consists of the multi-layer structure. Three Logan acoustic treatments were under investigation (see Figure 3). These are the trim applied to floor panel, the carpet from luggage compartment and the headliner. The Table 1 illustrates the features of each trim part.

2.2 Impedance tube

The measurement method is based on the transfer function method, which represents a complex ratio between the output and the input of a system, according to ISO 10534-2. In Figure 4 a schematic view of impedance tube is presented. At one end of the tube is mounted a sound source (loadspeaker) attached to a signal generator that generates the incident waves inside the tube. The other end of the tube is blocked by a mobile support where the test sample will be positioned. When the incident plane wave impinge the test sample, a part of incident sound energy is absorbed by the sample and the rest is reflected off the surface. The two microphones mounted near to the test sample in the tube wall measure sound pressure at two fixed location.

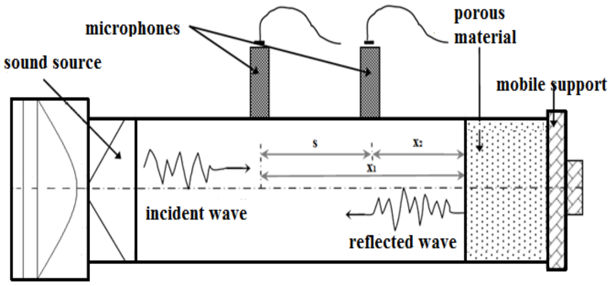


Fig. 4. Schematic view of impedance tube

The transfer function between the two microphones is obtained by means of a two-channel digital frequency spectrum analyzer.

The acoustic pressures of the incident wave p_I and the reflected wave p_R are, respectively:

$$p_I = p_I^0 e^{jk_0 x} \tag{3}$$

$$p_R = p_R^0 e^{-jk_0 x} \tag{4}$$

where p_I^0 and p_R^0 are magnitudes of p_I and p_R at the surface of the test sample, called the reference plane ($x = 0$); k_0 represents the complex wave number.

The acoustic pressure p_1 and p_2 at the two microphone fixed positions are:

$$p_1 = p_I^0 e^{jk_0 x_1} + p_R^0 e^{-jk_0 x_1} \tag{5}$$

$$p_2 = p_I^0 e^{jk_0 x_2} + p_R^0 e^{-jk_0 x_2} \tag{6}$$

where x_1 is the distance from sample surface to microphone 1, and x_2 is the distance from the sample surface to microphone 2.

By using relations (5) and (6) we can define the complex transfer function as follow:

$$H_{12} = \frac{p_2}{p_1} = \frac{e^{jk_0 x_2} + re^{-jk_0 x_2}}{e^{jk_0 x_1} + re^{-jk_0 x_1}} \tag{7}$$

where $r = \frac{p_R^0}{p_I^0}$ represents the complex value of the sound pressure reflection factor at reference plane. From relation (7) we get the following relation for r:

$$r = \frac{e^{jk_0 x_2} - H_{12} e^{jk_0 x_1}}{H_{12} e^{-jk_0 x_1} - e^{-jk_0 x_2}} \tag{8}$$

By using the value of r (8), the normal incidence sound absorption coefficient α and the normal surface impedance Z can be calculated as follow [18]:

$$\alpha = 1 - |r|^2 \tag{9}$$

$$Z = \rho c_0 \frac{1+r}{1-r} \tag{10}$$

where ρ is the density of the air, c_0 is the speed of sound in air and $Z_0 = \rho c_0$ represents the characteristic impedance of the air.

3. ACOUSTIC ANALYSIS

To assess by simulation the sound pressure level (SPL) at one location of interest - the driver's ear location, a simplified passenger cabin FE model was taken into consideration.

In order to reduce the interior noise, it is important, on the one hand, to understand the dynamics of the vehicle panels and how them interact with the air inside the vehicle cabin and on the other hand, to know the optimum acoustic treatments for each set of the sound package inside the passenger cabin.

The objective of this section is to investigate the effect of the sound absorbing material applied on the floor panel on the sound pressure level at the driver's ear location, in the frequency range from 0 to 300 Hz.

3.1 Finite element model creation

At the pre-processing stage of the finite element analysis (FEA), the following basic steps were completed in HyperMesh:

❖ Design the 3D geometry of the structural panels that looks like the interior cabin of the vehicle. The overall dimensions of the model are: 1200 mm x 700 mm x 782 mm. The thickness values of the panels are presented in Table1.

❖ Starting from this 3D geometry, the middle surfaces of the panels and the air cavity were discretized by using proper finite elements. For modeling the structural panels, CQUAD4 finite elements were used. These elements are 2D quadrilateral elements which have four corner nodes with six degrees of freedom (DOFs) per node: three translational displacements in X, Y and Z directions and three rotations about the X, Y, and Z axes. For the acoustic cavity, CHEXA finite elements were used. These are 3D elements bounded by six faces, having eight nodes with one degree of freedom per node: the acoustic pressure at that location.

❖ The sound absorbing material is defined by using the CHACAB (Connection Hexa Acoustic Absorber) card that defines the frequency – dependent structural acoustic absorber element in a coupled fluid – structural analysis [19], [20]. These elements can be shown as a mass attached to a spring and damper which are in parallel connection. The associated properties of the structural acoustic absorber element is defined by means of PACABS card, that allows us to specify the acoustic impedance of the porous material in terms of frequency dependent resistance (real part of the impedance) and frequency dependent reactance (imaginary part of the impedance) .

❖ Create the fluid - structure interface by using ACMODL card.

❖ Create isotropic and fluid materials using MAT1 and MAT 10 card images, respectively (Table 3).

❖ Create the associated properties (Table 2) by using PSHELL and PSOLID cards and then assign them to their structural and fluid elements.

❖ Apply the loads and the boundary conditions to the FE model.

❖ Define a frequency dependent dynamic unit load (force) at the geometric center of the firewall panel by using the DAREA load type and TABLED1 and RLOAD1 cards.

❖ Create a set of frequencies to be used in the response solution by using FREQ1 card.

❖ Create a node set for the output of results.

❖ Create a Radioss load step for the acoustic analysis.

❖ Specify a set of output for the frequency response analysis.

Table 2

Panel thicknesses

Panel name	*Thickness [mm]*
floor	0.9
front & back doors	1
firewall	0.8
windshield	5
roof	0.7
bulkhead	0.8

Table 3

Material properties

Structure	*Young's modulus*	*Poisson's ratio*	*Density*
Steel	2.1e11 Pa	0.312	7850 kg/m^3
glass	6.2e10 Pa	0.24	2300 kg/m^3
Fluid	*Speed of sound*		*Density*
air	343 m/s		1.21 kg/m^3

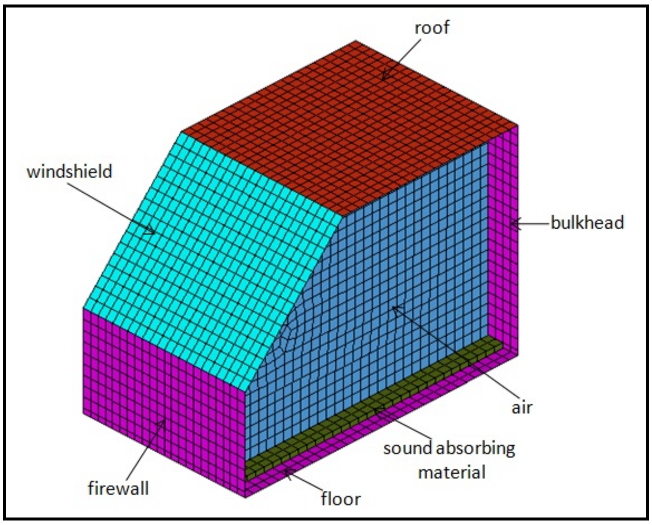


Fig. 5. The finite element model of the model

3.2 The equations of motion for the coupled fluid-structure system

At the interface between structure part and fluid part, the accelerations ($\ddot{u}$) on the structural grid points excite the air particles on the fluid part, and the pressures (p) on the fluid grids excite the structural part. This is a coupled problem between the vibration of the panels and the pressure inside the passenger cabin.

The acoustic analysis of the coupled system – the structure and the air - can be done either by direct integration or by modal frequency response analysis. The outputs of the analysis are:

- the responses for the structure part - the displacements and rotations at structural grid points
- the responses for the fluid part – the pressures at the fluid grid points.

After finite element discretization the equations of motion for the fluid part and for the structure part are as follows [3], [14]:

$$M_s\ddot{u} + B_s\dot{u} + K_s u = f_s + f_{sf} \tag{11}$$

$$M_f\ddot{p} + B_f\dot{p} + K_f p = f_f + f_{fs} \tag{12}$$

where:

❖ M_f and M_s represent the mass matrix for the fluid part and for the structure part, respectively.

❖ B_f and B_s represent the damping matrix for the fluid part and for the structure part, respectively.

❖ K_f and K_s represent the stiffness matrix for the fluid part and for the structure part, respectively.

❖ p and f_f are the acoustic nodal pressures vector and the external source vector acting on the fluid part.

❖ u and f_s are the structural nodal displacements vector and the external source vector acting on the structure part.

❖ $f_{fs} = A^T\ddot{u}$ is the vector of forces from the structure part that are acting on the fluid part.

❖ $f_{sf} = -Ap$ is the vector of forces from the fluid part that are acting on the structure part.

❖ A represents the interface matrix that describes the geometric relationship between the faces of the fluid elements on the surface of the air mesh and the structural nodes on the wetted surface of the structural mesh.

The differential equations of motion for the coupled fluid - structure system, can be written in the following matrix form:

$$\begin{bmatrix} M_s & O \\ -A^T & M_f \end{bmatrix}\begin{Bmatrix} \ddot{u} \\ \ddot{p} \end{Bmatrix} + \begin{bmatrix} B_s & O \\ O & B_f \end{bmatrix}\begin{Bmatrix} \dot{u} \\ \dot{p} \end{Bmatrix} + \begin{bmatrix} K_s & A \\ O & K_f \end{bmatrix}\begin{Bmatrix} u \\ p \end{Bmatrix} = \begin{Bmatrix} f_s \\ f_f \end{Bmatrix} \tag{13}$$

3.2 The results of the acoustic analysis

The vibration of the firewall panel was considered to be the most dominant source of noise in the cabin of the car, when the engine is running. Engine forces are transmitted to the vehicle panels through the engine mount locations. The amplitude variations of the force data can be obtained experimentally at each mount location while the engine is running at different speeds.

In current study, for the frequency response analysis, we have chosen a unit load along the direction of the length of the car, that is applied on a grid point of the firewall panel which is located near the geometric center of the firewall. The unit load is a sinusoidal or harmonic force in the frequency range of 0-300 Hz. The response is the sound pressure level variation as a function of a frequency-dependent force. During the post-processing with HyperGraph (module of HyperWorks), the pressure was converted to 'A' weighted sound pressure by using the relation (14), [3], [14].

$$SPL_{dB(A)} = 20\log\left(\frac{p/\sqrt{2}}{p_{ref}}\right) + A_{weighting} \tag{14}$$

where p is predicted by the Radioss solver and $p_{ref} = 20*10^{-6}\,Pa$ is the reference sound pressure in air.

The location of the response is a specific grid point inside the cabin, which represents the driver's hearing point.

The sound pressure level at the driver's ear is evaluated in two situations: with and without acoustic treatment applied to floor panel.

As can be seen in Figure 5, for the FE model without acoustic treatment, the highest sound pressure level is 88 dB(A) which is achieved at a frequency of 57 Hz.

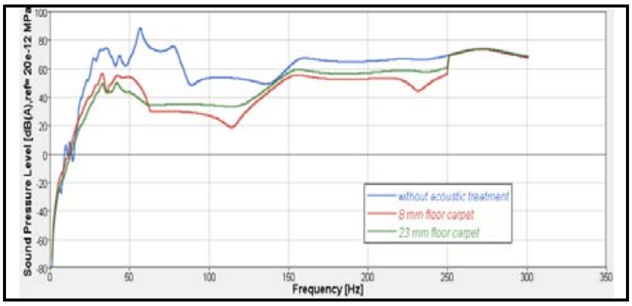


Fig. 6. Sound pressure level at driver's ear location as a function of excitation frequency, with/without acoustic treatment

The acoustic treatment is represented by a felt layer applied to the floor panel with the thickness of 8 mm and 23 mm respectively. The sound absorption coefficient values for the trim part are presented in Table 4.

Table 4.

The sound absorption coefficient values

Trim part	***Frequency [Hz]***	***Sound absorption coefficient***
8 *mm* floor carpet	16 ... 50	0.101
	63 ... 250	0.66
23 *mm* floor carpet	16 ... 50	0.309
	63 ... 250	0.66

5. CONCLUSIONS

In the present study it is shown the effect of the sound absorbing material on sound pressure level at the driver's ear location for a simplified passenger compartment. Three Logan acoustic treatments have been presented. A vibro-acoustic FE model of a simplified passenger cabin was utilized to simulate the sound pressure level at the driver's ear location due to the harmonic force applied near the geometric centre of the firewall panel in the frequency range from 0 to 300 Hz. Two cases were investigated: the first one in which the FE model is without acoustic treatment and the other one in which the sound absorbing material was attached to the floor panel.

From Figure 6 it can be observed that the sound pressure level for the FE models with acoustic treatment is less than the sound pressure level for the FE model without acoustic treatment.

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STUDII PRIVIND EFECTUL MATERIALELOR FONOABSORBANTE ASUPRA NIVELULUI DE PRESIUNE SONORĂ DIN INTERIORUL UNUI HABITACLU SIMPLIFICAT DE AUTOMOBIL

Rezumat: Confortul acustic din habitaclul automobilului reprezintă un domeniu important de cercetare pentru constructorii de automobile si adesea o modalitate de diferenţiere a brandurilor auto. Confortul acustic depinde de materialele, ce au o construcţie multistrat, din interiorul automobilului. Performanţa acustică a unui material este evaluată prin intermediul a doi parametrii acustici: coeficientul de absorbţie sonoră şi pierderea energiei sonore prin transmisie. În prima parte a acestui studiu, sunt prezentate 3 tratamente acustice aplicate in interiorul unui automobil marca Logan. În a doua parte a acestui studiu, a fost utilizată metoda elementelor finite (MEF) pentru analiza acustică a unui habitaclu simplificat de automobil. Pentru a simula nivelul de presiune sonoră de la urechea conducătorului auto, datorită excitaţiei aplicate asupra panoului tablier, care divizează compartimentul motor de compartimentul pasagerilor, a fost efectuată analiza modală a răspunsului în frecvenţă cu ajutorul solverului Radioss. Pentru a studia efectul materialului fonoabsorbant asupra confortului acustic, a fost investigat şi analizat nivelul de presiune sonoră din interiorul habitaclului simplificat cu şi fără tratament acustic aplicat.

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