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Numerical Analysis of an Auxetic Anti-tetrachiral Sandwich Panel Subjected to Steady-state Harmonic Base Motion

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Abstract In this paper, the authors show the results of numerical simulations representing the test of an aluminum sandwich panel with an auxetic anti-tetrachiral core on an exciter. Steady-state vibration analyses utilizing modal superposition (linear dynamics) were performed. The bottom of the panel had all the degrees of freedom constrained and excitation in form of base acceleration in the vertical direction was applied. The obtained results were in form of contour plots of selected output variables in the frequency domain. In addition, curves showing the variation of acceleration, velocity and displacement of a selected representative point in frequency were generated. The results were compared with those obtained for the panel with a non-auxetic core, in the form of a standard hexagonal honeycomb. It was found that the auxetic panel is not superior in the whole range of frequencies but a workflow useful in the design of sandwich panels for operating conditions involving vibrations was developed.

Keywords: steady-state dynamics, auxetic, sandwich panel, base motion, finite element analysis.

1. Introduction

Auxetic materials and structures are characterized by a negative Poisson's ratio. Because of that, they exhibit unintuitive deformation patterns – unlike regular materials, they become thicker when stretched and thinner when compressed. The existence of such materials was confirmed in 1920 but the name "auxetic" (from Greek "auxetikos" – tending to increase) was introduced in 1991 [1].

The large interest in auxetics started in the 1980s, along with the growing number of possible applications of these materials. They can be used for protective equipment, implants and prostheses or components for the aerospace industry, among others [2]. Researchers already designed many different auxetic structures topologies for use in various conditions. However, even the most basic unit cell shapes, such as re-entrant (introduced by Gibson et al. in 1982) or anti-tetrachiral, still require further investigation in terms of their response to some types of loads, especially dynamic ones. One of the most important discoveries in terms of auxetics was the development of a special method of turning regular foams into auxetic ones. This was achieved by Lakes in 1982. The method is based on the fact that particular thermo-mechanical loading conditions result in a change of foam microstructures in such a way that auxetic characteristics appear [1].

Sandwich (three-layered) structures, including panels, are commonly used for demanding applications where strong but lightweight components are needed. This is particularly important for the aerospace and defense industry. Sandwich panels typically consist of thin outer layers (usually made of lightweight metal such as aluminum) and a thick but porous core. The core can be made of foam but may also be in the form of a honeycomb structure, auxetic or non-auxetic [3,4]. The connection between the layers depends on the manufacturing technology but usually, cover plates and core are permanently connected, with or without the use of an adhesive. As an example of a manufacturing process leading to such bonding, one may consider extrusion or 3D printing.

Significant research has looked at the behavior of auxetic sandwich panels during indentation tests [5]. One of the first such studies was carried out by Chan and Evans [6] in 1998. The authors tested both conventional and auxetic foams and observed that the latter ones deform differently. The auxetic foam tends to flow towards the indenter, densifying and thus strengthening around the contact area. As a result, such foams exhibit significantly larger stiffness and smaller deflection under the same load.

The response of auxetic sandwich panels to dynamic loads (in the form of blast waves or projectile impact) was also covered. The authors performed such studies on the panels with the same geometry as the ones discussed in this article. The conclusion was that auxetics have increased resistance to these kinds of loads and thus can be used as protective structures, for example in the case of military vehicles [7,8]. Other researchers, such as Imbalzano et al. [9] and Novak et al. [10] also confirmed the potential of auxetics to be used for blast and ballistic protection. Nevertheless, the topic of vibration of auxetic sandwich structures is much less common in the literature. In 2015 Strek et al. [11] described the results of dynamic simulations performed on a sandwich panel with an auxetic core immersed in the filler material. In 2016, Mukhopadhyay et al. [12] performed free-vibration analyses on sandwich panels with a randomly irregular honeycomb structure used as a core. Strek et al. [13] in 2018 verified the mechanical impedance of a particular sandwich beam with auxetic metal foam core. In the same year, Nguyen et al. [14] investigated the nonlinear dynamic response and vibration of auxetic sandwich panels. Two works regarding the topic of vibration of auxetic sandwich panels were also published in 2020. Li et al. [15] used the finite element method to analyze large amplitude vibrations of auxetic panels with a functionally graded core. Tran et al. [16] performed dynamic analyses of sandwich panels with re-entrant core subjected to moving oscillator load on elastic foundation.

Sandwich panels can be used as vibration isolators for various machinery, including sensitive measuring equipment. The versatility of such panels arises from the fact that the geometry of their cores can be easily modified and adapted for particular operating conditions. However, the high number of parameters that can be adjusted makes it difficult to adapt the designs manually. Thanks to the simulation approach discussed in this article it is possible to make this process much more convenient and efficient.

The goal of this paper is to verify the dynamic behavior of a sandwich panel with an auxetic anti-tetrachiral core when compared with the non-auxetic equivalent. Steady-state vibrations under base motion excitation are considered and numerical analyses are performed to check the response of both sandwich panels. Such studies were not yet described in the literature since other researchers focused on different aspects of auxetic sandwich panels and their dynamic behavior, as explained in previous paragraphs. The goal is also to develop a new computational workflow that can be useful when designing sandwich panels for applications involving vibrations and to replace a physical test on an exciter with a numerical simulation.

2. Steady-state dynamics analyses with base motion acceleration and modal damping

The general matrix equation of motion in linear dynamics is given below [17,18]:

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{C}\dot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{F}, \tag{1}$$

where: $\mathbf{M}$ is mass matrix, $\ddot{\mathbf{u}}$ – acceleration vector, $\mathbf{C}$ – damping matrix, $\dot{\mathbf{u}}$ – velocity vector, $\mathbf{K}$ – stiffness matrix, $\mathbf{u}$ – displacement vector, $\mathbf{F}$ – external force vector. Before using a mode-based steady-state dynamics analysis, eigenfrequency extraction must be performed. The equation of motion for undamped free vibrations used in these studies takes the form [17,18]:

$$(\mathbf{K} - \lambda\mathbf{M})\mathbf{u} = \mathbf{0}, \tag{2}$$

where λ is eigenvalue.

Subsequent mode-based analyses involve the modal superposition method. It uses natural frequencies and mode shapes to evaluate the response of a structure to dynamic loads, assuming a linear problem. The deformations are calculated from combined mode shapes with each one multiplied by a scale factor. The displacement vector is defined as a sum of products of modal displacement and the generalized coordinate of each mode.

In the case of steady-state vibration analyses, it is assumed that both applied load and response are harmonic [18]:

$$\begin{cases} \mathbf{u} = \mathbf{u}^* = Ue^{i\Omega t} \\ \dot{\mathbf{u}} = \boldsymbol{i}\Omega\mathbf{u}^* \\ \ddot{\mathbf{u}} = -\Omega^2\mathbf{u}^* \\ \mathbf{F} = \mathbf{F}^* \end{cases}, \tag{3}$$

where: t – time, Ω – excitation frequency. The $*$ superscript indicates a complex quantity. Based on this assumption, one may formulate the harmonic equation of motion [18]:

$$-\Omega^2\mathbf{M}\mathbf{u}^* + \boldsymbol{i}\Omega\mathbf{C}\mathbf{u}^* + \mathbf{K}\mathbf{u}^* = \mathbf{F}^*. \tag{4}$$

The base motion excitation is a commonly used boundary condition in finite element analyses. It is utilized to simulate rigid body motions of the foundation of the structure being studied. It can be defined as a displacement, velocity or acceleration. If one of the first two types is used, the software uses differentiation to obtain the acceleration. Then it is converted into applied inertia loads [18]:

$$\mathbf{M}\ddot{\mathbf{u}}_{\mathbf{b}}, \tag{5}$$

where: $\ddot{\mathbf{u}}_{\mathbf{b}}$ – applied base motion acceleration.

As mentioned above, modes obtained from the eigenfrequency extraction procedure do not have damping accounted for. However, in most cases, at least a small damping is necessary for subsequent modal superposition analyses and thus a critical damping fraction is introduced. The damped natural frequency of each mode is given by [18]:

$$\omega_d = \omega_n\sqrt{1-\xi^2}, \tag{6}$$

where: ω_d – damped natural frequency, ω_n – undamped (original) natural frequency, ξ – critical damping fraction.

It is important to remember that numerical analyses introduce various approximations influencing the accuracy of the results. One of the most important issues is mesh density which needs to be selected in such a way that the solution does not change by more than a few percent when the mesh is further refined. On the other hand, a too large increase in mesh density may lead to the solution time becoming unacceptably long. As described in the next section, the authors accounted for this factor when performing the study. Another source of potentially large errors is an inaccurate description of the material behavior. In this case, there were no advanced material models since the simulations performed for this article assume linear material characteristics. The overall accuracy of the numerical simulations can be verified and improved with the help of physical testing but the ultimate goal is to avoid the necessity of performing such tests for each new design of the sandwich structure.

3. Research problem

For the article, numerical simulations were performed using Abaqus 2021 software. Mode-based steady-state dynamics analysis procedure described above was used. Two sandwich panels were tested – auxetic and non-auxetic. The purpose was to compare their dynamic response under the same loading. The auxetic panel had a core with anti-tetrachiral cells while the non-auxetic panel had a core in form of a regular hexagonal honeycomb structure [7-8]. The panels were designed in such a way that their overall and unit cell dimensions were nearly identical. However, some discrepancy was unavoidable due to the differences in geometries of these two types of unit cells [8]. Each panel had a size of 305x305x76 mm (including the two 5 mm thick cover plates). Unit cells had an approximate overall size of 26 mm. The geometry was based on that from previous studies [7-8]. A non-structural mass of 7 kg was applied to the top of each panel to simulate a piece of typical small laboratory equipment such as a vibration exciter [19].

Cover plates were meshed using three layers of solid C3D8 elements (linear hexahedrons) because of their significant thickness when compared with the cores. For the cores, shell S4 elements (linear quadrilaterals) were used. All shell elements had an assigned thickness of 0.76 mm [7-8]. A mesh convergence study was performed and it was found that for the selected mesh density the accuracy of the results is within an acceptable range (further double refinement of the mesh changes the results only by 1-2%). Geometries of both panels are shown in Fig. 1, including 3D views (a-b), side views with global dimensions (c-d) and unit cells (e-f).

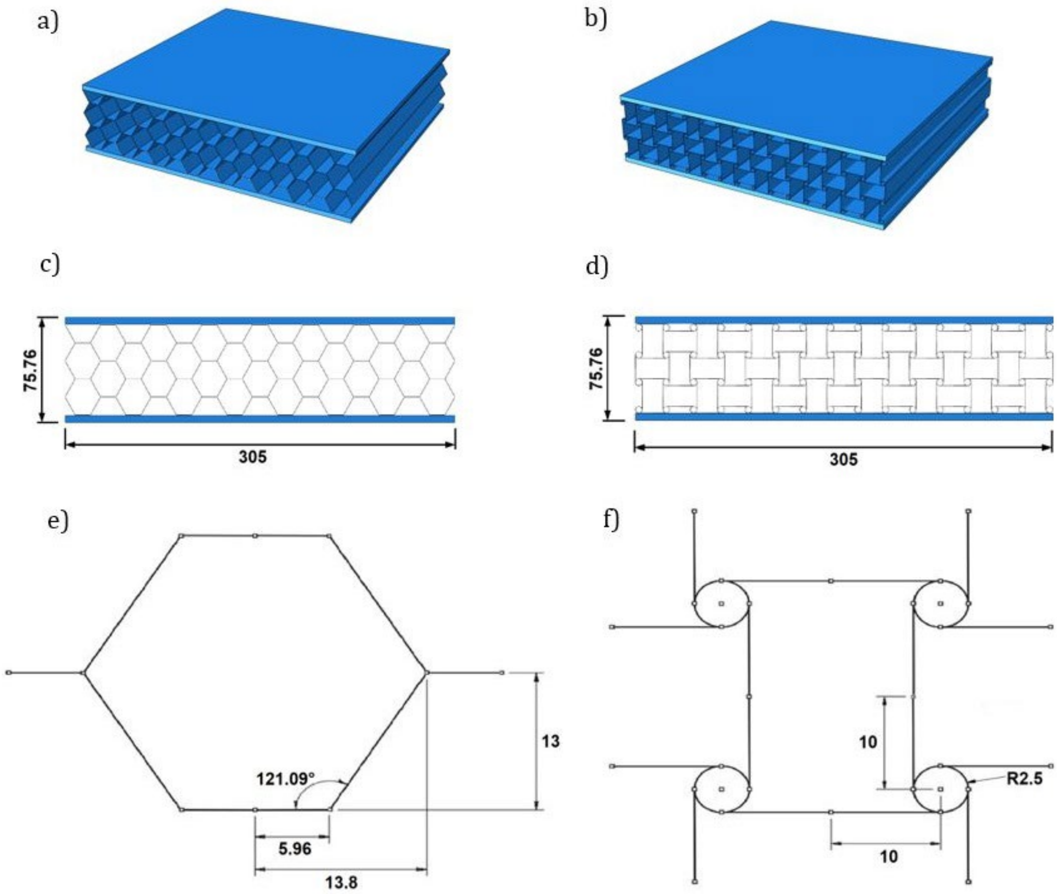

Figure 1. The geometry of the non-auxetic panel (on the left) and the auxetic panel (on the right).

It was assumed that the cores and cover plates of the sandwich panels are made of aluminum alloy 6061-T6. Properties of this material are listed in Tab. 1-2.

Table 1. Mechanical properties of Al 6061-T6 alloy [18].

Property	Value
Density	2700 kg/m^3
Young's modulus	68 GPa
Poisson's ratio	0.33

Table 2. Composition of Al 6061-T6 alloy [20].

Al	95.8 – 98.6 %	Cr	0.04 – 0.35 %
Mg	0.8 – 1.2 %	Zn	0.25 %
Si	0.4 – 0.8 %	Mn	0.15 %
Fe	0.7 %	Ti	0.15 %
Cu	0.15 – 0.4 %	other - total	0.15 %

The same analysis setup was used in both analyses. As mentioned in Sect. 1, it can be assumed that cover plates are bonded to the core and thus tie constraint was used to realize this permanent connection. The bottom surface of each panel was restrained only in the normal direction. Since modal superposition was used, the analysis consisted of two stages – eigenfrequency extraction and steady-state dynamics. Base acceleration in a vertical direction with a constant 100g amplitude, which can be considered as maximum amplitude used in the case of tests performed with small vibration shakers [21], was defined as harmonic excitation used in the steady-state vibration step. This boundary condition was defined to represent the conditions during the test on an exciter. The steady-state dynamics stage of the simulation covered frequencies between 1 and 4000 Hz and involved modal damping with a critical damping fraction of 0.02. The theory behind these concepts is described in Sect. 2.

Performed analyses belong to linear dynamics and thus none of the possible nonlinear effects (such as plasticity, damage or geometric nonlinearity) were considered. As described above, the goal of this study was to verify the behavior of auxetic and non-auxetic sandwich panel under the assumption of linear steady-state vibrations. However, a nonlinear dynamics approach can be considered for the future.

3. Results

The results of the simulations are shown below (Figs. 2-7).

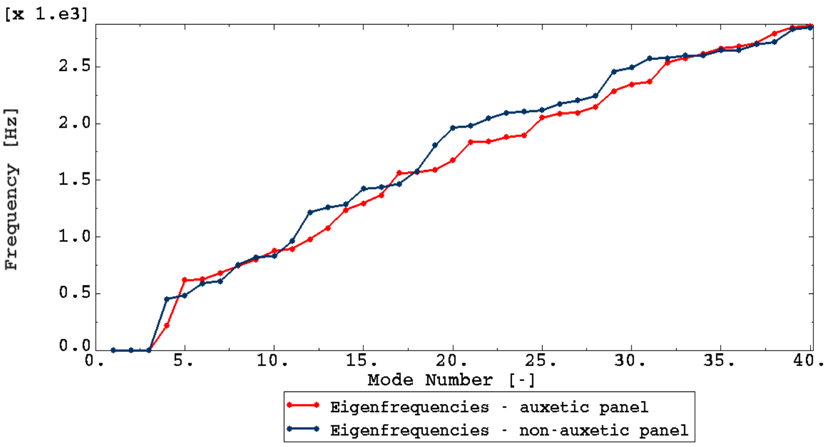

Figure 2. The distribution of eigenfrequencies for auxetic and non-auxetic panel.

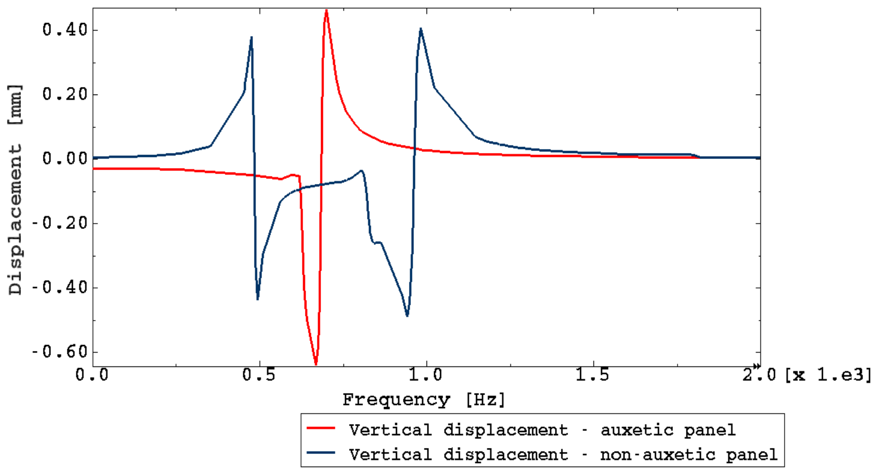

Figure 3. Vertical displacement of a node in the middle of the top surface of each panel as a function of frequency.

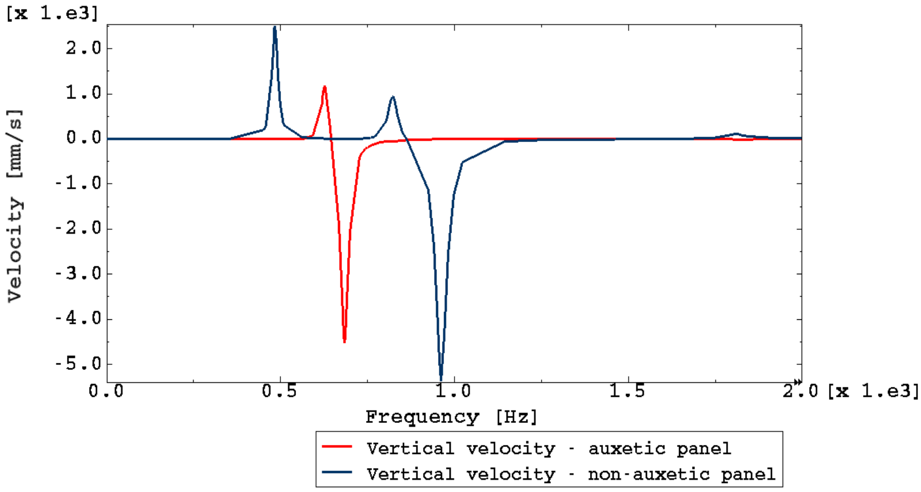

Figure 4. Vertical velocity of a node in the middle of the top surface of each panel as a function of frequency.

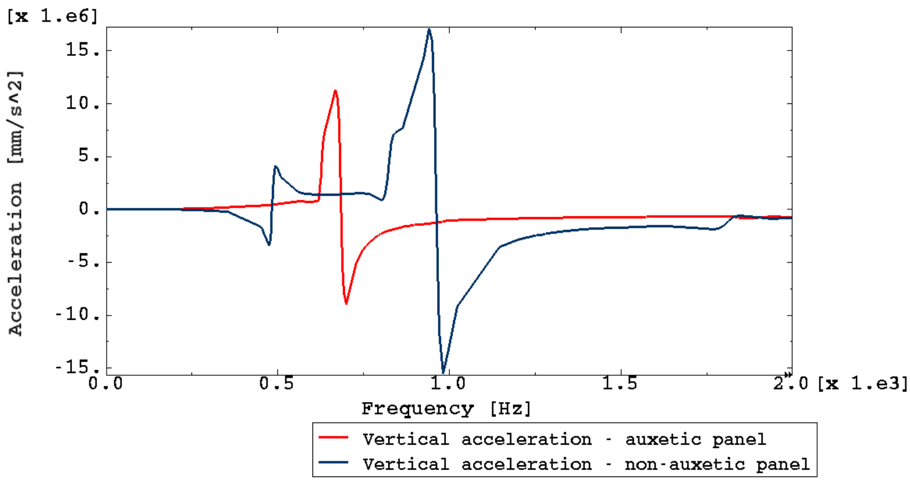

Figure 5. Vertical acceleration of a node in the middle of the top surface of each panel as a function of frequency.

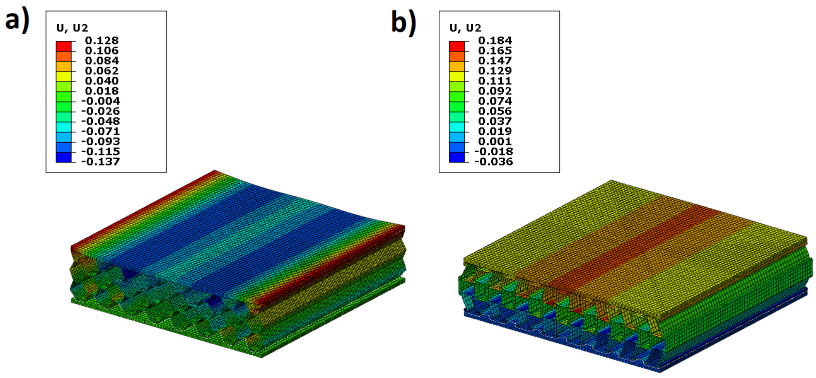

Figure 6. Vertical displacement (in mm) at a frequency of 759 Hz in the case of: a) non-auxetic panel, b) auxetic panel.

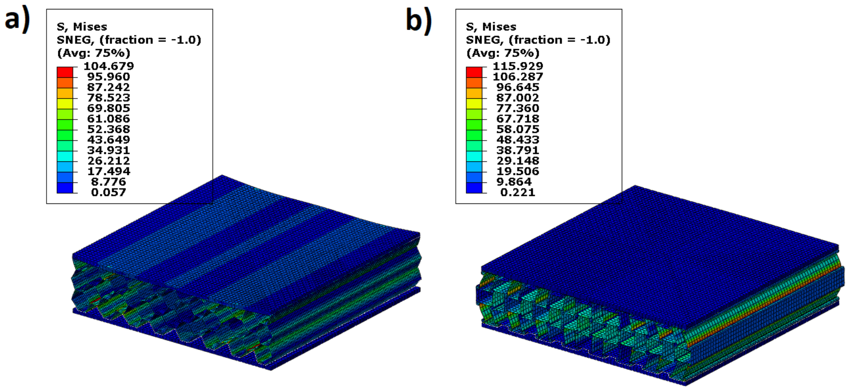

Figure 7. Von Mises stress (in MPa) at a frequency of 759 Hz in the case of:
a) non-auxetic panel, b) auxetic panel.

Obtained results indicate that there is no advantage of the auxetic panel over the non-auxetic one in the whole studied range of frequencies. However, significant differences in responses of both panels are clearly visible in all the studied results. The first of the presented results is the distribution of the natural frequencies for both panels (Fig. 2). Meaningful (not rigid body mode) eigenfrequencies were found in the range of around 200 – 2900 Hz. Plots of displacement, velocity and acceleration versus frequency (Fig. 3-5) show clear peaks at resonant frequencies. When comparing those peaks for the auxetic and non-auxetic sandwich panel, offsets in resonant frequencies can be seen. In the case of displacement, there are more peaks for the non-auxetic panel but the auxetic one has a larger maximum magnitude of displacement (26.44% difference). When considering velocities, the non-auxetic panel exhibits an additional peak and higher maximum magnitude (17.03% difference). In the case of acceleration, the situation is similar – there are additional peaks for the non-auxetic panel and the maximum magnitude is higher (41.02% difference). Results were also shown in the form of contour plots of displacement and von Mises stress for a selected frequency close to the resonant ones (Fig. 6-7). For this particular frequency, the maximum displacement and stress are higher in the case of the auxetic panel.

4. Conclusions

Results of the simulations show that the auxetic sandwich panel with an anti-tetrachiral core is not fully superior to the non-auxetic panel with a hexagonal honeycomb core in terms of resistance to vibrations in a given range of frequencies. However, there are specific frequencies for which the auxetic panel would be a better solution due to significantly lower deflection and stress. The studies described in this article might be particularly useful when a core has to be selected for a sandwich panel that will be subjected to a known range of harmonic excitation frequencies. Of course, simulations can be easily performed for other panel geometries, base acceleration magnitudes and frequencies. One could even develop a plug-in for Abaqus/CAE to easily change the conditions of the analysis and obtain clear results indicating which core will be suitable for a particular application. The development of such a tool might be of interest in a future article.

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Additional information

The authors declare: no competing financial interests and that all material taken from other sources (including their own published works) is clearly cited and that appropriate permits are obtained.

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