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Elastic properties of a novel cellular structure with trapezoidal beams

Weidong Liu [a,*], Honglin Li [a], Jiong Zhang [b], Hongda Li [c]

[a] College of Energy and Electrical Engineering, Hohai University, No. 8 Fochengxi Road, Nanjing 211100, PR China
[b] School of Civil Engineering and Architecture, Wuyi University, No. 22 Dongcheng village, Jiangmen 529020, PR China
[c] Key Laboratory of Fundamental Science for National Defense-Advanced Design Technology of Flight Vehicle, Nanjing University of Aeronautics and Astronautics, No. 29 Yudao Street, Nanjing 210016, PR China

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ABSTRACT

Cellular structures are potential candidates for cores of flexible sandwich skins that are required of low in-plane stiffness for morphing and high out-of-plane stiffness for load-bearing, as well as zero Poisson's ratio for one-dimensional morphing. In this paper a novel cellular structure of close-to-zero Poisson's ratio with trapezoidal beams is proposed to meet these requirements. The elastic properties of the structure were investigated through a combination of theoretical analysis and finite element validations. Then the effects of geometric parameters on the properties of the structure were studied and analyzed. Finally comparisons on the properties with a classical accordion cellular honeycomb were carried out. Results show that the analytical results are well matched with results of the finite element validations and the proposed structure is of a better in-plane morphing property than the conventional cellular structure. Small value of thickness-to-length ratio, large values of horizontal sub-beam proportion ratio along x-direction and vertical to trapezoidal beam thickness ratio are recommended for balancing the in-plane and out-of-plane properties.

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1. Introduction

Smart structures are increasingly employed on mechanical devices to obtain better performance in practical applications during recent years [1,2]. Wings and wind turbine blades with morphing structures and adaptive control systems are attracting more and more attention in the studies of advanced mechanical designs [3–6]. Flexible skin, which is one of the key technologies for designing morphing structures, is required of low stiffness in the morphing direction, as well as high stiffness to withstand the aerodynamic load. In studying flexible skins for morphing wings, Kudva and Bartley-Cho et al. investigated a through-the-thickness flexible honeycomb structure covered with a flexible silicone rubber as the external skin for a variable camber wing [7,8]. Andersen employed metallic ribbons as the support structure, which were also covered with a silicone rubber as the skin [9]. Kikuta, Perkins, Reed and Keihl et al. studied shape memory polymers as support for flexible skins [10–13]. Yokozeki and Thill et al. proposed a corrugated flexible skin for a wing undergoing one-dimensional morphing [14,15]. The materials and structures proposed in these studies were generally capable of generating required deformations, but there are still problems to be solved such as large volume, heavy weight, complicated manufacturing and operating processes. Daynes et al. proposed a structure using a through-the-thickness aramid honeycomb as the internal support of a morphing blade of a wind turbine, which was covered with carbon fiber reinforced plastics and silicone rubber as the external skins [3,16]. This structure was of excellent load-bearing capability but not light-weighted, and hard to be manufactured due to the complex airfoil surface shape. Lachenal et al. studied an elastic skin manufactured from woven balanced plain weave glass impregnated with an epoxy resin system [17]. This skin was mainly designed for the out-of-plane twisting, so it was not suitable for large deformation applications because of large stiffness. Olympio and Gandhi proposed a concept of flexible skin using cellular honeycomb structure as internal support to withstand aerodynamic loads and external flexible skin layer to maintain smooth appearance [18]. The study showed that sandwich structure consisting of a cellular core covered with a compliant face-sheet is an ideal option for passive flexible skin.

Cellular structures have already been widely studied with various theoretical and experimental methods in recent years and proved to be of good mechanical properties [19–25]. They are potential candidates for cores of sandwich flexible skins because of their excellent characteristics such as light-weight, satisfactory in-plane morphing and out-of-plane load-bearing capabilities. Gibson and Ashby studied the mechanical properties of a conven-

* Corresponding author.
E-mail address: liuweidong@hhu.edu.cn (W. Liu).

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Nomenclature

l	length in the x-direction of the trapezoidal beam
h_t	height in the y-direction of the trapezoidal beam
t_t	thickness of the trapezoidal beam
l_{hs}	length in the x-direction of the horizontal sub-beam
g_t	spacing in the y-direction between two adjacent trapezoidal beams
t_v	thickness of the vertical beam
l_v	$h_t + g_t$, height in the y-direction of the vertical beam
b_t	depth in the z-direction of the structure
h	h_t/l, height-to-length ratio of the trapezoidal beam
t	t_t/l, thickness-to-length ratio of the trapezoidal beam
g	g_t/l, spacing-to-length ratio of the trapezoidal beams
μ	l_{hs}/l, horizontal sub-beam proportion ratio along x-direction
η	t_v/t_t, vertical beam to trapezoidal beam thickness ratio
b	b_t/l, cell depth-to-length ratio
E	Young's modulus of the raw material
E_x, E_y, E_z	equivalent elastic moduli in the respective directions
G	shear modulus of the raw material
G_{xy}, G_{xz}, G_{yz}	equivalent shear moduli in their respective planes
ν	Poisson's ratio of the raw material
ν_{xy}, ν_{yx}	Poisson's ratios in the x–y plane
$\sigma_x, \sigma_y, \sigma_z$	tensile stresses applied to the unit cell in the respective directions
T, S	shear forces applied to the unit cell in the respective directions
M_i, P_i, Q_i	internal bending moment, axial and shear forces in the respective beam
δ_i	displacement in the respective direction
ρ_r	relative density of the structure

tional hexagonal honeycomb, and presented the non-dimensional in-plane equivalent moduli considering internal bending moments, axial forces and shear forces [26]. Quantities of cellular structures with various Poisson's ratios have been investigated focusing on elastic properties [27–35]. From the previous studies it can be seen that for structures of non-zero Poisson's ratio, strain and stress are generated in the non-loading direction other than the loading direction because of the Poisson effect, which limits their applications on one-dimensional deformations such as chord-wise, span-wise and trailing edge morphing. So developing structures with zero or close-to-zero Poisson's ratio for one-dimensional morphing applications is necessary. Olympio et al. proposed and studied a classical accordion cellular honeycomb with parallel vertical beams and in between V-type beams [36]. Bubert et al. fabricated a flexible skin supported by a reinforced accordion cellular honeycomb with V-type beams, and analyzed its in-plane equivalent moduli of two directions, but didn't discuss the modulus in the third direction and the shear moduli [37]. Liu et al. proposed a cellular structure of close-to-zero Poisson's ratio with in-plane corrugated cosine beams and studied its elastic and shear moduli [38]. These studies showed that this kind of cellular accordion honeycomb is conducive to one-dimensional morphing because of its close-to-zero in-plane Poisson's ratio, but there is still room for improvement in reducing the in-plane modulus. In addition, to provide better solutions for one-dimensional applications on morphing wings and morphing blades, it is needed to develop new structures of zero or close-to-zero Poisson's ratio and with lower in-plane elastic modulus.

In this paper, a novel cellular structure of close-to-zero Poisson's ratio with trapezoidal beams is proposed. The structural equivalent elastic modulus in the x-direction and shear modulus in the x–y plane were obtained by the Castigliano's second theorem considering internal bending moments, axial forces and shear forces. The lower and upper bounds of the equivalent shear modulus in the x–z plane were determined by the principles of minimum complementary energy and minimum potential energy, respectively. Moreover, the Poisson's ratios in the x–y plane, the equivalent elastic moduli in the y-direction and z-direction as well as the equivalent shear modulus in the y–z plane were analyzed. Finite element analysis were carried out to verify the theoretical predictions of the six moduli. The effects of geometric parameters on the moduli of the structure were also analyzed. Finally, the superiority of the morphing capability of the proposed structure is presented by comparisons with the classical conventional accordion honeycomb with V-type beams and suggestions on parameters setting for designing the honeycomb are provided considering the out-of-plane load-bearing capability.

2. Theoretical models

Fig. 1 shows the geometry of the proposed structure. As shown in Fig. 1(a), the structure is composed of parallel aligned vertical beams and trapezoidal beams periodically filled in the spaces between the vertical beams. Considering the periodical layout of the structure, a unit cell is employed for analysis as shown in Fig. 1(b). The geometric parameters of the unit cell are shown in Fig. 1(c), where l is the length in the x-direction of the trapezoidal beam, h_t is the height in the y-direction of the trapezoidal beam, t_t is the thickness of the trapezoidal beam, l_{hs} is the length in the x-direction of each horizontal sub-beam in the trapezoidal beam, g_t is the spacing in the y-direction between two adjacent trapezoidal beams, t_v is the thickness of the vertical beam, l_v is the height in the y-direction of the vertical beam. From the geometry of the structure, it is seen that $l_v = h_t + g_t$. In addition, the depth in the z-direction of the structure is set to be b_t, which is not shown in the figure but is used in the analysis of the structural properties. Thus, the elastic properties of the structure can be described with the following set of non-dimensional geometrical parameters: $h = h_t/l$ (height-to-length ratio of the trapezoidal beam), $t = t_t/l$ (thickness-to-length ratio of the trapezoidal beam), $g = g_t/l$ (spacing-to-length ratio of the trapezoidal beams), $\mu = l_{hs}/l$ (horizontal sub-beam proportion ratio along x-direction), $\eta = t_v/t_t$ (vertical beam to trapezoidal beam thickness ratio), $b = b_t/l$ (cell depth-to-length ratio).

It should be noted that, to avoid the overlapping along the y-direction of the horizontal sub-beams the geometric constraint $t < g$ must be satisfied. Also, to avoid the overlapping along the x-direction, the geometric constraint $l_x < l/2$ should be satisfied when $\mu > 1/4$, as shown in Fig. 2(a).

From Fig. 2(a) it is obtained that:

$$l_o = 2l_{hs} - l/2 \tag{1}$$

Then the angle α can be represented as:

$$\alpha = \arctan(h_t/l_o). \tag{2}$$

The length of l_a can be obtained as:

$$l_a = t_t/2/\tan(\alpha/2) \tag{3}$$

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