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VIBRATION DAMPING PROPERTIES OF POROUS MATERIALS

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Abstract. Porous materials are traditionally used in industry for their sound absorption and insulation properties. Over the past decade, more attention has been given to their elastic and damping properties. There is a particular interest in the automotive industry to replace heavy layers (consisting of free viscoelastic rubber layers) with felts or foams evidencing high damping capabilities. The goal of this work is to study the vibration damping performances of some porous materials. To that purpose, the viscoelastic mechanical properties of two foams are measured by means of a torsional rheometer, and numerical simulations are carried out to compare their vibration damping performances.

1 INTRODUCTION

Over the past decades, the control of noise and vibrations has become a standard requirement in the product development process, in particular in the automotive industry. To achieve targetted levels of noise and vibration in the vehicle, sound-absorbing , sound-insulating and vibration-damping parts are placed in critical areas of the vehicle. Classically, porous materials are used for sound absorption and sound insulation while viscoelastic materials provide significant vibration damping. The contribution of porous materials to vibration damping is generally neglected. The design of noise and vibration treatments needs to be balanced with lightness and cost requirements, which leads to the development of innovative materials and concepts. Therefore, there is a particular interest in the automotive industry to replace heavy layers, such as free viscoelastic layers, with felts or foams evidencing high damping capabilities [1].

The goal of this work is to investigate the possibility to use porous acoustic treatments to achieve structural vibration damping. To that purpose, the viscoelastic mechanical properties of two foams are measured by means of a torsional rheometer, and numerical simulations are carried out to compare their vibration damping performances.

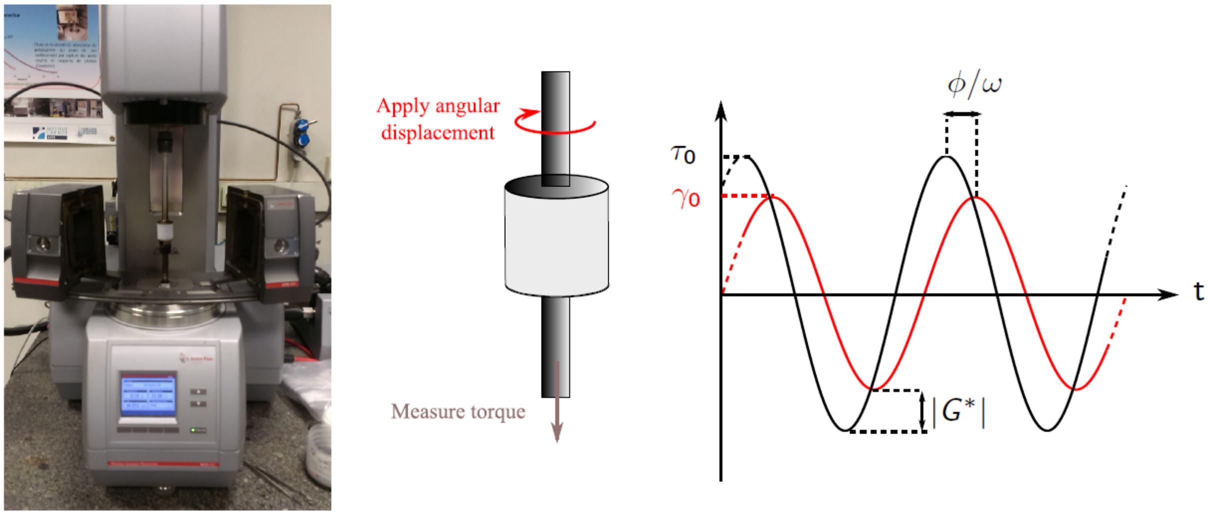

Figure 1: Oscillatory rheometer in torsion MCR 502 (Anton Paar).

In the first section, the characterisation procedure used to measure the complex shear modulus of several foams is described. This procedure is based on the measurement of complex shear modulus by means of a torsional rheometer on a reduced frequency range and at different temperatures, and the application of the time-temperature superposition principle. A fractional derivative model is then identified from experimental master curves to describe the frequency dependency of the material's properties.
In the second section, the vibration damping performance of both foams as a free layer damping treatment is determined through numerical simulation of a simply supported plate. In the finite element modelling, the foam is defined a 3D viscoelastic material, to allow an easy comparison with rubber materials classically used for vibration damping.

2 VISCOELASTIC PROPERTIES OF FOAMS

Two foams have been tested in this work: a melamine foam and a closed cell polyurethane foam. These foams are part of the five porous materials tested in an interlaboratory study on the dynamic elastic properties of poroelastic media [2]. The measurement procedure, based on [3], is described in section 2.1 and the results obtained after application of the time-temperature superposition principle are discussed in section 2.2.

2.1 Experimental set-up

Experimental measurements are performed with a rheomoeter MCR 502 apparatus from Anton Paar. A photo of the experimental set-up is given in Figure 1. Cylindrical samples of porous materials of 24 mm diameter and 25 mm thickness are cut using gasket punches. The samples are glued to the parallel plates of the rheometer using a two sided bonded tape. One end of the sample is glued to a fixed plate, connected to a torque transducer while the other end is connected to a harmonically excited plate. By

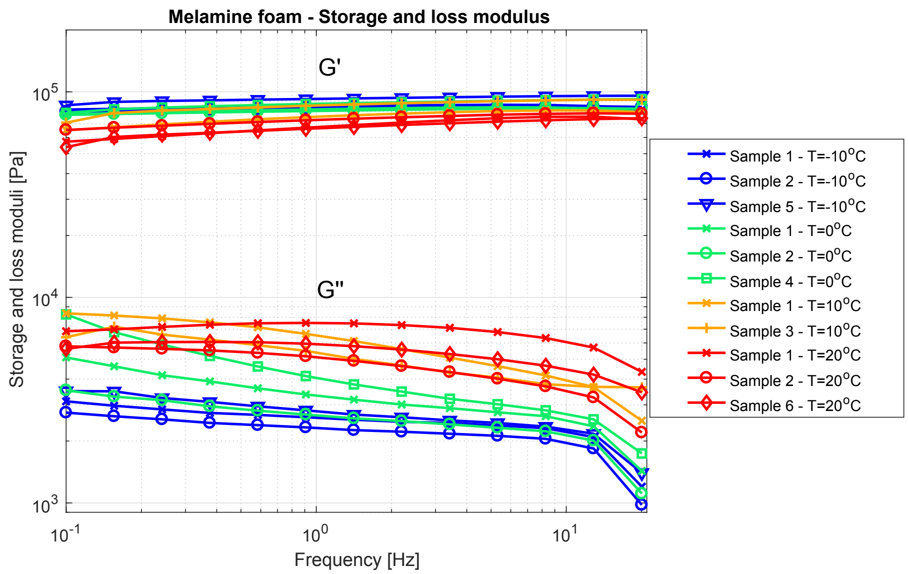

Figure 2: Storage and loss shear moduli for melamine foam samples.

measuring the stress time response τ_0 of a sample of material to an imposed strain γ_0, the rheometer allows the measurement of the complex modulus $G^*(\omega, T)$ as a function of angular frequency ω and temperature T:

$$G^*(\omega,T) = |G^*(\omega,T)|\exp(\mathrm{i}\phi(\omega,T)) = G'(\omega,T)+\mathrm{i}G''(\omega,T) = G'(\omega,T)(1+\mathrm{i}\eta(\omega,T)) \quad (1)$$

where $|G^*|$ and ϕ are respectively the amplitude and the phase of the shear modulus, G' and G'' are the storage and the loss shear moduli, η is the loss factor. In this work, the samples were excited in torsion in the frequency range $[0.1\text{ Hz}, 20\text{Hz}]$, and in the temperature range$[-10^o\text{C}, 20^o\text{C}]$, at 0.1% of dynamic strain to remain in the linear viscoelastic domain.

2.2 Time-temperature superposition principle

The storage and the loss moduli measured from samples at different temperatures of both materials are presented in Figures 2 and 3. One notices that the modulus of the melamine foam slightly depend on frequency and temperature while the polyurethane foam evidences a strong dependency on both parameters.

Direct measurements of the complex modulus are limited to a narrow frequency range. The time-temperature superposition principle can be used to extrapolate the dynamic properties over a broad frequency range at a chosen reference temperature [4]. This technique consists in applying shift coefficients to measured isotherms:

$$\begin{cases} f_{\mathrm{red}} = a_{\mathrm{T}}(T_i)f \\ |G^*(f_{\mathrm{red}},T_0)| = b_{\mathrm{T}}(T_i)|G^*(f,T_i)| \\ \phi(f_{\mathrm{red}},T_0) = \phi(f,T_i) \end{cases} \quad (2)$$

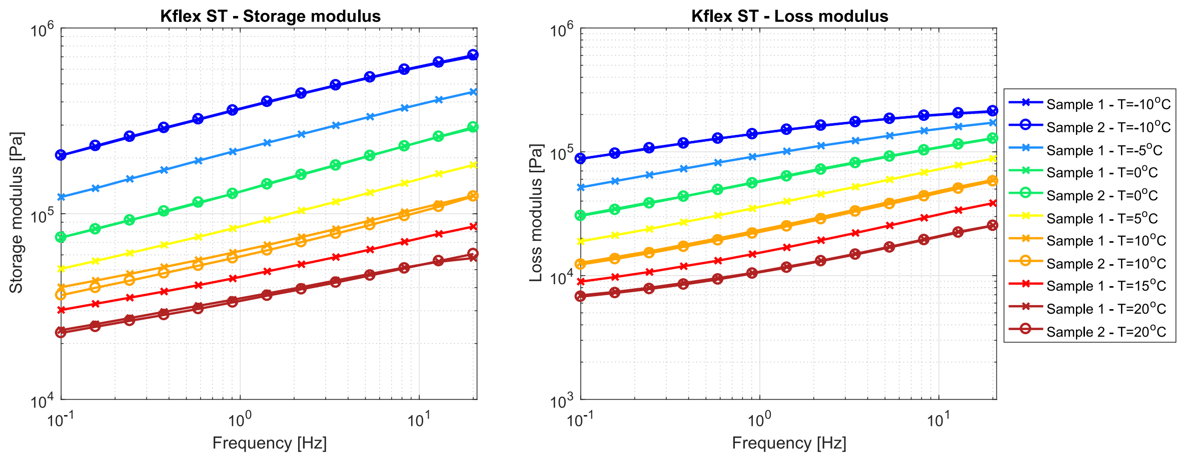

Figure 3: Storage and loss shear moduli for polyurethane foam samples.

where $a_{\mathrm{T}}(T_i)$ and $b_{\mathrm{T}}(T_i)$ are respectively the horizontal and vertical shift coefficients applied to the isotherm measured at the temperature T_i, T_0 is the reference temperature at which the dynamic properties are extrapolated and f_{red} is the reduced frequency after extrapolation. In this work, the shift coefficients are computed following the method described in [5], ensuring the fulfilment of the Kramers-Kronig relations which convey the causality condition. A reference temperature of 20^{o}C is chosen. The master curves obtained for both material are shown in Figures 4 and 5. The melamine foam is a porous material with very light damping, with a maximum loss factor of 0.12 at 0.25 Hz, while the polyurethane foam evidences moderate damping, with a maximum loss factor of 0.48 et 320 Hz.
A fractional derivative model is identified to describe the frequency dependency of both porous materials [6]:

$$G^*(\omega) = \frac{G_0 + G_\infty(\mathrm{i}\omega\tau)^\alpha}{1+(\mathrm{i}\omega\tau)^\alpha} \tag{3}$$

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 in the constitutive equations. The values of those parameters identified from the experimental master curves are:

- for the melamine foam:

$$G_0 = 4.79\ 10^4\mathrm{Pa} \quad G_\infty = 8.63\ 10^4\mathrm{Pa} \quad \tau = 0.132\mathrm{s} \quad \alpha = 0.43 \tag{4}$$

- for the polyurethane foam:

$$G_0 = 1.31\ 10^4\mathrm{Pa} \quad G_\infty = 2.11\ 10^6\mathrm{Pa} \quad \tau = 4.70\ 10^{-8}\mathrm{s} \quad \alpha = 0.30 \tag{5}$$

These parameters lead to a good description of the frequency dependency of the materials' shear modulus, as shown in Figures 4 and 5.

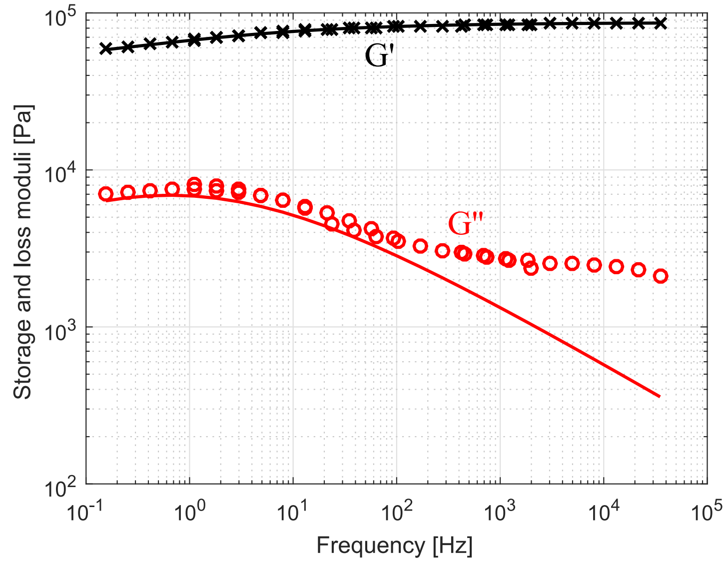

Figure 4: Dynamic shear modulus at 20^o C for the melamine foam. Measurements (markers) and identified viscoelastic model (straight line).

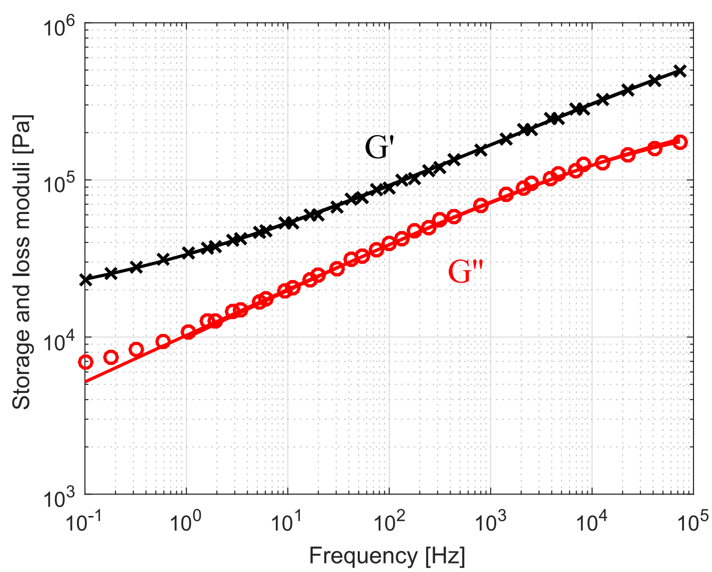

Figure 5: Dynamic shear modulus at 20^o C for the polyurethane foam. Measurements (markers) and identified viscoelastic model (straight line).

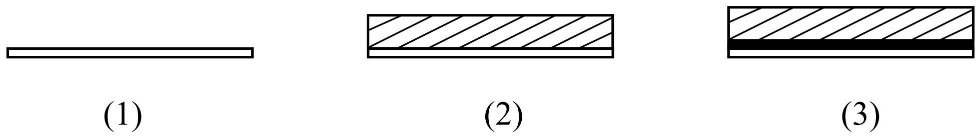

Figure 6: Three configurations under study. Aluminum panel (white), Rubber material (black), Poroacoustic material (hatched).

3 NUMERICAL SIMULATIONS

This numerical study explores the feasability of using the porous materials characterised in the previous section as a free layer damping treatment. Finite simulations on a simple test case aim at comparing the vibration damping performances of both foams to that of a rubber material classically used in the automotive industry.

3.1 Description of the test case

A typical vibration damping soundproof treatment applied in the automotive industry consists in a vibration damping layer (generally made of rubber) secured to the panel surface of the vehicle and a sound proof layer formed on the vibration damping layer [7]. This kind of treatment is classically applied on the floor of the vehicle [1]. To explore the vibration damping performance of foams, three structural configurations are considered:

1. Bare aluminum panel (Fig. 6.(1))
2. Aluminum panel + Poroacoustic material (Fig. 6.(2))
3. Aluminum panel + Viscoelastic rubber material + Poroacoustic material (Fig. 6.(3))

The dimensions of the aluminum panel are 0.42 m $\times$ 0.36 m $\times$ 3 mm. The thickness of the vibration damping layer and the sound proof layer taken in this study are respectively 1 mm and 10 mm. In the configuration n^{o}2, the poroacoustic material is modeled as a viscoelastic material, whose shear modulus depends on frequency, according to the fractional derivative model identified in the previous section and whose Poisson ratio is considered is arbitrarily taken as constant ($\nu = 0.3$ for the melamine foam and $\nu = 0.35$ for the polyurethane foam). In the configuration n^{o}3, the viscoelastic rubber material considered for the vibration damping layer is the one characterised in [8], which is a self-adhesive synthetic rubber from an automotive TIER supplier. The frequency dependency of its shear modulus is described by a fractional derivative model whose parameters are:

$$G_0 = 8.59\ 10^4\text{Pa} \quad G_\infty = 5.34\ 10^7\text{Pa} \quad \tau = 10^{-4}\text{s} \quad \alpha = 0.72 \tag{6}$$

In the last configuration, the poroacoustic material is modelled as an elastic material with structural damping, as it is usually done in vibroacoustic simulations. The shear modulus is then taken as constant and equal to the relaxed modulus G_0 (see Eq. (4) and (5)). A structural damping of 0.06 is considered for the melamine foam and of 0.37 for the polyurethane foam (these values correspond to the average damping ratio over the

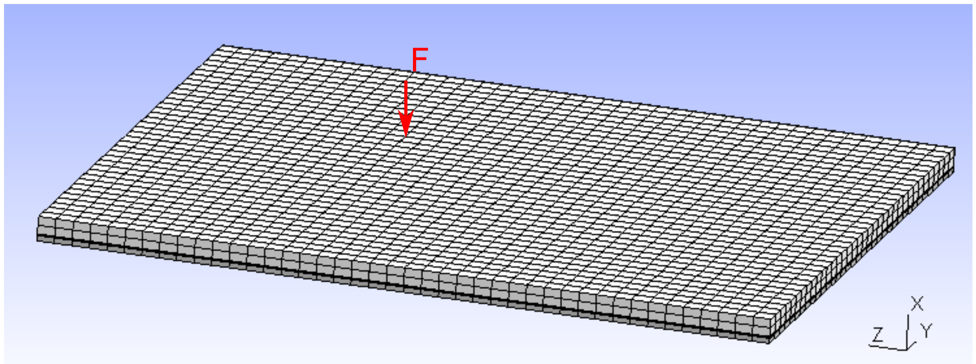

Figure 7: Finite element mesh for configuration 3 and location of the unit load point.

	Melamine foam		Polyurethane foam	
Configuration	2	3	2	3
Added mass (kg)	0.0151 kg	0.1814 kg	0.0726 kg	0.2389 kg
Added mass (%)	1.2%	14.8%	5.9%	19.5%

Table 1: Added mass of the vibration damping (soundproof) treatment for each configuration.

frequency range $[1, 800]$ Hz). The density of the melamine foam is 10 kg/m^3 and that of the polyurethane foam is 48 kg/m^3.
The panels are simply supported. Figure 7 shows the finite element mesh and indicates the location of the unit load point. In the finite element model, all layers are modeled with 20-node hexaedral element. Frequency responses are defined as the ratio between the displacement of the structure along the x-axis at the load point and the applied load. Frequency responses are computed on the frequency range $[0, 800]$ Hz as structural vibration issues generally occurs in the low frequency range.

3.2 Results and discussion

Figures 8 and 9 compare the frequency response of configurations 1 to 3 for the melamine foam and the polyurethane foam respectively. In both cases, the frequency response corresponding to the third configuration is more damped. It should be noted that the differences between both configurations 3 is due to the difference of density between the melamine foam and the polyurethane foam. The latter acts as a more efficient constraining layer for the viscoelastic damping layer.
However, one can notice that the polyurethane foam taken as a free layer damping (configuration 2 in Figure 9) introduces significant damping in the system. This result supports the possibility of tailoring porous materials that sufficiently damp structural vibrations while decreasing the overall weight of the car body, as evidenced by Table 1.

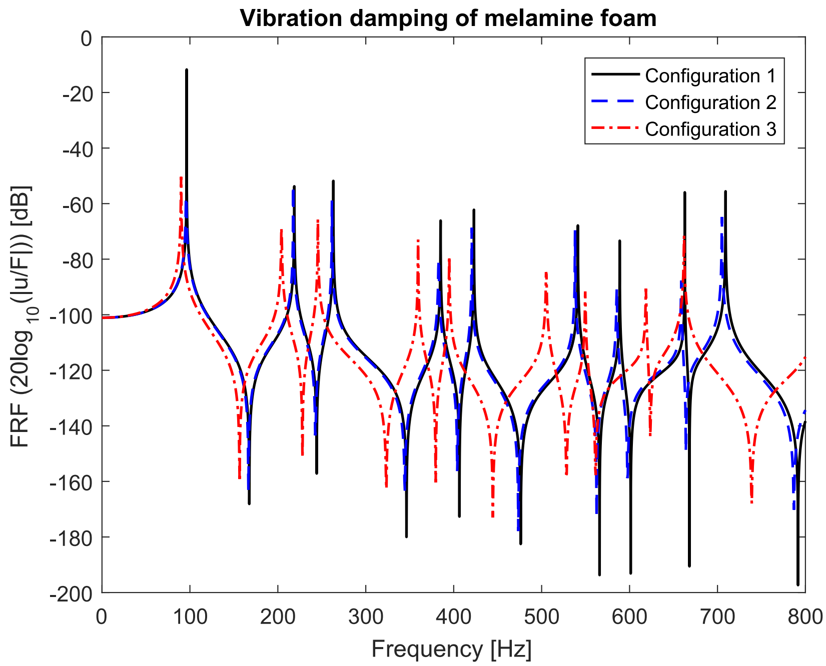

Figure 8: Frequency responses of the panel for configurations 1 to 3 using melamine foam.

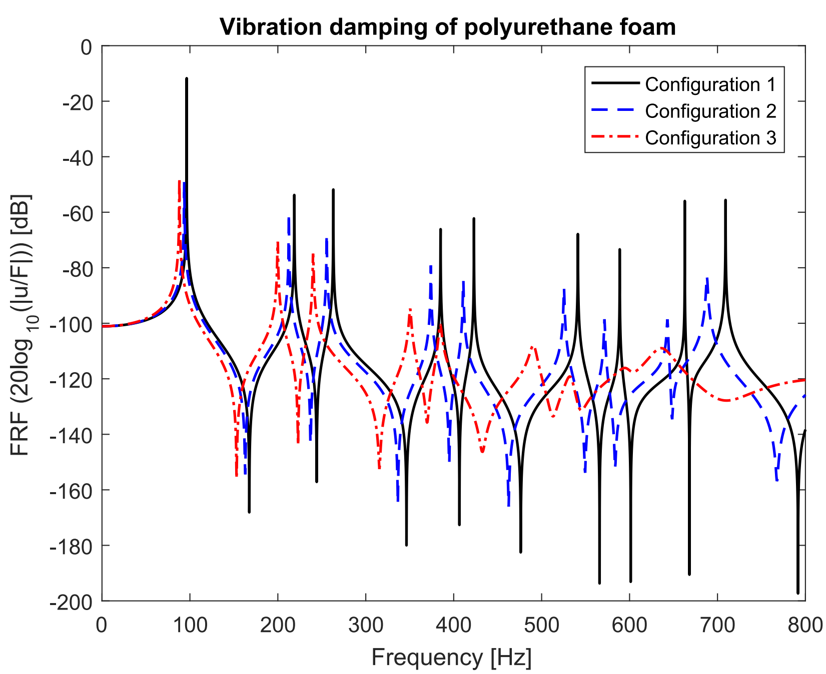

Figure 9: Frequency responses of the panel for configurations 1 to 3 using polyurethane foam.

4 CONCLUSIONS

The goal of this work was to investigate the possibility to use porous acoustic treatments to achieve structural vibration damping. Numerical simulations of multilayer panels were carried out using the viscoelastic properties of two foams identified through DMA measurements. Results show that the polyurethane foam under study taken could be used as a free layer damping treatment to significantly reduce structural vibrations.

In this work, only the viscoelastic properties of foams were under study. The next step is to introduce frequency-dependent elastic properties of foams in a elasto-poro-acoustic model and perform dynamic experiments on multilayer panels. Comparison of numerical and experimental structural responses would help understanding whether the viscoelastic properties measured by DMA could be solely attributed to the dissipative properties of the frame or be influenced by fluid-structure coupling. In [9], the fluid-structure coupling is considered lower in the case of cylindrical samples under torsion loading (as presented in this work), but to which extent?

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