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Effective topologies for vibration damping inserts in honeycomb structures

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ABSTRACT

The introduction of material into the void of honeycomb-like structures, such as foam, viscoelastic or particulate filling, has been credited with improving the damping properties of the honeycombs. Optimisation of such damping inserts has been investigated, and indicates that partial occupation of the void could be more efficient, on a density basis, than full filling. The main goal of this study is to explore fully damping in honeycomb cells with inserts from the point of view of minimal increase in density and location of inserts. In this paper, damping of vibrations in the plane is investigated using analytical, finite element and topological optimisation methods to find the best locations of a damping insert within the cell.

1. Introduction

Honeycomb sandwich panels, formed by bonding a core of a honeycomb between two thin facesheets are in wide use in aerospace, automotive and marine applications due to their well known excellent density specific properties [1–4]. There are many technological methods of damping vibrations including, use of inherently lossy materials such as viscoelastic materials, viscous and friction damping, and use of smart materials such as piezoelectrics [5–9]. Some have been applied to damping of vibrations in particular to sandwich panel and honeycomb structures, including viscoelastic inserts in the cell voids [10–11]. Complete filling of the cell with foam, viscoelastic or particulate fillers have all been demonstrated to improve damping loss in honeycombs [10–14]. However, the use of an additional damping material inside the core of a sandwich panel increases its mass which is often deleterious and may also lead to a significant change in dynamic properties. The work presented here explores the competing demands of vibration damping and minimum additional mass in the case of secondary inserts in honeycomb-like structures.

The behaviour of cellular cores structures filled with viscoelastic materials has been observed experimentally in [9] for the first time with a copper foam as a matrix and an elastomer as filling material. Filling of hexagonal cores with foam was then demonstrated for improved energy and impact absorption [15–18]. Foams have also been used to fill honeycomb structures with consequent improvement of damping properties [13,14]. However, adding foam into honeycomb structures significantly increases the density of the sandwich panel, even if foams themselves exhibit relatively good density specific properties. To avoid excessive increases in density, cells may be only partially filled with an insert. For example, Woody and Smith obtained an improvement of around 60% in damping loss factor by filling only selected cells within an array, adding less than 6% to the structure's mass [14].

Structures filled with particles, generally small metallic or glass spheres, provide energy dissipation by non-elastic impact and friction damping to the vibrating structure [19–21]. One of the advantages of this technique is to provide damping in any loading mode and over a wide frequency range, and with little change in stiffness of the structure [19]. However, this approach significantly increases the density of the sandwich. Depending on the application, different materials can be used as particle dampers, e.g. metals and polymers. Michon et al. proposed using viscoelastic particles [12], the dissipation of energy by viscoelastic deformation providing additional energy loss.

Complete occupation of a honeycomb cell void with a viscoelastic material has been shown to improve damping loss [10,11]. Viscoelastic master curves for hexagonal and re-entrant honeycombs with viscoelastic filler have been illustrated in [4]. It has also shown that the design of the insert has an important impact on the loss factor of a structure [22]. The more strain energy is dissipated by the insert the more efficient the viscoelastic insert is. Designs of viscoelastic inserts inside honeycombs which improved the damping properties have been patented [23]. This patent describes the damping improvement of honeycombs with, (i) a constrained layer of viscoelastic material within the ribs of the cell and (ii) a viscoelastic material inserted in the corner of the cell.

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Considering these results, it seems that the honeycomb filling method can be optimised by the use of specific core designs or specific inserts.

In order to minimise the added mass of honeycomb structures with damping inserts, the aim of this study is to find the optimal locations for ligaments made from a high damping material within the void of honeycomb cells, allowing for different cell geometries, and under a variety of in-plane loading cases reflecting the deformation of core honeycomb cells in a range of possible structural vibration modes.

2. Methods

The approach taken was to explore the deformation and strain in a ligament connecting parts of a honeycomb cell, via closed form relations, and then to identify the location which gave rise to the largest strain of a viscoelastic ligament for a range of differently shaped honeycomb cells. The solutions were then validated using finite elements. A FE topological optimisation was also undertaken to check whether ligament stiffness, which was ignored in the analytical model, had an appreciable effect.

2.1. Parametric analytical study of honeycomb cells loaded either axially or in in-plane shear

An analytical study was undertaken to identify the maximum relative displacement of the cell ribs inside various honeycomb unit cells, for in-plane axial and in-plane simple shear loading (Fig. 1). The effectiveness of any damping insert will be maximised if it is subjected to the largest deformations and strains available. The honeycomb cell can, in this sense, amplify the local strain experienced by an insert.

This approach ignores the stiffness of the viscoelastic insert assuming that the stiffness of the cell itself dominates, as is supported by Abd El-Sayed et al. [25]. This will be invalid for cases where the very stiff or large inserts are used.

Following Gibson and Ashby [2], deflection of the cell ribs under in-plane axial loading of the honeycomb can be modelled as bending deformation of a cantilever guided at its end (l ribs in Fig. 2). However, it must be noted that the bending-only deformation of the ribs described in [2] can be considered a valid assumption for slender cell walls and for internal angles θ not approaching 0° at which point beam stretching dominates behaviour [24]. For cells where θ approaches 0° the cells are effectively square, therefore highly anisotropic and in practice are generally avoided. Eq. (1) describes the vertical deflection of the rib, where P is the load normal to the beam as represented in Fig. 3, l the length of the beam, E the Young's modulus of the honeycomb material and I the second moment of inertia of the cell wall.

$$y = \frac{P \cdot l \cdot x^2}{4 \cdot E \cdot I} - \frac{P \cdot x^3}{6 \cdot E \cdot I} \tag{1}$$

where x and y are lengths in the local coordinate system of ribs in Fig. 3.

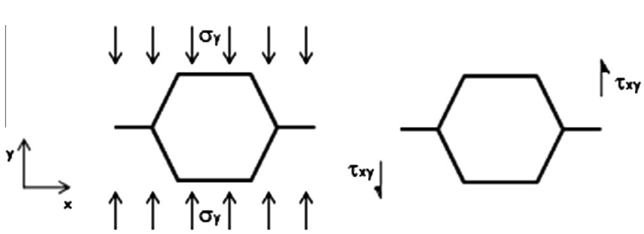

Fig. 1. Loading modes considered in the analytical model, In-plane axial loading (left) and In-plane simple shear loading (right).

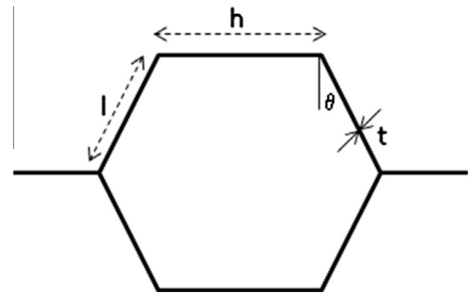

Fig. 2. Honeycomb cell with its parameters: h, l, t and θ.

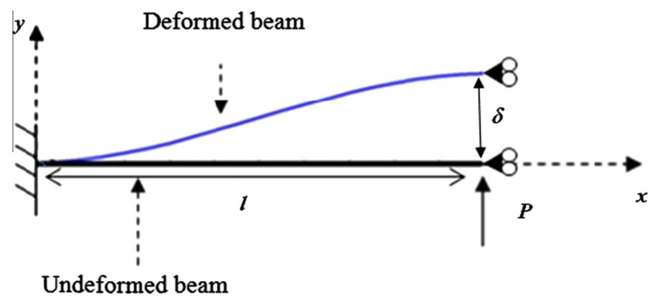

Fig. 3. Bending deflection of a cantilever beam under guided end conditions.

In-plane simple shear in the honeycomb was modelled by the bending deformation of the horizontal h ribs of the honeycomb cells using Eq. (1). The bending deformation of the oblique l ribs was taken into consideration as its deformation is negligible compare to the one of the horizontal h ribs in this specific loading [2].

Honeycomb cells were loaded under a global 1% strain ε_{global} both for the axial in-plane and shear loading. The load P for both in-plane axial and in-plane shear loading is given by Eq. (2), where δ is the deflection of an Euler–Bernoulli beam in its local coordinate system (Fig. 2).

$$P = \frac{\delta \cdot 12 \cdot E \cdot I}{l^3 \cdot \sin\theta} \tag{2}$$

Eqs. (3) and (4) show the expression of δ respectively for in-plane axial loading and in-plane shear.

$$\delta_{axial} = \frac{\varepsilon_{global} \cdot l \cdot \cos\theta}{\sin\theta} \tag{3}$$

$$\delta_{shear} = \varepsilon_{global} \cdot (h + l \cdot \sin\theta) \tag{4}$$

The deformations under these two loading modes were compared to the deformations obtained with Finite Element (FE) models. For this purpose a FE model of the honeycomb was constructed with the commercial FE software Ansys 11. Twenty Beam4 elements were used to model each beam, and boundary conditions simulating in-plane axial, in-plane simple shear and in-plane pure shear were considered. Fig. 4a shows the deformed shape of the honeycomb cells under 1% strain for the loading modes considered (displacement magnified by a factor 10). For both in-plane axial and shear loading condition the analytical model matches the FE results validating hypotheses made. Fig. 4b shows in particular that the deformed shape of the honeycomb cell when loaded in in-plane simple or pure shear at the same strain is identical.

A parametric search of all possible insert ligament locations was undertaken to identify the locations of the ligaments with maximal strain. This process is described in the following three steps: