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Active inflatable auxetic honeycomb structural concept for morphing wingtips

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Abstract

This paper describes a new concept of an active honeycomb structure for morphing wingtip applications based on tubular inflatable systems and an auxetic cellular structure. A work-energy model to predict the output honeycomb displacement versus input pressure is developed together with a finite element formulation, and the results are compared with the data obtained from a small-scale example of an active honeycomb. An analysis of the hysteresis associated with multiple cyclic loading is also provided, and design considerations for a larger-scale wingtip demonstrator are made.

Keywords: morphing wingtip, active honeycomb, inflatable structures, modelling, hysteresis

(Some figures may appear in colour only in the online journal)

Radius of the lower circular arc. Nomenclature rì Radius of the upper circular arc. Length of the oblique wall of a honeycomb r_2 а unit cell. Length of the contact surface between the С straight honeycomb cell walls and the tube. b Length of the straight wall of a honeycomb unit cell. Δh Distance between the midpoints of c and b. Top angle between the oblique and straight α Minimum value of the top angle. α_{\min} honeycomb walls. Minimum value of the lower angle. $\beta_{\rm min}$ Lower angle between the oblique and straight β Input work to the system. Win honeycomb walls. Output work from the system. W_{out} Angle between the two straight honeycomb θ cell walls. Work done by the pressurized air inside W_P the tube. Distance between the center of the tube and h Work done by the weight. the intersection of the lines related to the W_G straight honeycomb cell walls. Work related to the deformation of the W_H Initial (baseline) radius of the tube. R_0 honeycomb. Work related to the volumetric expansion of θ_0 Value of θ while the tube has a circular cross W_V the tube. section.

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ΔW Dissipated energy during the loading phase. η Loss factor.	W	Applied work during the loading phase.					
η Loss factor.	ΔW	Dissipated energy during the loading phase.					
	η	Loss factor.					

1. Introduction

Morphing technologies have demonstrated a clear ability to improve the general performance of aircraft [1, 2]. The morphing wingtip is a typical example of a shape-changing structure that improves the lift-to-drag ratio during climb by effectively changing the length of the wingspan [3, 4]. Additional advantages of using a morphing wingtip include the reduction of the induced drag at high speeds, the increase of the maneuverability by using the wingtip as an added surface control, and the reduction of the overall fuel consumption in high-altitude long-endurance flights. Another advantage provided by morphing wing tips is the possibility of reducing the parking size of airplanes, similar to the foldable wing tips used in carrier-based aircraft. Over the last two decades, new classes of structures and materials have been evaluated for morphing aircraft applications, including mechanical structures actuated by a motor [5, 6], inflatable systems [7], corrugated skins [8], multistable structures [9, 10], and shape-memory-alloy- (SMA) actuated components [11]. Honeycombs and cellular structures also constitute viable design options for morphing wing configurations [12, 13].

Honeycombs play an important role in reducing the mass of traditional airframe structures by providing a very high bending stiffness per unit weight, a feature which has also been extensively used in wingbox design [14–17]. The advantages of a honeycomb or cellular structure also lay in its deformability, which has been used to produce novel furniture designs [18] and various morphing configurations applied to wind turbine blades [13], spanwise [19–21] and chordwise morphing [22, 23], and variable camber wing designs [24]. Cellular morphing structures have also been used for deployable antennas [25] and morphing reflectors [26]; the latter exhibits a zero Poisson's ratio effect and negative stiffness under large deformations [27].

The majority of the morphing honeycomb solutions proposed so far consist of passive designs. However, some research groups have evaluated two active honeycomb configurations based either on SMA cores [28, 29] or shape memory polymers [30]. An actuation strategy adopted in morphing honeycombs consists of using a pressured fluid within the honeycomb cells. The concept of a pneumatic pressured honeycomb for a variable camber wing has been developed by producing an adaptive change via the application of either uniform [31] or differential pressures in different cells [32, 33]. Hydraulic tubes can also be used within honeycombs or segmented structures in cases such as biomimetic beam-steering antenna concepts [34, 35], robotic platforms [36–38], and prosthetic hands [39].

This paper describes a morphing honeycomb configuration with a negative Poisson's ratio (auxetic) topology, actuated by inflatable tubes. A sample unit cell of the active honeycomb is manufactured and tested, and relations between the input pressure in the inflatable tube and the overall deformation of the morphing structures are measured. Auxetic configurations feature a volumetric expansion of the solid under tensile loading and converse shrinking when the loading is compressive [40-42]. The type of auxetic honeycomb considered in this work has the classical re-entrant (butterfly) configuration [43]. As we show in the following paragraphs, the auxetic honeycomb is instrumental in avoiding the creation of external bumps during the morphing of the wingtip. An approximate analytical model is also developed to describe the behaviour of the output work of the active honeycomb versus the input pressures and other geometry parameters that define the unit cell. A finite element (FE) model is developed both for further benchmarks and to assess the limits of the validity of the analytical model. A reducedscale sample of the active honeycomb concept was also



Hinge Recovery Spring

Figure 1. Structural layout of the morphing wingtip.



Figure 2. Deformation mechanism of the proposed morphing wingtip.

produced, showing satisfactory agreement with the predictions provided by the model. The sample also shows some specific behaviour associated with the hysteresis of the system when undergoing cyclic loading, a feature that must be taken into account when dealing with different aspects of the morphing wingtip design.

2. Morphing wingtip concept

The honeycomb structural concept for the morphing wingtip developed in this paper is shown in figure 1. It consists of a wing box, the wingtip, a hinge system, and a recovery spring. A negative-Poisson's-ratio butterfly honeycomb with internal pressured tubes and a flexible composite skin make the cellular structure. The fundamental layout of the morphing wingtip demonstrator is as follows:

- 1. The wingtip is kept horizontal by a keyway, while the pressurized tube in the honeycomb helps sustain the loading.
- 2. When the pressure in the tube is reduced to zero, the wingtip leaves the keyway and flaps around the hinge under the action of the lift force and a recovery spring (figure 2). The skin follows the contraction of the cellular structure, maintaining the smoothness of the external



Figure 3. Unit cell model of the active honeycomb.

surface. The wingtip slots into another keyway to form an angle with the horizontal plane.

3. When air is pressurized into the tube, the wingtip leaves the keyway and overcomes the lift force and the recovery from the spring to return to a horizontal position. At the same time, the flexible composite skin is kept under tension.

The selection of a negative-Poisson's-ratio honeycomb for the cellular layout is clear from observing figure 2. When the wingtip flaps upwards, the honeycomb is compacted, and by auxetic effect the cellular core tends to deform toward the interior of the transverse wing section. On the contrary, the use of a positive-Poisson's-ratio honeycomb provides an external expansion of the structure during compression, reducing the flapping angle and adding drag by the creation of a bump.

3. Modeling

Two aspects considered for the active actuating honeycomb concept are the change in geometry during the deformation and the output work of the structure. The model described in this paragraph consists of a geometric part and a work-energy component. The deformations of the active honeycomb under different input pressures were also simulated using ABAQUS with linear elastic (LE) and hyperelastic properties of the tubes, respectively.

3.1. Honeycomb cell geometry

The geometry of a single cell of the active actuating honeycomb is shown in figure 3. The two circular tubes are fixed to the honeycomb's straight cell walls. When the straight walls are compressed, the tubes assume a new shape, with their section consisting of two circular arcs at the upper and lower sides and two straight lines at the left and right ends. There are four assumptions made to model this structure:



Figure 4. Relationships between c/R_0 , r_1/R_0 , and r_2/R_0 versus θ for h = 65 mm.

- 1. The honeycomb configuration and the tubes are assumed to be symmetric with respect to the interface line between the two tubes.
- 2. The section of a tube is symmetric with respect to the line that connects the centers of the two arcs. This line intersects the two straight cell walls at point O.
- 3. There is no sliding between the tubes and the honeycomb walls during pressurization and depressurization (i.e., the parameter, h, is a constant).
- 4. Hoop strains in the tubes are neglected.

When the circumference of the tube is constant, the following relation is satisfied:

$$\left(\pi - \frac{\theta}{2}\right)r_1 + 2c + \left(\pi + \frac{\theta}{2}\right)r_2 = 2\pi R_0. \tag{1}$$

From (1), it is possible to derive:

$$\tan\left(\frac{\theta}{4}\right) = \frac{r_2 - r_1}{c} \tag{2}$$

$$\frac{r_1}{r_2} = \frac{h - c/2}{h + c/2}.$$
 (3)

For a constant value of h, it is possible to derive from (1)–(3) the expressions of c, r_1 , and r_2 as functions of θ , and therefore reconstruct the change of geometry of the unit cell:

$$c(\theta) = \frac{\pi R_0 - \pi h \tan\left(\frac{\theta}{4}\right)}{1 + \frac{\theta}{4} \tan\left(\frac{\theta}{4}\right)}$$
(4)

$$r_1(\theta) = \frac{h - c(\theta)/2}{h + c(\theta)/2} r_2(\theta)$$
(5)

$$r_2(\theta) = \left(h + \frac{c(\theta)}{2}\right) \tan\left(\frac{\theta}{4}\right). \tag{6}$$

The behaviour of the nondimensional parameters c/R_0 , r_1/R_0 , and r_2/R_0 for h = 65 mm is shown in figure 4. When $r_1 = r_2 = 0$, the tube is compressed to a surface, and c

assumes its maximum value. However, this particular case is only theoretical; because of the presence of a finite thickness of the tube for c = 0 and $r_1 = r_2 = R_0$, the tube becomes circular, and the value of the θ for this particular case is defined as θ_0 .

From inspection of the unit cell geometry (figure 3), it is also possible to express the parameters α and β as functions of θ . From observing the triangle $A_1A_2B_2$, one can infer that $A_1B_2 = A_1A_2 \sin \alpha = a \sin \alpha$. Similarly, from triangle OA_1B_2 , it is possible to observe that $A_1B_2 = OB_2 \tan \frac{\theta}{2} = (A_2B_2 + OA_2) \tan \frac{\theta}{2}$. The segments A_1B_2 and OA_2 can be further expanded as:

$$A_2B_2 = A_1A_2 \cos \alpha = a \cos \alpha.$$

$$OA_2 = h - \Delta h - \frac{b}{2}.$$
(7)

Therefore, $A_1B_2 = \left(h - \Delta h - \frac{b}{2} + a\cos\alpha\right)\tan\frac{\theta}{2}$, and the following relation can be obtained:

$$\left(h - \Delta h - \frac{b}{2} + a \cos \alpha\right) \tan \frac{\theta}{2} = a \sin \alpha.$$
 (8)

Similarly, from triangle $A_3A_4B_1$, one obtains $A_4B_1 = A_3A_4 \sin \beta = a \sin \beta$. From triangle OA_4B_1 , one obtains $A_4B_1 = OB_1 \tan \frac{\theta}{2} = (OA_3 - A_3B_1) \tan \frac{\theta}{2}$. The segments A_3B_1 and OA_3 can also be expressed as:

$$A_{3}B_{1} = A_{3}A_{4}\cos\beta = a\cos\beta,$$

$$OA_{3} = h - \Delta h + \frac{b}{2}.$$
(9)

In this case, the segment A_4B_1 can be expanded as $A_4B_1 = \left(h - \Delta h + \frac{b}{2} - a\cos\beta\right)\tan\frac{\theta}{2}$. It is possible, therefore, to find the following recursive relation:

$$\left(h - \Delta h + \frac{b}{2} - a\cos\beta\right)\tan\frac{\theta}{2} = a\sin\beta.$$
(10)

The minimum values α_{\min} and β_{\min} are reached when the oblique cell ribs are tangential to the circular arcs of the tube:

$$\tan\frac{\beta_{\min}}{2} = \frac{\frac{b}{2} - \frac{c}{2} - \Delta h}{r_2}$$
(11)

$$\tan \frac{\alpha_{\min}}{2} = \frac{\frac{b}{2} - \frac{c}{2} + \Delta h}{r_{1}}.$$
 (12)

3.2. Work-energy model

Figure 5 shows a schematic diagram of a single-cell active honeycomb. The bottom cell wall is fixed, while the upper straight and oblique cell ribs are in constant contact with a rigid wall connected to a hinge at an angle of θ . The initial angle of the configuration is θ_1 . The length of the upper rigid wall is *L*. The distance between the center of the straight cell wall honeycomb and the hinge (point O) is *h*. The transverse width of the structure is *w*. The unit cell is subjected to



Figure 5. (a) Layout of the forces and pressures acting on the single unit cell. (b) Work of the forces and pressures present in the system. Quantities denoted in blue indicate the input energies, and quantities in black are related to the output energies.

loading by weight through a concentrated force located at the tip of the inclined rigid wall.

The model considers the adopted assumptions to simulate the actuation of the 'smart-stick' concept, which is inspired by the deformation mechanism of spider legs [36–38]. The whole system deforms under a rigid-type body motion produced by the inflation of the air inside the tubes (figure 5(a)). As a first approximation, the contribution provided by the restoring force of the tubes subjected to compression by an external load is neglected [36–38]. Therefore, the system receives as input work (W_{in}) the one generated by the inflation of the two tubes with air (W_P):

$$W_{in} = 2W_P. \tag{13}$$

The input work has to overcome the work done by the weight (W_G) , the one related to the deformation of the honeycomb (W_H) , the volume expansion of the tube (W_V) , and the work related to the tube's elastic expansion (W_E) . Therefore, the sum (W_{out}) of these energies is:

$$W_{out} = W_G + W_H + 2W_V + 2W_E.$$
(14)

Neglecting the influence of nonconservative forces (i.e., friction) in the system, by conservation of energy one can obtain:

$$W_{in} = W_{out}.$$
 (15)

The work of the pressured air is equal to the internal pressure in one tube $(P(\theta))$ multiplied by the contact area

between the tube and the wall, and the distance, $hd\theta/2$:

$$W_P = \int_{\theta_1}^{\theta} \frac{P(\theta)c(\theta)wh}{2} d\theta, \qquad (16)$$

where $c(\theta)$ can be obtained from equation (4).

The work done by the weight consists of two parts: the contribution from the loading weight (G), and the work done by the weight of the upper rigid wall (G_{wall}),

$$W_{G} = \int_{L\sin\theta_{1}}^{L\sin\theta} GdS + \int_{L\sin\theta_{1}/2}^{L\sin\theta/2} G_{wall}dS$$
$$= \left(G - \frac{G_{wall}}{2}\right) L\left(\sin\theta - \sin\theta_{1}\right).$$
(17)

Note that the contribution from the honeycomb cell wall is neglected because of the intrinsic low weight on the core. The contribution from the strain energy associated with the flexing of the honeycomb ribs (W_H) is also neglected because of the very weak in-plane stiffness induced by the low relative density of the core used for this design.

The angle of the system is θ_0 when the tube assumes a circular shape. For $\theta \leq \theta_0$, the work of the tube is equal to the work related to its volume expansion (W_V). The volume of the tube can be obtained from inspection as:

$$V(\theta) = \left[\left(\pi - \frac{\theta}{2} \right) r_1^2(\theta) + c(\theta) (r_2(\theta) + r_1(\theta)) + \left(\pi + \frac{\theta}{2} \right) r_2^2(\theta) \right] w.$$
(18)

Therefore, the work related to the volume change is:

$$W_{V} = \int_{\theta_{1}}^{\theta} P(\theta) \Delta V(\theta) d\theta$$
$$= \int_{\theta_{1}}^{\theta} P(\theta) (V(\theta + d\theta) - V(\theta)) d\theta.$$
(19)

If the pressure is kept constant, the work associated with the volume expansion depends only on the difference between the values of the initial and final volumes. Equation (19) can be therefore simplified into:

$$W_V = P(V(\theta) - V(\theta_1)).$$
⁽²⁰⁾

For $\theta > \theta_0$, the work on the tube is the work related to the elastic expansion of the tube itself. In that case, the tube will be dilated into a larger circle, and the corresponding mechanical work can be expressed as:

$$W_E = \int_{R_0}^{R} \frac{EA(R - R_0)}{R_0} 2\pi dR = \frac{\pi EA}{R_0} (R - R_0)^2.$$
 (21)

From inspection, $R = h \tan\left(\frac{\theta}{4}\right)$, and equation (21) can be therefore be expressed as:

$$W_E = \frac{\pi E A}{R_0} \left(h \tan\left(\frac{\theta}{4}\right) - R_0 \right)^2.$$
(22)



Figure 6. FE model of the system composed by the inflatable tubes and the auxetic unit cell. (a) Solid model. (b) View of the tubes when pressure is applied to the internal surface.

(b)

3.3. FE model

Figure 6(a) shows the solid model of the prototype structure, which was used to develop the FE mesh and simulation using ABAQUS 6.10. The lower (wooden) beam is fixed. The intersection line of the two beams is fixed in displacement and free in rotation, allowing the upper wooden beam to rotate freely around this line. Forces along the vertical direction are applied on the nodes of the right end of the upper beam to represent the loading weights. The honeycomb unit cell is fixed between the two beams. The compressed shape of the tube is calculated by using equations (4)–(6). The two tubes are in contact with the honeycomb walls and with each other, but no contact friction is considered in this analysis. A uniform pressure distribution is applied to the internal surface of the tube (figure 6(b)). The deformations of the honeycomb are obtained under different loading weights and input pressures. In this work, the tube is described by as having both LE and hyperelastic (Mooney-Rivlin) properties. Both the linear elastic isotropic material (Young's modulus 3 MPa [44], Poisson's ratio of 0.4) and the hyperelastic isotropic material $(C_{10} = 2.3 \text{ MPa}, C_{01} = 0.58 \text{ MPa} [45], \text{ and } D_1 = 0.001)$ are referred to the polyvinyl chloride (PVC) material of the tubes. Each tube was meshed using 9952 C3D20R elements (10 elements across the thickness of the tubes). The wooden plates (Young's modulus of 12 GPa, Poisson's ratio of 0.2) and the paper honeycomb (Young's modulus of 246 GPa, Poison's ratio of 0.10) were meshed using 2000 and 4976 C3D20R elements, respectively. The solver used consisted of a medial axis algorithm.

4. Displacement-pressure relation and experimental model

Figure 7 shows the unit cell active honeycomb demonstrator used for the experimental analysis. The tubes (in red) are made of PVC (Shenzhen Longxiang Electronic Co., LTD). The central radius of the tube is $R_0 = 2.5$ mm, with thickness



Figure 7. The frame and structure of the testing device.

equal to t = 1 mm. The tubes were glued between the honeycombs using 3M double-sided adhesive to avoid misalignment during the deformation. The negative-Poisson'sratio butterfly honeycombs (in orange) are made from paper glued on wood beams (in black) with $t_b = 0.3$ mm thickness, and the distance between the center of the tube and O is equal to h = 65 mm. The length of the upper rigid wall, OA, is L = 116 mm. The length of the honeycomb oblique rib is a = 8 mm, while the length of the straight rib is b = 32 mm. The two wood panels are joined by a steel hinge (in purple) with a width of w = 25 mm and a thickness equal to $t_m = 4$ mm a. The lower wood panel is fixed with a jaw vice (in blue). A Kevlar wire rope is used to suspend different weights (in green). The angle between the two wood panels is given by the following equation:

$$\tan \theta' = \frac{H_1 + \Delta H}{h}.$$
 (23)

During the actuation of the honeycomb, the input pressure, P', is kept constant at specific steps, and θ' is considered to be the final angle between the cell walls under the applied pressure, P'. The main mechanical sources of deformation in the system are considered to be the input air pressure and the recovery force of the tube. The primary output work is assumed to be generated from the load provided by the weight. Because the contribution provided by the tubes during the deformation may be considered small (i.e., both W_E and W_V are negligible), equation (16) can therefore be simplified as:

$$2\int_{\theta_1}^{\theta} P'c(\theta')whd\theta' = (G + G_{wall}/2)L(\sin\theta' - \sin\theta_1).$$
(24)

By differentiating both sides of equation (23), one obtains:

$$2whP'c(\theta') = (G_{\min} + G_{wall}/2)L\cos\theta'.$$
 (25)



Figure 8. ΔH -*P*' relations from the FE simulations using both the LE and hyperelastic tube models.

$$P' = \frac{\left(G + G_{wall}/2\right)L\cos\theta'}{2whc(\theta')}.$$
(26)

Equations (21) and (24) can be used to predict the input pressure for a needed vertical height, ΔH , on the honeycomb structure.

In the experimental case, the initial geometry parameters are $H_1 = 4 \text{ mm}$ and $\theta_1 = 3.52^\circ$. The distance, ΔH , is measured using a laser distance-measuring instrument (KEY-ENCE LK-GD500, accuracy: 0.01 mm) from a distance of 450 mm.

For this experiment, only one tube is used. The tube was observed to undergo some localized kinking, but no internal air blockage was apparent. One end of the tube was closed, and the opposite was connected to an air compressor pump. A pneumatic relief valve controlled the input pressure. The pressure was measured using an air pressure gauge (YB-150A, Hongqi Instrument Co. Ltd, range: 0–0.6 MPa, accuracy: 0.005 MPa). The distance, ΔH , is measured at increasing values of input pressure under different weights (600 g, 800 g, and 1000 g). The step for a single increase test is 20 kPa, and the step for the three cyclic tests is 40 kPa.

5. Results and discussions

Figure 8 shows the ΔH -P' relations predicted by FE using the LE and hyperelastic mechanical properties for the tubes, respectively. As expected, the higher the weight, the lower the maximum change in height ΔH . When the input pressure is larger than 0.1 MPa, it is apparent that the deformation increases linearly with the input pressure, and the deformation in the LE model is larger than the deformation obtained from the hyperelastic system. The LE model consistently provides a softer response than the hyperelastic model. The softening becomes significantly important when the input pressure is higher, and it reaches a 22% decrease in ΔH for pressure values of 0.28 MPa. This fact can be explained by observing that at the beginning of the pressure loading, the LE tubes first tend to expand horizontally (along the *h*-direction) without



Figure 9. Curves of p' versus ΔH for the experimental, FE hyperelastic material, and analytical models.

normal deformation, a feature that the hyperelastic tubes do not show. Therefore, the tubes defined by the linear elastic material tend to have a bigger deformation along the output displacement.

Figure 9 shows the general comparison of the ΔH -P' relations from the experimental, hyperelastic FE, and analytical models for three different weight configurations. It is apparent that the model tends to always be conservative against both the experimental output deformation and the FE deformation, especially when the input pressure is lower than 0.20 MPa. However, the analytical model tends to approach both the experimental and FE predictions at higher pressures. One can also see a very good agreement when the lowest weight loading condition (600 g) is applied. The FE model also tends to provide a softer response under 0.18 MPa, although it is stiffer than the one predicted by the analytical model. From observing figures 8 and 9, it is apparent that contact friction and stick-slip (neglected both in the analytical and FE models) play a significant role at lower levels of input pressure and external weight. Moreover, a precise modeling of both the geometry and the nonlinear geometric deformation of the tubes also appears to be an important aspect that the analytical model does not take into account at low P' and high G values. The analytical model is also limited by the rigidbody load transfer assumption typical of the smart-stick actuation unit. Another limit of the analytical model presented in equation (24) and represented in figure 9 is the lack of the resorting force from the compressed tubes. However, it is worth noting that in the present model and sample, only one inflatable unit was present, while the 'smart-stick' concept (and the related analytical model) tend to work efficiently when large numbers of smaller repeating units are connected together [36], similar to the configuration proposed in figure 1. High levels of input pressure show that stick-slip and contact friction are significantly less important, and the approximation used to describe the tubes in the analytical model (circular with linear isotropic material) is sufficient to provide an adequate comparison with the experimental results.



Figure 10. Displacement, ΔH , versus input pressure for different tube radii and loading weights.

The dependence of the pressure-displacement performance of the honeycomb actuator from the radius of the tube, R_0 , can be evaluated from the experimental tests shown in figure 10. All the curves show an initial marked increase of output displacement for small values of the input pressure, followed by a stiffening effect at larger values of p' due to a decrease of the produced ΔH . For a constant value of θ' , the contact length, $c(\theta')$, increases for higher values of R_0 (see equation (4)), and larger actuation forces could be obtained with larger values of $c(\theta')$ using the same input pressure, P'. As a result, the output, ΔH , increases with the augmentation of the radius of the tube, R_0 , by an average of 75% when the R_0 varies between 2.5 mm and 3.5 mm in the stiffening part of the displacement-pressure curve. For smaller input pressures, the sensitivity of the displacement variation, ΔH , versus the tube radius is even more significant, with a 2.75-fold increase of the displacement when the structure is subjected to a 1000 g loading.

Hysteresis effects are also apparent when the honeycomb cell is subjected to cyclic loading (figure 11). The curves corresponding to the unloading phase are at lower ΔH amplitude compared to the loading configuration, with a residual displacement left. It is also apparent that the first cycle provides the largest value of the residual displacements, while successive loading-unloading has a significantly smaller residual component, with almost all the slack in the system eliminated after the first cycle. The deflection of the honeycomb cell therefore depends not only on the input pressure, but also on the loading procedure used. In the system considered in this paper, the major component of the output work is used to oppose the deformation from the loading weight. Therefore, the energy absorbed can be considered to be related to the potential energy of the loading weight after a single sinusoidal cycle:

$$W_{An} = GL \left(\sin \theta_{1n} - \sin \theta_{1(n-1)} \right) = 1.2.3...$$
(27)

where *n* corresponds to the *n*th cycle. In the experimental configuration used in this work, $\theta_{10} = \theta_1 = 3.52^\circ$.

From equations (23) and (27) one obtains:

$$W_{An} = GL \left[\sin \left(ac \tan \left(\frac{\Delta H_{1n} + H_1}{h} \right) \right) - \sin \left(ac \tan \left(\frac{\Delta H_{1(n-1)} + H_1}{h} \right) \right) \right].$$
(28)

The values of the energy absorbed under different weights and calculated from equation (28) are listed in table 1. Note that the stored energy tends to decrease at each cycle. The configuration corresponding to the lowest weight (600 g) has a three-fold drop of energy absorbed between the first and third cyclic loadings, while for the highest weight, the energy lost tends to stabilize after the second cycle. The significant differences in terms of energy lost between the weight configurations is another indication that at lower mass loading, the system needs to be subjected to repeated cycles to oppose the slack in the layout and the stick-slip effects that occur between the contact surfaces of the inflatable tubes and the honeycomb walls. Another measure of the energy lost during the cyclic loading is the loss factor, η , defined as the ratio between the dissipated energy (ΔW) and the applied work (W) during the loading phase:

$$\eta = \frac{\Delta W}{W}.$$
(29)

Loss factors for each cyclic loading are also presented in table 1. Similar to the case of the energy dissipated, the loss factors in general tend to decrease between the first and third cyclic loadings. However, the loss during the cyclic loading is more uniform between the different weight configurations, with an average 50% drop between the first and third cycles. It is interesting to note that at constant cyclic loading, loss factors tend to differentiate very little between weights of 600 g and 800 g, and to increase significantly at the highest loading, up to 80% for the second (figure 11(d)) and third cycles.

The inflatable auxetic honeycomb concept relies on the use of a suitable elastomeric skin to follow the deformation behaviour of the cellular structure during morphing. From a design perspective, a suitable solution may be provided by matrix-dominated carbon-silicone elastomeric composites [19], which can provide up to 70%–80% tensile strain deformation. However, because of the auxetic butterfly honeycomb configuration with large negative Poisson's ratio values [46], an equivalent auxetic elastomeric skin may be used to guarantee the compatibility of the displacements between skin and core by Poisson's effect [47]. However, it must be noted that in this case, the maximum tensile strains may be reduced because of the high in-plane uniaxial/shear coupling existing in those skins. A common design issue in cellular structures for morphing wing sections is how to design the interface between the honeycomb ribs and the skins themselves. Although this topic goes beyond the scope of this work, we note that the problem has been partially solved in chiral wingbox configurations by using modified



Figure 11. The variation of ΔH versus the input pressure under three cyclic loadings for (a) 600 g, (b) 800 g, and (c) 1000 g. (d) A summary of the deformation, ΔH , for different weights corresponding to the second cyclic loading.

	600 g loading		800 g loading		1000 g loading	
Cyclic test #	W _{An} (mJ)	η(%)	W _{An} (mJ)	$\eta(\%)$	$W_{An}(mJ)$	η(%)
1	0.0209	10.83	0.0370	9.54	0.0620	19.39
2	0.0149	8.04	0.0135	7.84	0.0248	12.21
3	0.0074	5.51	0.0108	6.58	0.0023	9.96

Table 1. Energy dissipation during the cyclic loading.

circular-node layouts in the vicinity of the skin of the airfoil [48]. Not only is the cylindrical node typical of chiral auxetic configurations, but it has also been used to produce centersymmetric modified hexagonal honeycombs [49, 50] like the one used for the morphing concept proposed in this paper. Another aspect to be considered in the design of a realistic wing tip based on an inflatable honeycomb actuation is the choice of the fluid in the tubular structures. A hydraulics-based system would provide higher control displacement precision, but the weight penalty may be considerable in light of the potential aeroelastic problems stemming from the presence of the added mass from the hydraulic systems close to the tip of the wing. Pressurized air may be used instead, but that brings both a potential loss of control precision and added weight induced by the presence of heat exchangers to cool the spilled air from the aeroengine compressors. Nevertheless, the kinematics provided by the auxetic inflatable honeycomb for the morphing configuration is appealing because it offers the possibility of producing a continuous-profile wing variation without the creation of external bumps that cause unwanted aerodynamics consequences.

6. Conclusion

This paper has introduced the concept of a novel active honeycomb configuration based on inflatable tubes and an auxetic centresymmetric cellular topology. A phenomenological model from the work-energy equation has been developed to obtain an input pressure/output displacement relation for the actuation system. An experimental model has also been developed, and the results have been compared with the predictions from the analytical model and FE simulations representing the tube/auxetic cell prototype. An analysis of the hysteresis due to cyclic loading on the active honeycomb configuration has also been performed. The sample has shown the feasibility of the concept for possible wingtip morphing configurations, and the promise of using the auxetic active honeycomb for further designs.

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