https://dx.doi.org/10.22161/ijaers.5.7.17 ISSN: 2349-6495(P) | 2456-1908(O)

Methodology to Vibrational Noise Attenuation of Panels in Vehicles through Sound Absorption Materials

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Abstract**—***Thevibroacoustic comfort in vehicles is an important quality item and every day the carmakers look for new solutions for noise reduction and refinement of comfort.This study proposes the application of an experimental technique to determine the noise attenuation from the vibration of panels of a vehicle through the application of material for sound absorption.Sound absorbing materials are used in vehicles to attenuate high frequency noise, due to their characteristics. This study proposes the use of this type of material to attenuate noise of medium frequencies (100 – 600 Hz), predominant in structure-borne noise, complementing the existing ones, in order to refine the vibroacoustic behavior of the vehicle. Sound absorption materials are easy to handle, have a lower cost and require a short time for implementation.For the development of this work, a car cabin prototype was built using tubes and steel plates for the experimental tests.A finite element numerical model was created to obtain the vibrational behavior of the panels in this frequency range through modal analysis test. Experimental tests were performed on vibroacoustic frequency response function (FRF). It was observed in the tests performed that the application of sound absorption material attenuates significantly the vibration noise of panels in the range of medium frequencies, from 100 to 600 Hz. This methodology will allow the development of proposals for noise attenuation solutions and refinement of acoustic comfort with lower time and costs.*

Keywords**—Acoustic comfort, FRF, sound absorption, structure-borne noise, panels vibration.**

I. INTRODUCTION

The vehicles have several sources of noise and vibrations that work simultaneously in the various static and dynamic conditions in which the vehicles are exposed. In this way, solutions to problems related to noise and vibrations become complex because they presented wide frequency bands and a combination of transmission forms. Carmakers are looking for solutions for internal noise attenuation that contemplate short development time and lower cost.

The noise can be transmitted to the cabin through the air (airborne noise) and through the structure (structure-borne noise). Structure-borne noise originates from the vibrations that the structure receives, propagates throughout the body and the vibration of the panels generates noise in the internal cavity of the vehicle. The main sources of structural noise in a vehicle are the powertrain and the set of tires, wheels and suspension [7]. The structure-borne noise is perceived at frequencies up to 600 Hz, while airborne noise can be perceived in the range of 400 to 10000 Hz [4]. The graph of Fig. 1 shows the contribution of aerial and structural noise by frequency range.

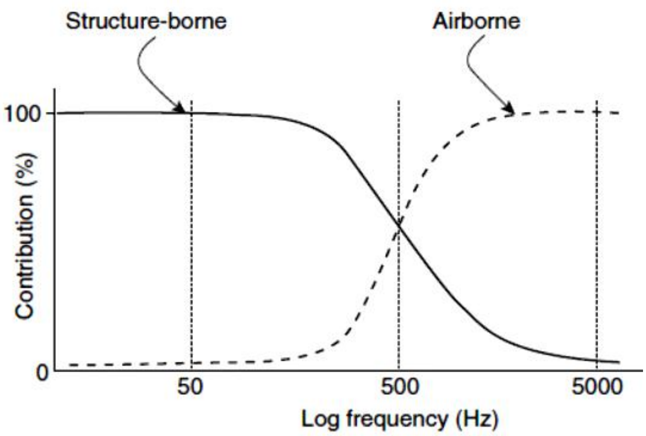

Fig.1 - Contribution of structure-borne and airborne noise in the overall noise of a vehicle[4]

https://dx.doi.org/10.22161/ijaers.5.7.17 *ISSN: 2349-6495(P) | 2456-1908(O)*

Several experimental techniques were developed for the characterization of vibroacoustic trajectories, using the Frequency Response Function (FRF) [2]. The Frequency Response Functions characterize a path through relations between physical quantities of two points. The entry or beginning of the trajectory can be understood as the stimulus to the system. The exit or end of the trajectory can be considered as the response of the system to the applied stimulus. FRF is the ratio of the output signal to the input signal in the frequency domain. Fig. 2 demonstrates a FRF scheme.

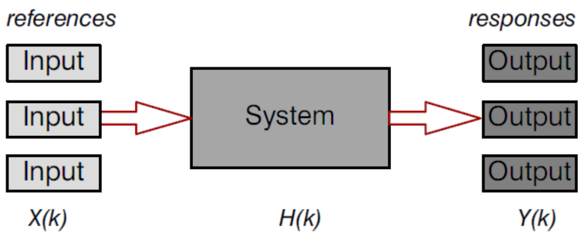

Fig.2: Frequency Response Function scheme.

Equations (1) and (2) demonstrate the FRF's calculation:

$$F(f) \times H(f) = X(f) \quad (1)$$

$$H(f) = \frac{X(f)}{F(f)} \quad (2)$$

Where *H(f)* is the Frequency Response Function, *X(f)* the output signal in the frequency domain and *F(f)* the input signal in the frequency domain [5].

The Coherence function is a quantity that relates the input and output signals and can be interpreted as the fraction of the output spectrum that is coming from the input spectrum. The function, for each frequency value, assumes zero value when there is no relation between the input and output signals and assumes value one when the output is fully correlated to the input. The coherence function (γ_{xy}^{2}) is defined as equation (3):

$$\gamma_{xy}^{2}(f) = \frac{G_{xy}(f)^2}{G_{xx}(f) \cdot G_{yy}(f)} \quad (3)$$

Where G_{xy} is the cross spectrum between the input and output signals and G_{xx} and G_{yy} the autospectrum of the input and output signals, respectively [5].

The vibroacoustic transfer paths are Frequency Response Functions that describe paths that originate in the vibration of the structure and the response refers to a point located in the space surrounded by air. From the application of a force on the structure at any given point, it radiates sound energy that is transmitted through the air to the point of the receiver. It is referred to as a hybrid path that has a stimulus in the structure and response observed at some point in the passenger compartment of the vehicle, thereby determining the noise transmitted by the structural route [5]. Fig.3 demonstrates the determination of structural noise.

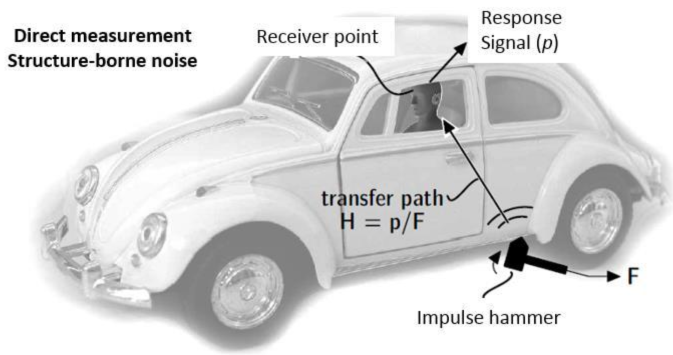

Fig. 3: Structure-borne noise measurement [10]

Solutions for structure-borne noise attenuation are more complex than solutions for airborne noise. Airborne noise is usually treated by materials that has insulation and absorption characteristics and are applied to the floor, firewall, engine region, and others [9]. These insulation works well for high frequency noise attenuation. A porous material is significantly more effective from the frequency range of 1000 Hz, as for sound absorption. Because of this behavior, its use is primarily for airborne noise treatment [1]. The graph of Fig. 4 shows a typical curve of sound absorption coefficient (α) as a function of frequency, of porous and fibrous sound absorbing materials installed on solid surface.

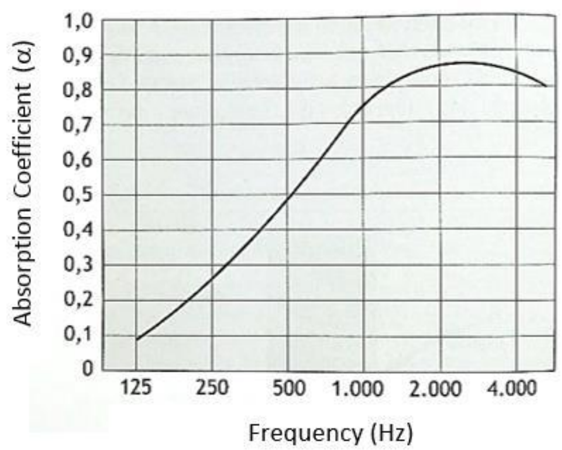

Fig. 4: Typical sound absorption graph for porous/ fibrous sound absorbing materials [1]

In order to attenuate the noise transmitted by the structure, the most effective forms require an optimization of the vibrational characteristics of the body, especially at the points of contact of the vibrational sources and in the supporting brackets or optimization of the elastic elements that are located between the source and the structure, which are the mounts. These components have other links, for example, the body has a controlled deformation in the event of a collision, the brackets and

https://dx.doi.org/10.22161/ijaers.5.7.17 *ISSN: 2349-6495(P) | 2456-1908(O)*

mounts must support the systems without breaking. Due to these links, solution proposals need to be subjected to various studies, which makes the development time is long and cost high. In this way, problems related to the noises transmitted by the structural route in vehicles can remain without solution after the end of the development of a product, compromising its quality.

Thus, this work investigates the noise behavior of vibrating panels when sound absorption material is applied inside the car cabin in order to achieve an attenuation that improves acoustic comfort.

For this study, a body prototype of a small car was built using tubes and steel plates and a numerical model of this body for the evaluation of the modal behavior of the panels. Vibroacoustic frequency response functions (FRF) were performed, with force input signal in body and the response of sound pressure level (SPL) response in car cabin. The tests were performed under the conditions of the body with and without insulation. The insulations used were porous blankets of textile material of automotive application.

II. METHODOLOGY

2.1 Body Prototype

For the development of this work, a prototype of a car body was built, with approximate dimensions of a small car hatch model. The structure is composed of square steel pipes welded, according to Fig. 5.

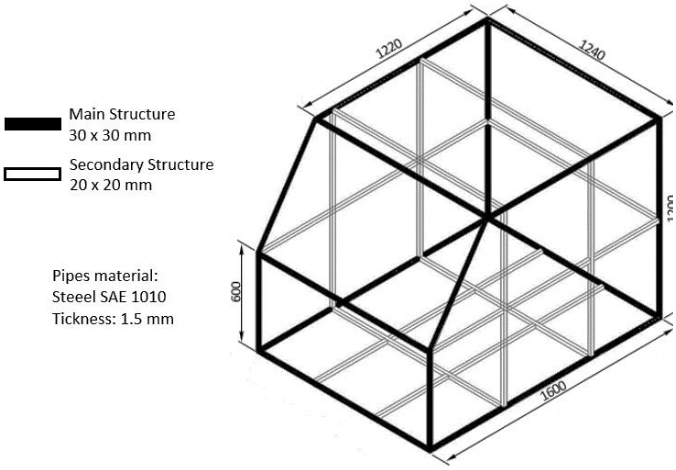

Fig. 5: Tubular structure of body prototype

The body is enclosed with steel plates with a thickness of 0.910 mm (SAE 1010) through rivets, sealed with silicone, except the front left side that is bolted to allow access to the interior, as shown in Fig. 6.The prototype is supported on four 6-inch casters to allow its locomotion.

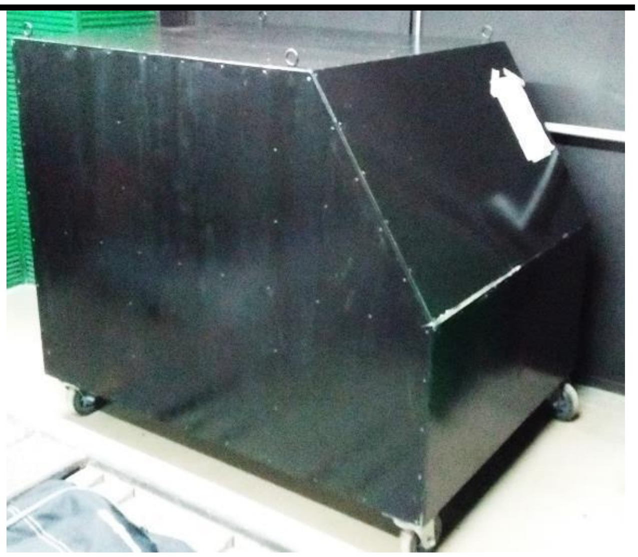

Fig. 6: Body prototype

2.2 Experimental Tests

The experimental tests for the development of the methodology consist of measurements of vibroacoustics Frequency Response Functions of the body, with excitation in the structure and response inside the cabin. The excitation is performed through an impact hammer with a force transducer and the response is measured through a microphone positioned in the region of the driver's right ear. Fig. 7 illustrates the FRF analysis system model.

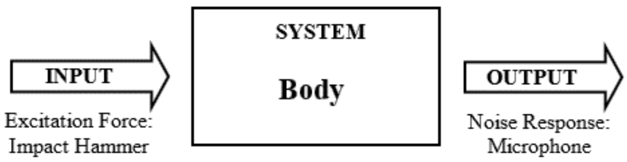

Fig. 7: Model analysis system

The excitation point in the body was defined in the lower left front region and was determined considering that it is a rigid point of the structure and region of important sources of structural noise, such as suspension and powertrain. The response point was defined in the region of the driver's right ear. Fig. 8 shows the measurement points.

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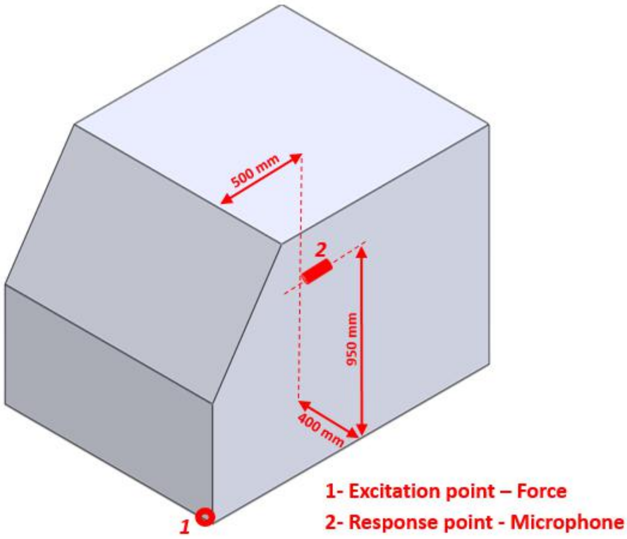

Fig. 8: Positioning of excitation and response points

The equipment used to the tests and analyzes are reported in Table 1.

Table.1: Equipment used for testing and analysis.

Equipment	Specifications	Manufacturer
Software	TestLab version 15	LMS Siemens
Analyser	Scadas Mobile 8 channels	LMS Siemens
Microphone	½" free field, 50 mV/Pa, mod. 46 AE	GRAS
Impact Hammer	2,25 mV/N, mod. 086C03	PCB

2.3 Tests Settings

The experimental tests were performed in two configurations, the first without insulation and the second with insulation applied on the floor, in the firewall and in the ceiling. The insulation consists in porous textile fibers blankets of automotive application from manufacturer Adler Pti, with grammage 1400 g/m^2, as shown in Fig. 9.

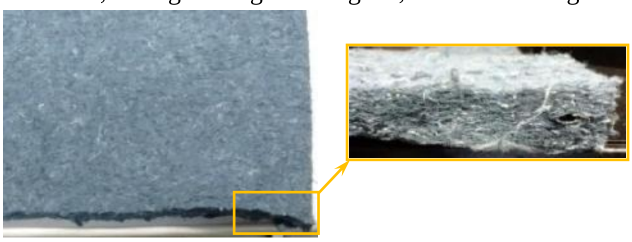
Fig. 9: Textile fiber porous insulation blanket 1400 g/m^2

The Fig. 10 shows the regions of application of the insulation in the body prototype.

Fig. 10: Blankets application: floor and firewall (a) and ceiling (b)

The applied area of insulation is 4.24 m^2 and the total insulation mass is 6.0 kg.

2.4 Virtual Modal Analysis

The body model FEM (Finite Element Method) was built using the software HyperMesh version 13.0. The types of elements used were the PShell for the body and CWeld for the welds. The Fig. 11 illustrates the numerical model.

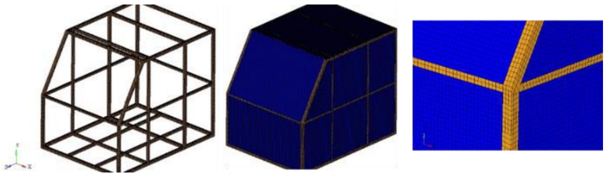
Fig. 11 – FEM model

III. RESULTS AND DISCUSSION

3.1 Body Frequency Response Function

Initially, the body FRF were obtained for the non-insulated configuration. The FRFs obtained are the sensitivity vibroacoustic functions. The higher sensitivity values indicate that the body responds more intensely to an excitation by frequency, so the higher the sensitivity value, the worse the acoustic's body behavior for a structural excitation. The graphic of Fig. 12 shows the result for the body without insulation.

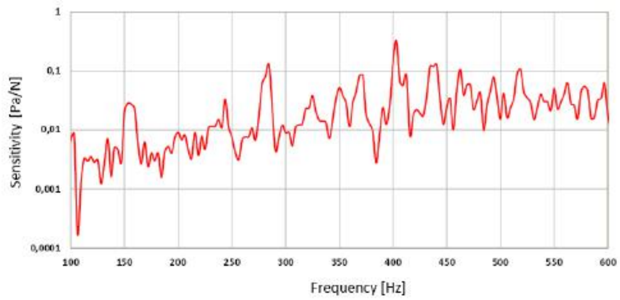

Fig. 12: Body vibroacoustic sensitivity function graphic

The peaks values that stand out in the graph were selected to determine the vibration modes. The frequency's values corresponding to the peaks are 156, 244, 285, 325, 350, 372, 403, 441, 462, 493, 519, 550, 562 and 581 Hz.

3.2 Body Modal Analysis

The FEM model analysis was performed using the OptiStruct 13.0 software, which determined the body modes in the desired frequency range for the study and which are presented in the Fig. 13.

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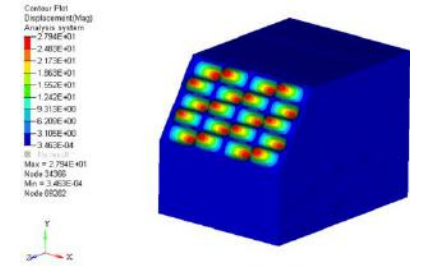
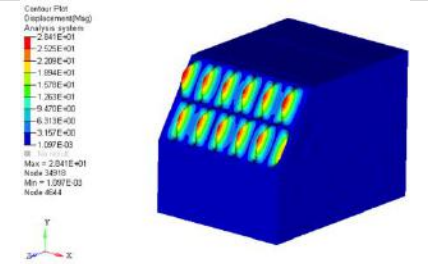
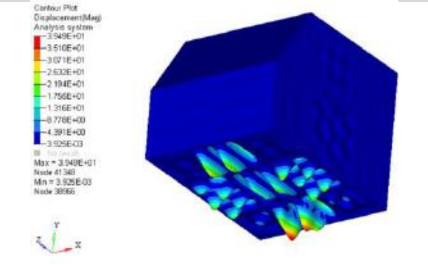
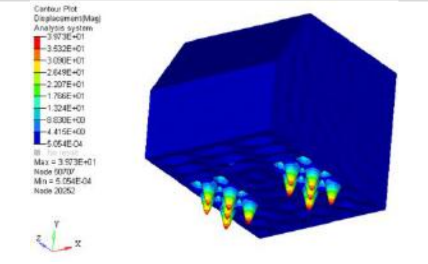
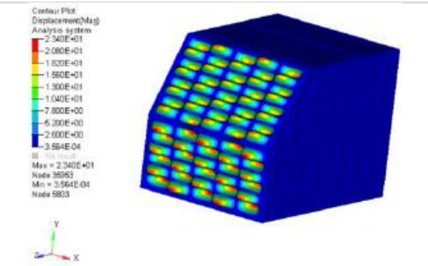
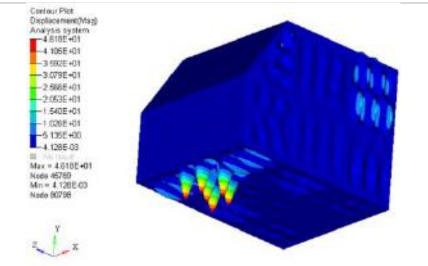
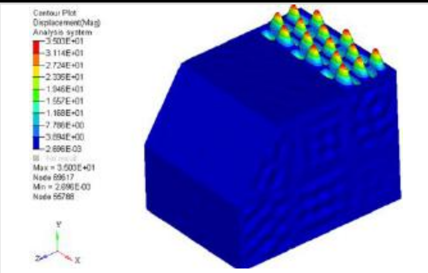
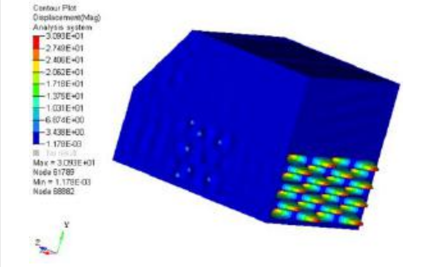
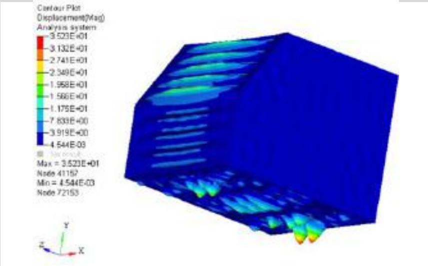
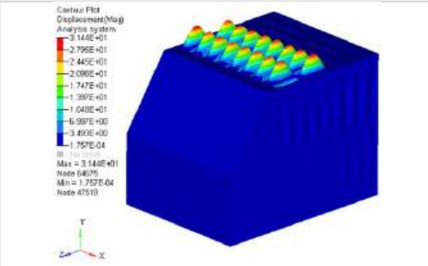
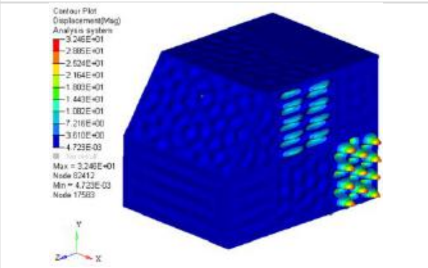
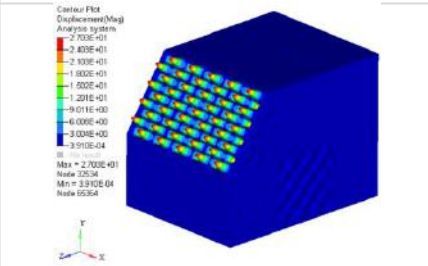
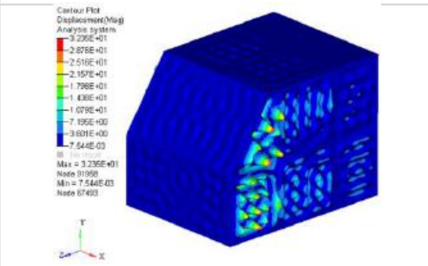
Frequency	Mode
156 Hz	
244 Hz	
285 Hz	
325 Hz	
350 Hz	
372 Hz	
403 Hz	
441 Hz	
462 Hz	
493 Hz	
519 Hz	
550 Hz	
562 Hz	

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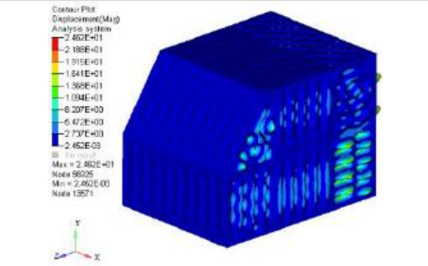
581 Hz	

Fig. 13 – Body vibrations modes

The modal analysis results shown in Fig. 13 indicate that the modes in the selected frequencies refer to the body panelsvibration.

3.3 Body Frequency Response Function with insulation

The comparative results of FRF vibroacoustics between the body with and without insulation 1400 g / cm^2 applied are presented below in Fig. 14.

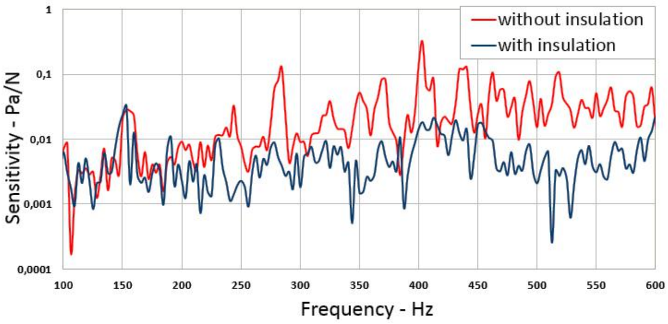


Fig. 14: Body vibroacoustic sensitivity with and without insulation

The results presented in the comparative graph of Fig. 14 demonstrate that significant structure-borne noise attenuation occurred with the application of insulation in the body, mainly in the previously selected frequency bands, related to the panels vibration.

The Table 2 shows the percentage of structure-borne noise attenuation in the selected frequencies, with the application of insulation.

Table.2: Structure-borne noise attenuation between body with and without insulation

Frequency [Hz]	Sensitivity [Pa/N]		Attenuation
	With insulation	Without insulation	
156	0,00204	0,02700	92%
240	0,00142	0,03288	96%
285	0,00391	0,12739	97%
325	0,00365	0,03861	91%
350	0,00157	0,03841	96%
372	0,00537	0,08406	94%
403	0,01853	0,32338	94%
441	0,01424	0,12920	89%
462	0,00953	0,10675	91%
493	0,00828	0,07842	89%
519	0,00274	0,10673	97%
550	0,00313	0,05097	94%
562	0,00560	0,06263	91%
581	0,00309	0,04647	93%

The reduction in vibroacoustic sensitivity indicates that the cabin has a lower noise level when subjected to structural excitation and consequent vibration of the panels. In this way, the application of porous insulation contributes significantly to reduce the noise emitted by the panelsvibration, as shown in Table 2.

The coherence functions of each configuration were analyzed to verify the relationship between the input signal and the output signal. In the two configurations tested, the values of coherence presented values above 0.9 in the whole analyzed frequency range, therefore, considered satisfactory.

IV. CONCLUSIONS

From the results presented in this paper, it is concluded that the noise attenuation from the vibration of the panels can be attenuated significantly with the use of sound absorption material.

Itshould be noted that materials for sound absorption are light, relatively low cost and require little time to build tooling for their production, which can make implementation viable in a vehicle even in a short time.

With the construction of the prototype of the body and based on the evaluation methodology developed, it will be possible to perform comparative tests of materials, regions of application of the insulation, among others, reducing time and cost in the development of proposals to improve acoustic comfort.

ACKNOWLEDGEMENTS

The authors would like to thank CAPES (Coordenação de Aperfeiçoamento de Pessoal de Nível Superior), PUC Minas (Pontifical Catholic University of Minas Gerais), Adler PTI and Fiat Chrysler Automobiles for support in this work.

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