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Mechanical wave filtering through meta-cushions: an experimental study on vibration attenuation performance

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Keywords: resonant metamaterials, wave filtering, vibration attenuation, monopile installation

Abstract

The increasing installation of large offshore monopiles for wind turbine foundations generates harmful underwater noise during pile driving, particularly at low frequencies. Existing noise mitigation systems target noise around 1000 Hz, however, their performance reduces for lower frequency noise expected for the upcoming monopiles. To address this, a metamaterial-based cushion, or meta-cushion, was proposed to filter waves generated by impact loads. Although the meta-cushion principle has been numerically verified, an experimental validation has not been performed yet. Moreover, the meta-cushion was designed for a single frequency range, requiring the development of a meta-cushion with multiple attenuation zones to address various resonant frequencies that could induce noise. In this paper, a graded meta-cushion prototype is manufactured and experimentally tested to validate its vibration attenuation capabilities. The numerical and experimental modal analyses show that the meta-cushion effectively attenuates vibration across single or multiple frequency ranges, particularly when viscoelastic properties are incorporated, from which the results demonstrate the ability of the meta-cushion in attenuating vibration from single and multiple frequency ranges related to small-sized monopiles (around 1000 Hz) and large-sized monopiles (lower than 1000 Hz). This study advances the meta-cushion towards practical application in mitigating undesirable vibrations during full-scale monopile installations.

1. Introduction

The increasing installation of offshore wind farms across Europe is boosted to meeting the 42.5% renewable energy target by 2030 [1]. To support such wind farms, monopiles are typically used as foundations due to their cost-effectiveness and ease of installation. Current monopile sizes range from 4 to 8 m and 25 to 40 m in diameter and length, respectively [2–4], however, future wind farms are expected to require even larger monopiles [5]. The installation process involves driving the monopile into the seabed through impact hammer blows, which can provide energy ranging from 150 kJ to 6600 kJ [6]. The high impact hammer blow induces longitudinal wave propagation, leading to pile resonance and consequently causing radial excitation of the pile wall. This radial motion interacts with the surrounding water, emitting noise waves that can travel long distances in marine environment, causing potential risks to marine life, such as hearing loss and even death [7–9].

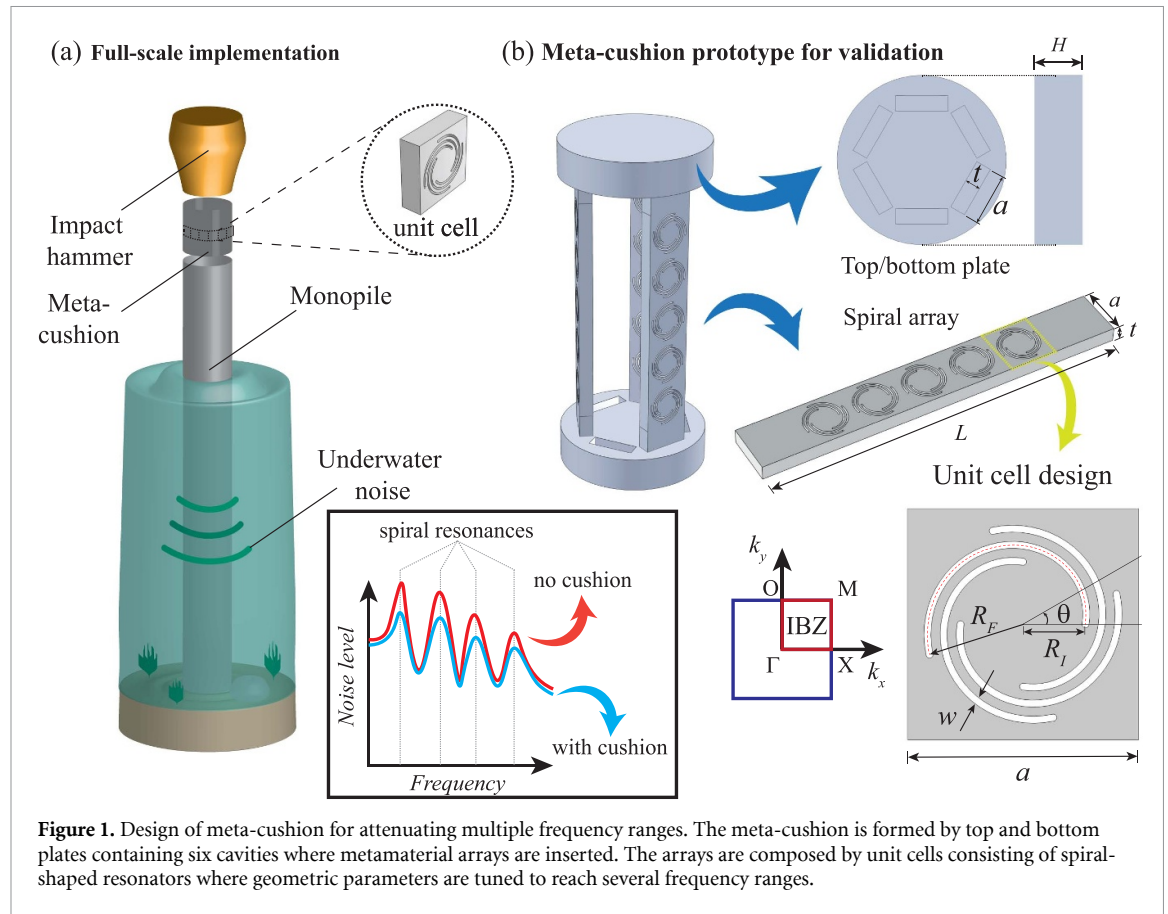
Because the frequency content of the generated noise depends on the pile ring frequency, which is related to the pile diameter, as well as on the characteristics of the impact load, the installation of larger monopiles results in noise dominated by low-frequency components [10]. For example, monopiles with diameters of approximately 8 m can generate underwater noise at frequencies below 1000 Hz. In this frequency range, current noise mitigation systems exhibit reduced effectiveness. For example, bubble curtains, which are the most commonly used noise mitigation system, can achieve noise reductions of approximately 25 dB at 1000 Hz. However, their effectiveness decreases at lower frequencies because larger air bubbles are required; such bubbles are unstable and tend to break up into smaller ones [11, 12]. Noise mitigation screens consist of double-walled steel pipes that create an air layer,

thereby inducing sound reflection due to acoustic impedance mismatch. As monopile sizes increase, the required dimensions of noise mitigation screens also increase, which can render this solution impractical. To improve low-frequency noise attenuation, combinations of multiple mitigation systems have been employed, although this approach increases both operational costs and installation time. To address the low-frequency underwater noise problem, our previous work [13] proposed the meta-cushion, a metamaterial-based structure designed to filter impact load energy, particularly in the lower frequency range (see figure 1(a)). The meta-cushion consists of periodic unit cells containing resonant elements that act as a spring-mass system. When the longitudinal wave generated by the hammer blows propagates through the meta-cushion in a frequency related to the high underwater noise, the resonant element moves oppositely to the propagation direction, prohibiting the transmission of such waves. The impact wave propagates along the pile wall, except at the frequencies filtered by the meta-cushion, inducing lower radial vibrations that radiate sound into the water. Since the radiated sound pressure is proportional to the pile wall surface velocity, a reduction in vibration amplitude at a given frequency corresponds to a comparable reduction in radiated sound pressure level, as observed in our numerical simulations. This resonant mechanism relies on the negative effective mass density, an unique metamaterial property widely explored for metamaterials with ability to attenuate vibrations [14–16]. While our previous work [13] demonstrated numerically the meta-cushion's functionality, experimental validation is still pending. Moreover, further research is also required to develop a meta-cushion capable of mitigating vibrations across multiple frequency ranges, addressing simultaneously potential monopile resonances that could generate high noise at various frequencies. Achieving multiple band-gap zones using resonators with distinct masses would lead to impractical designs at larger scales. To address this limitation, this paper investigates spiral-shaped resonators, which have been shown to produce deeper sub-wavelength band gaps than traditional spring-mass resonators [17–21]. This enhanced performance arises from their ability to achieve low dynamic stiffness without increasing resonator mass; instead, band gaps at different frequency ranges are realized by varying the effective spring length. Moreover, the unique geometry of spiral resonators enables the merging of multiple band gaps—such as Bragg scattering-type and locally resonant-type—thereby enhancing vibration attenuation [22].

This paper aims to experimentally demonstrate the vibration attenuation capabilities of the meta-cushion using a 3D-printed prototype (see figure 1(b)). The prototype contains spiral-shaped resonators with distinct geometric properties, chosen over those proposed by Vasconcelos *et al* [13] for their ability to generate band gaps in similar frequency range while using smaller unit cells, potentially reducing manufacturing costs for full-scale monopiles. Numerical and parametric analyses are performed to select the geometric features of spiral resonators inducing attenuation in distinct frequency ranges. The combination of a set of spiral resonators provides the ability of multiple band gaps, which could mitigate vibrations in different frequency ranges associated with high noise levels. The experimental validation is performed through modal impact testing, which provides frequency response functions (FRFs) that characterize the intrinsic vibration attenuation behavior of the meta-cushion independently of the specific excitation signal. This approach enables prediction of attenuation performance for arbitrary input spectra, including the broadband transient loads characteristic of pile-driving operations. Published measurements of pile-driving noise spectra for large-diameter monopiles [7, 8, 10] indicate dominant frequency content below 1000 Hz, which overlaps with the band gap frequencies targeted by the meta-cushion designs presented in this study. The present work introduces three distinct contributions: (i) the meta-cushion concept as an integrated structural device designed to be placed between the impact hammer and the monopile to filter specific frequency components from the impact wave; (ii) the first experimental validation of graded configurations combining multiple spiral resonator types within a single device, systematically evaluating all two-type and three-type combinations; and (iii) the demonstration that band gap predictions from infinite periodic analysis translate to measurable attenuation in finite, assembled meta-cushion structures. The results highlight the potential of a full-scale meta-cushion as a noise mitigation system during monopile hammering, which has not yet been implemented.

2. Design of the meta-cushion

As depicted in figure 1(a), the meta-cushion is placed between the hammer and the cushion to filter unwanted vibration related to high noise levels. To validate its principle this section presents initially the design process of the graded meta-cushion prototype for filtering vibration in multiple frequency ranges. The graded meta-cushion prototype is formed by three parts: the top and bottom plates acting as the cushion's cap, and the arrays containing five spiral-shaped unit cells, as depicted in figure 1(b). The plates contain six cavities where the arrays can be inserted. This allow the combination of arrays



containing single or multiple unit cell designs to attain single and multiple attenuation zones, respectively. The unit cell is formed by four identical spiral paths, spaced by an angle of 90° , which are used to define the inner mass and the curved beams connecting the mass to the external frame. By tuning the geometric parameters of the spiral paths, the unit cell can exhibit attenuation at multiple frequency zones.

The spiral path is inspired by the Archimedean spiral, which is characterized by having fixed distance between two successive turns. The Archimedean spiral can be described by the following equation in polar coordinates:

$$R(\theta) = R_I + b\theta, \quad (1)$$

where R_I and b are parameters defining the initial radius of the spiral and the spiral growth rate, respectively, and θ is the angle in rad, being θ_I and θ_F the angles where the spiral curve starts and ends, respectively. Parameter b can be expressed as

$$b = \frac{R_F - R_I}{2\pi n_t}, \quad (2)$$

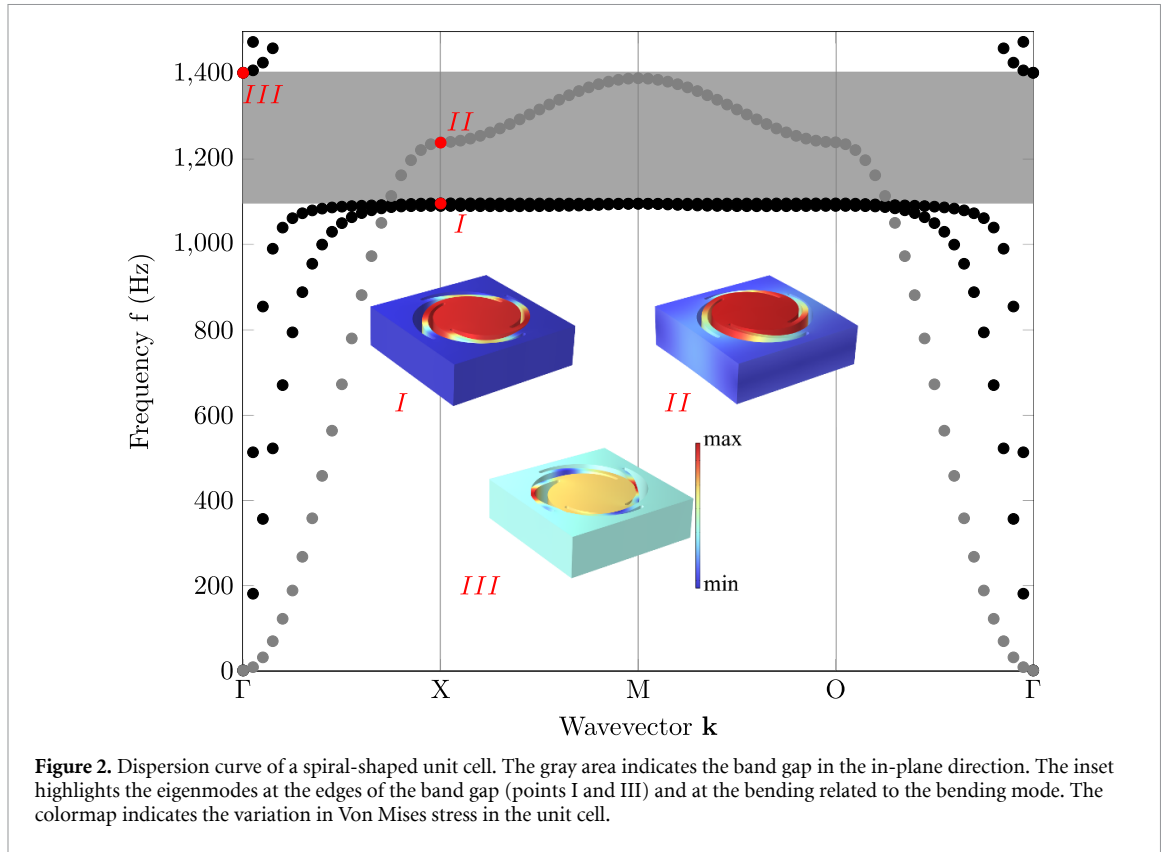
where R_F is the spiral's final radius and n_t is the desired number of turns.

Since the meta-cushion analysis is performed through COMSOL Multiphysics, it is convenient to define the spiral shapes by means of parametric curves. The spiral equations are then converted from polar to Cartesian coordinate system, where their corresponding parametric forms are given by:

$$\begin{aligned} x &= r \cos \theta \\ y &= r \sin \theta. \end{aligned} \quad (3)$$

Finally, the spiral's start angle and end angle are defined as θ_I and θ_F . The distance d between the turns and the gap g are defined as

$$\begin{aligned} d &= \frac{R_F - R_I}{n} \\ g &= d - w, \end{aligned} \quad (4)$$



where w is the width of the curved beams. The geometric parameters described here will be used to tune the dynamic features of the spiral resonator, which will be further described.

3. Numerical model

In this section, the numerical evaluation of the vibration capability of the meta-cushion is described. The analysis starts by investigating the resonant modes of the unit cell through dispersion curves. Then, the methodology for investigating the vibration attenuation performance of the spiral arrays and the assembled meta-cushion is presented, where FRFs are evaluated.

3.1. Dispersion curves of spiral resonators

The dynamic features of a metamaterial can be described by infinite periodic structures formed by unit cells, which represent the smallest periodic element from the structure. Wave propagation in infinite periodic structures is investigated through Bloch's theorem, which states that wave propagation can be expressed as the response of a wave propagating in a reference unit cell and the exponential term defining amplitude and phase change when waves propagate through other unit cells. Thus, the displacement field is expressed as:

$$\mathbf{u}(\mathbf{r}) = \bar{\mathbf{u}}(\mathbf{r}, \mathbf{k}) e^{i(\mathbf{k} \cdot \mathbf{r})}, \quad (5)$$

where $\mathbf{r}$ is the position vector, $\bar{\mathbf{u}}(\mathbf{r}, \mathbf{k})$ is a periodic displacement vector, $e^{i(\mathbf{k} \cdot \mathbf{r})}$ is a plane wave with $i = \sqrt{-1}$ and wave vector $\mathbf{k}$ that varies through the boundaries of the irreducible Brillouin zone (figure 1(b)). The finite element method is used to define the following eigenvalue problem,

$$[\mathbf{K}(\mathbf{k}) - \omega_n^2 \mathbf{M}(\mathbf{k})] \mathbf{u} = \mathbf{0}, \quad (6)$$

where $\mathbf{K}(\mathbf{k})$ and $\mathbf{M}(\mathbf{k})$ are the stiffness and mass matrices, and $\mathbf{u}$ is the eigenvector that provides the unit cell's modal shape related to the angular frequency ω_n . The eigenvalue problem in equation (6) is solved

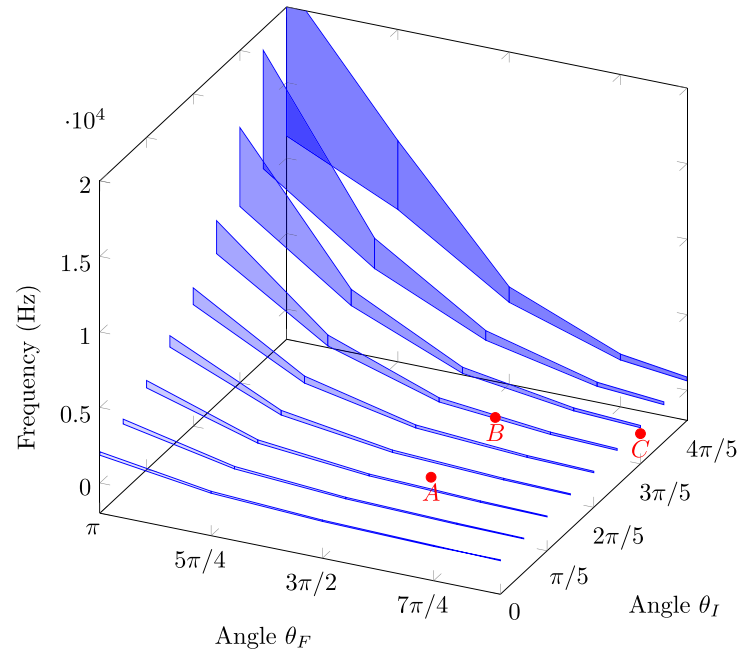


Figure 3. Influence of geometric parameters on band gap formation. The blue meshes represent the bandwidth identified in the studies. The selected designs A, B, and C are highlighted by the red dots.

Table 1. Geometric features of spiral-shaped unit cells constituting the meta-cushion.

	Geometric parameters			
	a [mm]	θ_I [rad]	θ_F [rad]	t_b [mm]
A	40	$\pi/4$	$8\pi/5$	1.2
B	40	$\pi/2$	$8\pi/5$	1.2
C	40	$3\pi/5$	$19\pi/10$	1.2

by providing the wave vectors $\mathbf{k}$, which will give the frequencies ω_n . These two variables will then constitute the dispersion curves of the unit cell, which will describe the response of the unit cell when wave propagates through it. Figure 2 displays an example of dispersion curves when solving the eigenvalue problem for an unit cell with the following geometric parameters: $a = 0.04$ m, $R_I = 0.008$ m, $R_F = 0.017$ m, $\theta_I = 3\pi/5$, $\theta_F = 8\pi/5$, $n_t = 0.8$, $w = 0.0012$ m. The black curves indicate eigenmodes excited in the in-plane direction. The eigenmodes related to points I ($k_x = 1$, $k_y = 0$ and $n = 1$) and III ($k_x = 0$, $k_y = 0$ and $n = 4$) are shown in the inset, where it is observed the resonator motion in the same plane of wave propagation (the colormap indicates the range between the minimum and maximum unit cell displacement). On the other hand, the gray curve represents modes excited in the out-of-plane, which it is indicated by bending motion of the unit cell from point II ($k_x = 1$, $k_y = 0$ and $n = 3$).

To explore the ability of multiple frequency range attenuation of the meta-cushion, it is necessary to design unit cells generating band gaps in different frequency ranges. The designs are defined here through a parametric study, which evaluates the band gap for various set of geometric parameters. In this study, we select the following parametric vector: $\theta_I = [0 : \pi/10 : \pi]$, $\theta_F = [\pi/2 : \pi/4 : 7\pi/4]$ (notice that the other geometric parameters of the unit cell are dependent on the varied parameters, therefore they are not considered in the parametric investigation, except the gap width w , which remains constant due to manufacturing limitations). Figure 3 shows the band gap when such parameters are varied. It is observed that the band gap position leverages when the initial angle θ_I increases, while its position decreases when the angle θ_F varies from π to $7\pi/4$. From this analysis, three types of spiral resonators exhibiting band gaps at distinct frequency ranges are selected, one design (B) with band gap around the attenuation zone of traditional noise mitigation systems (around 1000 Hz) and two designs with lower band gaps in potential frequency ranges for the next generation of monopiles (A and C). The selected geometries A, B, and C are indicated in table 1. For all designs, $n_t = 0.8$, $R_I = 0.008$ m, and $R_F = 0.017$ m.

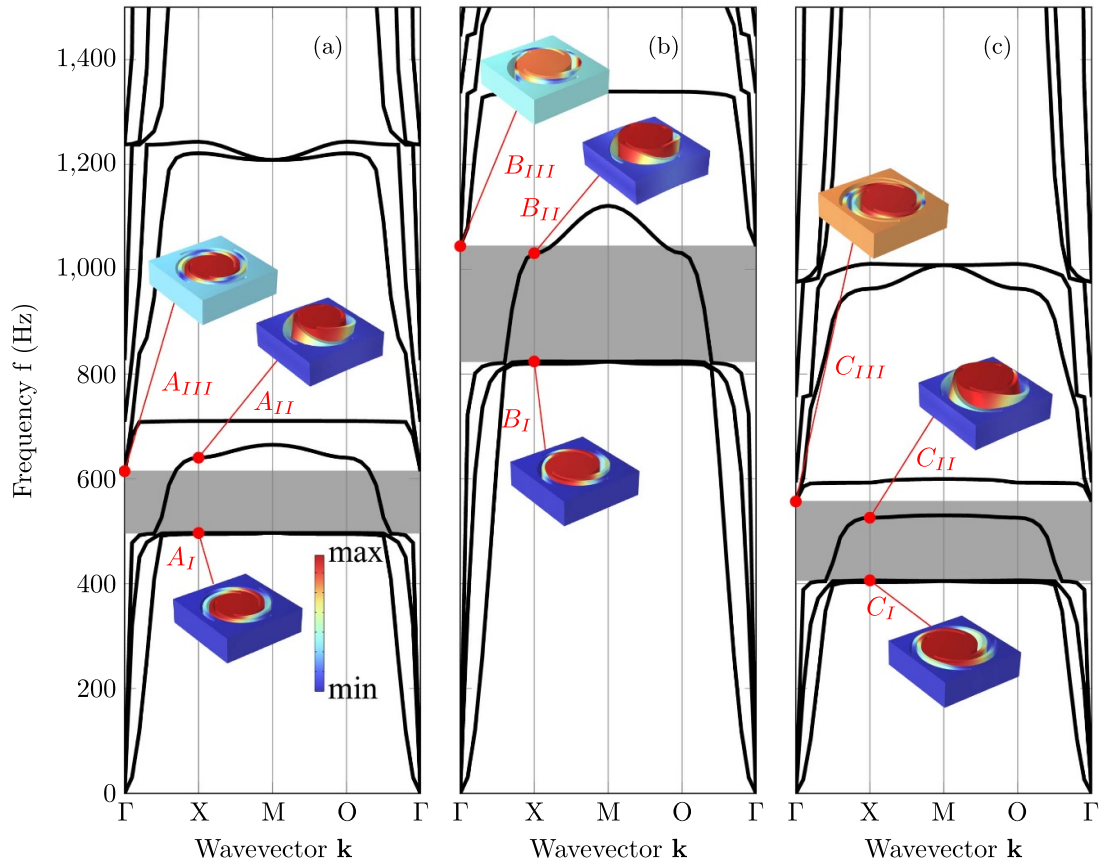


Figure 4. Dispersion curves of spiral-shaped unit cells of types (a) A, (b) B, and (c) C indicated in table 1. The gray area indicates the band gap associated to in-plane wave propagation. The curve inside the highlight band gap is related to out-of-plane modes as indicated by modes A_{II} , B_{II} , and C_{II} . The colormap indicates the Von Mises stress variation in each unit cell.

Figure 4 depicts the dispersion curves of waves propagating through an infinite domain formed by periodic spiral unit cells. Band gaps related to in-plane waves are identified by gray rectangles in frequency ranges varying from 496.87 Hz to 614.7 Hz (type A), from 824.05 Hz to 1043.97 Hz (type B), and from 406.5 Hz to 556.9 Hz (type C). This is indicated by the unit cell deformation modes at the bottom edge of the band gap (modes A_I , B_I , and C_I). The modes A_{II} , B_{II} , and C_{II} indicate the out-of-plane deformation of the unit cell, where the squared frame bands and the spiral resonator moves perpendicularly to the wave propagation direction.

3.2. FRF analysis

Although the dispersion curve analysis can well indicate the vibration attenuation zones in infinite periodic domains, it is essential to verify if the attenuation performance still remains in a finite structure, such as the spiral arrays and the assembled meta-cushion. The analysis of such structures is performed through FRFs, from which a time-harmonic excitation is applied in one end of the structure, while the response is measured in the opposite end. To calculate the FRFs, Fourier transform is applied in the input and output signals, from which the following expression is used:

$$\text{FRF} = 20 \log \left(\left| \frac{X(\omega)}{F(\omega)} \right| \right), \quad (7)$$

where $X(\omega)$ and $F(\omega)$ are the frequency spectrum of the displacement response and input signals (excitation force), respectively (notice that the logarithm term in equation (7) was introduced to express the attenuation levels in terms of decibels (dB)). FRFs are commonly used to indicate the behavior of a structure due to an applied excitation. If the FRF gives a positive infinite peak, this indicates high response due to excitation, which means a natural frequency of the structure. In the contrary, the response of the structure is low, which means a reduction in the input energy transmission.

The spiral arrays and assembled meta-cushion are modeled as a linear PLA solid with Young's modulus $E = 3.16$ GPa, density $\rho = 1250$ kg m⁻³, and Poisson's ratio $\nu = 0.33$. As the viscoelastic properties

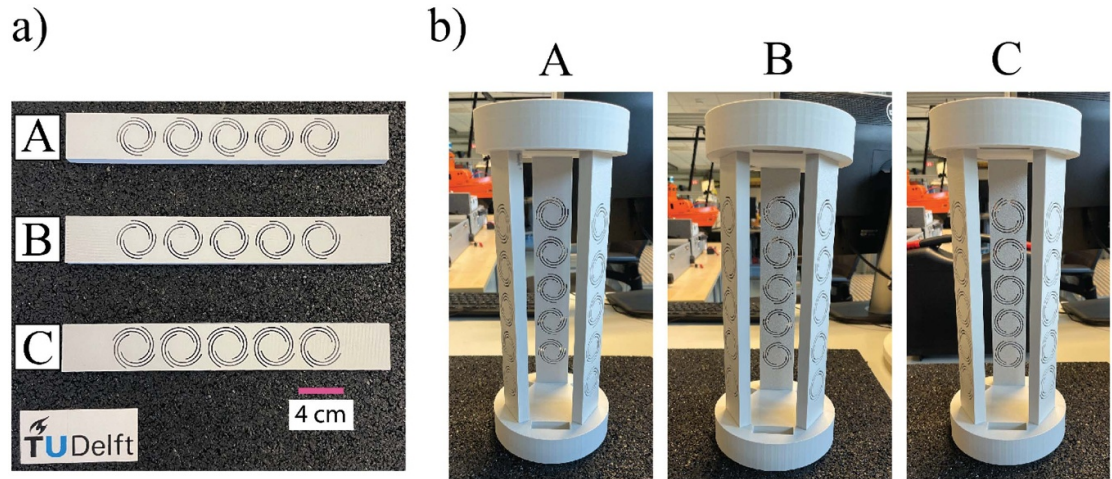


Figure 5. (a) Printed spiral arrays for assembling the meta-cushion. (b) Meta-cushions each containing three identical arrays of each type for experimental validation.

of PLA can change the vibration attenuation performance of the meta-cushion, the viscoelastic effects are then integrated into the numerical model by modifying the stress tensor, which is given by

$$\sigma_v = (K' + iK'')\epsilon_e, \quad (8)$$

where K' storage modulus, K'' loss modulus, ϵ_e is the strain tensor of the elastic model. The storage and loss moduli are determined through dynamic mechanical analysis (DMA), where a force oscillating at a set frequency is applied at PLA samples under varying temperatures [23]. Extensive literature exists on using DMA to obtain viscoelastic properties of neat and composite polymers [24–26]. For PLA samples manufactured under similar conditions as the ones presented in this work, the storage and loss moduli are, respectively, $K' = 2220$ MPa and $K'' = 112.9$ MPa.

Regarding the boundary conditions, the external excitation is represented by a time-harmonic point load $F(t) = F_0 \sin \omega t$ with amplitude $F_0 = 1$ N is applied at the center of the cross-section area of one array end (and of the top plate, for the meta-cushion), while acceleration is measured in the middle point at the opposite end (at the bottom plate, for the meta-cushion). The load frequency varies from 1 Hz to 1500 Hz, covering the attenuation range of interest. Stress-free boundary conditions are applied at the remaining array and meta-cushion surfaces. Mesh size has been defined by ensuring at least five elements per wavelength to capture wave propagation behavior in the frequency range of interest. Mesh refinement has been applied in the resonators' edges and vertices. The excitation load and the measured response are then replaced in equation (7) to obtain the numerical FRF, from which the results will be further compared to the experimental data.

4. Experimental analysis

The experimental analysis of the meta-cushion starts with evaluating the FRFs of the array types in table 1 individually. The arrays are made of PLA (figure 5(a)) and were fabricated using a FDM 3D manufacturing process. Once they have indicated the desired band gap, the arrays are combined through the top/bottom plates, also made of PLA through the same 3D printing process, to constitute the meta-cushion. Initially the meta-cushion is analyzed for arrays of the same type (figure 5(b)) and then for combined types. The setup used to calculate the FRFs is presented in figure 6, where the metamaterial structures are suspended by fishing wires. An excitation signal is applied in one structure end by a modal impact hammer (Type DJB of model IH/02) with sensitivity 2.58 mV N^{-1} , while an accelerometer (ICP® accelerometer model 35C22 with sensitivity $1 \text{ mV m}^{-1} \text{ s}^{-2}$) measures the response in the opposite end. The signals are sent from the data acquisition system Sirius-HD to a laptop, where the FRF is processed in the software DewesoftX. The FRF is evaluated from 1 Hz to 10 000 Hz, from which a frequency resolution of 5 Hz was chosen, but for sake of visualization the FRFs are shown until 1500 Hz, where the band gaps have been identified in the numerical simulations. The impact tests are performed five times to verify the consistency of the measured data.

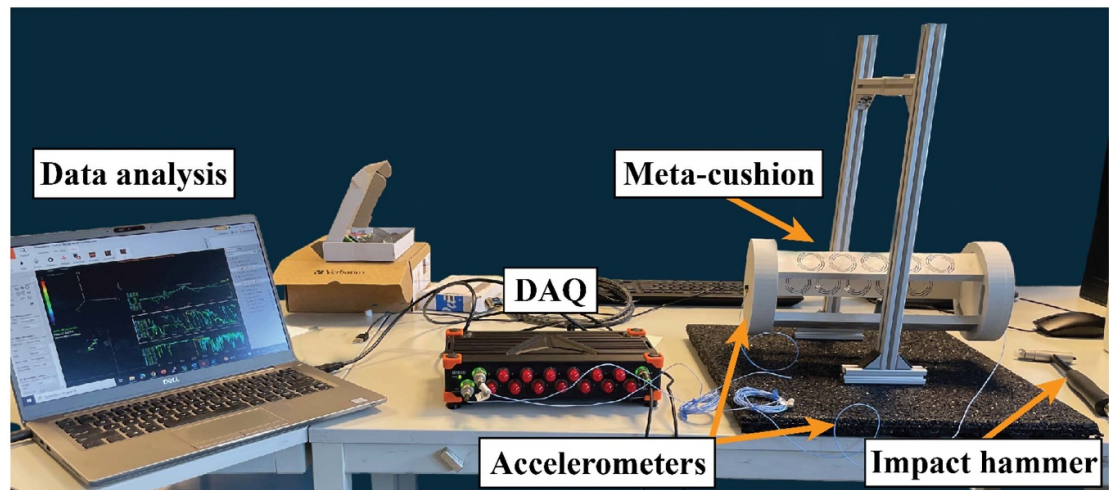


Figure 6. Experimental setup used to measure the frequency response functions (FRFs) of arrays and assembled meta-cushions.

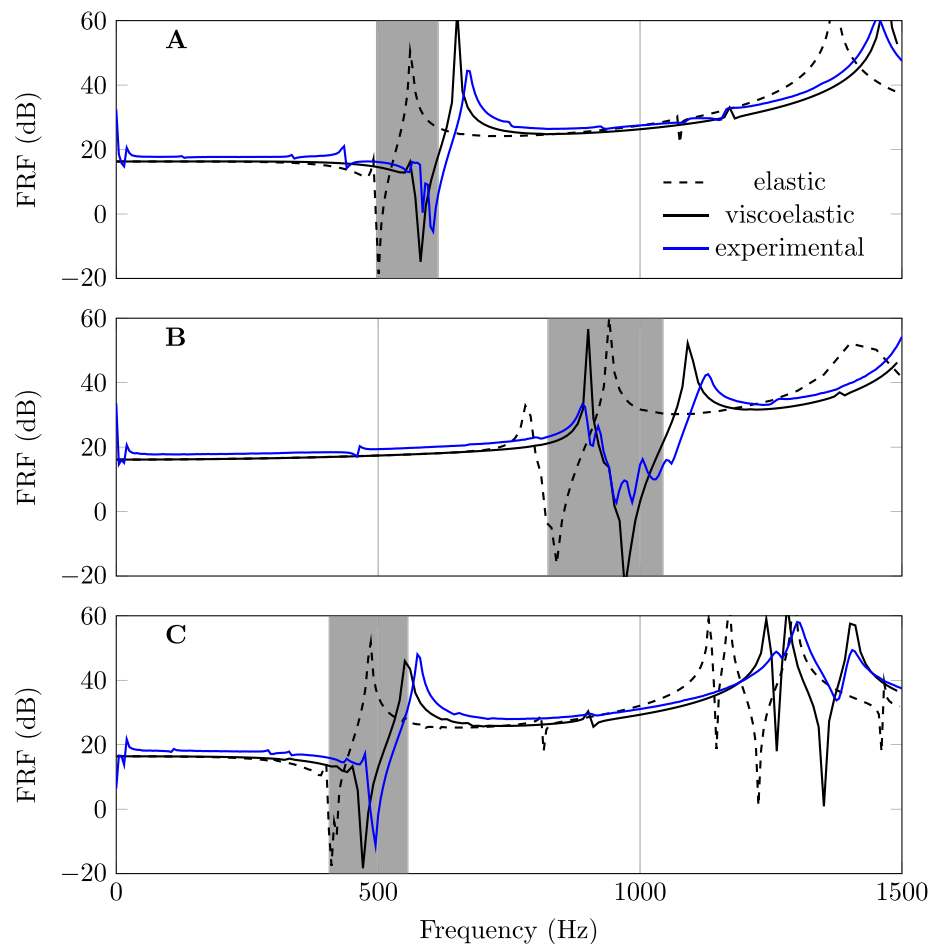


Figure 7. Comparison between the FRFs obtained through numerical and experimental analyses of spiral arrays. The numerical analysis was performed for both elastic and viscoelastic model of the arrays. The gray areas indicate the band gaps obtained via the dispersion curve analysis (see figure 4).

4.1. Validation of spiral arrays attenuation

Figure 7 presents the FRFs obtained numerically and experimentally for the three array designs indicated in table 1. The numerical elastic results indicate a reduction in the FRF within the frequency range near the band gaps predicted by the dispersion curve analysis (gray rectangles), confirming that the attenuation properties identified for infinite domains are applicable to the spiral arrays (finite

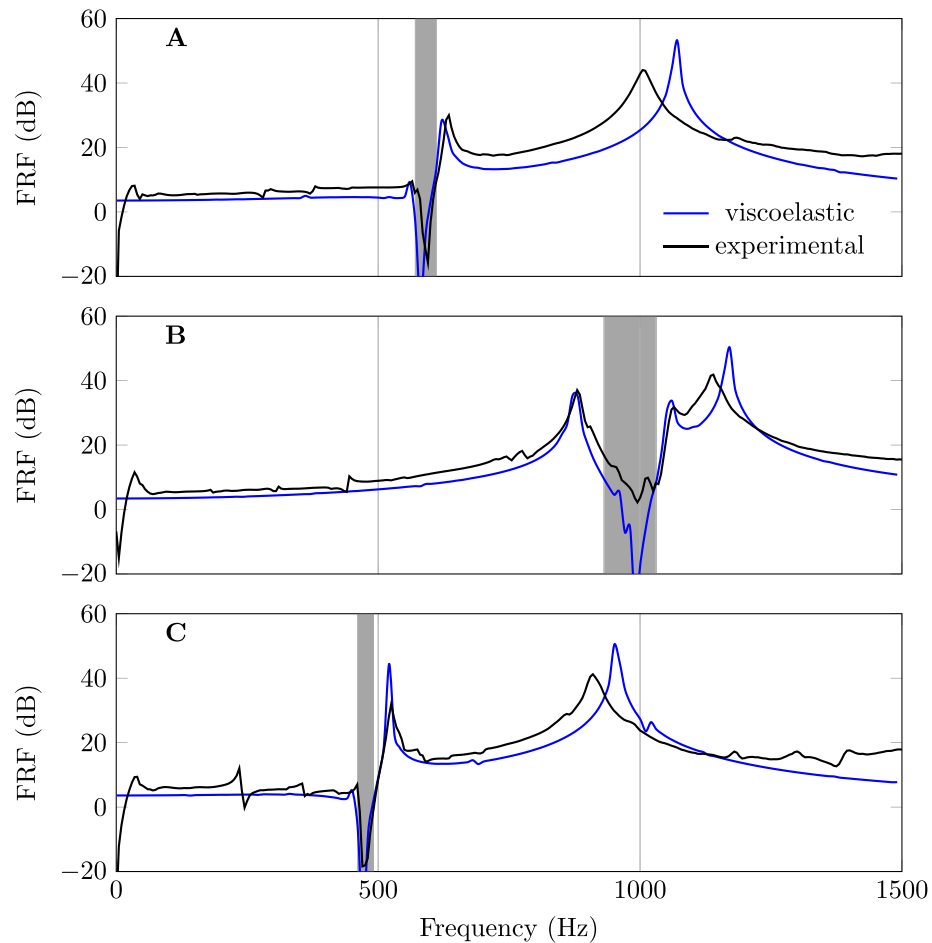


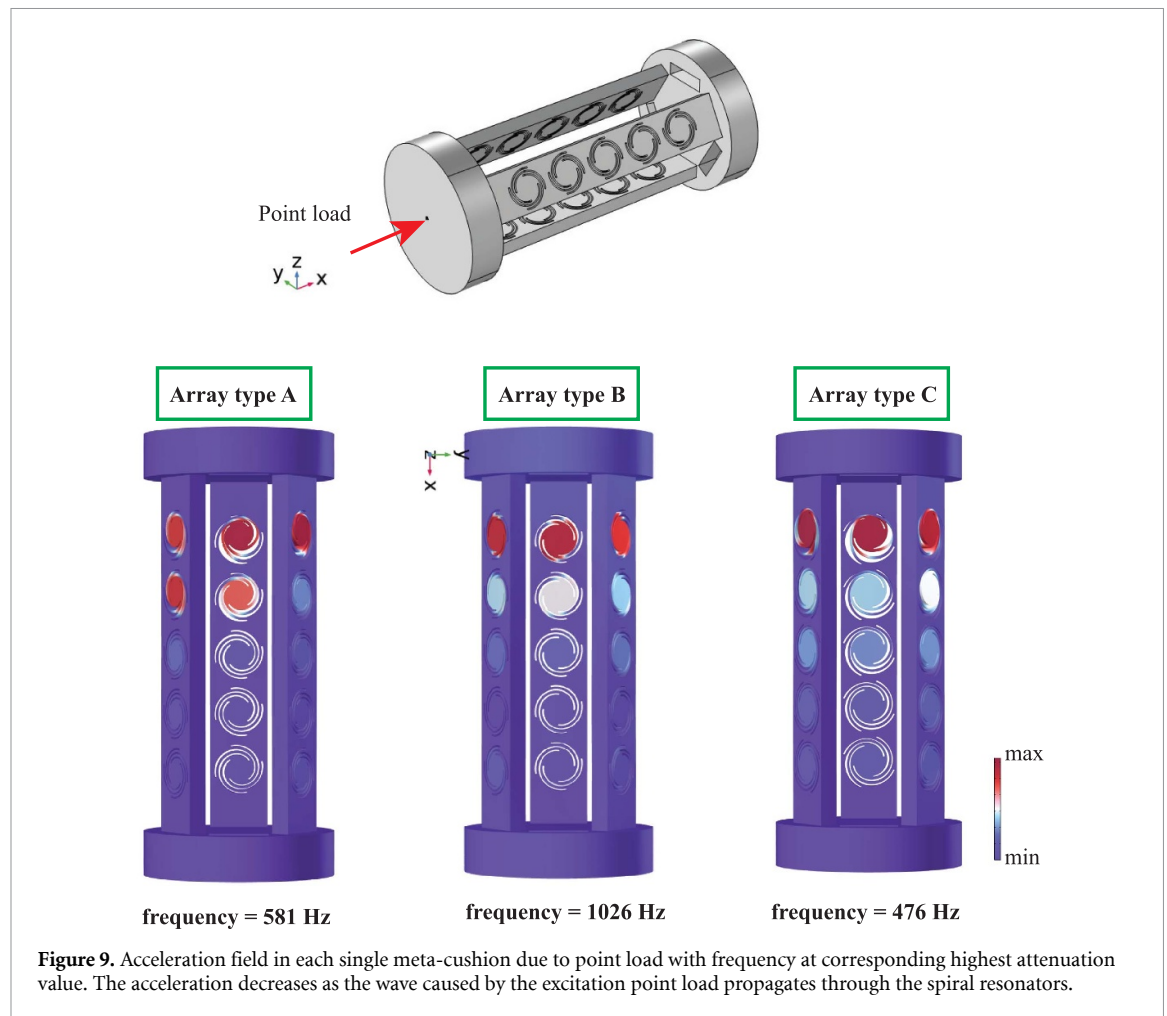
Figure 8. FRFs of meta-cushions formed by three spiral arrays of type A, B, and C. The gray areas indicate the attenuation zones identified in the viscoelastic numerical model displayed in figure 7.

Table 2. Meta-cushion configurations for verification of multiple attenuation zones. The number two indicates the quantity of arrays inserted in the cavities of the top/bottom plates.

Array type	Meta-cushion configurations			
	AB	AC	BC	ABC
A	2	2	—	2
B	2	—	2	2
C	—	2	2	2

domains). However, the numerical results are not aligned with the experimental measurements, which show the attenuation zone shifting towards higher frequencies. This discrepancy suggests that damping effects should be incorporated into the numerical model [27], particularly for high-frequency band gap design—notice that the elastic and experimental results of arrays A and C shows better agreement than results of array B. The numerical results with incorporated viscoelastic effects as described in equation (3.2) are shown in blue in figure 7, from which an improved agreement with experimental data is observed. Regarding the amplitude of attenuation, the experimental results reveal a reduction in attenuation level and a larger attenuation frequency range at higher frequencies in comparison to the elastic model, highlighting the frequency-dependent nature of viscoelastic materials [28–30].

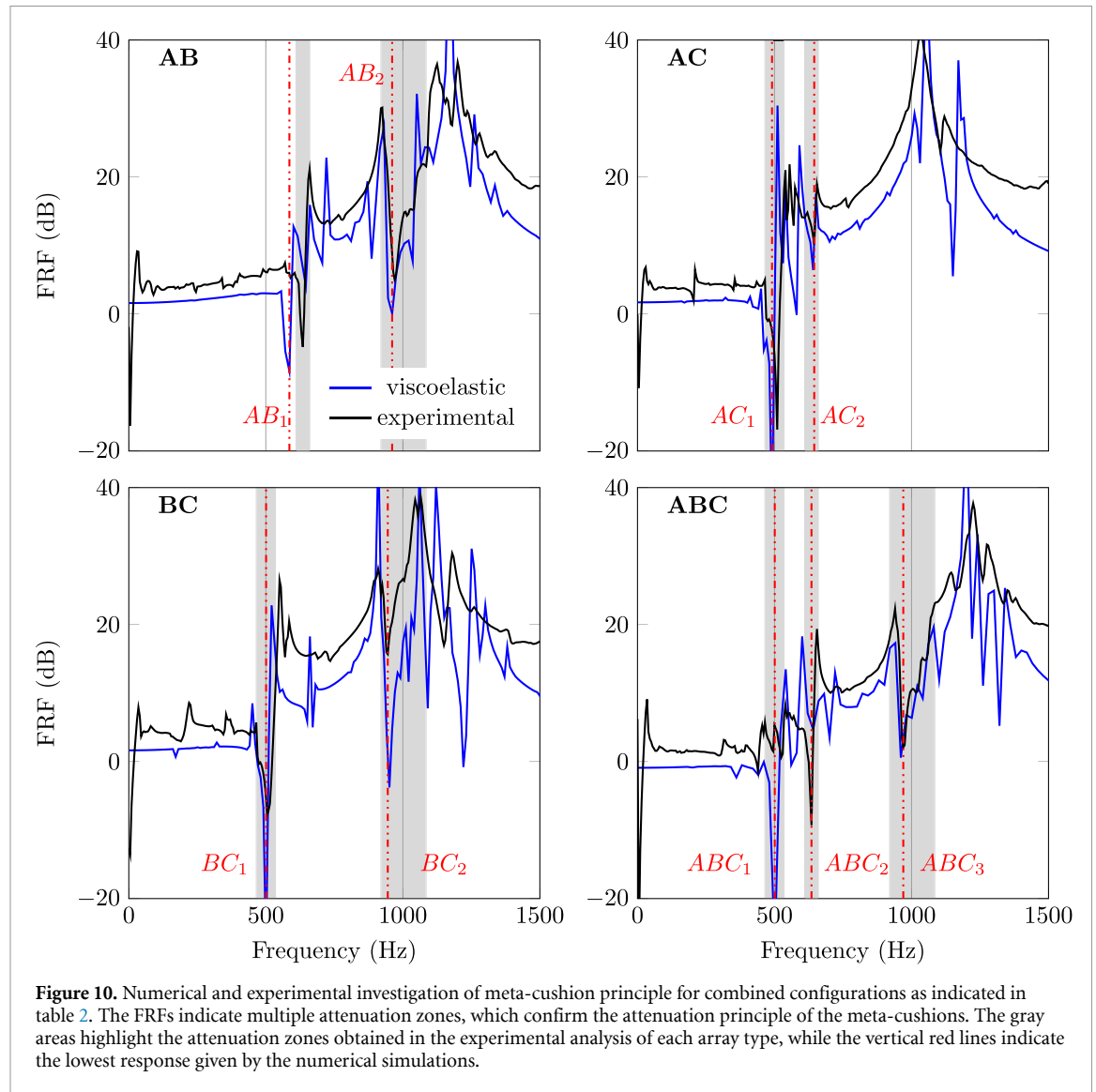
The remaining discrepancies between numerical and experimental results arise from three main sources. First, geometric tolerances in FDM 3D printing (± 0.1 mm on spiral arm width) shift the band gap center frequency by approximately $\pm 5\%$ – 10% , consistent with the observed experimental deviations. Second, the fishing-wire suspension approximates free-free boundary conditions but introduces small stiffness and damping at the attachment points, primarily affecting the response at low frequencies, where the rigid-body modes of the suspended structure interact with the structural response. Third,



measurement uncertainty arises from sensor positioning, signal noise, and excitation repeatability; however, they primarily affect the amplitude response rather than the band-gap formation. Averaging five impact tests is therefore used to reduce uncertainty related to excitation repeatability.

4.2. Verification of meta-cushion principle

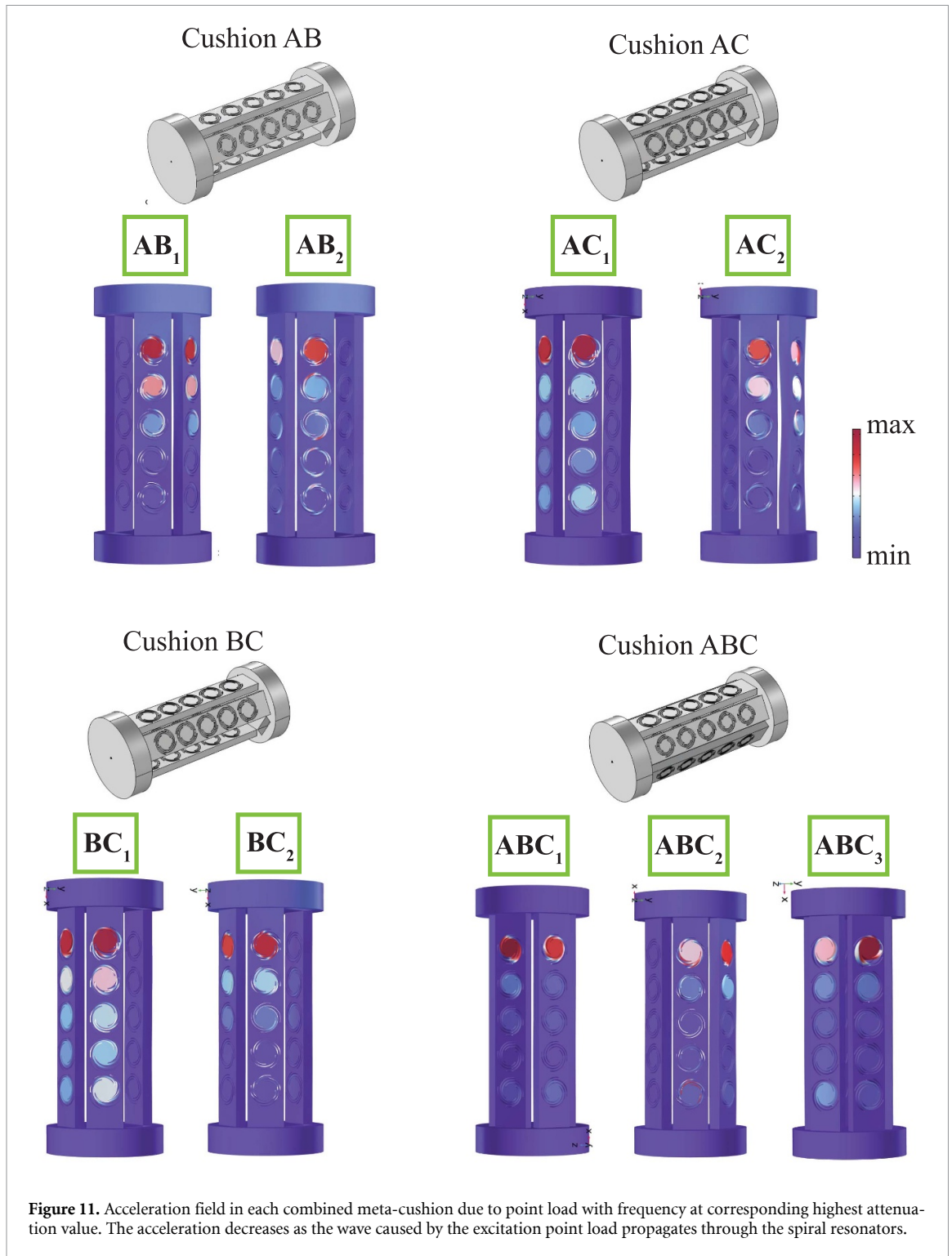
After demonstrating that the spiral arrays can filter waves within desired frequency ranges, it is crucial to assess whether this mechanism still remains when the meta-cushion is assembled. To verify this, three identical spiral arrays (as indicated in figure 5(b)) are inserted in the top and bottom plates with 130 mm of diameter and 30 mm of height H . The modal impact testing is conducted where the excitation load is applied in the center of the top plate, with the response measured at the center of the bottom plate. The experimental measurements are then compared with the attenuation zone predicted by the viscoelastic model of the arrays (as indicated in figure 7) and with a numerical model of the meta-cushion with same boundary conditions. Figure 8 displays the FRFs for each meta-cushion, with the gray areas indicating attenuation zones of the array model and the blue curves indicating the FRFs of the numerical models of the meta-cushion. The fractional bandwidth (ratio between band gap width and center frequency) for each resonator type is about: 7% (571–611 Hz) for type A, 10% (931–1031 Hz) for type B, and 6% (461–491 Hz) for type C. Maximum attenuation within each band gap ranges from 15 to 25 dB relative to adjacent resonant peaks. By comparison, conventional bubble curtains achieve 7 to 25 dB attenuation around 1000 Hz but lose effectiveness below 500 Hz due to bubble instability. The meta-cushion targets precisely this sub-1000 Hz range, suggesting potential as a complementary or alternative mitigation approach for low-frequency noise. Although peaks are observed in other frequency zones, for both numerical and physical model of the meta-cushion, the attenuation zones remain consistent with those identified in the spiral array analysis. Despite that, both models indicate a key feature of locally-resonant metamaterials, where the vibration attenuation performance relies on the resonators



instead of the periodicity. The resonant behavior of each single meta-cushion model is illustrated in figure 9. The colormap indicates the acceleration field due to a harmonic point load with frequency associated to the highest attenuation value identified in the numerical simulation. Notice that the highest acceleration value occurs in the spiral resonators near the point load and the acceleration reduces along the array length.

In the final step of the experimental analysis, the principle of multiple attenuation zones of a graded meta-cushion is verified. With the three types of spiral arrays and six cavities available in the top/bottom plates, four graded meta-cushion configurations are investigated, as indicated in table 2, from which two attenuation zones are expected in configurations AB, AC and BC, and three attenuation zones are expected in configuration ABC.

The FRFs for the graded meta-cushion configurations are presented as black curves in figure 10, where gray rectangles highlight the attenuation ranges observed in the experimental investigation of the single arrays. The blue curves indicate the numerical results of the combined cushions. In figure 10, two main drops for configuration AB are identified at frequencies ranging from 620 Hz to 640 Hz and from 940 Hz to 1045 Hz, aligning well with the attenuation zones seen in the single array analysis. However, the second band gap shows approximately 6 dB less attenuation compared to the numerical results, which can be attributed to the stronger damping effect at higher frequencies. While the band gaps become wider, their attenuation amplitude is reduced. Band gaps have been also identified at the remaining configurations; 470–520 Hz and 625–645 Hz for configuration AC, 470–525 Hz and 935–955 Hz for configuration BC, and 485–515 Hz, 620–640 Hz and 955–1040 Hz for configuration ABC. In these configurations, the lowest band gap agrees well with the numerical results in terms of amplitude



and width, while the highest band gap shows reduced attenuation, especially for configuration BC—approximately 20 dB difference. The extra peaks identified for frequencies higher than 1000 Hz, particularly for configurations BC and ABC, can be attributed to the increase in wave reflections due to the addition of more arrays in comparison to the case of a single-cushion. Another reason stands for possible modal coupling effects, since the impact load may excite several modes. The deformation pattern of each configuration is obtained by numerical models illustrated in figure 11. The patterns are identified in each frequency related to the highest attenuation for each configuration, which are highlighted by the red vertical lines in figure 10. Notice that the spiral resonators are mostly excited at their corresponding frequencies, while outside those, the resonators move slightly. This confirms the ability of the meta-cushion in filtering waves from distinct frequencies.

Interestingly, the experimental responses of configurations AC and ABC, with band gaps in close frequency ranges, show a reduction of the resonant peaks presented between the attenuation zones, which can be related to band gap merging and damping effects [30, 31]. The merging principle can be applied to increase the width of band gaps in low-frequency ranges, which have demonstrated to be narrower for arrays type A and C than for array type B. In configuration ABC, the shift of the highest peak above 1000 Hz is related to an increase in stiffness in comparison to the other configurations, which only contain four arrays. The increase in stiffness of configuration ABC is also indicated by the low FRF amplitude for frequencies near zero, indicating then low acceleration of the structure. The analysis presented here confirms the meta-cushion capability to attenuate vibrations across multiple frequency zones, demonstrating its potential in reducing multiple pile vibrations during their installations in an offshore environment.

5. Conclusions

This study demonstrated the effectiveness of the meta-cushion in attenuating vibrations across multiple frequency ranges, as shown by both experimental and numerical analyses. Experimental results for individual arrays targeting specific frequencies aligned well with numerical models, particularly when viscoelastic effects were considered. Similar behavior was observed for single and combined meta-cushions. While the meta-cushion provided broader attenuation zones near 1000 Hz than the numerical model, the performance was slightly reduced due to frequency-dependent damping. The experimental validation of combined meta-cushions confirmed the principle of multiple vibration attenuation. Although the low-frequency band gaps were narrow, the merging of adjacent gaps using different spiral shapes showed promise for enhancing low-frequency attenuation. Some extra peaks and shifts in the band gaps suggest the need for a refined viscoelastic model, but the numerical models remain reliable for assessing attenuation performance and can serve as a basis for developing new designs without requiring experiments for each prototype.

Future research will focus on refining numerical models by incorporating viscoelastic effects more accurately, particularly in scenarios involving multiple resonant effects. This will improve the prediction of band gap location and attenuation performance. Additionally, regarding the transition to full-scale implementation, the resonant frequency of the spiral unit cell scales as $f_r \propto (1/a)\sqrt{(E/\rho)}$, where a is the unit cell dimension, E is the Young's modulus, and ρ is the material density. For steel ($E \approx 200$ GPa, $\rho \approx 7850$ kg m⁻³) relative to PLA ($E = 3.16$ GPa, $\rho = 1250$ kg m⁻³), maintaining the same band gap frequencies requires scaling the unit cell size by approximately a factor of 3.2. This yields a unit cell of 127 mm, which is feasible for monopile applications with diameters of 4–8 m. The non-dimensional geometric parameters (w/a , n_t , θ_t , θ_F) can be preserved to maintain the band gap behavior, providing a direct design pathway for full-scale steel meta-cushions [13]. Using this scaling approach, meta-cushions can be tested in an offshore environment. With respect to the manufacturability in steel, modern manufacturing techniques (CNC machining, laser cutting, wire EDM) can produce spiral geometries at these dimensions. Finally, the top and bottom plates of the meta-cushion serve as load-bearing elements and can be designed with sufficient thickness to withstand the compressive stresses from pile-driving impacts.

The present study demonstrates the vibration filtering capability of the meta-cushion at laboratory scale. The transition to full-scale offshore implementation requires further research, including scaling analysis for steel designs, fluid-structure interaction modeling for acoustic validation as discussed in our previous work [13], structural integrity and fatigue life assessment under representative impact loads, and field testing in realistic offshore conditions. Here, the effectiveness of the meta-cushion in mitigating monopile vibrations and reducing underwater noise during pile installation will be measured against baseline data from installations without any noise mitigation system. These steps constitute the roadmap for bringing the meta-cushion from laboratory validation to practical noise mitigation application.

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Data availability statement

The data that support the findings of this study are openly available at the following URL/DOI: <https://doi.org/10.4121/45d5dc49-5b4c-4152-97f3-29105a6e7f59.v1> [32].

Conflict of interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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