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Design and fabrication of a metamaterial with resonators to dissipate mechanical vibrations

Diseño y fabricación de un metamaterial con resonadores para disipar vibraciones mecánicas

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Abstract

Mechanical vibrations have been a recurring problem in manufacturing industries, so, its control and isolation are a focus of much research. Metamaterials are a new concept about artificial materials with stunned properties. It means that the material's cellular architecture is designed to improve its mechanical properties, including damping capability. This work proposes a novel metamaterial focus on improving its damping capability. Its cellular architecture is based on a structure to support load and resonators to dissipate vibration energy. The resonators are designed and studied around a specific range of frequencies by a numerical model. Physical models are fabricated by additive manufacture and tested to determine the transmissibility coefficient. The results of experimental tests are reported, and those demonstrate that the damping capability is enhanced by de metamaterial proposed.

Keywords: Metamaterial; cellular architecture; vibrations; resonator; transmissibility.

Resumen

Las vibraciones mecánicas han sido un problema recurrente en las industrias manufactureras, por eso, su control y aislamiento son foco de muchas investigaciones. Los metamateriales son un nuevo concepto de materiales artificiales con propiedades sintonizadas. Esto quiere decir que la arquitectura celular del material es diseñada para mejorar sus propiedades, incluyendo la capacidad para amortiguamiento. Este trabajo propone un nuevo material centrado en el mejoramiento de su capacidad de amortiguamiento. Su arquitectura celular está basada en una estructura para soportar carga y con resonadores para la disipación de energía de vibración. Los resonadores están diseñados y estudiados alrededor de un rango de frecuencia específico a través de un modelo numérico. Un modelo físico es fabricado con manufactura aditiva y ensayado para determinar el coeficiente de transmisibilidad. Los resultados de las pruebas experimentales son reportados, y estos demuestran que la capacidad de amortiguamiento ha mejorado por el metamaterial propuesto.

Palabras clave: metamaterial; arquitectura celular; vibraciones; resonador; transmisibilidad.

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1. Introduction

Mechanical vibration poses a prevalent challenge in production systems. The objective of numerous studies is to mitigate vibration amplitudes or isolate productive systems. Fundamental isolation is achieved through material damping. In rigid materials, the damping property enables the reduction of mechanical vibrations to a limited extent.

Metamaterials are synthetic materials engineered to exhibit unconventional mechanical properties that diverge from those found in natural materials. So, a metamaterial can be designed to improve, for example, its damping capability based on a cellular architecture that lets to do that.

This work presents a brief review about metamaterials, isolation materials, and resonators applications topics. In material and methods section a metamaterial is designed. First, four cellular architectures are proposed to be studied, and then, resonators are design. Both constitute, properly configured, the cellular architecture of the metamaterial proposed. This metamaterial is designed and studied by numerical model, and experimental studies are also conducted. Finally, the results section reports on the isolation capability of the metamaterial across a defined frequency range.

2. Literature review

Metamaterials are artificial materials designed and constructed with the aim of obtaining in them unusual mechanical properties which are not peculiar to materials in their natural state [1]. Examples of such properties are negative refraction, negative permeability, negative thermal expansion coefficient, and negative Poisson coefficient, among others ([2], [3]). Due to the variety and expectation of applications that these materials may have, in the last decade the scientific community has carried out numerous theoretical, numerical, and experimental investigations in this field [4]. Different applications of this type of material recorded in the literature include invisibility [5], attenuation of mechanical energy [6], [7], [8], the ability to perfectly focus a light point [9] and acoustic black holes [10].

In view of this wide range of possibilities for use in the field of metamaterials, in recent years elastic-acoustic materials have been created, among other things, consisting of defined periodic structures [11]. These have unique properties such as attenuation of vibration amplitudes, negative permeability and negative mass or density, among others [6].

One of the most important operating characteristics of elastic-acoustic metamaterials is the existence of unwanted frequency bands, ranges in which the propagation of certain waves is filtered or attenuated. This has allowed them to be used in different high and low frequency vibratory energy dissipation applications. Unwanted frequency bands are usually dissipated by two mechanisms, Bragg scattering and local resonance. By means of Bragg dispersion it is not easy to control low frequency bands [12], due to the dimensioning of the structure matrix [13]. While local resonance can be used for the dissipation of vibratory energy at low frequencies [14]. Elastic-acoustic metamaterials obtain the local resonance behavior due to the addition of resonators in the cell structure [15]. These resonators are activated when they reach their natural frequency [16]. There are many types, such as translational and rotational resonators. These resonators generate a negative effective density and a negative effective elastic modulus respectively [7].

Due to the effectiveness of these materials and their designs for sound insulation and vibration attenuation, they have been used in different applications. One of these is for wave attenuation in pipes, where resonators were added to dissipate bending waves. These waves are in a specific frequency range and impair the operation of the system [17]. Experiments have also been conducted to see the effect of buckling on elastic-acoustic metamaterials. In the work reported by Wang et al [18], resonators with a metal core were used in an elastic matrix and it was identified that deformation could activate or deactivate the resonator to control a range of frequencies of interest.

As expected, in metamaterial applications with local resonators it is essential to know the natural frequency at which these resonators operate and thus determine the frequency range at which they operate properly [19]. Therefore, to design or calculate the local frequency of the resonator, it is necessary to perform a modal analysis using the finite element method [20]. This method allows to know some static and dynamic properties of the metamaterial by different types of numerical analysis such as static [21], harmonic [22], PSD response [23], among others. The results of these studies allow to analyze the relationship between these properties and the cellular structure of the metamaterial, and even to graphically visualize the system's response to different operating conditions. Likewise, assisted design, and models, allow making decisions regarding the manufacturing processes of this type of materials, being additive manufacturing an interesting manufacturing process for this type of materials.

Additive manufacturing is one of the methods of manufacturing metamaterials, it stands out for its ability to manufacture, with thermoplastic polymers, pieces of geometric complexity and characteristics of flexibility. In addition, this technique optimizes the consumption of raw material [24]. PLA is used in this type of manufacturing because it is economic, has good mechanical properties and its raw material is derived from renewable resources [25]. These materials have been developed commercially with different types of reinforcements that allow it to improve its physical-mechanical properties. Therefore, the proposed cellular architectures of metamaterials, additive manufacturing technologies and the development of manufacturing input materials have allowed metamaterials to begin exploring a wide range of applications, including vibratory control and isolation.

The presence of vibrations in production systems with amplitudes outside the permissible range can generate failures that involve inefficient production, financial losses, and even risk the health of operators. Companies present problems with the treatment of direct transmission vibrations. Therefore, the need arises to implement isolators or attenuators to reduce, as much as possible, the magnitude of force or motion transmitted by the source [26]. To determine the efficiency of the isolators, the transmissibility test, used to determine the amount of energy transmitted from a disturbance source to the test object, is implemented [27].

This article proposes 4 designs of a metamaterial, with a cellular architecture combined with resonators to dissipate low frequency mechanical vibrations, in a range of 27 Hz to 33 Hz. The local resonance method was used to dissipate energy. The resonators were designed as suspended masses, to dissipate energy by means of the displacement of the resonant mass [19]. The four proposals have been studied and evaluated, as part of the methodology, to select a metamaterial with better capacities to dissipate vibration energy. The results of the transmissibility coefficients are presented, as well as the responses of the metamaterial to the excitation of the test pieces in a range of frequencies of interest.

3. Materials and methods

The Figure 1 presents, through a flow diagram, the steps that make up the methodology used to conduct the present research, methodology described below:

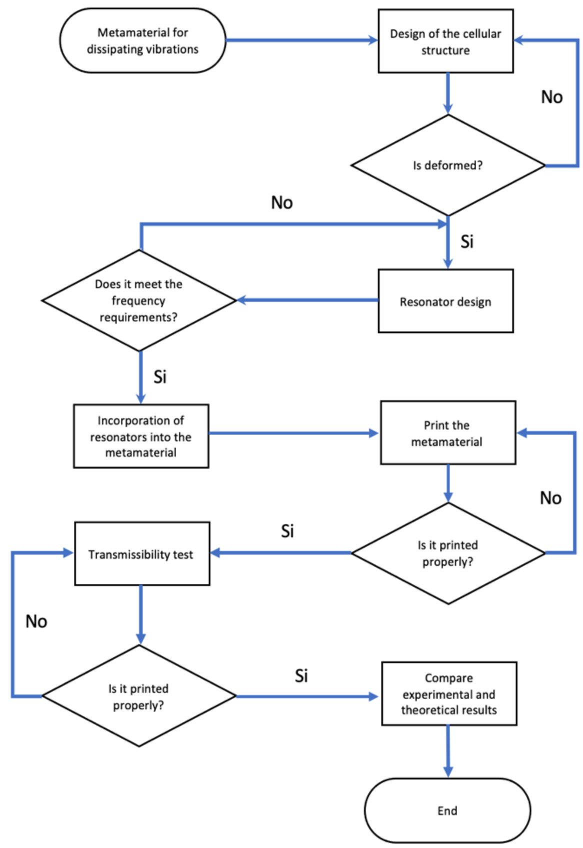

Figure 1. Flow diagram of metamaterials with resonators. Source: self-made.

3.1. Design of the cellular structure of metamaterial

Four alternatives of cellular structures have been developed. These proposals were based on parameters such as geometry, dimensions, and rigidity [4], the latter to ensure its carrying capacity. The proposed structures have been designed and manufactured in 100X100X20 mm, so that they can be tested on a test bench to estimate the coefficient of transmissibility. The structures developed and analyzed are as follows:

a) *Circular cellular structure*: it consists of a circular matrix of 3X3, see Figure 2(a), with a diameter of 34 mm, with a depth of 20 mm and a space between circles of 1 mm.
b) *Square cell structure*: it consists of a pattern of 8 squares, see Figure 2(b), with dimensions of 32X32 mm, depth of 20 mm and thickness of 1 mm. The distribution of the squares is alternately to improve their load capacity.
c) *Hexagonal cell structure*: it consists of a pattern of 8 regular hexagons, see Figure 2(c), which is inscribed in a circle of 32 mm, with 18.48 mm edge, have a depth of 20 mm and a thickness of 1 mm.

d) *Octagonal cell structure*: it consists of a pattern of 8 regular octagons, see Figure 2(d), with equal edges of 13.25 mm, inscribed in a circle of 32 mm, have a depth of 20 mm and a thickness of 1mm.

The four alternatives at the top and bottom have a support plate with dimensions of 100X20X1 mm, which provides stability to each of the structures and serves as a support and anchor for the transmissibility test.

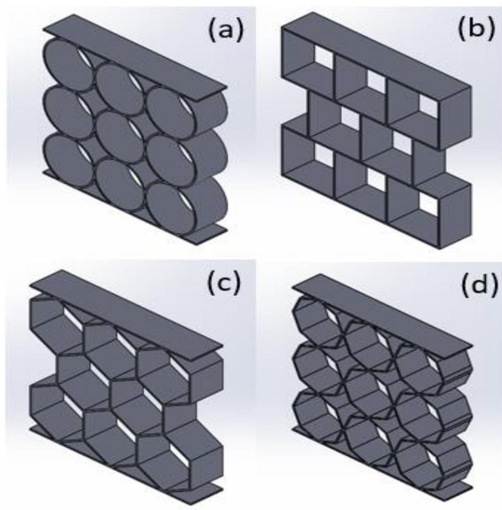

Figure 2. Cellular structures of the metamaterial (a) Circular structure (b) Square structure (c) Hexagonal structure (d) Octagonal structure. Source: self-made.

3.2. Design of resonators

Metamaterials can harbor resonators by improving the efficiency of the resonator for dissipating the energy of mechanical vibrations [16]. Resonators act when the excitation frequency is near or coincides with the natural frequency of the resonant mass.

3.2.1. Estimated natural resonator frequency

The resonators are made up mainly of 2 elements: the support (a think bar) and a hanging mass. The calculation of the natural frequency considering the ratio between the moment of inertia of the suspended mass and the bar holding it. The Figure 3 shows the free body diagram of the generic resonator concept, where: *M* is the suspended mass, *m* the mass of the support bar, *cm* the center of mass, *l* the length of the support and *O* the pivoting point of the system. The mass suspended has a specific geometry with its inertia moment.

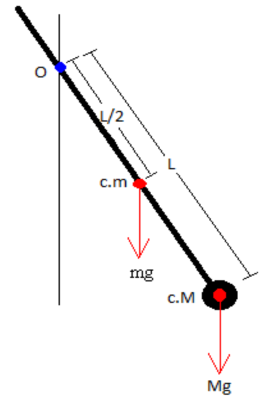

Figure 3. Free body diagram of the generic resonator concept. Source: self-made.

The design of the resonator begins with the estimation of the first natural frequency of the resonator, from equation (1)

$$f = f_0 \sqrt{\frac{g(2ml + Ml)}{2(I_m + I_M)}} \tag{1}$$

Where f_0 is an initial excitation frequency, *g* the acceleration of gravity, and finally *Im* and *IM* the moments of inertia of the suspending element and the suspended mass respectively.

3.2.2. Alternatives for the resonators

It is stated that the design of the resonators should be such that they act dissipating the energy of vibration in a frequency range between 27 Hz and 33 Hz. Similarly, the design considers that the resonator will be housed, embedded, in a geometry that restricts its dimensions and its movement. Four resonator alternatives are proposed and studied. Each of them is described below:

a) Hexagonal resonator: Formed by an irregular hexagon with the following external dimensions: 2 edges of 16 mm and 4 diagonal edges of 6.32 mm. The hexagon is hollow, with a thickness of 2 mm and a depth of 20 mm, see Figure 4(a).
b) Rectangular resonator: This resonator has a rectangular shape, with interior dimensions of 6X10 mm and exterior dimensions of 12X16 mm, with a depth of 20 mm, see Figure 4(b).
c) Trapezoidal resonator: This alternative has a trapezoid shape with external dimensions: short base of 14 mm, long base of 21 mm and height of 12 mm. The trapezium is 2 mm thick, 20 mm deep, see Figure 4(c).
d) Triangular resonator: This resonator is composed of an isosceles triangle, with external dimensions of:

base 20 mm and height 16 mm, and interiors of: base 10 mm and height 8 mm, with a depth of 20 mm, see Figure 4(d).

All alternative resonators are sustained by a support of dimensions: 12 mm long, 0.96 mm wide and 0.96 mm deep.

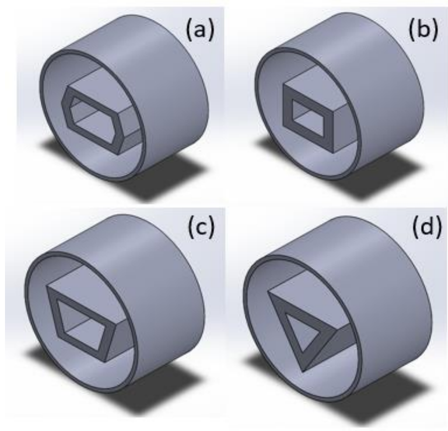

Figure 4. Resonator alternatives (a) Hexagonal resonator (b) Rectangular resonator (c) Trapezoidal resonator (d) Triangular resonator. Source: self-made.

3.3. Test resonator

The precision in the measurement of the natural frequency of resonators was validated by means of a modal analysis in the laboratory. This analysis was performed, by designing a test resonator. Which has a natural frequency and larger size. This facilitates data collection in experimental modal analysis.

The test resonator is designed using a scaling process, using SolidWorks software. The final features of the resonator are as follows: first natural frequency close to 100 Hz and trapezium geometric shape. It has dimensions of 18 mm smaller base, 36 mm larger base, 22 mm high, with a depth of 20 mm and a thickness of 6 mm. Which is attached to a support 36 mm long, 5 mm wide and 5mm deep. Inscribed in an outer diameter circle of 102 mm, with a thickness of 3 mm. The Figure 5 shows the isometric and frontal view of the proposed test resonator.

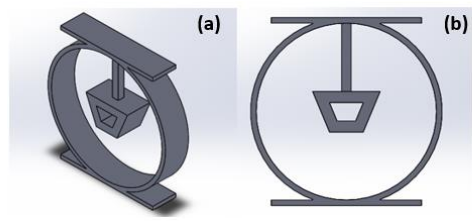

Figure 5. CAD model of the test resonator
(a) Isometric view of the test resonator
(b) Front view of the test resonator
Source: own elaboration.

3.4. Numerical analysis of the test resonator

The numerical modal analysis of the test resonator was performed using Ansys software. Different natural frequencies were identified, the value of the first natural frequency was 99.01 Hz. The deformation analysis was performed at this frequency and a minimum deformation of 0.0215 mm, and a maximum deformation of 10.716 mm was obtained, as seen in Figure 6.

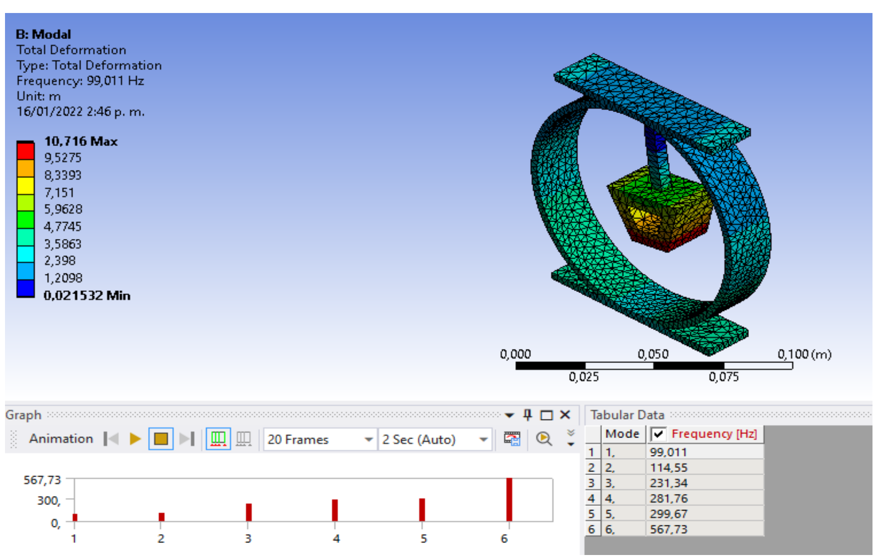

Figure 6. Modal analysis of the test resonator performed at Ansys. Source: self-made.

3.5. Physical model of the test resonator

To perform the experimental modal analysis (EMA), it is necessary to manufacture the physical model. This process was performed with additive manufacturing, using PLA filaments, using a 3D printer. The physical model of the test resonator is presented in Figure 7.

Figure 7. Physical model of the test resonator. Source: self-made.

Experimental modal analysis (EMA) of the test resonator: For the realization of the EMA it is necessary to simulate the free – free condition from the system. For this, the piece in study is suspended through elastic elements that allow it to vibrate freely, see Figure 8. The study piece is excited through an impact generated and measured with a hammer 5850B of Dytran Instruments.

The vibratory response of the part is measured using a piezoelectric accelerometer, sensors 3056D2 of Dytran Instruments INC, of 100 mV/g. The structure is struck at different points previously selected to have an adequate mapping of the frequencies and modal forms. The system used is an OROS Or35, 8 channels, with software for data acquisition and modal analysis, the last one allows visualizing modal forms from the mapping of excitation points and planned measurement.

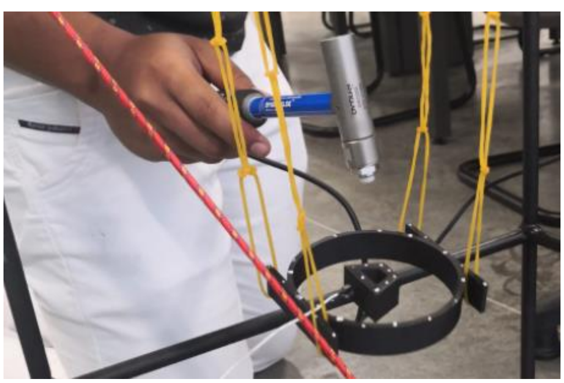

Figure 8. Mounting of the resonator modal analysis. Source: self-made.

At the end of all impacts, the software calculated the natural frequency of the model which was 98.35Hz. See Figure 9, by the modal Or35 software.

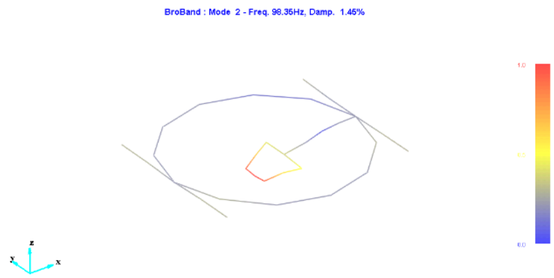

Figure 9. Modal analysis result of the test resonator. Source: self-made.

The numerical and experimental results about natural frequencies have been compared. The resulting error, generated by the numeric model, is less than 1% estimated by equation (2). This model allows the projection of resonator designs at the implementation scale for the metamaterial.

$$\left|\frac{measured_{value} - estimated_{value}}{estimated_{value}}\right| * 100 \qquad (2)$$

Cellular structures with resonators: The 4 structures were combined: circular, square, hexagonal, and octagonal; with the 4 alternative resonators: hexagonal, rectangular, trapezoidal, and triangular, for a total of 16 alternatives, as shown in Figure 10.

3.6. Numerical model

The proposed cellular architectures have been previously studied through a numerical model implemented in Ansys. This model allows to establish the response of the metamaterial in condition: static, harmonic excitation and free vibration condition - free for modal analysis.

Static analysis allows to observe the behavior of each of the alternatives when exposed under a load. When performing this simulation, the program shows the deformation suffered by the material, showing its critical points [21]. In this way, and since the material must withstand a load in operation, it is evaluated that the static deformations of the material are in a range that allows to operate properly.

The Figure 11 shows the deformation of each of the alternatives when exposed to a static load of 30N evenly distributed on the upper face of the material.

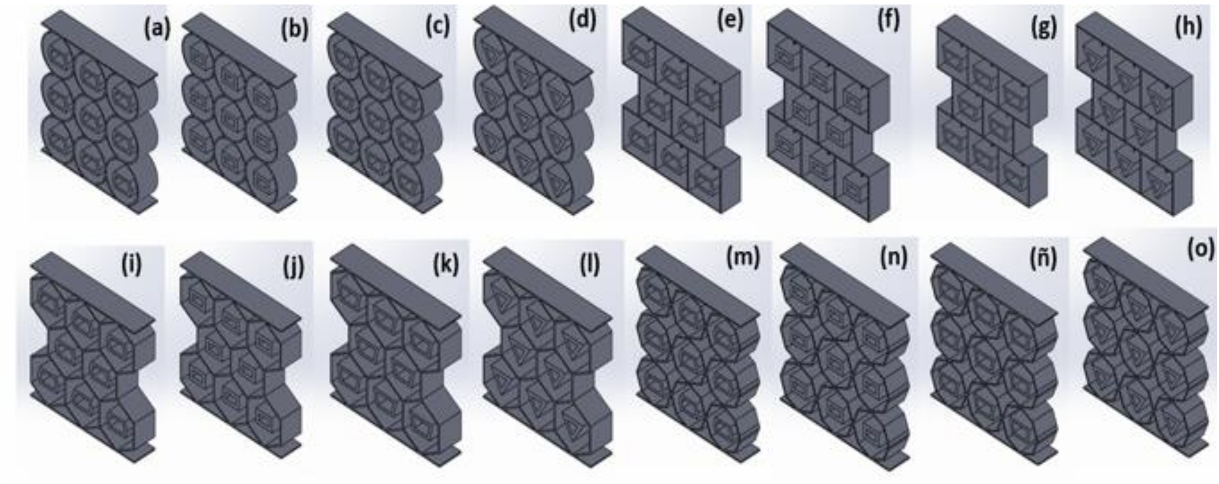

Figure 10. Model of cellular architecture with resonators. Source: self-made.

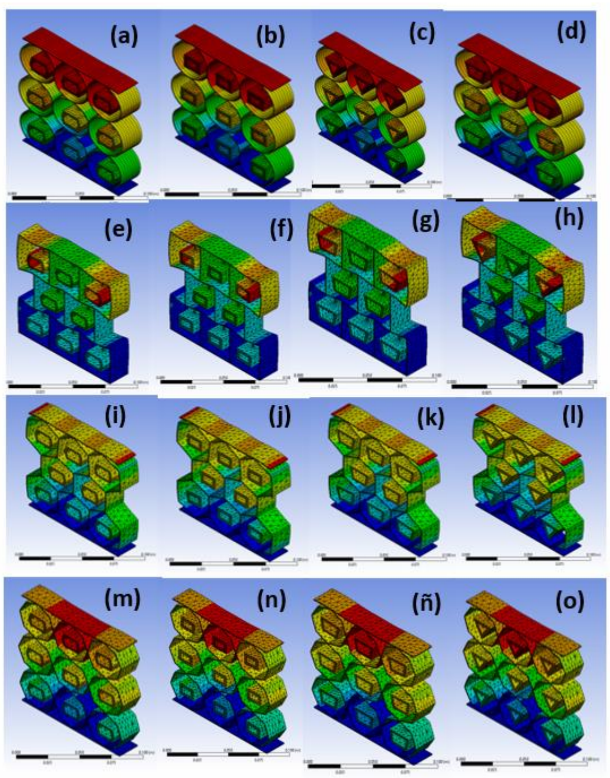

Figure 11. Static analysis of structures with resonators. Source: self-made.

Modal testing, this analysis allows estimating the natural frequencies, vibration modes and damping, of each of the alternatives, and observing the deformation of the material to each of the natural frequencies found [20].

The interest focuses on the natural frequency found in the study frequency range, and for which the resonator was designed. The Figure 12 shows the modal behavior of resonator structures at the first natural frequency.

Harmonic analysis, in mechanical vibrations is a method used to analyze the vibratory behavior of a system. It consists of breaking down the total vibration into a series of vibrations of different frequencies, known as harmonics. This allows to analyze each frequency independently and determine how each contributes to the total vibratory behavior of the system. Harmonic analysis allows us to calculate a ratio of amplitude, speed, or acceleration vs frequency, which allows us to observe the system's response to different frequencies [22].

This study analyzed deformation vs frequency; in Table 1 we can see the maximum deformation suffered by each of the alternatives to a given frequency.

In natural frequencies the alternatives reach their maximum deformation. The most deformed alternative is the structure d and resonator b with a value of $3{,}7428 * 10^{-3}$ m, and the least deformed is the structure a) and resonator b) with a value of $5{,}5689 * 10^{-6}$ m.

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Selection of alternatives: Since the study presents several alternatives of metamaterials, a selection of those that have, in general, a better behavior in face of vibratory isolation in the established frequency range is made. To do this, and from the numerical analyses studied of each alternative, a weighting and prioritization matrix is used, which has led to continue the study with the following proposals: circular and hexagonal architecture, both without resonator and with trapezoidal resonator.

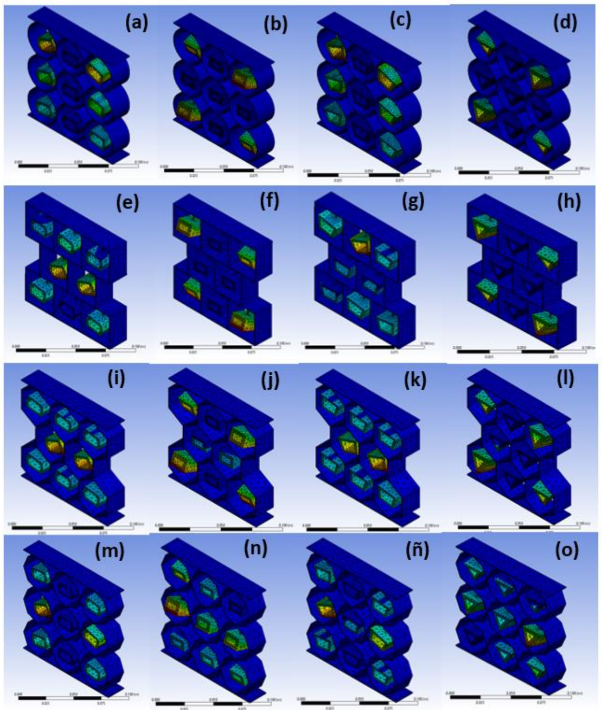

Figure 12. Modal analysis of structures with resonators As evidenced in Table 1, the frequencies are around 30 Hz, the smallest is the structure b and resonator b with 28,844 Hz and the largest is the structure c and resonator d with 30,859 Hz. Source: self-made.

3.7. The physical model

The selected prototypes were manufactured using additive manufacturing with PLA threads, using an ENDER 3 V2 Creality reference 3D printer.

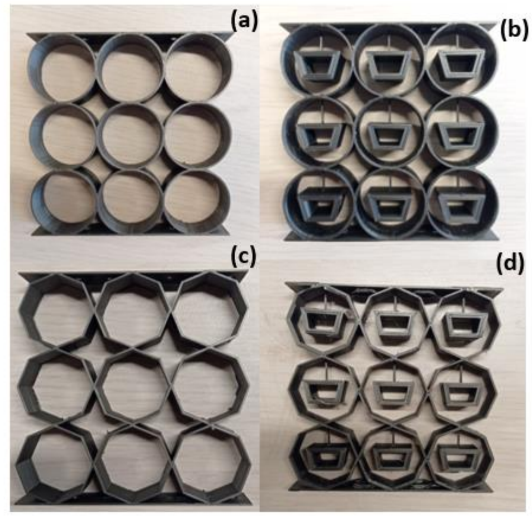

Figure 13. Manufactured alternatives: (a) Circular structure (b) Circular structure with hexagonal resonator (c) Octagonal structure (d) Octagonal structure with hexagonal resonator.

The circular and octagonal structures were printed, with and without resonators. The resonator has a trapezium shape, see Figure 13.

Table 1. Harmonic analysis of structures with resonators

Harmonic analysis of structures with resonators			
Structure	Resonators	Maximum deformation [mm]	Deformation frequency [HZ]
Circular	Hexagonal	$2{,}0224\text{x}10^{-5}$	29,534
	Rectangular	$0{,}5689\text{x}10^{-5}$	29,263
	Trapezoidal	$2{,}3056\text{x}10^{-5}$	30,269
	Triangular	$1{,}6989\text{x}10^{-5}$	30,150
Square	Hexagonal	$0{,}0162\text{x}10^{-5}$	29,696
	Rectangular	$0{,}5540\text{x}10^{-5}$	29,844
	Trapezoidal	$2{,}5255\text{x}10^{-5}$	29,362
	Triangular	$1{,}7300\text{x}10^{-5}$	29,700
Hexagonal	Hexagonal	$1{,}6067\text{x}10^{-5}$	30,309
	Rectangular	$6{,}1897\text{x}10^{-5}$	29,919
	Trapezoidal	$2{,}0247\text{x}10^{-5}$	30,293
	Triangular	$0{,}0022\text{x}10^{-5}$	30,859
Octagonal	Hexagonal	$0{,}3050\text{x}10^{-5}$	30,072
	Rectangular	$0{,}0374\text{x}10^{-5}$	29,821
	Trapezoidal	$0{,}0158\text{x}10^{-5}$	30,038
	Triangular	$0{,}2831\text{x}10^{-5}$	30,742

The objective is to compare the dissipation energy in both cases, structure with resonators vs structure without resonators.

3.8. Experimentation

This process focuses on obtaining the transferability coefficient through the mechanical vibration measurements.

The Mass Transferability Coefficient (MTC) is a measure of how much vibration is transmitted from an input point to an output point. It is calculated by comparing the output vibration amplitude with the input. When this coefficient gives results less than 1, it implies that the wave propagation medium is absorbing or dissipating the wave's vibration energy, equation (1), where F_1 and F_2 are vibration amplitudes of input and output signals.

$$MTC = \frac{F_2}{F_1} \tag{1}$$

Figure 14. Assembly of the test bench to obtain the coefficient of transmissibility. Source: self-made.

A test bench is proposed and used in this research to measure the MTC. The bench is composed of a support structure and coupling plates that allow the study piece to be adjusted appropriately to measure the vibrations of excitation and those transmitted through the part, see Figure 14. Likewise, the structure allows the coupling of the excitation system.

The bench and the test include next materials: a 100mV/g and 50g range sensors, as well as an 8-channel Oros Or35 analyzer, which processes purchases and analyzes the signals, and a mini-Shaker to excite the study iece, see Figure 15.

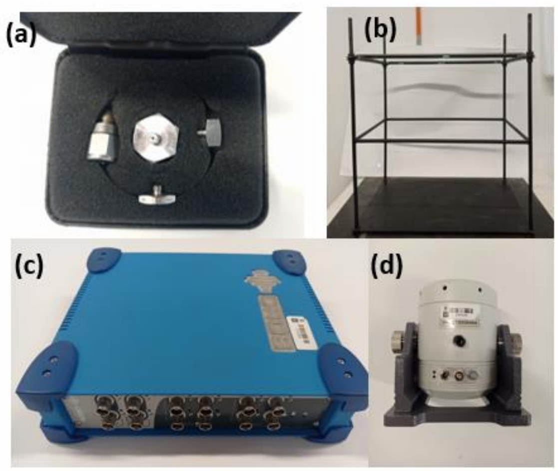

Figure 15. Necessary materials for laboratory tests. (a) 3056D2 Dytran Instruments INC Sensors (b) Metal structure (c) Analyzer - Oros OR35 (d) D) Mini-Shaker Sentek Dynamics. Source: self-made

The shaker is mounted, and the plates are fixing, as can be seen in Figure 14. The shaker is configured with a dynamic load of 30 N, a scan of frequencies of 10 Hz up to 40 Hz and with a voltage of 100 mV. Both the vibration on the top plate, on which the shaker excites, and the response on the bottom plate to which the studio piece transmits vibrations are measured. These measurements are those required for the calculation of the coefficient of transmissibility as expressed in equation (1).

4. Results and discussion

The results of the transmissibility test of cellular architectures with resonators and without resonators are reported. The objective is to evaluate the resonator's contribution to the energy dissipation capacity of the metamaterial.

The behavior of the coefficients of transmissibility of the cellular structures of metamaterial through the expression presented in equation (2), is plotted for each structure proposed.

$$1 - MTC \tag{2}$$

For circular structure: Figure 16 shows the ability to dissipate energy of this structure, in both cases with and without resonators. The circular structure without resonators exhibits low energy dissipation in the frequency range of interest, generally less than 34%.

However, when the structure is provided by resonators it improves its energy dissipation capability. The Figure 16 shows that the best performance is obtained about 30 Hz, dissipating around 65% of the vibration energy.

For octagonal structure: Figure 17 shows an unfavorable behavior in energy dissipation capability of the structure without resonators. In the frequency range of interest, from 27 Hz to 33 Hz, the system has limited dissipation capability, with energy dissipation dropping to a minimum of 20% at 31 Hz. This cellular structure with the implementation of resonators improved its energy dissipation capability. The Figure 17 shows that the maximum dissipation capability is around 27 Hz dissipating 69% of the vibration energy.

5. Conclusions

Four cellular architectures were proposed. Each one of them was designed to improve its capability to dissipate vibration energy. Furthermore, an additional element was designed and attached to the cellular structure with the goal of improving its performance in dissipating mechanical vibrations.

Four resonators were proposed and studied by numerical models and experimental method. A design process was developed for a specific range of frequencies, it means that de resonator can be tuned in the design step. The numerical model was successfully validated and then used to integrate the resonator into the base of the cellular structure. The trapezoidal resonator design demonstrated superior performance and was therefore selected and integrated into the cellular structures.

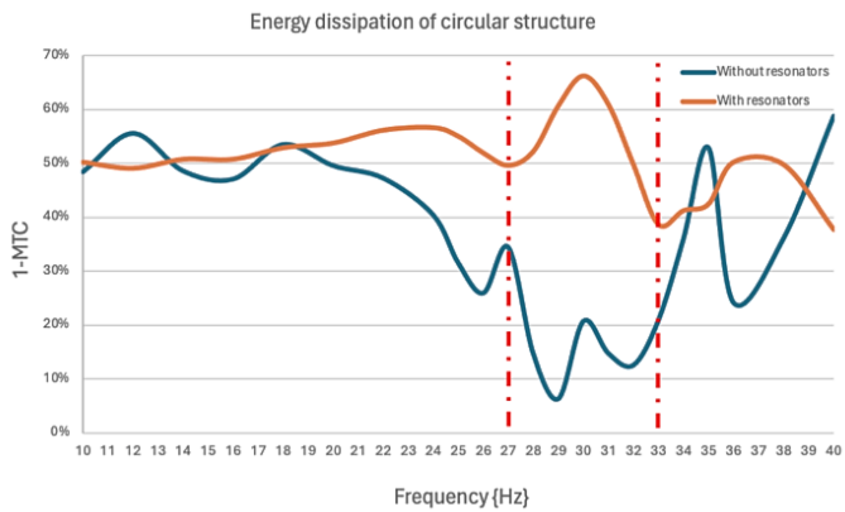

Figure 16. Energy dissipation of circular cell structure without and with resonators. Source: self-made.

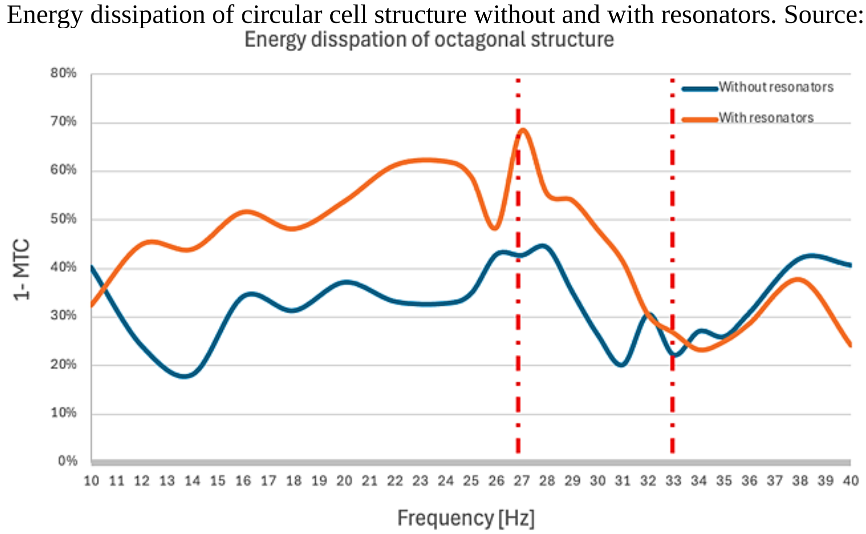

Figure 17. Energy dissipation of octagonal cell structure without resonators. Source: self-made.

The inclusion of resonators in the cellular structure demonstrated that the metamaterial significantly enhances its ability to dissipate vibration energy, particularly within the frequency range for which it was designed (27 Hz to 33 Hz). In the circular structure with resonators, energy dissipation reached 66.18%, while in the octagonal structure with resonators, it was 69.61%.

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Autor Contributions

H. G. Sánchez-Acevedo: Methodology, Project Administration, Writing of Original Draft, Writing-Review, and Editing. A. Velásquez-Vargas: Data Curation, Formal Analysis, Investigation, Methodology, Validation, Writing of Original Draft, Writing-Review, Editing. J. Gómez-Castellanos: Data Curation, Formal Analysis, Investigation, Methodology, Validation, Writing of Original Draft, Writing-Review, Editing

Conflicts of Interest

The authors declare no conflict of interest.

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