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A METHODOLOGY FOR THE PERFORMANCE EVALUATION OF DIFFERENT SURFACE DAMPING TREATMENTS

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ABSTRACT

The acoustic comfort of passengers is a concern for the automotive industry in which surface damping treatments are broadly used. The viscoelastic materials present in the treatment of bodies of passenger cars add about 10 kg of material. In this work, different free layer surface damping treatments are compared: 1) body in white, 2) body treated with a commonly used synthetic viscoelastic material: i) viscoelastic material based on bitumen (asphalt melt) or ii) a new viscoelastic liquid applied sprayable damper, and 3) a composition cork materials. The accelerance maps experimentally obtained are used to assess the performance of the surface damping treatment. It is also proposed a topology optimization for the surface damping treatments application in a simplified vehicle model. The methodology proposed obtained significant vibration level reductions of the body, coupled with a reduction of the application area and in the weight of the material.

Keywords: damping treatment, performance evaluation, acoustic, optimization.

1. INTRODUCTION

Nowadays, the acoustic quality inside a passenger vehicle may no longer be viewed by consumers as merely a refinement, but as a fundamental criterion which may have a decisive role at the time of purchase. Thus, in the highly competitive automobile industry, investments in the development and improvement of surface damping treatments (SDT) are currently increasing. However, these benefits must be paid, if not directly, at least in terms of additional fuel consumption due to the added mass. In this context, one challenge is to optimize the components of the vehicles for comfort with the need to reduce mass and cost.

In most cases, synthetic polymeric viscoelastic damping layers are used for passive vibration control. These materials dissipate considerable amounts of energy per cycle of oscillation, thus increasing the damping factor of the structure (Nashif et al., 1985). Several studies have been conducted regarding the performance of viscoelastic materials in various areas of passenger transportation, such as: aerospace (Rao, 2003), railway (Fan, 2009) and automotive (Lilley et al., 2001 and Rao, 2003).

The SDT on passenger vehicles usually consists of the application of a free (unconstrained) layer of a viscoelastic material, usually designated as free layer surface damping treatment (FL-SDT), on the panels of a vibrant body to reduce the sound radiation due to the propagation of waves (Balmès and Germès, 2002). Traditionally, and for an automobile, the damping material is applied on the doors, roof, dash, floor and back panels since the interior sound field depends on the vibration of the body panels (group of finite plates that constitute the cabin).

Usually, asphalt melt sheets are used on FL-SDT of body panels, see Fig. 1. This is an asphaltic-based compound material that allows for the manufacturing of asphaltic damping layers with target weight that can be supplied to the assembly line in stacks of individually shaped parts.

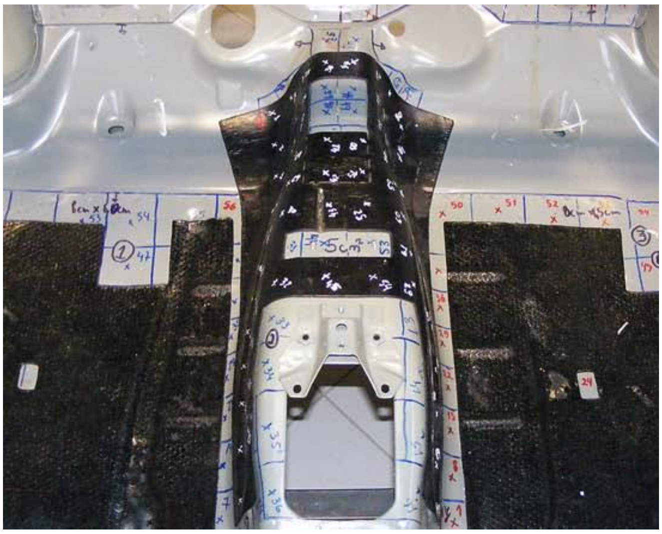


Fig. 1 – Asphalt melt sheet applied on floorpan of an automobile.

These asphaltic materials generally present some operational problems. Their high flow under high temperature restricts the use of these to horizontal surfaces. Furthermore, these materials exhibit low levels of damping performance relative to the others materials. However, while the performance is lower, they present lower costs compared to the others materials used to reduce floorpan vibration levels.

Nevertheless, damping material for passenger vehicle body, floor and dash panels have been undergoing significant advancements. One of these advancements is a sprayable material called Liquid Applied Sprayable Damper (LASD). LASD materials have three different chemistry bases: epoxies, sprayable asphaltic and water-based epoxies. However, the primary problem of LASD is the cost of the material, which is higher relatively to the asphalt melt sheets. Yet, the main advantage of LASD is related to cost savings associated to robotically spray-application, allowing the product to be repeatability applied (Nakkash et al, 2001) with various thicknesses, further allowing the optimization of the material geometry and location.

However, in the case of the floor of an automobile, FL-SDT asphaltic layers are still the most common, and are applied evenly and over most of the floor surface, see Fig. 1, contributing to approximately 10 kg to the total mass of the vehicle (Bevan et al., 2005).

It is clear that the benefits of the SDTs must be paid in terms of additional mass, resulting in additional fuel consumption. In this context, one challenge is to balance the comfort with the need to reduce mass and cost. This may be achieved, at least, in one of two ways. The first consists of reducing and optimizing the area subjected to the SDT.

Aiming to reduce the coverage area and the mass of material applied to the body, studies to evaluate the vibroacoustic behaviour of bodies and shape optimization of positions and application materials have been developed (Subramanian, 2003; Siavoshani et al., 2007 and Subic et al., 2005).

The second consists of using a light-weight material, e.g., cork composition. Note that while common viscoelastic material sheets, e.g., asphaltic melt sheet have a mass density varying between 1200 kg m^{-3} and 1600 kg m^{-3} (Furukava, 2009) and LASDs have a mass density of approximately 1000 kg m^{-3} (Furukava, 2009), there are cork compositions with a mass density of approximately 200 kg m^{-3} (Policarpo et al., 2013). Hence, for applications subjected to a wide range of temperatures (where common viscoelastic material may lose their properties) like nearby engine parts, exhaust systems, etc., the use of composition cork damping layers may be seen as a promising low mass density, thermal and acoustic insulation solution, which provides a damping capacity over a wide temperature range (Dickinson et al., 1995).

In this sense, a commonly used synthetic viscoelastic material and a composition cork materials will be considered in numerical simulations of a FL-SDT of the floor of a simplified car body in white (BIW). A finite element model is developed for topology optimization of the SDT applied to the floor of the car. Indeed, part of the present work is a natural sequence for the results obtained by (Furukava et al., 2009), (Maia et al., 2011) and (Policarpo et al., 2012a, 2012b).

Furthermore, an experimental method for mapping the levels of vibration amplitude regions of parts of the vehicle (a Fiat/Palio) for an optimized application of viscoelastic material in its structure is presented. This survey is conducted in a clean BIW. Three different damping material configurations are compared: 1) no structural damping, 2) FL-SDT using viscoelastic material based on bitumen (asphalt melt) currently used in the treatment of passenger vehicles on a large scale; 3) FL-SDT using a new viscoelastic material, LASD/Acusticol.

Results show a significant improvement in vibration reductions verifying the efficiency of surface damping treatment.

2. FREE LAYER SURFACE DAMPING TREATMENT

Free or unconstrained surface damping treatment consists of coating one or both sides of the structure with a damping material (Nashif et al., 1985 and Mead, 1999). So, whenever the structure is subjected to cyclic flexure, the damping material is subjected to tension-compression deformation parallel to the plane of the structure (see Fig. 2, where H_1 and H_2 are the thicknesses of the structure and of the damping material, respectively).

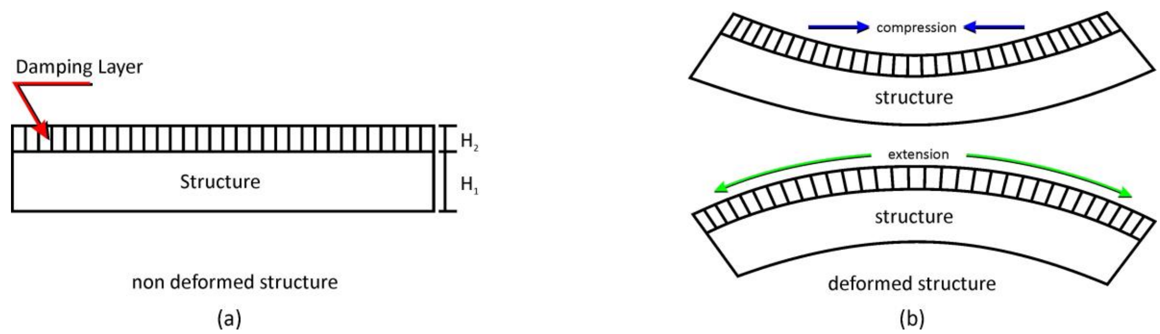


Fig. 2 – Free layer surface damping treatment: a) Undeformed structure; b) Deformed structure in flexure.

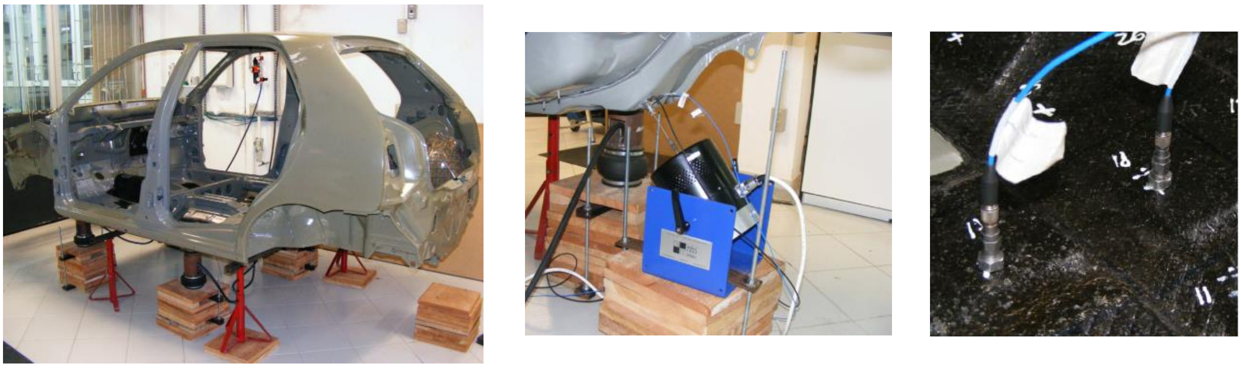
The elasticity modulus of the damping material is of a complex nature, considering its ability to dissipate energy during the load cycle. It is expressed as $E^{*}=E^{'}(1+i\eta)$ where E^{*} is the complex modulus, $E^{'}$ is the storage modulus, η is the loss factor and the product $E^{'}\eta$ is the loss modulus representing the capacity to dissipate energy (Nashif et al., 1985 and Mead, 1999). Although not considered in this work, note that for the damping materials both storage and loss modulli vary with temperature and frequency.

3. EXPERIMENTAL METHODOLOGIES AND IMPROVMENTS

3.1. Automobile Body in White (BIW)

A BIW, i.e., a bare body after welding but before painting, of a Fiat Palio vehicle, illustrated in Fig. 3 a), is used to assess the performance of two different FL-SDTs (asphalt pads and LASD/Acusticol material).

A vibration shaker, see Fig. 3 b), is used to excite the structure, following the guidelines from Fiat (2004). Sensing transducers, i.e., a force transducer, see Fig. 3 b), and accelerometers, see Fig. 3 c), are used to measure the force effectively applied to the structure and the response accelerations, respectively.



a) b) c)

Fig. 3 – BIW, excitation and sensing equipment: a) BIW of the Fiat Palio used in the study; b) Vibration shaker and force transducer; c) Accelerometers.

An accelerance mapping is created to access the vibration levels within subdomains of the BIW. It is understood by subdomains the components (plates, sheets) which internally are: tightly coupled, do not have large variations and geometric elements or ribs (beams on the top) (Subramanian, 2003). In this case, this BIW was divided into 15 subdomains, which were analysed separately. Besides facilitating the visualization of individual regions where there are higher accelerance levels, this mapping aims to identify areas in each subsystem where should actually be added some FL-SDT. However, in this study, only the results of one subdomain (the floor of the driver's side, see left side of Fig. 1) are presented.

Measurements were performed in the frequency range from 0 to 1000 Hz discretized into 6400 spectral lines, with a maximum resolution of 0.15625 Hz, and the number of samples for the average was 150, with rejection of overload and maximum overlap. In the signal, analysis software (Brüel & Kjær Pulse) was applied using the Hanning window in all measurement channels. The excitation was generated by a random signal (white noise) so that the structure had a flat response on the excitation frequency up to 1600 Hz.

This procedure was adopted for three different configurations. A configuration consisting in the asphalt pads that were applied at the factory, see Fig. 5. These pads were then stripped, leading to the other configuration that consists of the BIW without SDT, see Fig. 4. After mapping without damping, regions were defined for the application of the LASD/Acusticol material, which is the third configuration, in order to reduce the high accelerance levels obtained, see Fig. 6.

3.2. Results

This mapping procedure was applied to the BIW without any damping material, see Fig. 4.

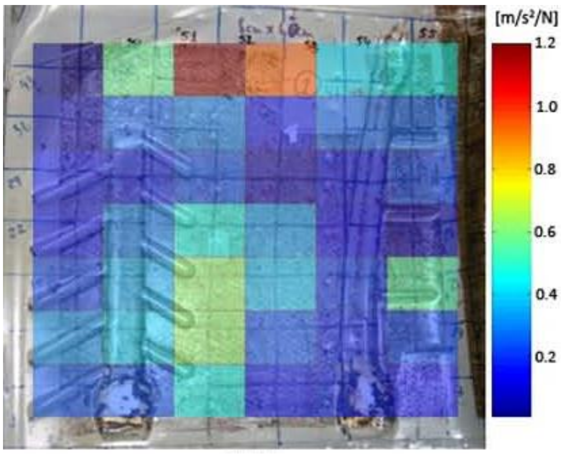


Fig. 4 – Accelerance mapping for the configuration without damping.

From Fig. 4, it is possible to observe that the majority of the higher accelerances are located at the third column (counting from left to right) and at the top line.

The next mapping was applied considering the asphalt pads applied in the factory configuration, Fig. 5. In the measurements conducted in the vehicle, care was taken to reproduce the conditions of damping that occur in the factory production line. Thus, the mounting of asphalt pads occurred at the car factory.

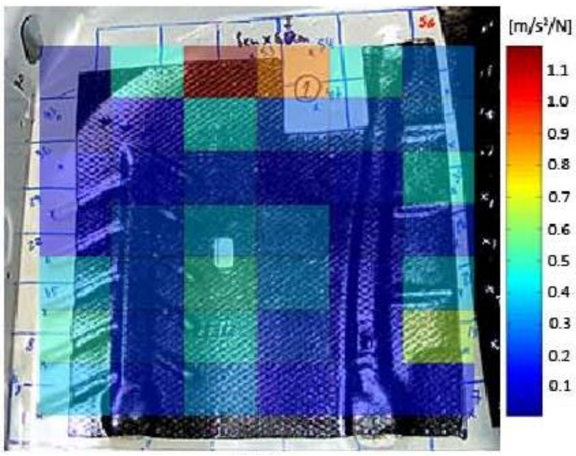


Fig. 5 – Accelerance mapping for the asphalt pads configuration.

From Fig. 5, it is possible to observe an acceleration decrease in the second, fourth and fifth columns (counting from left to right), relatively to the case without damping, see Fig. 4.

Next, an experimental optimization procedure of the application areas of a new soundproof LASD/Acusticol material in regions defined by the mapping of the undpamed configuration (see Fig. 4) was conducted, see Fig. 6. The experimental procedure of optimization aims at reducing the application area and determining the correct position for applying material based on mapping accelerance levels of the subdomain, to achieve an overall level of vibration at/or below the already reached one with the asphalt pads and if possible with a considerable reduction of the mass of added material.

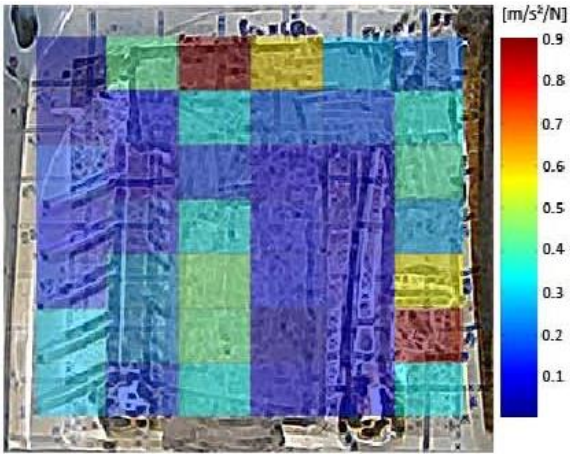


Fig. 6 – Accelerance mapping for the optimized configuration with LASD/Acusticol material

From Fig. 6 it is possible to observe a general acceleration decrease, relatively to the case without damping (see, Fig. 4), as the maximum acceleration reading decreased from 1.2 m N^{-1} s^{-2} to 0.9 m N^{-1} s^{-2}. In comparison with the asphalt pad configuration (see, Fig. 5), an acceleration decrease from 1.1 m N^{-1} s^{-2} to 0.9 m N^{-1} s^{-2} is observed.

From these accelerance mapping results, a comparison in terms of the amplitude of the accelerance as a function of frequency (considering seven of the most relevant frequencies) for the three configurations is presented next.

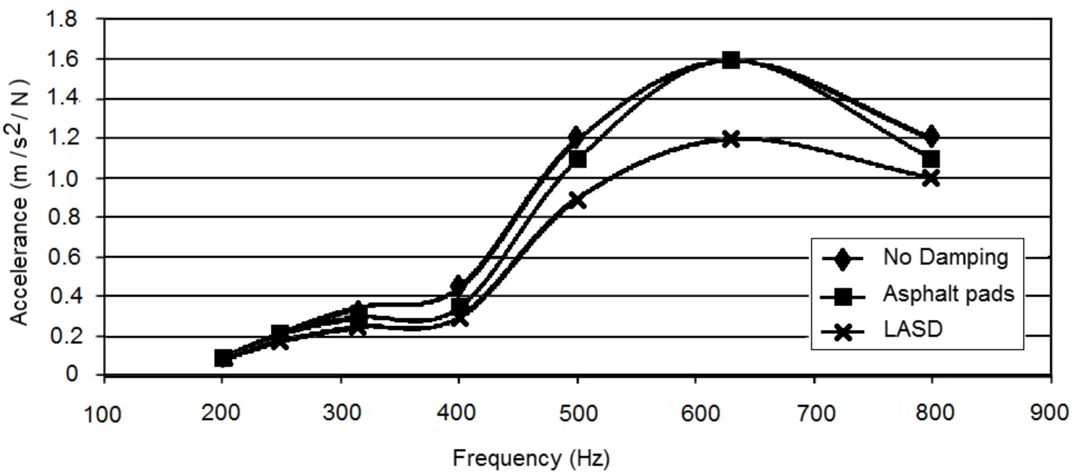


Fig. 7 – Accelerance as a function of the frequency for the three configurations.

Based on Fig. 7, it is possible to conclude that the LASD SDT is in fact more effective than the asphalt pads in this frequency range. Relatively to the undamped configurations, both the asphalt pads and LASD SDTs show significant improvements in decreasing the amplitude of the accelerance at those frequencies.

Thus, it was possible to obtain significant reductions in the vibration levels of the subdomain relatively to the original setting. Furthermore, with the LASD SDT a reduction of 26.3% could be obtained in the application area, and of 53.7% in the weight of added material.

4. NUMERICAL SIMULATIONS

The finite element method (FEM) is used to study the FL-SDT in the floor of a simplified car BIW considering two types of damping materials: i) a synthetic viscoelastic material; ii) and a composition cork. Furthermore, a structural topology optimization problem is formulated to find an optimized layout of the uniform thickness of FL-SDT spatially distributed over the vibrating structure.

The equation of motion for a multi-degree-of-freedom (MDOF) system, with N degrees of freedom and assuming that the dissipative mechanism is hysteretic (Maia et al., 1997), is expressed as

$$\mathbf{M}\,\ddot{\mathbf{x}}(t)+\mathbf{K}\left(1+\mathrm{i}\ \eta\right)\mathbf{x}(t)=\mathbf{f}(t), \tag{1}$$

where $\mathbf{M}$ and $\mathbf{K}$ are the $N \times N$ mass and stiffness matrixes, respectively; η is the loss factor; $\mathbf{f}(t)$ is the time dependent harmonic excitation force vector applied to the system; $\mathbf{x}(t)$ is the displacement response vector of the system and $\ddot{\mathbf{x}}(t)$ is the acceleration vector. To obtain the natural vibration mode shapes $\mathbf{\Phi}$ and natural frequencies of the system, one must solve Eq. (2) and the steady-state response of the linear elastic structure subjected to the axial load $\mathbf{F}_{\omega_{ap}}$, with excitation frequency ω_{ap}, is expressed by Eq. (3).

$$\left(\mathbf{K}\left(1+i\ \eta\right)-\lambda^2\mathbf{M}\right)\mathbf{\Phi}=\mathbf{0}, \tag{2}$$

$$\left(\mathbf{K}\left(1+i\ \eta\right)-\omega_{ap}^2\mathbf{M}\right)\mathbf{u}=\mathbf{F}_{\omega_{ap}}. \tag{3}$$

These analyses allow the necessary response characterization needed for the optimization problem of the spatial distribution of the damping layer.

The structural optimization problem studied here consists of finding an optimized layout of the uniform thickness of FLD treatment spatially distributed over the vibrating structure. The objective function chosen for this study is the root mean square of the maximum harmonic vibration response displacement $u(\omega)$ of one or more points N^* of the structure for a specific frequency range $[\omega_{\min},\omega_{\max}]$ from an initial value ($\omega_{\min}$) to a final value ($\omega_{\max}$), which contains one or more resonances.

The design variable $p(x)$ indicates the existence (p = 1) or absence (p = 0) of added damping layer. To relax the 0-1 problem to a continuum problem in]0,1], p can be relaxed to a “material relative density” value instead of the two extreme values. To simplify, it is here considered a piecewise constant parameter in each element p_i. To avoid computational problems, a value close to zero (e.g., $p_{\min}=10^{-6}$) is used instead of zero. A topology optimization problem can be formulated as:

$$\begin{aligned} &\min_{p_i} \left[\left[\sum_{j=1}^{N^*} \left(\max_{\omega \in [\omega_{\min}, \omega_{\max}]} |u(\omega)| \right)_j^2 \right] \Big/ N^* \right]^{1/2}, \\ &s.t. \\ &\sum_{i=1}^{N_{elem}} p_i \leq \text{Constant} \quad \text{and} \quad 0 < p_i \leq 1. \end{aligned} \tag{4}$$

where N_{elem} is the total number of elements to which the unconstrained damping layers are applied.

By relaxing the problem, intermediate values of p are allowed, which can be understood as a "porous" material, with properties that can be computed by

$$E_i = p_i^n E; \qquad \eta_i = p_i^{n-1}\eta; \qquad \rho_i = p_i^{n-2}\rho. \tag{5}$$

These properties follow the SIMP idea see for e.g., (Bendsøe and Sigmund, 2003), where the stiffness is proportional to density in the power n (greater than one). This penalizes intermediate values of p_i and for $n \geq 3$ the result of the optimization is mostly "black-and-white" ($p=1$ and $p=0$, respectively) for static compliance objective functions.

A constrained nonlinear optimization methodology, a sequential quadratic programming (SQP) algorithm, is used (Cottle et al., 1992). In SQP, a sequence of quadratic problems are solved. To overcome the task of determining second order derivatives, a *quasi-Newton* method is used. The gradients of the objective function are obtained by an adaptive finite difference method. For the numerical optimizations an iterative procedure with the following steps is defined: 1) initialize by giving initial design values; 2) run a harmonic analysis for the initial design; 3) start the optimization loop. After writing the values of the current iteration design variables to a file, it will call the finite element analysis in batch mode and will return to the optimization routine the current calculated values of the design variables; and 4) after satisfying the stopping criteria (constraints or predefined stopping parameters), run a harmonic analysis for the final design.

4.1. FEM Modeling and parameterization

A FEM of a simplified car BIW, illustrated by Fig. 8, is used to study the performance of the FL-SDT (applied to the floor of the car) using a synthetic viscoelastic material and a composition cork.

For the damping layer (see Fig. 2), solid finite elements (FEs) defined by four nodes and having three degrees of freedom at each node, translations in the nodal x, y, and z directions were used. The remaining components (e.g., roof, hood, doors, etc.) were modeled using Reiser-Midlin plate FE, defined by four nodes with six degrees of freedom at each node.

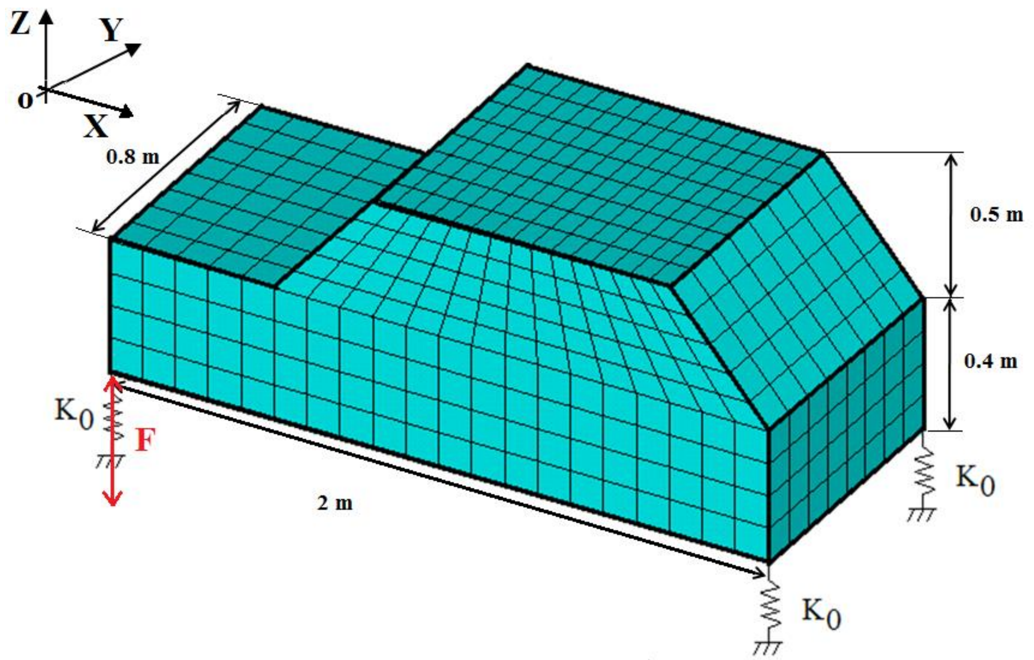


Fig. 8 – Finite element discretization of a simplified model of a car.

The structures is modeled considering 20 finite elements (FEs) in the X direction and 8 FEs in the Y direction. Even though one could allow several elements throughout the thickness of the damping layer, such an approach would require a significantly larger number of elements. Furthermore, as the optimization process may involve many finite element runs, the time required would be computationally larger. Thus, at this step only one element was used for the damping layer through the thickness. It is verified that this captures the essential features of the unconstrained damping layer behavior.

The geometric parameters of the elements are the thicknesses H_1=0.01 m and H_2=0.002 m of the structure and of the damping layer, respectively. A uniform steel plate with a thickness of 0.01 m was adopted as the shell of the car. To simulate the suspension, four tension-compression springs (no dampers) with a spring constant K_0 = 30 kN m^{-1} are applied. A steady-state (harmonic) force of 100 N is applied at point (F), as indicated in Fig. 8, and the displacements are determined at point r, see Fig. 9 a).

The properties of the materials are given in Table 1. Note that the complex longitudinal modulus of the cork agglomerate was determined using a hybrid analytical-experimental technique presented by some of the authors (Policarpo et al., 2013).

Table 1 – Material properties.

	E' (MPa)	ρ (kg/m^3)	η
Stainless steel	210×10^3	7850	0.001
Synthetic material	1	1350	0.44
Cork agglomerate NL 20 (Policarpo et al., 2013)	17.9	210	0.135

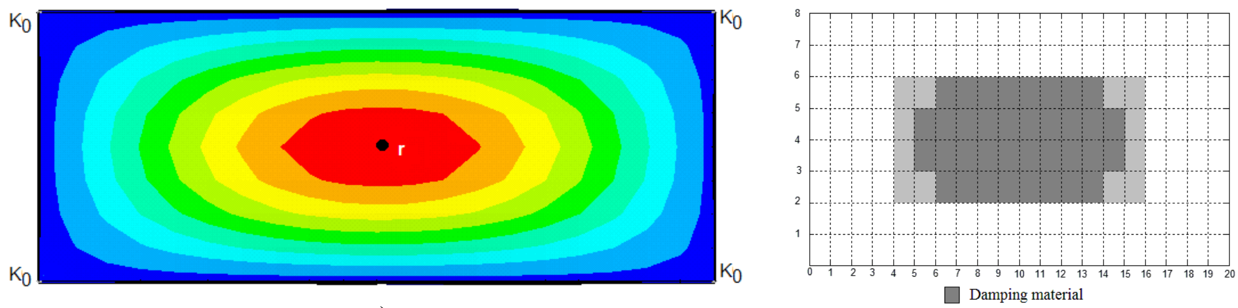
4.2. Simulation results

In this work, a finite element model of a simplified car BIW model is considered. It is harmonically excited at point F with the displacements determined at point r, as illustrated in Fig. 8 and Fig. 9 a). The goal is to minimize the objective function in a frequency range that contains the first vibration mode of the floor [20-60 Hz] subjected to a maximum of 30% of the total damping treatment area.

The results obtained are presented in Table 2 and are illustrated in Fig. 9.

Table 2 – Simplified car BIW model results.

Type of damping treatment →	FL-SDT: synthetic		FL-SDT: cork	
	Displacement	Mass	Displacement	Mass
Initial (without damping)	15.78 mm	1256.00 kg	15.78 mm	1256.00 kg
Before optimization (p_i =0.5)	9.34 mm	1258.16 kg	14.55 mm	1256.34 kg
Optimized	1.57 mm	1257.30 kg	5.03 mm	1256.20 kg
Optimized variation relative to the initial design	90.05%	+0.10%	68.12%	+0.02%



a) b)

Fig. 9 – Floor of the car: a) 1st Natural vibration mode (with indication of point r); b) Optimized material distribution.

After application and optimization of the damping treatments, see Fig. 9 b), the displacements were minimized by approximately 90% and 68% (see Table 2) for the FL-SDT with a synthetic viscoelastic material and a composition cork material, respectively. This was achieved with approximately 0.10% (0.86 kg) and 0.02% (0.14 kg) of additional mass for the synthetic viscoelastic and composition cork materials, respectively. It can be observed that the optimized material layout, see Figure 9 b), is similar in both FLD and CLD treatments (due to a maximum of curvature at the tip, while shear is constant along all beam). Thus, in this case the use of the synthetic viscoelastic material results in a decrease of 130% in terms of displacement and 500% in terms of additional mass, relatively to the composition cork material.

5. CONCLUSION

The proposed experimental methodology was used in a BIW of a Fiat Palio vehicle to assess the performance of two different FL-SDTs (asphalt pads and LASD/Acusticol material). With it, it was possible to obtain significant reductions in the vibration levels of the body relatively to the original setting, a reduction of 26.3% in the application area, and of 53.7% in the weight of the added material.

Regarding the simulation results of the SDT of the floor of a simplified car BIW, these show that the displacements were reduced by nearly 90% and 68% for synthetic viscoelastic and composition cork materials, respectively. However, this was accomplished with approximately 0.10% and 0.02% of additional mass for synthetic viscoelastic and composition cork materials, respectively.

Note that there are significant assumptions made here that can be re-addressed in future work, like the frequency and temperature dependence of the material, as well as the objective functions addressed by other authors.

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