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Effect of embedded resonators in sound insulation panels - A numerical study

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ABSTRACT

Acoustic metamaterials are a current topic of research for many applications. One such application is related to the definition of new sound insulation solutions which avoid well know dynamic phenomena such as the coincidence effect. The present paper studies the behaviour of a specific metamaterial-based solution in which specific elements are embedded in the sound insulation panel itself, forming a structure with complex dynamic behaviour. The main goal is to perform an initial study on the possibility of developing a simple strategy to incorporate resonant structures in wall panels, with the aim of reducing the effect of the critical frequency (coincidence effect) on the sound insulation of such panels. Both 1D and 3D periodic Finite Element models are used in this study, identifying not only the dispersion curves associated with the different modes of the structures but also trying to predict the global sound reduction provided by this complex element. A parametric study is presented to illustrate the concept and to evidence the expected benefits in terms of sound reduction.

Keywords: FEM, Noise reduction, Metamaterials with embedded micro-resonators
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1. INTRODUCTION

The acoustic performance of partition panel elements is usually based either in their mass or in a layered absorptive structure, including materials with different properties (external layers and internal absorbing materials, for example). Recently, increased sound insulation in plate-like elements has been achieved by using different classes of acoustic metamaterials [1-2], such as, plate-type acoustic metamaterials, spring-mass resonators, phononic crystals, elastic metamaterials, membranes in perforated plates or in honeycomb structures, locally resonant sonic materials, metasurfaces or perforated plates. Different research works have been published regarding the topic of using metamaterials to improve sound attenuation, namely [3-5]. In other works, by very active research groups, for example [6-10], solutions consisting of arrays of resonant structures introduced in different types of panels were analysed numerically and experimentally.

In fact, the use of local resonant elements, periodically arranged in a matrix, allows blocking sound waves beyond the limit of the conventional mass density law [11]. Schwan et al. [12] presented the results of the reflection phenomena in an elastic half-space with a “resonant surface” over which linear oscillators are distributed. The authors demonstrated that the surface motion comes to zero in the resonating direction around the oscillators’ eigenfrequency and the surface impedance may be isotropic or anisotropic, according to the type of oscillator.

Xiao et al., in two complementary works [13-14], studied first the propagation of flexural waves (and associated band gaps) in locally resonant thin plates made of 2D periodic arrays of attached spring-mass resonators on thin homogeneous plates, and then they studied the sound transmission loss of these metamaterial-based thin plates. The plane wave expansion method was extended to deal with such plate systems and the band gaps’ structures were analysed. The diffuse field sound transmission loss was numerically computed, with higher values being achieved by the periodic metamaterial plates than bare plates (with the same surface mass density) at frequencies within the mass-law region and the coincidence region.

Claeys et al. [15] assessed the potential of using periodic tuned resonators (point mass or spring-mass resonant elements) to generate vibrational stop bands in infinite and finite plates. Through unit cell modelling, they investigated the vibro-acoustic behaviour of the periodic structure and compared the attenuation factors for both interference and resonance based stop bands. The authors also used the wave based method to numerically simulate a finite plate with a periodic grid of tuned resonators and noticed the presence of stop bands in certain frequency ranges of the response of these finite structures. Stop band behaviour was also studied in plates with tuned resonators, both on vibration and on acoustic levels [6]. It was shown that care should be taken in designing stop bands for vibro-acoustic applications since the characteristic mode split in resonance based stop band materials can drastically influence the radiation efficiency of those materials. To obtain good acoustic stop band behaviour, tuned resonators should be designed above the coincidence effect.

Also based on the plane waves expansion method, Oudich et al. [5] presented two general analytical approaches to study both the sound transmission loss (STL), through a homogeneous elastic plate (thick or thin) decorated with a square periodic array of spring-mass resonators, and its band structure. The STL acoustic performance of this metamaterial plate was analysed for any normal or oblique sound waves incidence, while the metamaterial dispersion behaviour described the vibrational motions of plate and helped to understand the physics associated to sound radiation by the plate. The authors showed that high STL values at sonic frequency range can be attained, which can be useful and efficient for sound and vibration insulation applications, and that the critical frequency effect can be overcome if the spring-mass resonators are appropriately tuned.

In the present work, the incorporation of periodic resonators in the design of partition panels is sought, in order to maximize their acoustic performance without compromising the operability with higher weights. Following an initial previous study, regarding the numerical analysis of partition panels with micro-resonator-type metamaterials and possibilities for materialization [16], we investigate here the concept of periodic metamaterial panel developed by embedding, in existing panels (wood-fibre), small structures consisting of an elastic material around a mass. The main goal is to perform an initial study on the possibility of developing a simple strategy to incorporate resonant structures in wall panels, with the aim of reducing the effect of the critical frequency (coincidence effect) on the sound insulation of such panels. Making use of a general and versatile numerical method, as the Finite Element Method (FEM), implemented in one and three dimensions (1D and 3D), and profiting from the benefits of the periodicity observed in many metamaterials, a numerical conceptual study is presented on the noise reduction achieved by periodic partition panels.

2. NUMERICAL MODELLING

Noise reduction of a single partition panel, along the range of frequencies of interest in building acoustics, is known to be controlled by the stiffness of the panel, at the lowest frequencies, by the mass of the panel, in the mid frequencies range, and by the presence of critical frequencies due to the coincidence effect, at higher frequencies. In current partition panels, this last effect can be particularly penalizing for the global noise reduction. Thus, using metamaterials related concepts, panel configurations that could mitigate this effect are pursued The use of local resonant elements and their periodic distribution, has been demonstrated as a promising methodology to enhance the insulation properties of current acoustic materials, allowing blocking sound waves beyond the limit of the conventional mass density law and addressing other limiting physical phenomena.

2.1 One-Dimensional Finite Element Model

Initially, a simple uncoupled 1D numerical model, based on the FEM [15], has been used with the aim of estimating the sound reduction of a partition panel and numerically verify the effects observed when periodic resonant elements are added to the panel surface. The analytical advantages of the periodicity observed in the analysed metamaterial configuration are here taken into consideration, leading to very fast and efficient computations since a complete spatial discretization of the system is not needed, when numerically analysing periodic partition panels.

For the case of the 1D concept, the numerical model is represented in Figure 1, with the panel being excited by incident plane waves (with different incident directions in respect to the vertical panel, θ) and being modelled as infinite in the longitudinal (vertical) direction. In this approximate model, the strategy of incorporating periodic resonators is performed by adding discrete vibrating mass-spring resonant elements (1 degree of freedom dynamic systems) to the surface of the vertical panel, as illustrated in Figure 1.

Therefore, the following equation of motion can be solved by a classical 1D formulation of the FEM [15], computing the displacement vector, **u**, with the panel being discretized by Timoshenko beam elements (which are known to account for shear deformations),

$$\mathbf{K\,u} - \boldsymbol{\omega}^2\mathbf{M\,u} = \mathbf{F}, \tag{1}$$

where **K** and **M** represent the usual FEM stiffness and mass matrices, **F** represents the external forces and $\boldsymbol{\omega}$ the angular frequency given by $\boldsymbol{\omega}=2\pi f$.

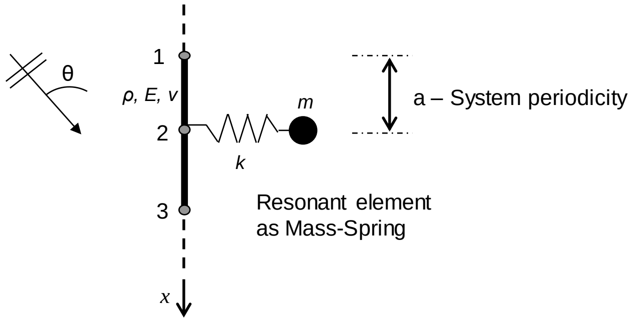


Figure 1 – Schematic representation of the 1D FEM modelling methodology, with periodic panelised system and mass-spring resonant elements.

In the numerical modelling of the periodic panelised system, we can use the Floquet-Bloch theory ([6], [18]) in order to define periodic boundary conditions. Thus, a unitary (isolated) cell of the periodic system can be modelled, resulting in reduced computation times and computer resources. In the numerical 1D FEM model, the following relations have to be prescribed, between displacement valued of adjacent nodes of the periodic system being modelled:

$$u_1 = u_2.\,\mathrm{e}^{-\mathrm{i}k_x a} \tag{2a}$$

$$u_3 = u_2.\,\mathrm{e}^{\mathrm{i}k_x a} \tag{2b}$$

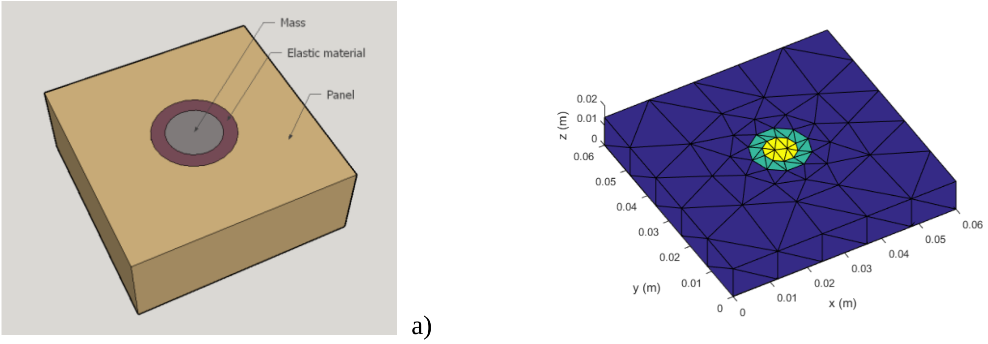
with a representing the spatial periodicity of the system and k_x representing the tangential wavenumber in the periodic direction, given by $k_x = \omega/c \cos\theta$, and c corresponding to the sound propagation speed.

For this case of simple systems, with homogeneous panels incorporating periodic discrete mass-spring resonators attached to the panel surface, the presented numerical 1D FEM methodology can be very efficient in the computation of the vibration pattern of this conceptual periodic systems.

2.2 Three-Dimensional Finite Element Model

In order to make it possible to analyze insulation systems with more complex configurations, namely, with the incorporation of internal (or embedded) resonant structures as schematically represented in Figure 2a), with internal masses embedded in elastic materials and incorporated in the panel plane, a 3D analysis was also performed. In fact, the same ideas referred above were taken into account while developing and implementing the 3D FEM model. The same equation of motion (Equation 1) is now being solved by a 3D formulation of the FEM [15], with the periodic panel with embedded resonators being modelled as a unitary periodic cell that is repeated along the two longitudinal directions (x and y) on the panel plane. Given the periodicity of the system, the Floquet-Bloch theory is once again used, with Equations 2a and 2b being complemented to account for both periodicities along the *x* and *y* directions.

Consequently, the unitary cell corresponds now to a small part of the insulation panel, incorporating an elastic material that enables the vibration of a mass, with a schematic representation given in Figure 2) and the correspondent 3D FEM discretization mesh being presented in Figure 2b).



a) b)

Figure 2 – Conceptual model of panel with embedded resonant structures: a) scheme of the unitary cell with internal resonator; b) proposed unitary cell numerical discretization of a panel with periodically distributed embedded resonators

In this case, the more complex vibrational behaviour of the mematerial panel was efficiently computed by the 3D FEM periodic model.

2.3 Sound reduction and Dispersion curves

Once the solution of the 1D FEM or 3D FEM models, in terms of nodal displacements of the vibrating surfaces are obtained, the radiated sound by the elastic panel can approximated by:

$$|p_{rad}| = \rho c|v| \tag{3}$$

where p_{rad} represents the radiated sound pressure, ρ the air density, and v the velocity of the vibrating panel.

Then, the sound reduction curve achieved by the periodic panel has been numerically evaluated by the sound transmission loss, given by the difference of two sound pressure levels, namely the sound pressure level corresponding to the incident plane wave impinging the panel and the sound pressure level determined by the sound radiated by the periodic panel, which can be written by

$$R = 20\log\left(\frac{|p_{inc}|}{|p_{rad}|}\right). \tag{4}$$

In both cases, oblique incident waves have been taken into account, with inclination angles from $\theta = 0$ to $\theta = \pi/2$, corresponding to varying directions from tangential incident waves to normal incident waves, respectively.

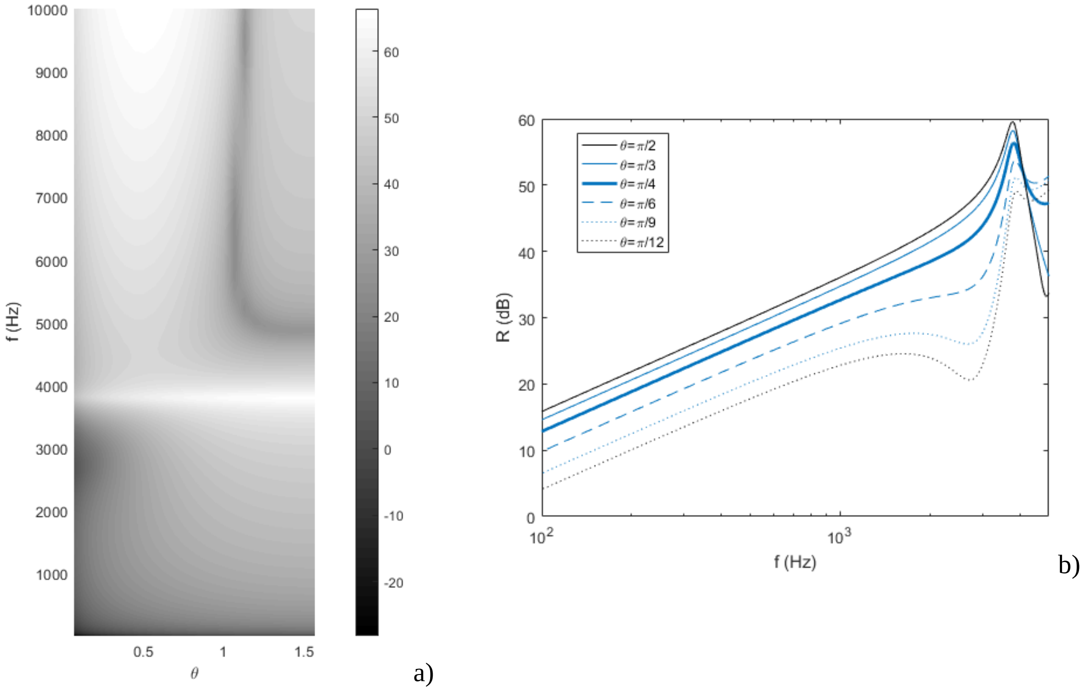
On the other hand, the metamaterial dispersion curves describe the vibrational motions of the periodic panel and can help understanding the physics related to the panel sound radiation. They represent the relation between angular frequency and the tangential wavenumber, and they identify the modal propagation/radiation behaviour of the metamaterial panel and the correspondent band structure. These curves are determined by the solution of the eigenvalue problem of the FEM matrix of the periodic unitary cell, getting the angular frequencies for each wavenumber. The presence of a stop band or band gap, which corresponds to a range of frequencies where all acoustic/elastic waves cannot propagate through the periodic system, is noticed by the absence of eigenvalues in the frequency range and it can be determinant when assessing the insulation behaviour of this type of system.

3. NUMERICAL RESULTS

3.1 1D FEM Results

Let us initially consider a homogeneous thin particle board panel (8 mm thick) where a set of mass-spring resonators is periodically attached to its surface. This conceptual metamaterial panel solution is here numerically analysed with the previously described 1D FEM model, for the case of resonators tuned in a resonance frequency of f_{res} = 3800 Hz, which is near the critical frequency of the thin panel.

Figure 3a) illustrates the sound reduction provided by the metamaterial panel for the complete set of incidence angles, from θ=0 to θ =π/2, with the latter corresponding to normal waves exciting the panel, in the frequency range up to 10 kHz.



a) b)

Figure 3 – Parametric analysis performed with 1D FEM model of panel with attached elastic resonators, for varying sound wave incidence angles, θ: a) noise reduction levels for varying (θ,f); b) noise reduction curves for 6 sound wave incidence angles.

In this figure, the variation of the sound reduction can be observed in a 2D coloured plot for varying incidence angles. The favourable contribution of the resonance of the discrete periodic elements attached to the panel surface is clearly visible (lighter grey tones correspond to higher sound reduction) whatever the incidence angle, but also some higher incidence angles with a sound reduction dip (darker grey tones correspond to lower sound reduction) associated to the coincidence frequency. In Figure 3b), the sound reduction curves, computed for the metamaterial panel can be observed in detail, for six specific sound incidence angles. As expected, the maximum sound reduction occurs at the resonance frequency of the periodic resonators.

Next, if we focus in the sound incidence angle of π/4, the 1D FEM results are presented in Figure 4. On the left side (Figure 4a)), the computed dispersion curves of the metamaterial panel can be observed as well as the dotted line corresponding to the incident sound wave for θ=π/4. The band gap formation is shown to occur in the approximate frequency range from 3800 to 5000 Hz, with the separation of the resonant and propagation dispersion curves. This cancelation effect can also be observed in the comparison illustrated in Figure 4b), with the sound reduction curves of the bare panel and the panel with attached resonators plotted with the theoretical mass law prediction. For the analysed incidence angle, the coincidence frequency dip in the noise reduction curve of the bare panel is being cancelled by the periodic tuned resonator attached to the panel surface.

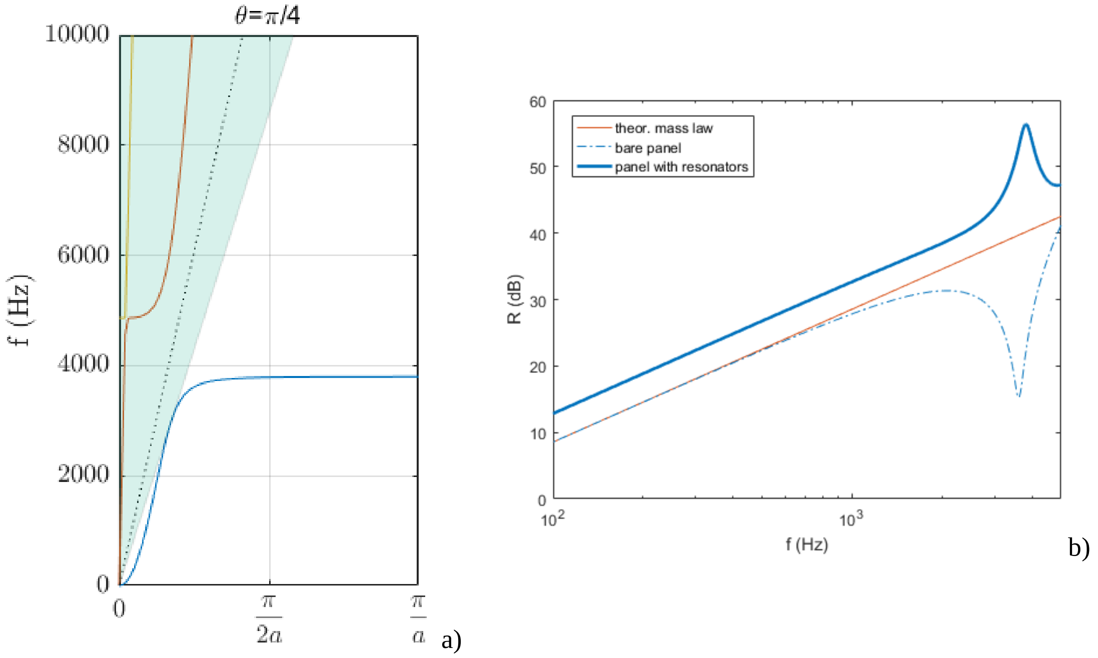


Figure 4 – Application of the 1D FEM model to panel with attached elastic resonators, for a sound incidence angle of π/4: a) dispersion curves and band gap formation; b) computed sound reduction for bare panel and panel with resonators, and theoretical mass law.

3.2 3D FEM Results

For the second set of results, a numerical FEM analysis of the 3D configuration illustrated in Figure 2 was performed, based on a unitary cell that is periodically repeated along both *x* and *y* directions. In this second configuration of an insulation panel, the resonant structures are embedded in the panel plate, with an interior mass confined by an elastic material. Thus, the 3D character of the vibrational and dynamic behaviour of the system can only be analysed in detail by a 3D numerical model. Due to the periodicity of this conceptual system, the previously described 3D FEM model can efficiently perform both the analyses of the metamaterial dispersion curves and the sound reduction through the periodic panel.

Let us now consider a homogeneous particle board panel (14 mm thick), with a set of resonators periodically embedded along its plane, forming a regular grid with 6 cm periodicity constant in both *x* and *y* directions. In this conceptual model, the resonant structures are formed by a cylindrical heavy mass (with 8.4 mm diameter) embedded in an elastic material (with exterior diameter of 15 mm).

The dispersion curves computed for the periodic unitary cell of are illustrated and can be compared in the case of a bare panel (Figure 5a)) and in the case of the same particle board panel with the embedded resonant elements (Figure 5b)). The complex structure of the vibrational modes is clearly visible in the case of the metamaterial configuration, and the formation of a small band gap slightly above 2000 Hz is observed, corresponding to local resonance observed when the embedded mass is vibrating in the transverse direction to the panel.

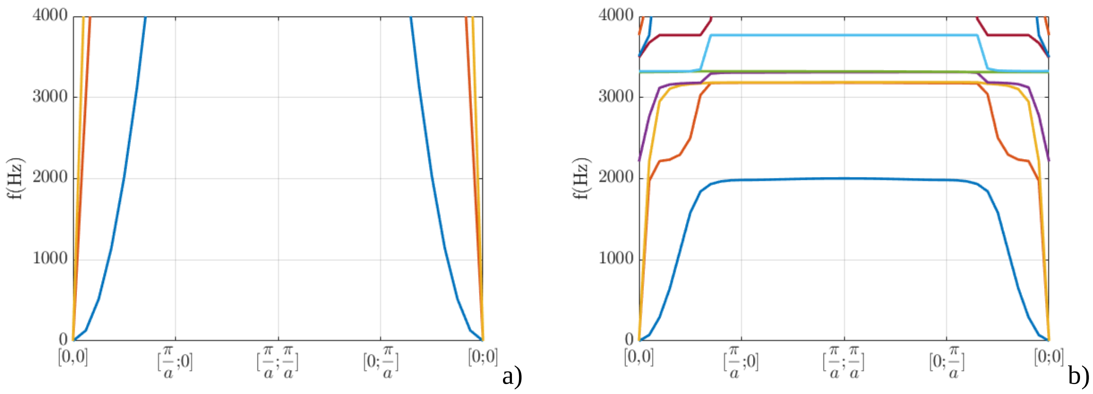


Figure 5 – Panel with embedded resonant structures (see Figure 2), dispersion curves obtained with 3D FEM model for unitary cell: a) bare panel: b) panel with embedded resonant structures.

Then, a brief parametric analysis was carried out, using the 3D FEM model, and some results are presented in Figure 6. In these plots, the estimated sound reduction achieved by the panels, for an incidence angle of $\theta = \pi/4$, is compared while assessing the influence of different Young modulus of the elastic material that confines the vibrating mass (Figure 6a)) and of different mass density of the cylindrical embedded mass (Figure 6b)).

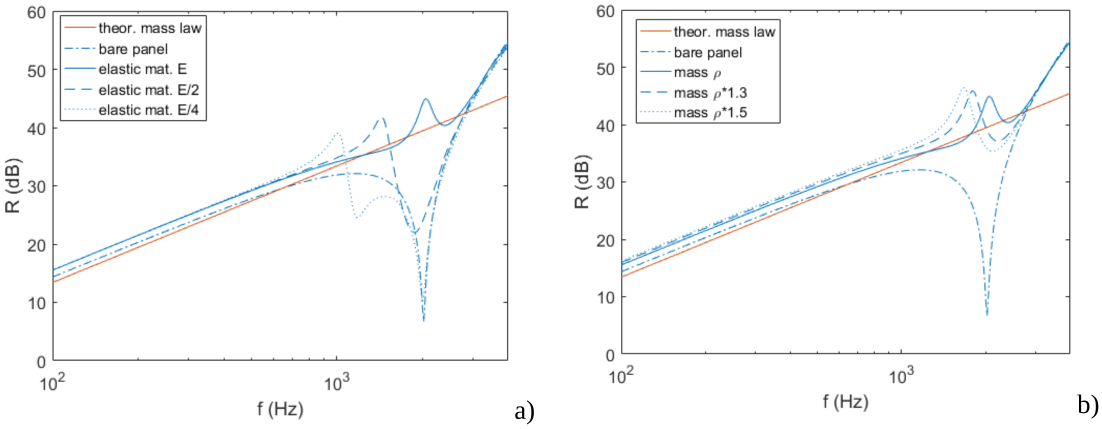


Figure 6 – Panel with embedded resonant structures (see Figure 2), numerical analysis with 3D FEM model of estimated sound reduction: a) with varying Young modulus of the elastic material; b) with varying mass density of embedded mass.

For reference purposes, the theoretical mass law for a homogeneous panel is also included in the plots, as well as the sound reduction curve of the bare panel, which exhibits a very marked dip around 2000 Hz corresponding to the panel's critical frequency. In both plots, one can observe the effect of adjusting the properties of the locally resonant structures, in order to achieve the best cancelling results of the coincidence effect that greatly penalizes the sound insulation of the bare panel.

4. CONCLUSIONS

In the present work, a preliminary numerical study of the effect of the presence of periodic resonant elements in the configuration of sound insulation panels has been performed. Two FEM models have been implemented, in 1D and 3D, in order to evaluate the dispersion curves of these metamaterial panels as well as estimate their sound reduction for different sound incidence angles. Given the periodic character of the proposed panels, the numerical analysis was based on the unitary cell that is being periodically distributed along the panel surface, which results in very efficient computations. As conceptual models, in the 1D case, mass-spring resonators were considered to be attached to the surface of the insulation panel, and, in the 3D case, the resonant structures correspond to embedded masses confined by elastic material. The vibrational and dynamic behaviour of these periodic systems has been illustrated and the practical interest of adjusting the resonant elements seems to be promising for cancelling the coincidence effect observed in insulation panels. Further studies are now concerning different possibilities of materializing these concepts.

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