

Self-Tuning Vibration Absorber Made With In-Vacuo Structured Fabrics

Lisa Ortis, Daniel Casagrande, Paolo Gardonio, Emiliano Rustighi, Ciro Malacarne, Matteo Perini

Abstract. This paper presents a study on the design and implementation of a self-tuning beam-like vibration absorber. The beam element is fixed to a middle post and it is made by a core structured fabric encased in a deflated plastic bag. The bending stiffness of this beam can be suitably varied by modulating the vacuum in the bag. In this way the resonant response of its flapping vibration resembles the dynamics of a mass-spring-damper absorber, whose fundamental resonance frequency can be duly adjusted by varying the level of vacuum in the bag. The paper presents an extensive experimental analysis of the tuning features and vibration control performance generated by a prototype absorber, which is fixed on a plate hosting structure excited by a broadband random disturbance. Two tuning principles are investigated: the first relies on the so-called “global cost function” given by the minimisation of the plate total flexural kinetic energy; the second relies on the so-called “local cost function”, which is based on the maximization of the vibration power absorbed by the flapping vibration of the beam-like element of the absorber.

Key words: Tuneable Vibration Absorber, Adaptive Absorber, Self-Tuning, Structured Fabrics, In-vacuo structures.

1 INTRODUCTION

Traditionally and up to 1980s, mostly passive control strategies were used to control the structure-borne and air-borne noise of lightweight structures. But as the need to reduce the

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ecological impact of machineries, in terms of raw materials and energy consumption, progressively increased over time, the vibration response and sound radiation of these machineries also changed. The increasing stiffness to mass ratio of the newer materials leads to two effects: the acoustic wavelength and the flexural wavelength coincide at lower frequencies [1]; and the low modal overlap frequency band of the flexural response is extended to higher frequencies [2], [3]. There is then, an increment of the structure- and air-borne sound-radiation in an extended low-frequency band in which passive control strategies are less effective, unless heavy vibration and insulation treatments are employed [4], [5], [6]. This led to the development of active and semi-active control strategies [7], [8], [9], [10], [11], [12]. Active control systems, mostly used in the transportation sector can generate large vibration reductions even when the dynamic response of the hosting structure undergo large changes, but they are more complex and costly [13], [14], [15]. Semi-active vibration control systems on the other hand, can deliver good performances even for significant changes in the structural response and are less complex and costly [10], [12].

Tuneable vibration absorbers (TVAs) have been widely studied and implemented extensively during the last century [16], [17], [18], [19]. These are usually composed of a seismic mass fixed on an elastic suspension [19], [20], [21], [22]. Three parameters are crucial to achieve good performances controlling the resonant response of a flexural mode of a structure: the position of the TVA, its natural frequency and its damping. The TVA should be mounted in an antinodal point of the target mode, the natural frequency of the TVA should be about the same as the target mode and its damping should be set to maximise the vibration absorption [3], [23]. Many studies have been conducted to propose tuning laws that find optimal values for the stiffness and damping of a TVA mounted on a simple structure [17], [18], [19], [24], [25], [26], [27], [28], [29], [30], [31]. Typical values for the seismic mass are in the range of 5% to 15% of the mass of the hosting structure and for the damping factor of the TVA, which usually is not so critical, are between 0.1 and 0.2. On the other hand, the natural frequency of the TVA, which is related to its mass and stiffness, should be finely tuned in the vicinity of the frequency of the target mode [31].

This paper presents experimental results of a prototype TVA attached to a thin plate structure to control the resonant response of a target flexural mode when it is being excited by a broadband random disturbance. The TVA is composed of in-vacuo structured fabrics contained in an airtight plastic film enclosure. The stiffness of the prototype can be tuned by modifying the pressure inside the plastic bag. The structured fabric is a regular network of 3D printed interlocked hollowed trusslike rigid particles [32], [33], similar to medieval chain armours and is manufactured using a 3D printer [34], [35]. When air is drawn out of the bag, the pressure generated by the deflated plastic film forces the particles to jam [36], [37], [38], [39], [40], [40], [41]. The coated fabric now behaves as a rigid body, whose stiffness is proportional to the vacuum pressure [32], [33]. The dynamic of the prototype resembles that of a classical mass-spring-damper absorber with tuneable fundamental resonance frequency [42], [43] using a vacuum controller to reach the stiffness value required for a properly tuned TVA. Two control laws are implemented to tune the TVA to control the resonant response of the first flexural mode of a plate model structure: a “global cost function”, which is set to minimise the time-averaged total flexural kinetic energy of the plate; and a “local cost function”, which is set to

maximise the time-averaged vibration power absorbed by the TVA [23], [31]. To minimise the response of a target mode, the two cost functions are derived with reference to the kinetic energy and vibration power functions filtered in a small frequency band centred at the resonance frequency of the target mode.

The paper is structured in five sections. Section two describes the experimental set-up and the proposed prototype TVA. Section three introduces the global and local tuning cost functions. Section four presents a comprehensive experimental analysis of the vibration control performance of a single TVA, tuned to target the first mode, on the flexural response of the structure excited by a random disturbance. This analysis gives insights on the effect of the proposed TVA on the power absorption of the TVA and on the flexural kinetic energy of the structure. Finally, section five reports the conclusion of this study.

2 PLATE WITH THE IN-VACUO STRUCTURED FABRIC TVA

Figure 1a,b shows the model structure used to investigate the tuning and vibration control performance features of a TVA composed by a beam-like in-vacuo element fixed to plate. The plate is made of steel and has dimension $L_x \times L_y = 668 \times 443$ mm and thickness $h_p = 1.8$ mm. As can be seen in Figure 1c, the plate is excited by a transverse point force exerted by a shaker at position $x_f = 0.245L_x$, $y_f = 0.105L_y$ and is equipped with one TVA at position $x_{TVA} = 0.455L_x$, $y_{TVA} = 0.300L_y$. The post of the TVA is fixed to the plate via an impedance head to measure the force and velocity at the base of the TVA, which have been used to derive the vibration power absorbed by the TVA. Also, the stinger of the shaker is fixed to the plate via a force cell used to measure the reference force excitation employed in the experiments.

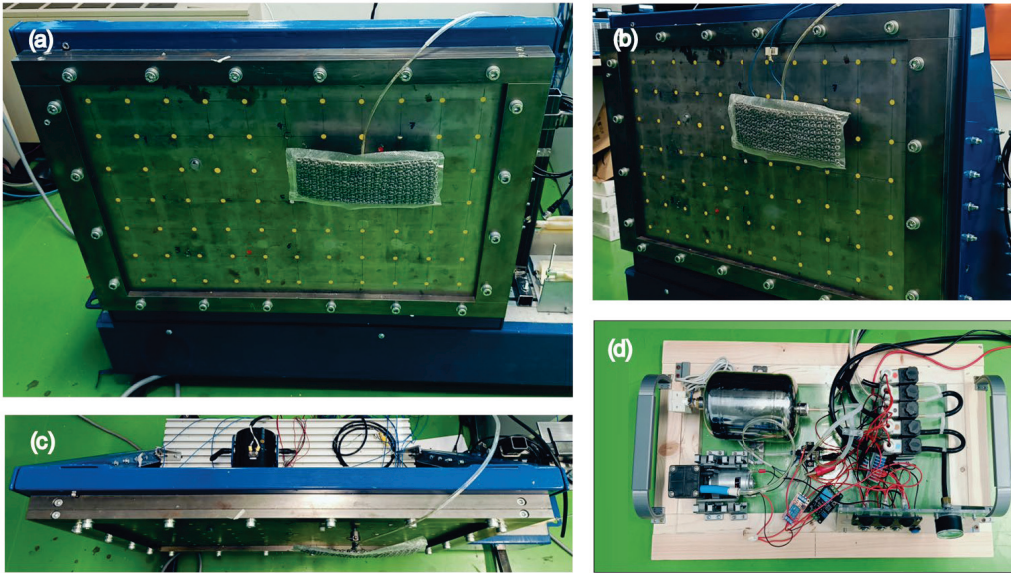


Figure 1: Experimental setup: (a) front and (b) lateral views of the plate with the beam-like vibration absorber made with the in-vacuo structured fabric; (c) top view of the plate and shaker that generates the force excitation; (d) vacuum control plant.

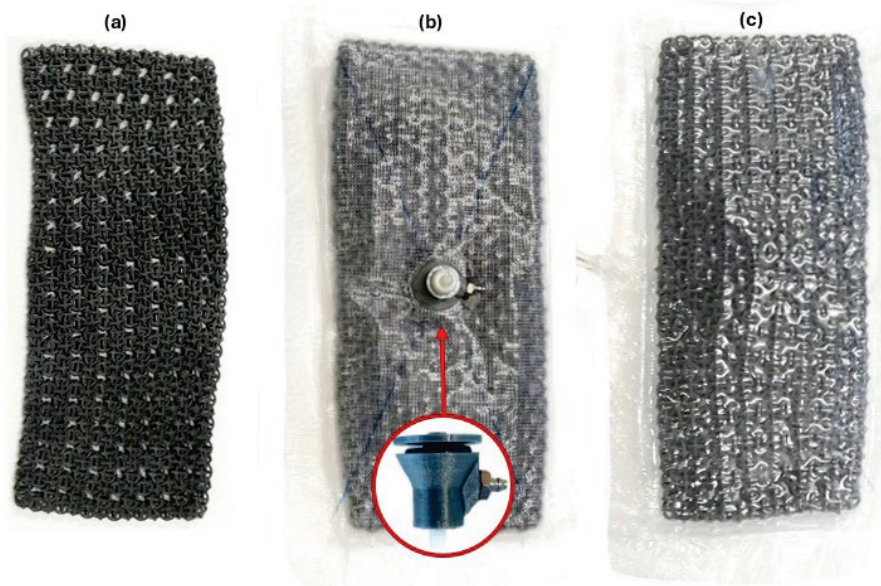


Figure 2: (a) Core fabric made of spherical-octahedral particles. (b) Bottom view of the deflated beam-like structure core fabric encased on a sealed plastic skin equipped with a post-vacuum junction in the middle. (c) Top view of the deflated beam-like structure.

As shown in Figure 2, the TVA is formed by a beam-like composite structure encompassing a core structured fabric and a sealed plastic skin, which is fixed in the middle to a post that works as vacuum junction too (Figure 2b). The core fabric is formed by interlocked rigid truss-like hollowed particles with spherical-octahedral shape [33]. This material shows a fluidlike state due to the mobility of the loosely interlocked grains (Figure 2a). However, when it is enclosed in the vacuum bag, the uniform pressure exerted by the plastic skin forces the truss-like hollowed grains to jam and lock such that the whole composite beam becomes solid (Figure 2c) [36], [37], [38], [39], [40], [41], [44]. At the grain scale, the jamming of the truss-like hollowed particles is characterized by compressive friction [36], [37], [38] that arise both at convex and non-convex contact surfaces (e.g. see page 241 - graph 3c,d in Ref. [32] and Ref. [40], [45], [46]). Nevertheless, at the larger scale of the composite beam, the convex and non-convex contacts generate resistances to compression and to traction respectively [32], which are essential to develop a resistance to bending. For instance, the four-point bending tests presented in Ref. [33] showed that the bending stiffness of a beam-like specimens like that used to fabricate the TVA for this study can be increased by about 40% by raising the vacuum pressure from 20 kPa to 80 kPa. Also, Ref. [43] showed that when the specimens are equipped with a middle post, the vibration of the base post produces a fundamental flapping vibration mode of the beam structure whose resonance frequency can be raised by about 20% when the vacuum pressure is brought up from 5 kPa to 80 kPa.

Ref. [47] showed that this flapping vibration mode resembles the dynamic response of classical vibration absorbers, which can be modelled in terms of a base and a seismic mass

connected via a spring and damper in parallel. Indeed, the spectrum of the absorber base impedance Frequency Response Function (FRF) is characterised by a resonance peak at the fundamental resonance frequency due to the flapping vibration of beam-like element, such that, at this frequency, the absorber effectively drains vibration energy from the hosting structure.

3 GLOBAL AND LOCAL TUNING COST FUNCTIONS

Two tuning laws are investigated in this study. The first relies on a global cost function that accounts for the time-average of the total flexural kinetic energy of the panel, whereas the second is based on a local cost function given by the vibration power absorbed by the TVA. To focus the control on the resonant response of a target r -th mode, the two functions are filtered in a small frequency band $\Delta\omega$ centred at the r -th flexural resonance frequency of the plate. Here, the two cost functions have been derived first in the frequency domain and then integrated over the small frequency band $\Delta\omega$. For instance, the spectrum of the time-averaged total flexural kinetic energy of the plate has been derived with the following expression:

$$\bar{K}(\omega) = \frac{1}{T} \int_0^T \int_S \frac{1}{2} \rho h \dot{w}(x, y, t)^2 dS dt \cong \frac{M}{4} \sum_{n=1}^N |Y_n(\omega)|^2 F^2. \quad (1)$$

Also, the time-averaged vibration power absorbed by the TVA has been derived from the following formula:

$$\bar{P}(\omega) = \frac{1}{T} \int_0^T f_a(t) \dot{w}_a(t) dt \cong \frac{1}{2} \text{Re}\{T_a^*(\omega) Y_a(\omega)\} F^2, \quad (2)$$

The two cost functions were then averaged in the $\Delta\omega_r$ frequency band centred at the r -th flexural resonance frequency of the plate, such that:

$$\bar{K}_r = \frac{\Delta\omega_r}{N_c} \sum_{i=1}^{N_c} \bar{K}(\omega_i), \quad (4)$$

$$\bar{P}_r = \frac{\Delta\omega_r}{N_c} \sum_{i=1}^{N_c} \bar{P}(\omega_i). \quad (5)$$

Here N_c is the number of samples of the kinetic energy and power spectra in the frequency band $\Delta\omega_r$ and ω_i is the frequency at the i -th sample. Also, in Eqs. (1), (2), T is the period of the time-harmonic vibration, $()^*$ indicates the complex conjugate and ρ , h , S , M are respectively the density, thickness, surface area and mass of the plate. Also, $Y_n(\omega) = \dot{w}_n(\omega)/f(\omega)$, $Y_a(\omega) = \dot{w}_a(\omega)/f(\omega)$ are the mobility FRFs measured respectively between the $N = 9$ accelerometers and shaker force cell and between the TVA impedance head and the shaker force cell. Moreover, $T_a(\omega) = f_a(\omega)/f(\omega)$ is the force transmissibility FRF measured between the TVA impedance head and the shaker force cell. Here $\dot{w}_n(\omega)$, $\dot{w}_a(\omega)$, $f_a(\omega)$ and $f(\omega)$ are the complex amplitudes of the time-harmonic velocities and forces defined in the following complex forms $\dot{w}_n(\omega) = \text{Re}\{\dot{w}_n(\omega) \exp(j\omega t)\}$, $\dot{w}_a(\omega) = \text{Re}\{\dot{w}_a(\omega) \exp(j\omega t)\}$, $f_a(t) = \text{Re}\{f_a(\omega) \exp(j\omega t)\}$ and $f(t) = \text{Re}\{f(\omega) \exp(j\omega t)\}$, where $j = \sqrt{-1}$ and ω is the circular frequency. Finally, $F = |f(\omega)|$ is the amplitude of the force excitation exerted by the shaker. The global cost function has been

used as a reference metric to identify the optimal tuning of the TVA that guarantees the control of the overall flexural response of the plate at the r -th flexural resonance frequency. Instead, the local cost function has been considered as a practical metric that can be easily measured at the TVA position and thus used to tune online the TVA.

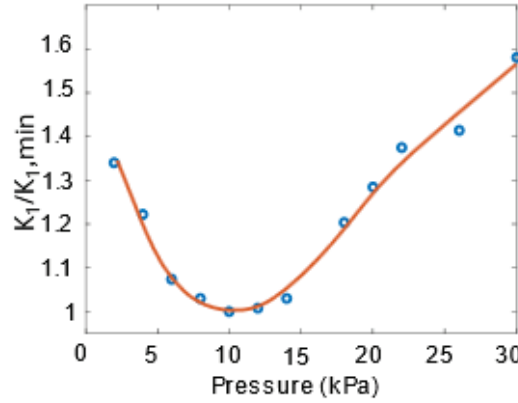


Figure 3: Normalised time-averaged total flexural energy of the panel with the vibration absorber filtered in a small band centred at the first flexural resonance frequency of the panel with respect to the vacuum pressure in the beam-like element of the absorber.

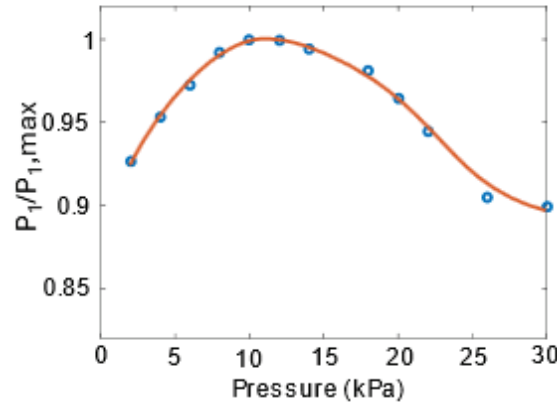


Figure 4: Normalised time-averaged electric power absorbed by the vibration absorber mounted on the panel filtered in a small band centred at the first flexural resonance frequency of the panel with respect to the vacuum pressure in the in the beam-like element of the absorber.

Figures 3, 4 show the two cost functions with reference to the resonant response of the first flexural mode such that $\omega_1 = 32$ Hz and $\Delta\omega_1 = 14$ Hz. The graph in Figure 3 shows that, the time-averaged kinetic energy, that is the spatially averaged flexural response, is characterised

by an inverted bell shape with a minimum for a vacuum pressure of 10 kPa. In parallel, the graph in Figure 4 shows that the time-averaged vibration power absorbed by the TVA is characterised by a bell shape whose maximum is at the same vacuum pressure of 10 kPa. Hence, as found in Ref. [23], the TVA can be swiftly tuned simply by setting the vacuum pressure in the beam element of the TVA to maximise the vibration power absorbed by the TVA itself. This is a rather important finding, since it provides a way to tune online the TVA without the need to measure the overall flexural vibration of the hosting structure; in this case of the plate. This finding complies with similar outcomes obtained for TVA made with a shunted coil-magnet seismic transducer [23] and with a shunted piezoelectric transducer [48], [49]. Indeed it is expected that, as proposed in [48], [49] the self-tuning can be conveniently implemented with a model-free extremum seeking control algorithm [50], [51], [52], [53], [54].

4 SPECTRA OF VIBRATION RESPONSE AND POWER ABSORBED

The spatially averaged flexural vibration of the plate and the vibration absorbed by the TVA are now analysed in the frequency domain with respect to the spectrum of the plate flexural kinetic energy and the spectrum of the vibration power absorbed by the TVA, which were defined in Eqs. (1), (2). The black line in Figure 5a shows that, in the 20 to 150 Hz frequency range, the spectrum of the flexural kinetic energy of the panel without TVA is characterised by 7 resonance peaks, which are well separated from each other. Hence, at each frequency, the response is controlled by the resonant response of a specific flexural mode of the plate. The blue line shows that, when the plate is equipped with the TVA set to maximise the vibration power absorption in the $\Delta\omega_1 = 14$ Hz bandwidth centred at the nominal resonance frequency of the first flexural mode, $\omega_1 = 32$ Hz, the first resonance peak is replaced by the typical double peaks generated by vibration absorbers. The two peaks have equal amplitude, which indicates the TVA is optimally tuned [17], [18], [19], [20], [21], [22]. Indeed, the amplitude of the two peaks is about 18 dB lower than that of the single peak for the plain panel.

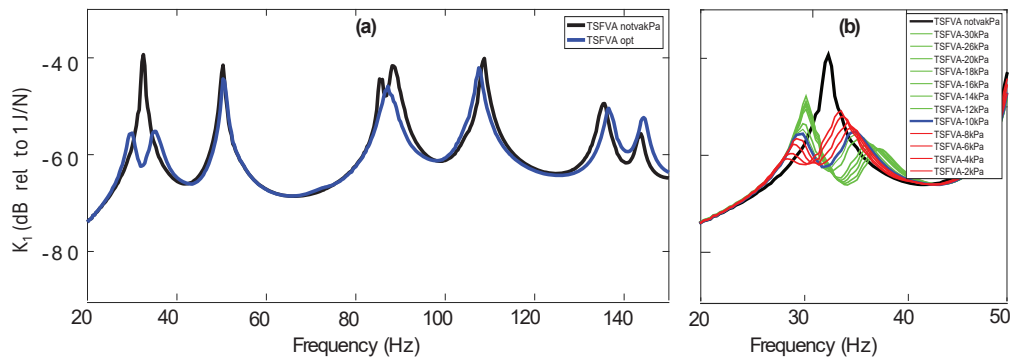


Figure 5: (a) Time-averaged total flexural kinetic energy of the plate (black line) and plate with the vibration absorber optimally tuned (blue line). (b) Magnified plot in the vicinity of the target resonance frequency for the cases where the vibration absorber is under-tuned (green lines), optimally tuned (blue line) and over-tuned (red lines)

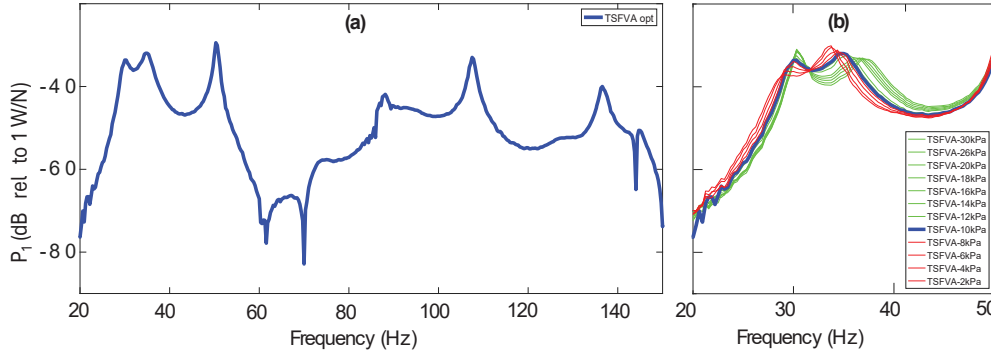


Figure 6: (a) Time-averaged vibration power absorbed by the optimally tuned vibration absorber mounted on the plate (blue line). (b) Magnified plot in the vicinity of the target resonance frequency for the cases where the vibration absorber is under-tuned (green lines), optimally tuned (blue line) and over-tuned (red lines)

The blue line in Figure 6a shows that the spectrum of the vibration power absorbed by the TVA has a wide-band pick centred at $\omega_1 = 32$ Hz, which is characterised by two apices. This confirms that indeed there is a significant vibration absorption effect from the resonant response of the first flexural mode of the plate. The spectrum shows narrowband peaks at the higher resonance frequencies of the plate, indicating significant vibration absorptions at these frequencies too. As discussed in Ref. [47], the dynamic response of the TVA made with the invacuo beam element can be modelled in terms of a base and seismic mass which are connected via a spring and damper. At the fundamental resonance frequency, the two masses are characterised by large counter-vibrations such that damper effectively “pumps” vibration energy from the base vibration. At higher frequencies above the fundamental resonance frequency, the seismic mass vibrates progressively less and becomes a sort of sky-hook reference. Hence the damper still produces an energy pumping effect, which however is controlled only by the motion of the base mass and thus it is bound to be relatively large at resonance frequencies of the hosting structure only. This explains why, there is a somewhat large vibration absorption at higher resonance frequencies of the plate structure. Indeed, the large resonant vibrations enhance the vibration of the base mass such that the damper effectively absorbs energy.

To conclude these analyses, the graphs in Figure 5b and 6b show the magnified spectra of the flexural kinetic energy and vibration power absorption in the vicinity of the first resonance frequency. Here the spectra are reported for the cases where the vibration absorber is under-tuned (green lines), optimally tuned (blue line) and over-tuned (red lines). The graphs clearly show how the optimally tuned TVA produces the equal amplitude double peak effect, whereas the under-tuned TVA is characterised by a small-large pair of peaks and the over-tuned TVA is characterised by a large-small pair of peaks [47].

5 CONCLUSIONS

This paper has studied the tuning and vibration control effects of a beam-like TVA made with a beam element mounted to a middle post, which is formed by a structured fabric encased in a deflated plastic bag. Two tuning principles have been investigated, which rely on a “global cost function” given by the minimisation of the plate total flexural kinetic energy, and a “local cost function”, based on the maximization of the vibration power absorbed by the TVA. The study has shown that the global cost function is characterised by an inverse bell shape whereas the local cost function has a bell shape with respect to the vacuum pressure in the in the beam-like element of the TVA. The minimum and maximum values of the two functions occur at the same level of vacuum. Hence the local cost function can be effectively used to tune the absorber without the need of complex sensing arrangements that detect the vibration of the whole hosting structure. The study has also demonstrated that the proposed TVA can generate reductions of the order of 18 dB of the resonance peak of the target mode.

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