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Highly orthotropic panels structural stability, Farrar and Bloch waves theory

Farrar and
Bloch waves
theory

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Abstract

Purpose – The purpose of this paper is to present the studies to develop structural highly orthotropic panels. These types of panels provide the capabilities required for structural morphing. They are highly deformable in one given direction; and in the perpendicular direction, one must be compliant to sustain the internal loads per certification stiffness and strength requirements.

Design/methodology/approach – Neither classic orthotropic panels nor regular cellular solids Bloch wave theories are strictly applicable on their study. A combination of computational method, MATLAB-coded, to analyze stability and classic structural beam theory is studied. Then, non-linear finite element method models are developed for an aerospace control surface application; their results are compared with reported Bloch wave sequences on periodic cellular solid panels.

Findings – The stability along stiffer direction is a requirement to obtain a continuous deformation and plasticization sequence of the cell rows in the perpendicular direction. A sample panel is sized and 3D modeled, and then produced using additive layer manufacturing process to demonstrate the initial stages of a validation and verification campaign.

Originality/value – This paper provides a new method to mechanical characterize highly orthotropic panels.

Keywords Buckling, Additive manufacturing, Bloch waves, Highly orthotropic panels, Morphing, Structures, Farrar theory

Paper type Research paper

1. Introduction

Structural morphing is an active research field that is intended to develop more efficient solutions. The available literature contains good examples of potential applications for morphing structures into wind turbine blade applications (Lachenal *et al.*, 2013). In the case of aircraft structures, there have been published studies about their application in high-lift wing devices (Thill *et al.*, 2008) and wing tips (Cheung *et al.*, 2014). In particular, previous studies have proposed the application of these types of morphing solutions as novel rudders that morph during deployment (Castillo-Acero, Cuerno-Rejado and Gomez-Tierno, 2014; Castillo-Acero, Gomez-Tierno and Cuerno-Rejado, 2014). Such novel rudders are the subject of this paper. Recently, the European Union (EU) FP7 Clean Sky program has made some of the research on structural morphing that has been undertaken public. The design of high-lift devices that can curve during their deployment is feasible using current state-of-the-art structures and commercially available actuators (Pecora *et al.*, 2014). The panel skins of such devices present different technical challenges for their development. At the 2014 ILA Berlin Air Show, the partners in the EU FP7 SARISTU “Smart Intelligent Aircraft Structures” project made public a solution based on a bi-material type of panel – the result of combining rigid aluminum stiffeners with elastomeric members – that can be curved while maintaining adequate mechanical properties.

The studies presented in this paper are the results of research concerning the definition of a different type of structural panel. These panels are based on highly orthotropic

Figure 1.
Different panels' behavior depending on cell configuration

Figure 2.
Cellular panels

repeating cells whose flexibility, chord-wise, is tuned as required for a device's kinematic deployment. At the same time, the panels' span-wise directions offer adequate stiffness and strength to sustain aerodynamic loads. These panels are a particular case from the more generic family of cellular solids (Gibson and Ashby, 1997). Figure 1 depicts the different Poisson macro behaviors of panels depending on the configuration of their cells. These behaviors range from standard effects in panels to zero-Poisson phenomena which have the possibility of developing auxetic responses.

Regular cellular solids, constituted by infinite periodic cells, are traditionally studied for stability under 2D compression forces (Wilcox, 1978). The particularizations of different types of cells can be found in different publications. The cases of isotropic circular and hexagonal lattices have been considered in numerous reference works and have been analyzed under different types of loading (Okumura *et al.*, 2002). Generally, it has been concluded after testing (Papka and Kyriakides, 1999) that, under uniaxial compression loading, these types of cellular panels always follow the same sequences of events (Gibson and Ashby, 1997). Figure 2 depicts (Papka and Kyriakides, 1999) this reported panel stress-strain response for two different cellular panels: there is an initial nearly linear response (Point 1), then deformation concentrates in a single cell row that starts to plasticize as a local mode that involves (Point 2), and this cell row is perpendicular to the compression force direction. This row shear collapses with an unsymmetrical failure mode and stays deformed. Then the next row experiences the same failure mode mechanism and that spreads from row to row, following the sequence in Points 4-7. It is relevant that the behavior is very similar for two types of cell materials with different stress-strain curves and mechanical properties, different Young moduli, and different yield and rupture stresses. This cellular panel macro behavior is positive and stores deformation energy, as occurs in aircraft tail bumpers. The use of this macro behavior in other airworthy aircraft structures, that require no yield up to limit load and then no rupture to

Notes: Circular left-hexagonal right, stress strain curves, compression uniaxial loading
Source: Papka and Kyriakides (1999)

ultimate load, requires that there be no permanent deformation in the first row, due mainly to shear-induced instability, as will be summarized later in this paper.

The buckling behavior of orthotropic plates has also been well-reported in numerous publications in the extant literature (Johns, 1971; Lekhnitskii, 1987; Kim *et al.*, 2009), but there have been no previous studies on highly orthotropic plates. The present work summarizes a new set of studies that have combined classical beam theory in mechanical engineering, with compression buckling stability based on Farrar (1953). The initial results appear to form a basis for simulating highly orthotropic panels' behavior under loading with non-linear finite element method (FEM) models. This new methodology intends to fill the gap in the analysis of highly orthotropic panels based on repeating basic unitary cell n -times, as depicted in Figure 3.

The set of parameters – pitch, distances, thickness, etc. – are defined to provide panel stability. The stability must be achieved to deal with buckling under compression loading as an initial requirement. This study, in the present case, is based on MATLAB-coded Farrar theory. Fulfilling this, requirement is of utmost importance to enable the design of morphing directions that accommodate any induced deformations without reaching failure stresses.

The close relationship between the load compression behavior of a panel and its geometric structure has been studied for flexible foams (Rusch, 1969). For highly orthotropic panels, the geometry definition of the cells has been studied to cope with an initial set of stiffness requirements. The selection of adequate geometrical values to achieve the required strength-stiffness and avoid permanent deformation has been studied in the context of classic continuous beam theory. An application for a commercial transport aircraft control surface was modeled cinematically in CATIA V5 to obtain the internal loads and stiffness required. Highly non-linear Abaqus FEM models have been developed to study panel behavior. Abaqus is an implicit finite element solver that has been reported to provide accurate results for these types of studies (Kumar and Gurvich, 2010; Geisler *et al.*, 2010; Asprone *et al.*, 2013). A representative panel specimen has been manufactured with a novel, commercially available 3D printing system to initiate tests-based validation and verification phases.

2. Methodology

The material considered for the panels was selected based on the commercial availability for final additive layer manufacturing or 3D printing. The raw material considered for designing and manufacturing the panels in this study was VisiJet M3-X. This material can be processed easily by a standard 3D printing device (Table I).

The highly orthotropic panels can present different configurations, depending on the cellular selection to be repeated. All the cases under study presented common features, i.e. springs-like type members in the deformable direction, and stiffeners in the perpendicular one, as is depicted in Figure 4.

Figure 3.
Zero-Poisson panel
and basic cell
geometry parameters

In all the studied cases, the stiffness properties were obtained; in-plane orthotropic $E_1, E_2, G_{12}, \nu_{12}, \nu_{21}$; out-of-plane $E_3, G_{13}, G_{23}, \nu_{13}, \nu_{31}, \nu_{23}, \nu_{32}$. The configuration named “V-Plane” has been characterized already (Barbarino *et al.*, 2011). This study incorporated the novel “Double-C” configuration that, while keeping the basic common geometric features as the V-Plane, showed performance improvements in relation to the “V-Plane” variant.

We considered an equivalent constant thickness “ t ” orthotropic panel with the same dimensions, width and length as the panel under study. The equivalent panel incorporated the stiffeners as well. The original highly orthotropic panel stiffeners had to provide adequate support to the springs in the perpendicular direction. This is a condition for any kind of loading and panel deformation state. Therefore, in order to assure panel stability, the stiffeners thickness “ t_s ,” height “ d ,” and the pitch between them “ b ” had to be able to provide stability to the overall panel. The requirement for stiffener stability was a key finding of this study. The critical failure mode for these panels was buckling under compression loading. The applied load flows were set to cope with an application objective. The Farrar (1953) theory was selected to provide allowable compression buckling stresses. It was programmed in MATLAB. This code took the internal compression loads and the geometry constraints as inputs, and performed calculations that yielded an optimum set of geometrical parameters for the structure. Farrar’s theory states that the optimum structure is one in which the compression stresses at the beginning of the initial buckling, σ_B , match the corresponding Euler buckling as a column, σ_E . In this way when the local buckling was induced, then the global buckling of the entire panel started.

The Farrar theory also defines a reference stress, σ_{ref} , and a mean stress, σ_m , because the reinforced skin stress distribution is not uniform. The Farrar factor, F , is defined as the ratio between the mean and the reference stress, concluding that the higher the value of F , the more optimum the structure results will be in terms of weight. “ N ” is the applied compression flow in the panel. Then the condition of simultaneous local and global buckling is imposed, and both must be equal to average stress: $\sigma_m = \sigma_B = \sigma_E$. Then F is obtained

Table I.
Visijet M3×3D
printable material
properties

Properties	Condition	Visijet M3-X
Density @ 80°C (liquid)	ASTM D4164	1.04 g/cm ³
Tensile strength	ASTM D638	49 MPa
Tensile modulus	ASTM D638	1,340 MPa
Elongation @ break	ASTM D638	3.3%
Yield strength	ASTM D638	20 MPa
Poisson coefficient	ASTM D638	0.3

Source: 3D Systems

Figure 4.
Highly orthotropic
panels’ configurations
studied

solely as a function on the geometry parameters, specifically the parameters t_s (stringers thickness), t (equivalent skin thickness), b (distance between stiffeners), d (stiffeners' height). The difficulty of applying this theory to a real case is found in the combination of equations and graphs. For this study, the graphs, for each Farrar factor F -level, were MATLAB-modelled (Figure 6). So, starting with the known values of $R_b = d/b$ and $R_t = t/t_s$, and the graph $F = f(R_b, R_t)$, the values of all the other geometric parameters as functions of the panel length, L_{eq} , were the following:

$$\sigma_B = \frac{\sigma_B}{\sigma_0} f_1 \left(\frac{A_s}{bt}, \frac{t_s}{t} \right) 3.6E \left(\frac{t}{b} \right)^2 \quad (1)$$

$$\sigma_E = \frac{\pi^2 E}{2L_{eq}} \left[\frac{\rho}{b} f_2 \left(\frac{A_s}{bt}, \frac{t_s}{t} \right) \right]^2 b^2 \quad (2)$$

where A_s is the transversal stringers area, height “ d ” and thickness “ t_s ”:

$$A_s = d \cdot t_s \quad (3)$$

where L_{eq} is the length of the skins and stringers, for the present case application purpose, pitch between ribs. It is assumed that ribs provide a stable support; E is the equivalent modulus along panel stiffer direction; σ_0 is the compression skin buckling as a plate:

$$\sigma_0 = 3.6E \left(\frac{t}{b} \right)^2 \quad (4)$$

$$t = \frac{1}{F(1+R_b R_t)} \left(\frac{N \cdot L_{eq}}{E} \right)^{1/2} \quad (5)$$

$$b = 1.103 \frac{\sqrt{F(1+R_b R_t)}}{\left[R_b^3 R_t (4+R_b R_t) \right]^{1/2}} \left(\frac{N \cdot L_{eq}}{E} \right)^{1/2} \quad (6)$$

$$F^4 = \frac{\sigma_B}{\sigma_0} \left(\frac{A_s}{bt}, \frac{t_s}{t} \right) 3.6\pi^2 \left(\left[\frac{\rho}{b} \left(\frac{A_s}{bt}, \frac{t_s}{t} \right) \right] \right)^2 \frac{1}{\left(1 + \frac{A_s}{bt} \right)} \quad (7)$$

Figures 5-12 depict the Farrar plots, obtained using MATLAB, for the different geometry parameters: t , d , b and t_s , and for two different panel lengths.

3. Unit cell analysis

Once the stability of the longitudinal stiffeners was assured, the selection of the cell was critical for defining the spring-like members of the highly orthotropic panel configuration properly. The previous stability study provided an equivalent stiffened orthotropic panel that fitted the strength requirement in one direction. The continuous beam analysis provided the geometry required on the deformable direction to reach the design goals in terms of adequate stiffness for large deformations.

We let E_0 be the modulus of elasticity of the basic material selected for the panel. The derivation of the panel elasticity modulus in the three main directions of the singular cell could be expressed as a function of E_0 . These panel elasticity modules were the result of

Figure 5.
MATLAB Farrar
plots, t -values
for $L = 200$ mm

Figure 6.
MATLAB Farrar
plots, t -values for
 $L = 100$ mm

making an analogy of the actual panel under study with a solid one of constant thickness, exhibiting anisotropy behavior in the three main directions. It was observed that the flexible direction elastic modulus was E_1 in the perpendicular direction within the plane of the panel E_2 , and that E_3 pertained to the stiffness of the panel in the perpendicular direction.

The selection of these Young modulus target values depended on the application of the panel. The panel had to follow the extension kinematics of morphing chord-wise, and the stiffness and strength required span-wise. This requirement was translated into a target value of two orders of magnitude of difference between E_1 and the E_2, E_3 pairing. Then, a general target value of 10 percent E_0 for these two last modules was selected:

$$E_2 \cong 0,1E_0; E_3 \cong 0,1E_0$$

Supposing an applied force, P , in the second direction, the deformation of the panel in the direction of this applied force could be expressed as a function of the width of the panel, b ,

Figure 7.
MATLAB
Farrar plots, d for
 $L = 200$ mm

Figure 8.
MATLAB Farrar
plots, d for
 $L = 100$ mm

Figure 9.
MATLAB Farrar
plots, b for
 $L = 200$ mm

Figure 10.
MATLAB Farrar
plots, b for
 $L = 100$ mm

Figure 11.
MATLAB Farrar
plots, t_s for
 $L = 200$ mm

the equivalent thickness, t_{eq} , and the equivalent elastic modulus, E_2 . However, the deformation could also be expressed in terms of the thickness, t , height, and d :

$$\varepsilon_2 = \frac{P}{ct_{eq}E_2} \quad (8)$$

$$\varepsilon_2 = \frac{P}{tdE_0} \quad (9)$$

If the problem was simplified, assuming that the equivalent thickness was the same as the basic stiffener, and by extension all the cell thicknesses were constant, it could be

Figure 12.
MATLAB Farrar
plots, t_s for
 $L = 100$ mm

concluded that:

$$E_2 = \frac{d}{c} E_0 \quad (10)$$

Imposing the target Young modulus condition, then:

$$d \cong 0, 1c \quad (11)$$

Similar reasoning for the derivation of E_3 in relation to E_0 could be followed. In this case, the other in-plane panel dimension was named c :

$$\varepsilon_3 = \frac{P}{bcE_3} \quad (12)$$

$$\varepsilon_3 = \frac{P}{t(b+c)E_0} \quad (13)$$

$$E_3 = \frac{t(b+c)}{bc} E_0 \quad (14)$$

Then, another geometrical constraint for the panel definition was concluded:

$$b \cong 20t \quad (15)$$

The expression of the latest equivalent elastic modulus, along the most flexible main direction, required using continuous beam theory, symmetries and simplifications, in particular the Euler-Bernoulli theory had to be valid, and deformations had to be low enough to assume that sections perpendicular to the deformation line would stay perpendicular after loading. It was assumed that there was a constant cell thickness that was at least one order of magnitude lower than the typical cell dimension. The first basic cell considered was composed by straight beams. The study then focused on analyzing the segment AB (Figure 13).

The point B displacements and rotation could be expressed as a function of the point A ones:

$$u_B = u_A + \theta_A(y_B - y_A) - \int_A^B \frac{M_f(s)}{E_0 I_z} (y_B - y) ds \quad (16)$$

$$v_B = v_A + \theta_A(x_B - x_A) + \int_A^B \frac{M_f(s)}{E_0 I_z} (x_B - x) ds \quad (17)$$

$$\theta_B = \theta_A + \int_A^B \frac{M_f(s)}{E_0 I_z} ds \quad (18)$$

Point A was in the symmetry plane and horizontal displacement and rotation were zero:

$$u_B = \frac{1}{E_0 I_z} \int_0^l (M_A + P s \cos \theta_0)(s - l) \sin \theta_0 ds \quad (19)$$

$$\theta_B = \frac{-1}{E_0 I_z} \int_0^l (M_A + P s \cos \theta_0) ds \quad (20)$$

The displacement of point B in the y -direction was restricted to:

$$0 = v_A + \frac{1}{E_0 I_z} \int_0^l (M_A + P s \cos \theta_0)(l - s) \cos \theta_0 ds \quad (21)$$

We let P be the applied force along the spring extension. The problem was reduced to obtain the value of the four unknown internal loads at A and B points, M_A , H_A , M_B and V_B . There were three equilibrium equations; therefore, the hyper-statics degree of the problem was one:

$$H_A = -P \quad (22)$$

$$V_B = 0 \quad (23)$$

$$M_B = -M_A - Pl \sin \theta_0 \quad (24)$$

The energy equation provided a new equation to solve the hyper-statics problem. The deformation energy was expressed as a function of the bending moment at the extreme

Figure 13.
Basic cell sketch and
internal loading

A point, M_A , and then a minimum energy state was imposed to obtain its value:

$$E = P \cdot u_B + M_B \cdot \theta_B \quad (25)$$

$$\frac{dE}{dM_A} = 0 \rightarrow -\sin \theta_0 \frac{l^2}{2} + 2M_A \frac{l}{P} + l^2 \sin \theta_0 + \cos \theta_0 \frac{l^2}{2} = 0 \quad (26)$$

$$M_A = \frac{-Pl}{4}(\sin \theta_0 + \cos \theta_0); \quad M_B = \frac{Pl}{4}(\cos \theta_0 - 3 \sin \theta_0) \quad (27)$$

The deformation at the extreme B point, and the springs' stiffness could be expressed as:

$$\varepsilon_B = \frac{Pl^2}{24E_0I_z} \sin \theta_0 \{3 \tan \theta_0 - 1\} \quad (28)$$

$$k = \frac{P}{u_B} = \frac{24E_0I_z}{l^3 \sin \theta_0 \{3 \sin \theta_0 - \cos \theta_0\}} \quad (29)$$

This expression could then be incorporated into a minimum panel weight to sustain a critical load, P_{cr} :

$$W = W(b, c, d, t, \theta_0) = \rho \left(\frac{w}{b} + 1 \right) \frac{12c}{kE_0t^2} \left(\frac{2P_{cr}}{\pi} \right)^2 \quad (30)$$

The equivalent elastic modulus in "1" direction, could be expressed as:

$$E_1 = E_0 \frac{2td^2}{bl^2} \frac{1}{\sin \theta_0 \{3 \tan \theta_0 - 1\}} \quad (31)$$

For the purpose of establishing an order of magnitude, and θ_0 geometries between 15° and 80° , then: $t \cong d \cong 0$, $1l \cong 0$, $1b \rightarrow E_1 \cong 10^{-3}E_0$. The panel could be designed to present a relative stiffness in one direction, two orders of magnitude lower than in the other two main directions.

A second-cell geometry, based on a circular central segment (Figure 14) has also been considered for geometrical definition.

This hyper-static structure was solved by particularizing the more general Rayleigh-Ritz method (Courant and Hilbert, 1989). The curved beam segment bending moment was expressed as a function of the hyperstatic unknown, M_B , after which zero rotation was imposed at point A:

$$\frac{E_1}{E_0} = \frac{1}{3} \frac{t^3}{cR^2} \frac{\sin \theta_0}{(2\theta_0 + \sin 2\theta_0 - \sin^2 \theta_0 / \theta_0)} \quad (32)$$

As: $t \cong 0$, $1c \cong 0$, $1R$ then $E_1 \cong 10^{-3}E_0$ then. The same elastic modulus relation was achieved.

The last configuration was of a mixed straight and curved segment (Figure 15).

Figure 14.
Curved cell study

Figure 15.
Mixed cell study

Following the same approach:

$$\frac{E_1}{E_0} = \frac{t^3(R\theta_0 + l)(l \cos \theta_0 + R \sin \theta_0)}{c(a_1 R^3 l + a_2 R^2 l^2 + a_3 R l^3 + a_4 l^4 + a_5 R^4)} \quad (33)$$

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theory

The constants in this expression were:

$$a_1 = 30 \cos \theta_0 + 6\theta_0 - 12\theta_0 \cos^2 \theta_0 \quad (34)$$

$$a_2 = 12(1 + \theta_0 \sin \theta_0 \cos \theta_0 - \cos^2 \theta_0) \quad (35)$$

$$a_3 = 3(\theta_0^3 - \theta_0 \cos \theta_0) \quad (36)$$

$$a_4 = \cos^2 \theta_0 \quad (37)$$

$$a_5 = (\theta_0 \sin \theta_0 \cos \theta_0 + \theta_0^2 - 12 \cos^2 \theta_0) \quad (38)$$

$$a_4 = \cos^2 \theta_0 \quad (4.30).$$

As: $t \cong 0$, $l \cong 0$, $1R$ then $E_1 \cong 10^{-3} E_0$ then as in the previous cells' cases.

4. Structural application, design, FEM models and manufacturing

The structural feasibility of this design has been verified by integrating the morphing skin structure in solid CATIA V5 models for kinematic mechanical analysis and simulation on a novel rudder structure for a commercial transport aircraft (Castillo-Acero et al., 2014). Figure 16 depicts some of the geometrical details of these 3D models.

The stiffness and loading for this application have been obtained with kinematics analysis based on CATIA V5 models (Figure 17).

This structure was FEM-modeled using Abaqus to obtain the internal stresses distributions (Figure 18), just for deflection-induced external loading. The plasticized zone was concentrated mainly in the longitudinal stiffeners (red areas). This stress concentration was coupled with the longitudinal internal forces when the structure was also loaded with aerodynamic and inertial loads. This explains the importance of stability on the stiffeners. The stress concentrations on the stiffeners were mainly shear-induced and, when combined with compression stresses, overall panel instability appeared due to the stiffeners' out-of-plane displacement, which collapsed the structure.

The maximum von Mises stresses and the logarithmic strains in the highly deformable panels have been extracted from different Abaqus executions for the control surface deflected between five and 30 degrees. Table II includes the values.

There was an initial linear observation of panel stresses with lower deflections, up to 10° (Figure 19). Then a plastic behavior between 10° and 15° was obtained, before a second linear answer between 15° and 30° appeared. It was noticed that the material plasticity appeared between 10° and 12° deflection. It resulted in an analogy between predictions for this FEM model on the resultant panel stress-strain curves in relation to those reported for cellular solid panels' Bloch wave sequences, but with some remarkable differences. There was an initial linear response, then first material plasticization

Figure 16.
Morphing rudder
applications of
the highly
orthotropic panels

Figure 17.
Control surface
kinematics studies,
displacements
at 30° deflection

Figure 18.
Von Mises
stresses at highly
deformable panels
at 30° deflection

Deflection	5°	10°	15°	20°	25°	30°
<i>Upper panel</i>						
$\sigma_{VM_{max}}$	10.3	19.8	19.9	20.9	22.2	23.5
LE(10 ⁻³)	8.1	16.5	26.5	36.5	46.1	55.0
<i>Lower panel</i>						
$\sigma_{VM_{max}}$	10.5	19.8	19.9	20.5	21.9	23.3
LE(10 ⁻³)	7.9	16.4	28.1	38.6	48.7	59.0

Table II.
Maximum Von Mises
stresses and
logarithmic strains in
the highly deformable
panels for the
control surface
deflected between
5 and 30 degrees

appeared in one row of the panel, and a new elastic response with a more relaxed secondary Young modulus appeared. Therefore, a design principle based on reinforcing the initial row that plasticizes, to increase the principal Young modulus answer and to delay the appearance of the secondary elastic zone could be concluded as being adequate. Such would be needed to ensure that no rupture stress occurred in the plasticized area.

An optimization in terms of thickness and height for the rest of the cells' rows would provide a continuation of these studies for designs predicated on minimum material weight performance in the future. The generation of a similar Bloch wave plasticization sequence,

Figure 19.
Maximum Von Mises
stresses at highly
deformable panels

on repeated circular and hexagonal cells solid panels, for example, could provide a minimum material and weight panel configuration, while still fulfilling the requirements for maintaining panels' integrity.

The 3D printing "Double-C" representative panel in the selected material VisiJet M3-X was performed, using dimensions of $230 \times 230 \times 30$ mm and a typical wall thickness of 2 mm (Figure 20).

The 3D printer used was a ProJet® 7000 MP whose operation is based on stereo-lithography technology. In total, 210 grams of material was used. The time consumed was around seven hours. The time to produce the panel was proportional to the amount of material used. Even if 3D printing technologies in the future were to enable a better cycle performance, and the weight was not critical for any particular application, the need to minimize the amount of material would keep on being of importance, from a serial manufacturing process productivity point of view.

Currently panel specimens are ready for campaign testing to complete validation. A verification campaign will allow the comparison of the empirical results with the analytical ones.

5. Conclusions

The approach described above is valid for designing a highly orthotropic panel, based on the repetition of a modular geometrical cell, to cope with two orders of magnitude stiffness ratio requirements, in the two in-plane main directions, and in terms of strength requirements. Setting the stiffeners' configuration for stability is required for designing the panel and subsequently the unit cells. This method has served to size and produce highly deformable panels that configure part of a morphing rudder structure for a commercial transport aircraft that is compliant with its strength and stiffness requirements. Numerical analysis based on non-linear FEM models, predicts a similar Bloch wave behavior to that observed in regular cellular solids, constituted by infinite periodic cells. The panels tested here developed a secondary elasticity response. Further studies are required to optimize deformable direction cells' geometry, from the viewpoints of weight and 3D manufacturing process productivity. A testing campaign is still required to initiate validation and verification stages. These tests should include static and fatigue loading, and also examine the environmental effects on mechanical properties degradation. Further studies will be required to identify new materials that are more suitable for the mechanical requirements of optimized panel configurations.

Figure 20.
3D-printed panel
specimen

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