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Damping performance of porous materials through dynamic analysis

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ABSTRACT

Porous materials are known to be effective in absorbing noise and eliminating sounds while viscoelastic materials are lightweight materials known to significantly reduce structural vibration. Hence, the study of porous viscoelastic materials is a promising field for applications on passive noise and vibration control in

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many industries such as the automotive and aerospace. The goal of this work is to see whether the dissipative properties of foams can be characterized by DMA measurements and to investigate the influence of the fluid-structure interaction in porous materials on their damping performance. In this regard, dynamic experimental tests of simply supported panels with a free-layer of porous material and finite element simulations of the covered panel are analysed. Their comparison can shed light on this problem and reveal if damping effects are mainly due to the viscoelastic matrix or are also influenced by the fluid-structure interaction. As a consequence, this allows a better understanding of the complexity of the system and helps modelling the materials behavior properly.

Keywords: Porous Materials, Vibration Damping, Viscoelastic Properties
I-INCE Classification of Subject Number: 47

1. INTRODUCTION

In the last few years, porous materials have been widely used for noise and vibration control in several applications such as civil, aerospace and automotive industries. Their energy dissipation mechanisms have been proved useful for sound absorption and damping purposes. When mounted in sandwich structures, their performance can even be improved by the combination of both materials properties.

Therefore, the characterization of their material properties becomes indispensable for predictions of their acoustics and mechanical behavior, and consequently, for their appropriate use. As pointed out by [1], many efforts have been done recently to develop measurements methods to assess their parameters, ranging from quasi-static to dynamic ones. However, in a more recent work [2], several characterization methods for dynamic viscoelastic properties were performed and compared, showing some discrepancies between their estimations.

The purpose of this work is to investigate if the viscoelastic properties of a polymeric foam can be estimated by DMA measurements, more precisely by a torsional rheometer, and also study the influence of fluid-structure interactions on their damping performance. To achieve this goal, dynamical measurements are performed on simply supported panels composed of an elastic layer with and without a free-layer of porous material. Afterwards, the frequency response function (FRF) of the experimental results obtained are compared with the FRF provided by a finite element model using previously measured viscoelastic parameters via a DMA machine.

This work is organized as follows. Section 2 briefly explains some basic concepts related to porous materials and the model adopted. Then, Section 3 describes the experimental set-up. Finally, Section 4 presents the experimental and numerical results, followed by concluding remarks.

2. FUNDAMENTALS

2.2.1. Basic Concepts of Porous Materials

Porous materials present a great range of shapes and forms. Basically, they can be divided into two main groups: granular materials such as piles of sand, and porous solids such as foams [3]. They can be found in nature, or be manufactured as a natural result of the fabrication process or as a desire for some mechanical and/or physical properties [4].

Porous solids, in particular, are composed of a matrix (solid skeleton) and a porous space. Their behavior depends on the size, arrangement and shape of the pores in addition to the porosity and the composition of the matrix itself [5]. Furthermore, these two phases may influence each other and fluid-structure interactions phenomena may occur such as dissipation processes which can be beneficial for certain applications [6].

In the context of this work, the porous solid material presents a polymer skeleton (which will be described in subsection 3.2) and as a consequence, it exhibits viscoelastic behavior such as viscoelastic solid materials [11]. In this regard, their properties show temperature and frequency dependencies, and viscoelastic phenomena such as creep, relaxation and hysteresis can occur.

2.2.2. Model Formulation

Under the paradigm of modelling porous materials, it is assumed herein that the viscoelastic dissipation mechanism is the main source of damping and consequently, only the solid skeleton behavior is relevant. Therefore, the porous material studied is considered as a monophasic viscoelastic one.

In order to explore their damping behavior, two structural configurations are evaluated as shown in Figure 1. The first configuration is a bare aluminium panel (elastic behavior), whereas the second one is an aluminium panel (elastic behavior) bounded with a free-layer of porous material (viscoelastic behavior).


(a) Bare panel
(b) Two-layered panel

Figure 1: Configurations under study

On both cases, the discretization of the equations of motion via the finite elements methods leads to

$$[\boldsymbol{K} - \omega^2 \boldsymbol{M}]\boldsymbol{U} = \boldsymbol{F}, \tag{1}$$

where $\boldsymbol{K}$ is the global stiffness matrix, $\boldsymbol{M}$ is the global mass matrix, $\boldsymbol{U}$ is the displacement vector and $\boldsymbol{F}$ is the force vector.

For the case of an elastic material with structural damping such as the first configuration, the stiffness matrix $\boldsymbol{K}$ can be expressed as

$$\boldsymbol{K}^* = (1 + j\eta)\boldsymbol{K_E} \tag{2}$$

where η is the structural damping coefficient of the structure.

For the second configuration, due to the addition of the porous layer, the global stiffness matrix from Equation 1 becomes frequency-dependent as follows

$$\boldsymbol{K}(\omega) = \boldsymbol{K_E} + G^*(\omega)\boldsymbol{K_V} \tag{3}$$

where $\boldsymbol{K_E}$ and $\boldsymbol{K_V}$ are, respectively, the global stiffness matrix related to elastic and viscoelastic components, and $G^*(\omega)$ is the complex shear modulus of the viscoelastic material [11]. Any viscoelastic model can be used to described the frequency-dependent complex shear modulus such as generalized Maxwell and fractional derivative models.

The complex shear modulus $G^*(\omega)$ herein is described by the four-parameter fractional derivative model [7] shown in Equation 4.

$$G^*(\omega) = \frac{G_0 + G_\infty(j\omega\tau)^\alpha}{1 + (j\omega\tau)^\alpha} \tag{4}$$

where G_0 and G_∞ are, respectively, the relaxed and unrelaxed shear moduli, τ is the relaxation time and α is the order of the fractional derivative model. It is worthwhile mentioning that these four parameters must follow thermodynamical constraints:

$$G_\infty > G_0 > 0,\ \tau > 0 \,\text{and}\, 0 < \alpha < 1 \tag{5}$$

Furthermore, the frequency response of both configurations are computed by direct method that solves Equation 1 at each frequency of interest.

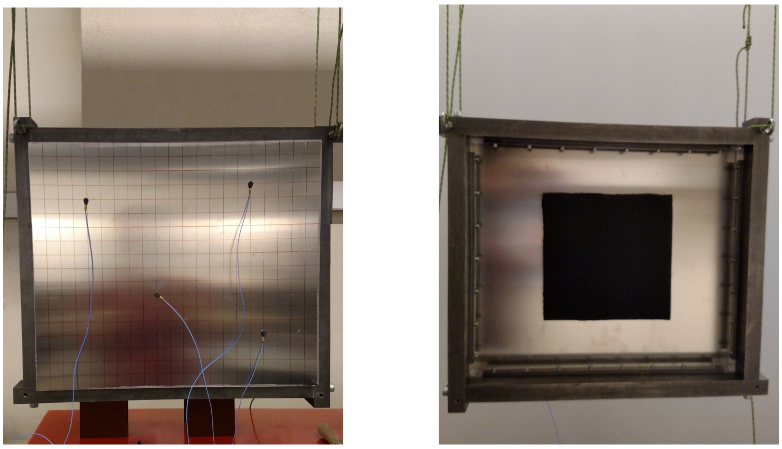
3. EXPERIMENTAL PROCEDURE

3.3.1. Description of the Experimental Set-up

The experimental rig chosen for the present characterization is composed of simply supported panels mounted on a frame, four accelerometers and an impact hammer. The latter is used to apply a punctual force to excite the panel and the structure's response is measured by the four lightweight accelerometers glued on the bare panel side through bee wax.

As previously mentioned, two testing configurations are evaluated in order to verify if the porous material could be applied as a free layer damping treatment to reduce structural vibrations. The first configuration consists of an aluminium plate with dimensions 420 × 360 × 3 *mm*, which was built following the description found in [8]. The second configuration, in turn, is obtained by gluing a free-layer of porous material with dimensions 200 × 200 × 25 *mm* centered on the top of this aluminium plate.

Figure 2 shows the positions of the four accelerometers glued on the bare panel side, and the free-layer of porous material glued on the aluminium plate, representing the second configuration.



(a) Bare panel side

(b) Configuration 2

Figure 2: Experimental set-up: simply supported panels

The experimental frequency response function (FRF) measured is the accelerance. This FRF is expressed in terms of acceleration and is defined as the ratio between the acceleration measured by accelerometers and the force applied to the structure through the hammer. Equation 6 expresses its magnitude in decibel scale (dB).

$$FRF_{dB} = 20 \log \left(\left| \frac{\hat{\mathrm{A}}(\omega)}{\hat{\mathrm{F}}(\omega)} \right| \right) \tag{6}$$

where $\hat{\mathrm{A}}(\omega)$ and $\hat{\mathrm{F}}(\omega)$ are, respectively, the Fourier transforms of the acceleration and the force measured.

As structural vibration issues usually occurs in the low frequency range, measurements are performed up to $800Hz$, with a frequency step of $0.5Hz$, at room temperature. It is worthwhile mentioning that the frequency response of each configuration was measured ten times to verify the reproducibility and repeatability of these measurements. The results shown in section 4 are the average values.

3.3.2. Tested Materials

The porous material studied is a closed-cell polyurethane (PU) foam which is a lightweight material characterized by its good sound proofing, thermal insulating and shock absorbing properties [3]. This foam has been previously characterized by [9] and its fractional derivative model parameters, shown in Eq. 4, are depicted on Table 1.

Table 1: Fractional derivative model parameters

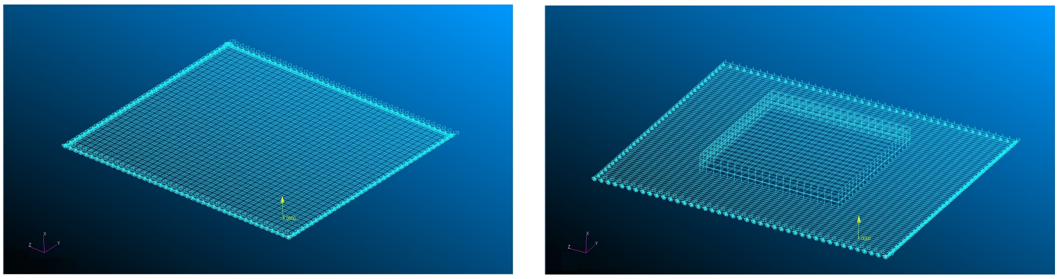
G_0 [Pa]	G_∞ [Pa]	τ [s]	α
1.31×10^4	2.11×10^6	4.70×10^{-8}	0.30

To limit the potential effects of spatial heterogeneity, the foam layer used has been cut off from the same small-sized block of the material used in [9]. Furthermore, the mass density (ρ) of this PU foam is 48 kg/m^3 and its Poisson's ratio (ν) is considered as a constant equal to 0.35 [2].

Additionally, the aluminium plate used on the experiments has Young's modulus $E = 69\ GPa$, a mass density $\rho = 2700\ kg/m^3$, a Poisson's ratio $\nu = 0.35$ and a structural damping coefficient $\eta = 0.001$ (considered constant as a function of frequency).

4. RESULTS AND DISCUSSION

In the finite element models for both configurations evaluated, all layers are modelled with a 20-node hexahedral element, leading to 32931 (Configuration 1) and 52272 (Configuration 2) degrees of freedom. Figure 3 shows the finite element meshes adopted as well as the location of unit load applied (which is similar to the location of the force applied in the experimental measurements).



(a) Configuration 1 (b) Configuration 2

Figure 3: Finite element mesh

The frequency responses are computed within the frequency range of 0-800 Hz at four nodes corresponding to the location of each accelerometer previously mentioned in subsection 3.1. Figure 4 shows these numerical results for each configuration tested. It is possible to observe that the free-layer of the porous material studied introduced damping in the structure. This is more evident after approximately $400Hz$.

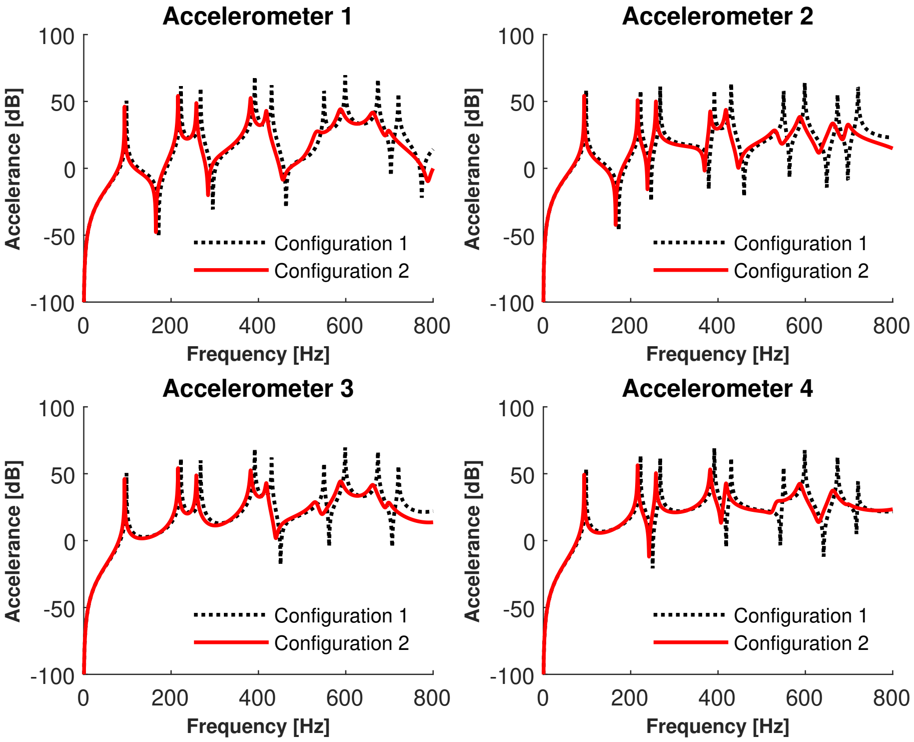


Figure 4: Simulation results

The experimental results obtained for each configuration are compared in Figure 5. As in the numerical results, the second configuration is more damped than the first one, evidencing the possibility of using porous materials to damp structural vibrations. The addition of this free-layer of polyurethane foam only increased the total weight by 3.9%.

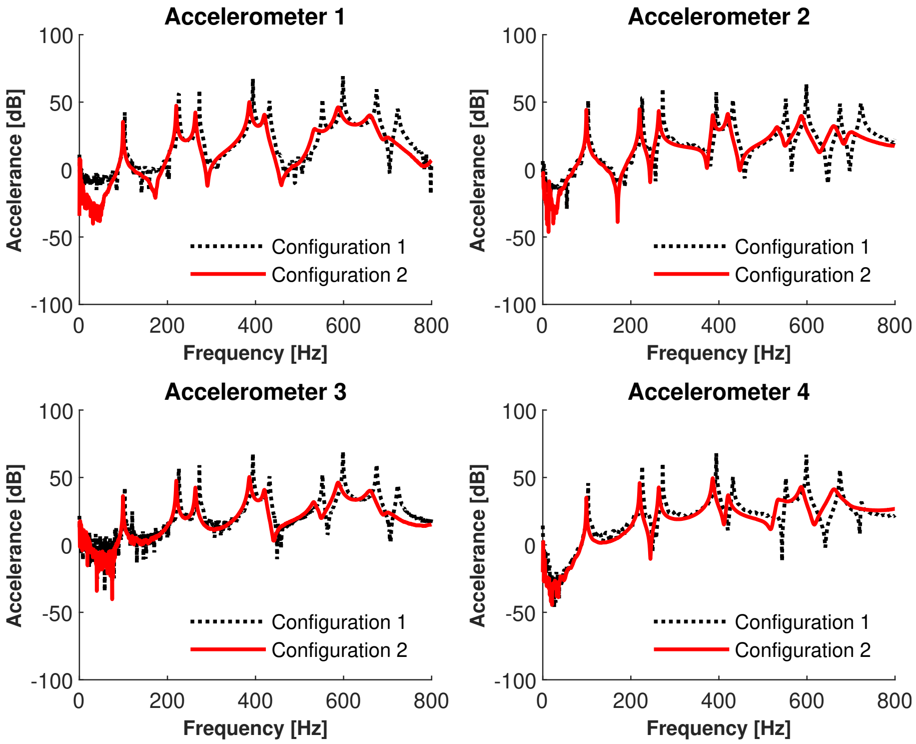
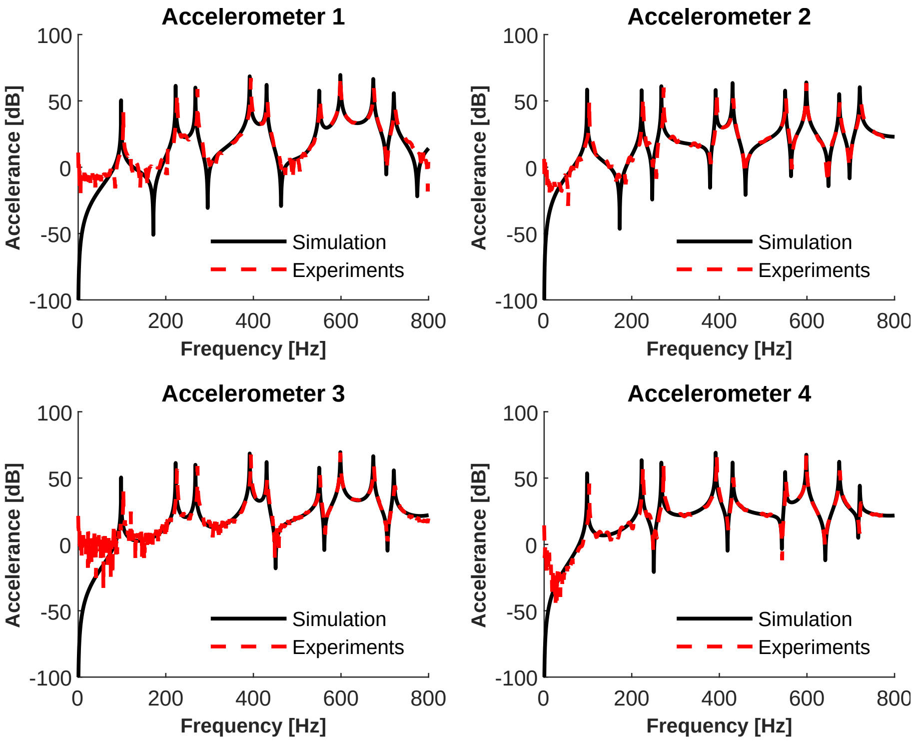
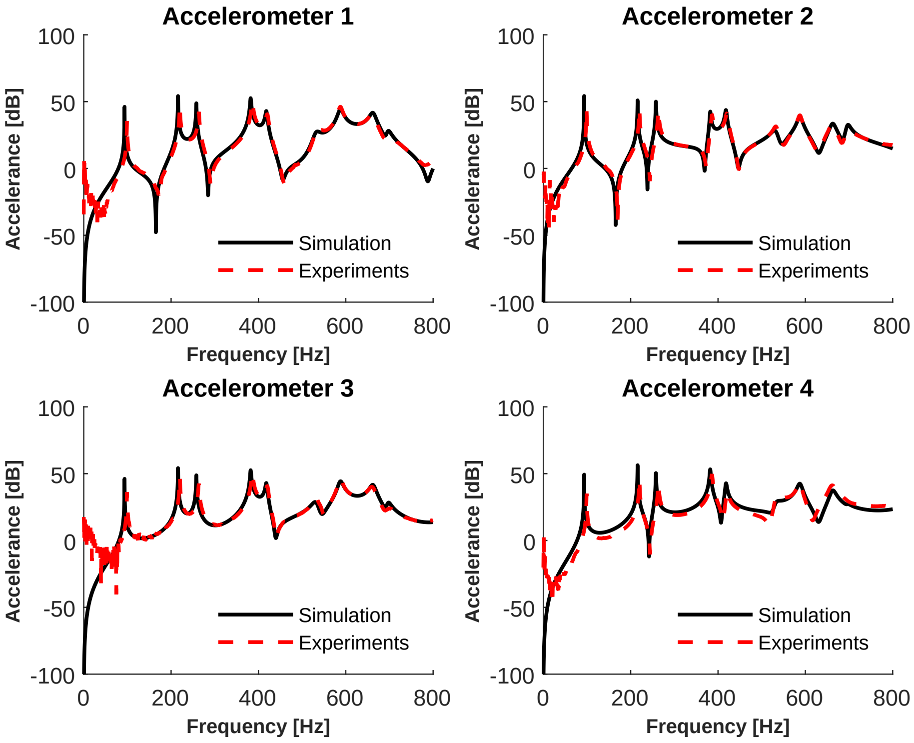


Figure 5: Experimental results

Figure 6 compares the experimental and numerical results. For both configurations, very good agreement can be observed. The magnitude of the FRFs and the resonant frequencies are quite similar, especially for the first configuration. However, it is also possible to note that the experimental result of the second configuration presents a little more damping than its numerical one and some discrepancies at lower frequencies. This may be due to noise and imprecisions in the acquisition of experimental data.



(a) Configuration 1



(b) Configuration 2

Figure 6: Comparison between the experimental and numerical frequency responses

Nevertheless, these results show that viscoelastic properties of this polymeric foam can be appropriately estimated and one can have good predictions of its mechanical behavior when modelling it as a solid viscoelastic material. It is worthwhile mentioning that DMA measurements performed in [9] were carried out using the torsion mode. This usually gives good estimations as it ensures constant volume along the test and the influence of fluid-structure interaction on measurements is limited, as pointed out in [1].

5. CONCLUSIONS

Experimental measurements and numerical simulations were performed on simply supported panels to verify the possibility of predicting the mechanical behavior of polymeric foams through the use of viscoelastic properties measured by DMA in torsion mode. The damping induced by this foam in structural vibrations is also evaluated. Through the results, one can observe that the introduction of a porous material on the plate reduced the amplitude of vibration without having a significant increase on the system's mass. Also, the numerical and experimental data have a good agreement for all configuration indicating that the viscoelastic properties measured by DMA can be used on FE models to predict damping effects with a relevant precision at low frequency.

The next step is to extract the parameters of the fractional derivative model using these experimental results by the formalism of inverse problem and to compare with the ones estimated using DMA measurements, enabling insights on the accuracy of the methods. Furthermore, experiments at high frequencies are required to identify the limitations of this approach consisting of modelling the porous layer as a solid viscoelastic material. As frequency increases, the modelling of other dissipation phenomena may be necessary to obtain good agreements between numerical and experimental results.

6. ACKNOWLEDGEMENTS

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7. REFERENCES

[1] Luc Jaouen, Amélie Renault, and Mickael Deverge. Elastic and damping characterizations of acoustical porous materials: Available experimental methods and applications to a melamine foam. *Applied Acoustics*, 69(12):1129–1140, 2008.

[2] Paolo Bonfiglio, Francesco Pompoli, Kirill V. Horoshenkov, Mahmud Iskandar B.Seth A. Rahim, Luc Jaouen, Julia Rodenas, François Xavier Bécot, Emmanuel Gourdon, Dirk Jaeger, Volker Kursch, Maurizio Tarello, Nicolaas Bernardus Roozen, Christ Glorieux, Fabrizio Ferrian, Pierre Leroy, Francesco Briatico Vangosa, Nicolas Dauchez, Félix Foucart, Lei Lei, Kevin Carillo, Olivier Doutres, Franck Sgard, Raymond Panneton, Kévin Verdiere, Claudio Bertolini, Rolf Bär, Jean Philippe Groby, Alan Geslain, Nicolas Poulain, Lucie Rouleau, Alain Guinault, Hamid Ahmadi, and Charlie Forge. How reproducible are methods to measure the dynamic viscoelastic properties of poroelastic media? *Journal of Sound and Vibration*, 428:26–43, 2018.

[3] Alexander H.-D. Cheng. *Poroelasticity*, volume 27. Springer International Publishing, 1 edition, 2016.

[4] C. T. Herakovich and S. C. Baxter. Influence of pore geometry on the effective response of porous media. *Journal of Materials Science*, 34(7):1595–1609, 1999.

[5] Olivier Coussy. *Mechanics and Physics of Porous Solids*. John Wiley & Sons Ltd, first edition, 2010.

[6] Laith Egab, Xu Wang, and Mohammad Fard. Acoustical characterisation of porous sound absorbing materials: a review. *International Journal of Vehicle Noise and Vibration*, 10(1/2):129, 2014.

[7] R. L. Bagley and P. J. Torvik. A Theoretical Basis for the Application of Fractional Calculus to Viscoelasticity. *Journal of Rheology*, 27(3):201–210, 1983.

[8] Olivier Robin, Jean Daniel Chazot, Romain Boulandet, Marc Michau, Alain Berry, and Noureddine Atalla. A plane and thin panel with representative simply supported boundary conditions for laboratory vibroacoustic tests. *Acta Acustica united with Acustica*, 102(1):170–182, 2016.

[9] Lucie Rouleau, Alain Guinault, and Jean-François Deü. Vibration Damping Properties of Porous Materials. *6th European Conference on Computational Mechanics (ECCM 6)*, (7):1–9, 2018.

[10] Leitao Cao, Qiuxia Fu, Yang Si, Bin Ding, and Jianyong Yu. Porous materials for sound absorption. *Composites Communications*, 10(April):25–35, 2018.

[11] John D. Ferry. *Viscoelastic Properties of Polymers*. Third edition, 1980.