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Free vibration of actual aircraft and spacecraft hexagonal honeycomb sandwich panels: A practical detailed FE approach

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Abstract. This work presents a practical *detailed* finite element (FE) approach for the three-dimensional (3D) free-vibration analysis of *actual* aircraft and spacecraft-type *lightweight* and *thin* honeycomb sandwich panels. It consists of calling successively in MATLAB®, via a developed user-friendly GUI, a detailed 3D *meshing tool*, a macro-commands language *translator* and a commercial FE *solver* (ABAQUS® or ANSYS®). In contrary to the common practice of meshing finely the faces and core cells, the proposed meshing tool represents each *wall* of the actual hexagonal core cells as a *single* two-dimensional (2D) 4 nodes *quadrangular* shell element or *two* 3 nodes *triangular* ones, while the faces meshes are obtained simply using the nodes at the core-faces interfaces. Moreover, as the same 2D FE interpolation type is used for meshing the core and faces, this leads to an *automatic* handling of their required FE *compatibility* relations. This proposed approach is applied to a sample made of *very thin* glass fiber reinforced polymer *woven* composite faces and a *thin* aluminum alloy *hexagonal* honeycomb core. The unknown or incomplete geometric and materials properties are first collected through direct measurements, reverse engineering techniques and experimental-FE modal analysis-based inverse identification. Then, the free-vibrations of the actual honeycomb sandwich panel are analyzed experimentally under different boundary conditions and numerically using different mesh basic cell shapes. It is found that this approach is accurate for the first few modes used for pre-design purpose.

Keywords: free vibration; composite; hexagonal honeycomb; sandwich panel; detailed FE model

1. Introduction

Sandwich structures can be seen as a sub-set of layered composites (Birman and Kardomateas 2018). They have lower lateral deformation, higher buckling resistance and higher natural frequencies than others (Vinson 2001). Besides, they are known to have high thermal resistance, high strength-to-weight ratio, good energy and sound absorption, often low cost manufacturing and to represent a compromise between stiffness and lightness (Mackerle 2002). Therefore, sandwich constructions are popular for alleviating the weight, noise or vibration of many industrial structures such as in aerospace, aeronautics, automotive, rail, civil, marine, electronics and packaging. Their cores can be metallic (aluminum, steel, titanium) or in paper while their faces can be metallic, composite (graphite, carbon, glass) or in paper. In particular, for modern aircraft and

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spacecraft structures, the former are often in aluminum or NOMEX®, while the latter are in graphite-, carbon- or glass-epoxy multilayered or woven composites (Birman and Kardomateas 2018). Besides, these honeycomb sandwich panels are seen to be very suitable for embedding additional non-structural (thermal, electric, power, etc…) multi-functions (Sairajan *et al*. 2016).

The modeling approaches used for simulating honeycomb sandwich structures can be grouped in three categories (Noor *et al*. 1996): (i) *detailed* finite element (FE) three-dimensional (3D) models, for which the actual geometry of the faces and honeycomb core are represented using two-dimensional (2D) shell or/and solid elements; (ii) 3D or quasi-3D *continuum* models, for which the faces and core (or each layer of them) are replaced by an *equivalent homogeneous orthotropic continuum*; (iii) *plate/shell* 2D models, for which first (FSDT)- or/and third (TSDT)-order shear deformation (or classical plate-CPT) theories are used. In this case, the kinematics approximations are either *global* (equivalent single layer-ESL) or *discrete* (layer-wise-LW). In comparison to continuum and plate/shell models, detailed FE ones were rarely investigated until the 1990s; indeed, only one journal article was cited in the above review (Noor *et al*. 1996) of 800 main references and 559 supplementary ones. This may be due to the fact that the computational effort associated with them increases rapidly with increasing the core cells number (Burton and Noor 1997). However, it is thought here that this is rather due to meshing finely each core cells wall. This is here avoided through a *simplified* detailed FE approach, thus contributing to reduce this major disadvantage. On the other hand, it is well known that static and free-vibration responses of a range of honeycomb sandwich panels are highly sensitive to the equivalent continuum models transverse shear stiffness variations (Burton and Noor 1997). Besides, the latter assessment showed that detailed FE models, taken as standard of comparison, did not confirm either of the underlying constant strain/stress hypotheses used, respectively, in the upper/lower bound energy estimation approach of equivalent continuum models. In the opposite of detailed FE models, 2D plate/shell ones are the most used for the analysis of laminated composite and sandwich structures (Sayyad and Ghugal 2015), but rarely of honeycomb ones. Indeed, only one reference among 391 was cited about honeycomb sandwiches. Besides, analytical and experimental comparisons of CPT (Kirchhoff), FSDT (Mindlin) and TSDT (Reddy) showed that neither of these theories is adequate for the flexural free vibration analysis of aluminum honeycomb panels (Yu and Cleghorn 2005, Yongqiang and Zhiqiang 2008, Yongqiang and Dawei 2009).

Comparatively to other types of analyses, free vibration of honeycomb sandwich panels attracted less attention in the open literature (Yu and Cleghorn 2005), although related early analytical formulas (Ueng 1966, Bert 1967), experiments (Raville and Ueng 1967) and continuum FE models (Ahmed 1971) go back, respectively, to the mid-1960s and early 1970s. However, detailed FE 3D models were proposed firstly by Chamis *et al*. (1988) much later, by the late 1980s, but for simulating thermo-mechanical/structural behaviors of fiber composite sandwich structures, for which the above discussed three types of models have been addressed. Those for the free-vibration analysis appeared in the open literature only from early 2000s on (Liu and Zhao 2002). In the latter reference, the aluminum alloy honeycomb core was either assumed quasi-orthotropic or its cell walls meshed with plate elements to reflect the geometric nature of the hexagonal cells. Noticeably, the mass of the adhesive was added to that of the aluminum alloy faces, while the modulus and thickness of the faces were not adjusted, meaning that the additional stiffness was neglected. Besides, it was found that the detailed FE model predictions of the mode shapes were closer to the experimental ones than those from the continuum FE model. Moreover, in contrary to the latter, the former had the capability of predicting the local mode shapes.

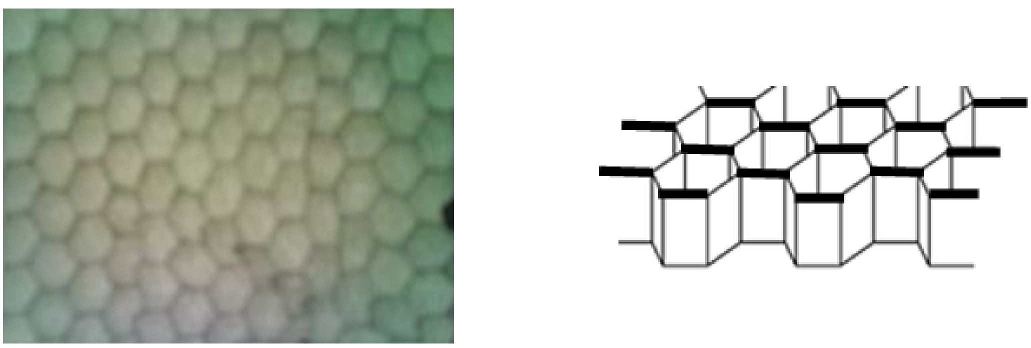
Detailed FE 3D models for the free vibration analysis of aircraft and spacecraft hexagonal honeycomb sandwich panels found wider interests from early 2010s on (Boudjemai *et al*. 2012). Indeed, in the latter, the first three natural frequencies of an aluminum faces-titanium honeycomb core sandwich cantilever plate were obtained using classical experimental modal analysis (with hammer exciter and accelerometer sensor), continuum and detailed FE, by PATRAN/NASTRAN®, models. For the latter, the QUAD4 element-based FE mesh of the skins and core were made separately then assembled, but no details on the latter issue were provided. Compared to experimental values, the latter model's errors were, respectively, around 4% and 10% for the first two and third frequencies. This was attributed to the huge effect of the clamping system on the measured values. Later, Li *et al*. (2013) proposed a special approach for which the sandwich plate composite laminated faces were discretized with 4-noded quadrilateral elements and LW theory, while the aluminum core was discretized with 8-noded solid elements within full (detailed), equivalent (continuum) or local (a combination of previous ones) models. Noticeably is that the degrees of freedom (DOF) of the LW theory are the same as the brick elements and the displacements variables at the upper and lower surfaces of the faces appear in the governing equations so that the faces-core interfaces compatibility relations were used for their assembly while ensuring the displacements continuity. It was found that higher natural frequencies and mode shapes, obtained by the local and continuum models, are inaccurate compared with those of the detailed one. Besides, it was concluded that, compared to the detailed model, the errors of the local and continuum ones result mainly from the equivalent material properties of the honeycomb. Recently, continuum and detailed FE, with ANSYS®, 3D models were also compared by Sakar and Bolat (2015), but against the only measured *first* mode of the free vibration of an aluminum honeycomb sandwich cantilever *beam*. Here, in the detailed FE 3D model, the faces and core were meshed by 8-noded shell elements. Noticeably, in the analyses, the beam length was determined according to the cells number. Both models provided increasing error with increasing the core thickness; in particular, the error was around 2.5% for a core thickness of 6 mm or 10 mm but was higher than 10% for a 15 mm thick core. More recently, the only measured (using a hammer exciter and a Laser sensor) *first* natural frequency was also used by Chenini *et al*. (2017), as a reference to investigate the kinematics effect on aluminum alloy (hard) and NOMEX® (soft) honeycomb sandwich cantilever *beams* vibration using continuum and 3D detailed FE, with ABAQUS®, models. Noticeably, the faces and core geometries were modeled under the 'Part' module and their assembling was under the 'Assembly' module with the option 'Depending' in order to facilitate the meshing of each component 'Part' using C3D8R FE. However, the detailed FE 3D model fundamental frequency errors were high, as they were a little bit less than 10% for aluminum (hard) core and a little bit higher than 13% for the NOMEX® (soft) core.

From the above literature analysis, it can be noticed that mainly metallic honeycomb sandwich structures were experimented in free vibration, often under cantilever boundary conditions (BCs), and corresponding detailed FE 3D models have fine meshes of the honeycomb cell walls. It is then the purpose here to consider experimenting free vibration (using different exciters and sensors) of glass fiber-reinforced polymer (GFRP)-Epoxy woven *composite* faces and *hexagonal* aluminum alloy core actual sandwich panels, used for modern aircraft and spacecraft structures, under different BCs. The geometric and materials data are initially unknown and, thus, they have to be determined, through direct measurements, reverse engineering techniques and modal experimental-numerical inverse identification, as *a priori* inputs to the sandwich panel numerical free-vibration analysis through the proposed practical, simplified and detailed FE 3D approach. Implemented via a user-friendly MATLAB® graphic user interface (GUI), the latter consists of a *meshing tool* that in-plane propagates a basic cell shape, a macro-commands language *translator* that generates FE simulation input-files and a commercial FE *solver* (ABAQUS® or ANSYS®). The approach is practical because it is simply implemented using commercial codes and can handle in blind (unknown or incomplete) material and geometric data. It is also simplified and detailed because it represents each wall of the actual hexagonal core cells as a *single* 2D 4 nodes *quadrangular* shell element or *two* 3 nodes *triangular* ones, while the GFRP-Epoxy *woven composite* faces meshes are obtained simply using the nodes at the core upper/lower interfaces. As the same 2D FE interpolation type is used for meshing the core and faces, this leads to an automatic handling of the FE compatibility relations required at the sandwich interfaces.

Therefore, in the following, the first section is devoted to the geometric and material properties description of the investigated modern aircraft and spacecraft-type actual hexagonal honeycomb sandwich panel. Then, the conducted experimental modal analyses under different measurement configurations and BCs are described and discussed. Next, the practical, simplified and detailed FE 3D approach is presented, together with the resulting numerical frequencies and mode shapes, and their correlations with the experimental values, for different commercial FE softwares and mesh basic cell shapes implementations. As a closure, the proposed practical detailed FE approach and its application to the actual honeycomb sandwich panel findings are summarized, and the foreseen extensions and applications are given as perspectives.

2. GFRP-Epoxy woven composite faces-hexagonal honeycomb core sandwich panel

The modern aircraft and spacecraft actual sandwich panel to be investigated here is constituted of a honeycomb core, with hexagonal cells which walls are made of a *very thin* aluminum alloy foil, to which are adhesively bonded to its major surfaces two GFRP-Epoxy woven composite faces. It is usually produced in two steps: the aluminum hexagonal honeycomb manufacturing and the GFRP-Epoxy woven composite faces (which might be prepared *a priori*) bonding. For the two main hexagonal honeycomb manufacture processes, the expansion process that uses stacked sheets and the corrugated one that uses corrugated sheets, the first step is based on foils bonding. This leads automatically to *doubling* the thickness of the hexagonal cell walls that are in the bonding plane (see Fig. 1). To be realistic, this manufacturing process-induced thickness doubling should be considered by the detailed FE 3D modeling approach; however, this feature is often neglected.



(a) Hexagonal cells geometric irregularities (b) In-plane bonding double thickness walls (in bold)

Fig. 1 Faces-core assembly-process induced cells geometric irregularities and double thickness walls

As the sample's geometric and materials data are unknown, it is the objective of this section to identify them for the developed detailed FE 3D models inputs. They can be classified into: easily measurable (with good accuracy), influential (important), uncertain and correctible (through test-model updating). These 4 categories are separately investigated hereafter.

2.1 Measurable characteristics

After de-bonding the GFRP-Epoxy woven composite faces from the aluminum alloy hexagonal honeycomb core, the latter's thickness is measured using a caliper as 4.2 mm, while the combined face and adhesive glue thickness is measured as 155 μm *approximately*. However, the core hexagonal cells are not perfectly regular. Their geometry depends on the core positioning during the faces bonding so that different irregularities may appear, as illustrated in Fig. 1a. In particular, the cells radius is *approximately* 3 mm while their height is 5.2 mm. The latter are later measured through counting the number of cells along a given in-plane dimension of the sandwich sample. After removing the adhesive glue and cutting some fragments from the center of the cell walls, the latter's constituting aluminum foil thickness is measured as 45 μm using a Vernier gauge. Thus, including the glue, of thickness 5 μm, between the two bonded wall foils, the double thickness is 95 μm (2 x 45 μm + 5 μm). However, this cell walls glue *small* thickness is neglected. The assembled sandwich panel has a whole thickness of 4.6 mm, while its surface mass density is 96.2 mg/cm^2. The latter is calculated from the ratio of the measured sample mass of 9.16 g to its area of 5.20 x 18.3 = 95.16 cm^2. It's worth noticing, from the above measurements, that the GFRP-Epoxy woven composite faces and hexagonal honeycomb cells walls are *very thin* and *very light*.

2.2 Influential characteristics

The influential characteristics are those that have important effects on the sandwich free-vibration *modal parameters*. This concerns its constituents' *mass* and *stiffness*. Therefore, on one hand, while the core mass can be obtained with a *minimal error* by *assuming* the mass density of its constituting aluminum alloy *generically* as 2815 Kg/m^3, those of the GFRP-Epoxy woven composite faces and adhesive glue are *unknown*. Considered together, the *generic* GFRP-Epoxy composite mass density of 1900 Kg/m^3 will be corrected from the precisely weighing of the sandwich sample and its FE model computed value. On the other hand, the sandwich panel constituents' stiffness parameters are particularly important. Again, while, the stiffness of the hexagonal honeycomb core can be obtained *without significant error* by *assuming generic* aluminum alloy Young's modulus of 70 GPa and Poisson's ratio of 0.3 for its cell walls isotropic behavior, those of the faces and adhesive glue are completely *unknown*. As the proportion of the woven composite fibers is the same along the fabric in-plane perpendicular directions, a *simplified isotropic* behavior is considered for the two parameters (*effective* Young's modulus and Poisson's ratio) modal test-FE model *inverse* identification.

2.3 Uncertain characteristics

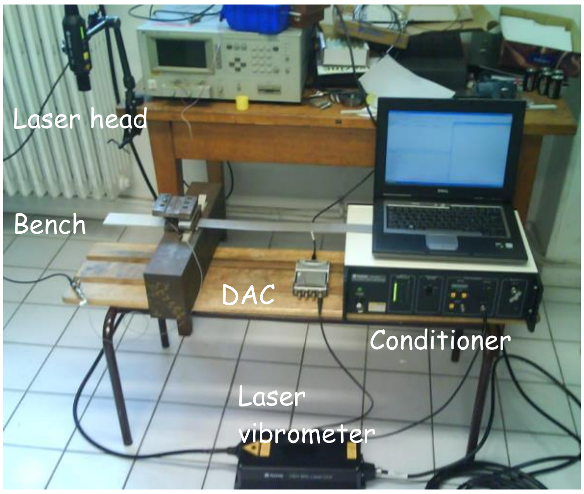
The proportions of the composite fabric Epoxy resin and glass fibers, and the glue, are *unknown* and *unmeasurable*. Only the fibers *identical* proportions along the in-plane *perpendicular* (0°, 90°) directions can be set *certainly*. Thus, each face thickness, surface mass and density are *unmeasurable*. This is because the faces internal (bonded) surfaces are irregular, due to the remaining glue on the core after de-bonding of the faces from the core. The de-bonding residual glue on the faces and core prevents also from making precise weighing of the sandwich panel constituents. The faces' mass and stiffness are then *uncertain*. The free-vibration-based mixed test-model inverse identification, through updating (or correcting) the latter's *generic known* parameters, is then hereafter considered.

2.4 Updatable characteristics

The properties of the aluminum alloy hexagonal honeycomb core can be *assumed* without much uncertainty (see previous sub-sections), while the stiffness of the woven composite faces can be *updated* through modal test-model correlations. Their surface mass can be deduced through measuring the mass of the sandwich panel. However, the composite faces thickness and density should be *approximated*. The former was given in the above first sub-section, while the latter is updated as 2200 Kg/m^3, from the generic value given in the above second sub-section, by including the glue mass and correlating the precisely weighed sample and FE model calculated masses. Besides, the woven composite faces Poisson's ratio is *assumed* as 0.17 (that of a generic GFRP-Epoxy composite), while its Young's modulus is inversely identified as 19 GPa, through correlating (updating) the FE model fundamental frequency with the measured one (560 Hz) of the sandwich panel under *free* BCs.

3. Experimental modal analyses

The experimental modal analyses aimed first to update the influential stiffness parameters of the woven GFRP-Epoxy composite faces that remain uncertain, then for validating the numerical results provided by different developed meshes (basic cells and FE shapes, edge effects) and softwares (ABAQUS®, ANSYS®) implementations. The used experimental setup is shown in Fig. 2.



(a) Acquisition system

(b) Excitation system

Fig. 2 Experimental setup for the free-vibration of the honeycomb sandwich panel (here cantilevered)

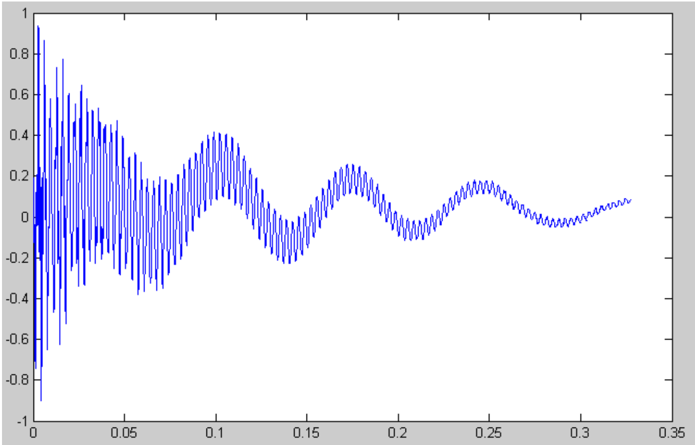
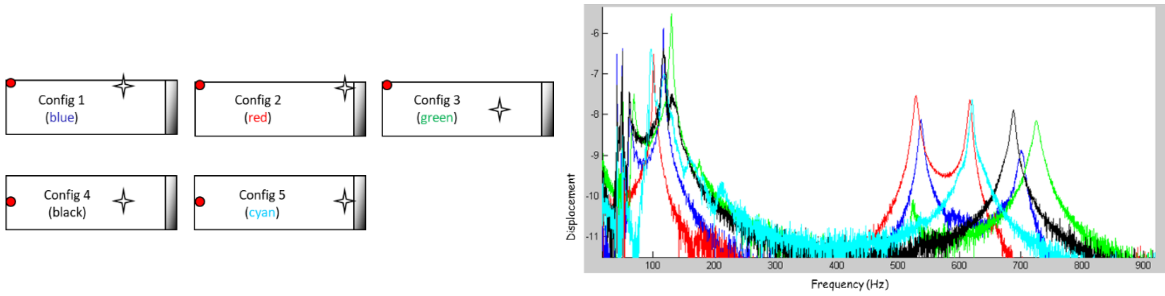


Fig. 3 Illustration of the high damping of the honeycomb sandwich panel (here simply-supported-SS; abscissa: time in s, ordinate: voltage in V)



(a) Excitations (red discs) and measurements (stars) normal to the clamped (shaded area) panel

(b) First few modes Fourier Transfer Functions (FTF) frequency responses

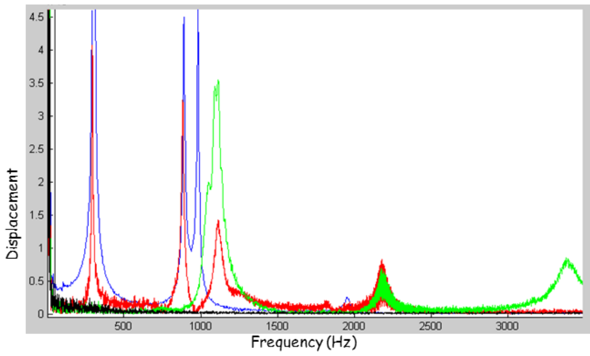
Fig. 4 Illustration of the variability and dependence on the exciter/sensor positions of the frequency response of the honeycomb sandwich panel (here cantilevered)

While the experimental extraction of structural modal properties (mode shapes and frequencies) is usually easy, that of the present GFRP-Epoxy woven composite faces-hexagonal honeycomb core sandwich panel is rather challenging. This is due to its lightweight and fragility that result into high damping, as illustrated in Fig. 3, and repeatability issues under different excitations and measurements positions, as shown in Fig. 4. Therefore, as the classical shock hammer exciter is efficient only for extracting the first few modes, a piezoelectric stack actuator is used for exciting some configurations. For both excitation types, a laser vibrometer is used as displacement sensor because the classical accelerometer sensing approach is much disturbing for the present honeycomb sandwich panel. The frequency measurements are hereafter discussed for various BCs.

3.1 Cantilever configuration

Due to the very high uncertainty-induced actual non-perfect clamping condition, in contrary to the modeled ideal one, the cantilever (or clamped-free) configuration is not recommended for free-vibration-based mixed experimental-numerical FE model updating and correlation (see Chevallier *et al*. 2009, Hamdi *et al*. 2014) in order to inversely identify materials properties (Benjeddou *et al*.

(a) Honeycomb sandwich panel SS configuration



(b) FTF frequency responses (colors indicate measurements at different points as in Fig. 4a)

Fig. 5 SS honeycomb sandwich panel frequency responses under various exciter/sensor positions

Table 1 Honeycomb sandwich panel measured and ABAQUS® frequencies (Hz) under free and SS BCs

BCs	Mode	**1**	**2**	**3**	4	5	6	7	**8**
Free	Type	x-z bending	torsion	x-z bending	torsion	x-z bending	-	-	-
	Measured	560	NM *	NM *	NM *	NM *	-	-	-
	ABAQUS®	563.31	1182.9	1541.1	2438	2986.3	-	-	-
	Err (%) **	**0.59**	NS ***	NS ***	NS ***	NS ***	-	-	-
SS	Type	x-z bending	x-z bending	torsion	x-z bending	**x-y** bending	torsion	x-z bending	torsion
	Measured	300	982	1110	1950	NM *	2180	3400	3950
	ABAQUS®	311.63	988.7	1148.3	2272.7	2314	2445.5	3724.8	3958
	Err (%) **	**3.88**	**0.68**	**3.45**	16.55	NS ***	12.18	9.55	**0.20**

*Non-measurable (at present cond.). **Err (%) = 100*(ABAQUS®-Measured)/Measured. ***Non-significant.

2013, Benjeddou and Hamdi 2016, Hamdi and Benjeddou 2017). Besides, it is difficult to excite this configuration, not only with the shock hammer but also with the piezoelectric stack actuator. Indeed, using various positions of the latter (see Fig. 4a), the measurements show high variability and dependence of the frequencies on the positions of the exciter, as illustrated in Fig. 4b. Therefore, this configuration is not retained for the modal test-FE model updating-based inverse identification of the faces' stiffness properties.

3.2 Simply-supported configuration

To realize SS BCs, two edges of stiff wood round quarters were first bonded, using thin beads of adhesive glue, to the ends of one face of the honeycomb sandwich panel, then clamped (glued) to a heavy mass as shown in Fig. 5a. To avoid the exciter influence on the free-vibration of this SS configuration, the piezoelectric stack actuator was positioned on the edges level, very close to the adhesive glue beads. The corresponding FTF frequency responses and frequency results are shown in Fig. 5b and in the 7th raw of Table 1, respectively.

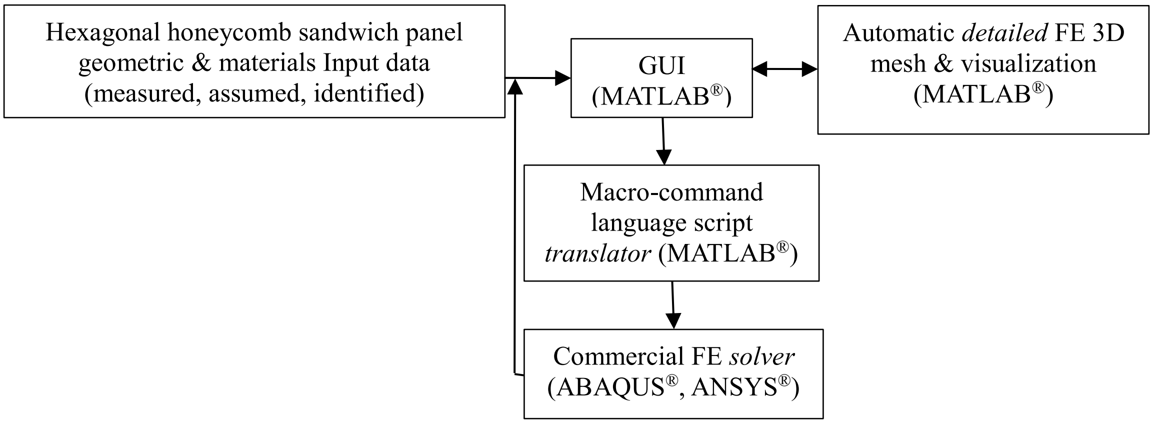


Fig. 6 Flowchart of the proposed practical, simplified and detailed FE 3D modelling approach of the actual GFRP-Epoxy woven composite-aluminum alloy hexagonal honeycomb sandwich panel

Table 2 Measured, assumed and identified honeycomb sandwich panel geometric and materials properties

Sandwich*	Length (L)	Width (l)	Thickness (H)	Length cells (N_L)	Width cells (N_l)
	18.3 cm	5.2 cm	4.6 mm	25 (counted)	10 (counted)
Core	Cell length (A)	Cell width (B)	Cell wall thickness (t)	double wall thickness (2t)	Cell side (a)
	*6 mm	*5.2 mm	*45 μm	*90 μm	*3 mm
	Material	Thickness (h)	Young's (E_c)	Poisson's (υ_c)	Density (ρ_c)
	Aluminum	*4.2 mm	**70 GPa	**0.3	**2815 Kg/m^3
Faces	Material	Thickness (t_{fg})	Young's (E_f)	Poisson's (υ_f)	Density (ρ_f)
	Composite	*155 μm	***19 GPa	**0.17	***2200 Kg/m^3

* Measured. ** Assumed as generic data. *** Identified via test-FE correlation.

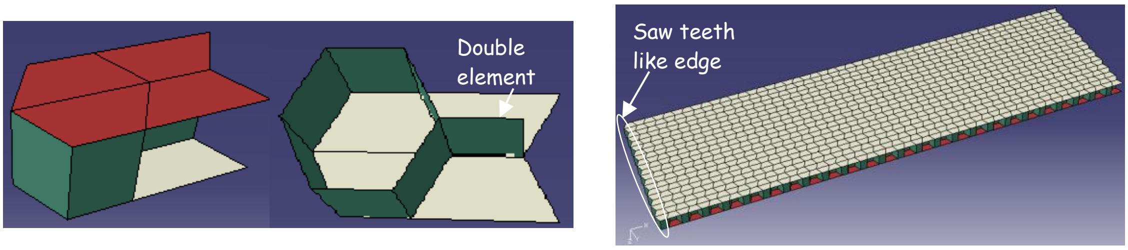
3.3 Free configuration

The free-vibration of the hexagonal honeycomb sandwich panel under completely free BCs has been measured through the emitted sound by a shock after a *free fall*. The only extracted natural frequency is the *fundamental* one that is 560 Hz as listed above in the third row of Table 1. It is used for the test-FE model updating in order to get the woven composite faces' effective stiffness parameters, in particular their Young's modulus given above. It's worth mentioning that the eventual panel-air vibro-acoustic coupling during this free fall test was neglected.

4. Detailed FE 3D approach-based analyses

The flowchart of the proposed practical, simplified and detailed FE 3D approach is shown in Fig. 6 and a summary of its input geometric and materials parameters and their measured, assumed or inversely identified values that are used for meshing and simulating the actual GFRP-Epoxy woven composite faces-aluminum alloy hexagonal honeycomb core sandwich are given in Table 2.

After collecting the *actual* GFRP-Epoxy woven composite-hexagonal aluminum alloy



(a) Hexagonal honeycomb core mesh basic cell (b) Detailed FE 3D mesh of the sandwich panel

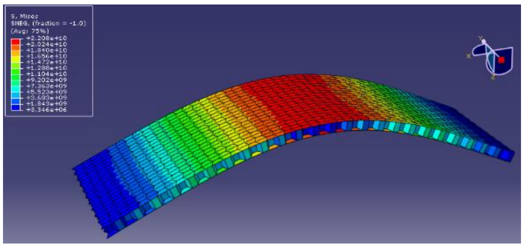
Fig. 7 Hexagonal honeycomb sandwich panel detailed 3D mesh by the developed MATLAB® mesh tool

honeycomb sandwich panel geometric and materials input data, through measurements, assumptions and identifications (see Table 2), the here proposed practical FE 3D approach represents *parametrically* and geometrically *in detail* its three layers (2 faces sandwiching a core) using a *single* 4-noded *quadrangular* or *two* 3-noded *triangular shell* 2D FE for each hexagonal face *wall*. The upper and lower nodes of the core elements are also used for meshing the top and bottom faces of the sandwich so that the FE compatibility relations at the core-faces' interfaces are automatically fulfilled. For this purpose, via a user-friendly MATLAB® GUI, an *automatic* detailed FE meshing tool, that can also visualize all FE model details (nodal and element matrices, resulting mesh and BCs) and a macro-command language script (MATLAB®), that translates the geometry, material, BCs, FE type and analysis input data to a commercial FE solver (ABAQUS®, ANSYS®) have been developed and used, first for the vibration-based experimental-numerical inverse identification of the unknown faces' stiffness parameters, then for the panel's further (wider frequency band, various BCs, mesh basic cell and FE shapes and softwares) numerical free-vibration analyses. Different implementations for ABAQUS® and ANSYS® are hereafter discussed.

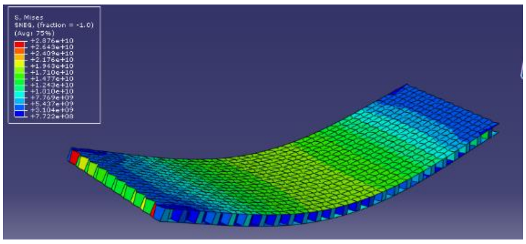
4.1 ABAQUS® implementation

The first implementation of the above (Fig. 6) detailed FE modelling approach was for the commercial solver ABAQUS® Standard; i.e., the macro-command script was written in a language useable by the latter via a specific *input file*. The developed MATLAB® automatic meshing tool propagates the core-faces basic cell, shown in Fig. 7a, along the length (x) and width (y) directions until having the mesh of the whole sandwich panel shown in Fig. 7b.

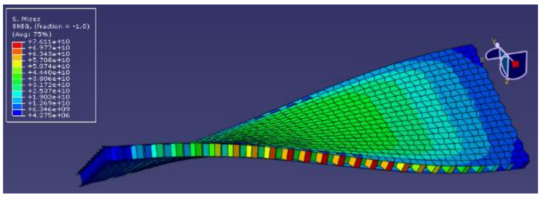
It's worth mentioning that the equivalence of nodes at the repeated cells interfaces is made at each propagation direction. Besides, the faces' elements are defined directly from the nodes on the top and bottom edges of the core wall elements. This allows ensuring automatic handling of the required faces-core interfaces' FE compatibilities and continuities. The cell walls of single thickness are modeled with a single 4-noded *shell* FE with reduced integration (S4R), having 6 DOFs per node (3 translations and 3 rotations), while those of double thickness are here modelled by superposing two elements; this simplifies, in particular, the number of elements properties. However, in contrary to the actual panel, the chosen *closed* basic cell (Fig. 7a) leads to saw teeth like edges (see white ellipse on left side of Fig. 7b). This may affect the computed frequencies since it leads to less mass and stiffness, as assessed later. It may also explain partially the deviations between measured and computed frequencies (see Table 1). The input file for ABAQUS® contains in the same part, the panel full mesh and three sets of elements; one set of



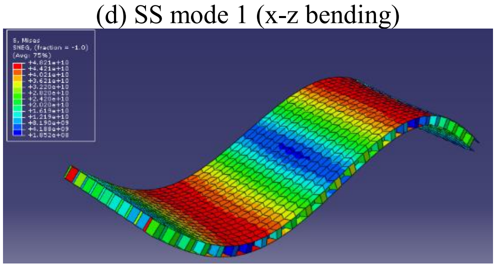
(a) Free mode 1 (x-z bending)



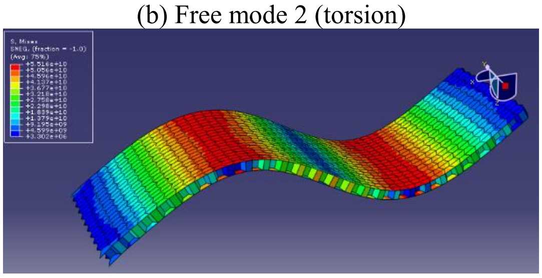
(d) SS mode 1 (x-z bending)



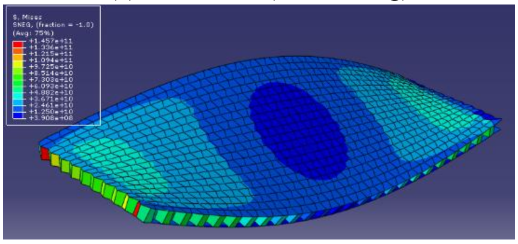
(b) Free mode 2 (torsion)



(e) SS mode 2 (x-z bending)



(c) Free mode 3 (x-z bending)



(f) SS mode 3 (torsion)

Fig. 8 Hexagonal honeycomb sandwich panel simulated first 3 modes for free (left) and SS (right) BCs

elements per face and one set for those of the core. These sets serve to apply the related properties. The detailed FE 3D model is loaded to ABAQUS® via the function 'import/load'.

The main input parameters for the developed MATLAB® automatic mesh tool of the sandwich panel are the core thickness (distance between the faces), the hexagonal honeycomb cells side, the number of core cells along its length (x) and width (y) directions and the name of the input file for ABAQUS®. It's worth mentioning that two additional *homothety* scaling factors (coefficients of homothety-Chom) along x (ChomX = 4/5 for the mesh in Fig. 7b) and y (ChomY = 1 for the mesh in Fig. 7b) directions are introduced and can be used optionally. Their role is to make the sandwich panel's actual (measured) length and width equal to those computed from the hexagonal cell in-plane dimensions and the number of cells (see Table 2) in the same directions. In fact, there are actually only 10.5 cells in the panel width direction, while the corresponding calculated number is 11 cells. Besides, the calculated number of cells along the panel length is only 21, compared to the actual 25. These differences between the measured (counted) and calculated number of core cells are due to the geometric irregularities of the hexagonal honeycomb cells, as discussed above (see also Fig. 1a). That's why the user is given the option to privilege either the actual or calculated panel dimensions. Depending on the selected option, the results may differ, as assessed later. This is also one of the main reasons for the observed deviations between the measured and computed frequencies under different BCs as shown in Table 1. From the latter, it can be observed that the here proposed detailed FE 3D modelling approach, applied to the present actual GFRP-Epoxy

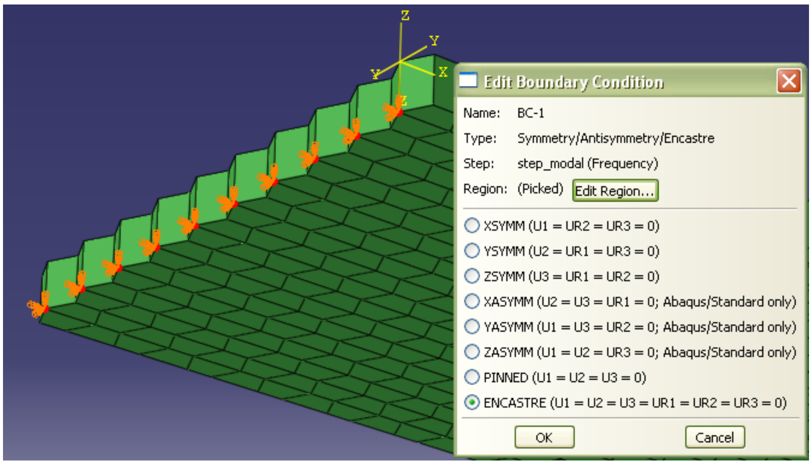


Fig. 9 Modelling of the adhesive glue beads for the actual SS configuration of Fig. 5a

woven composite faces-aluminum alloy hexagonal honeycomb core sandwich panel, is very accurate for the first few (3) modes (see Fig. 8), particularly the torsion ones.

The SS discrepancies of some higher modes (4, 6 and 7) frequencies, visible in Table 1, may be also due to the modelling of the actual adhesive glue beads (see Fig. 5a) by a row of clamped nodes (see Fig. 9) at each supported side. Indeed, this may not be reflecting the sandwich panel actual mechanical fixtures supposed to be simple supports.

4.2 ANSYS® implementations

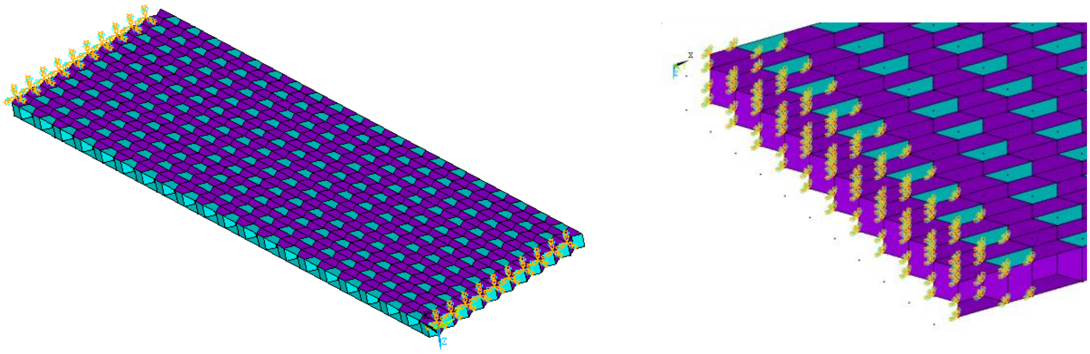
A second independent implementation was made for ANSYS® commercial FE solver. It uses a different (simpler) basic cell, shown in Fig. 10a, for generating the honeycomb core automatic detailed 3D mesh. The sandwich faces and core hexagonal cell walls (1 FE per wall and the in-plane bonding double thickness was not considered for implementation simplicity reasons) were meshed using 4-noded SHELL63 FE, having 6 DOFs per node (3 translations and 3 rotations). Besides, here the MATLAB® translator generates a text file, containing macro-commands, compatible with the ANSYS Parametric Design Language (APDL®), which can be read within ANSYS® Utility Menu through its 'Read Input from…' function.

This new implementation was used to analyze the free-vibration of the actual GFRP-Epoxy woven composite-aluminum alloy hexagonal honeycomb core sandwich panel under free, SS and cantilever BCs which are modelled as shown in Fig. 11. The latter shows that the above (see Fig. 7b) non desirable saw teeth-like edges effect is also present here.

The simulation results, denoted as ANSYS1, for the first five modes (see Fig. 12) of the actual GFRP-Epoxy woven composite-aluminum alloy hexagonal honeycomb core sandwich panel under free BCs are given in Table 3. The latter shows that they are less than 4% of relative deviation from ABAQUS® reference results, as given in Table 1 and repeated here in the 3rd row of Table 3 for comparison clarity. This over-estimation is due to neglecting the walls double thickness. Besides, the fundamental frequency is less than 4.5% from the measured reference one (see Table 1). These favorable comparisons validate ABAQUS and ANSYS1 implementations. However, both suffer from the resulting saw teeth-like edges and have no GUI.

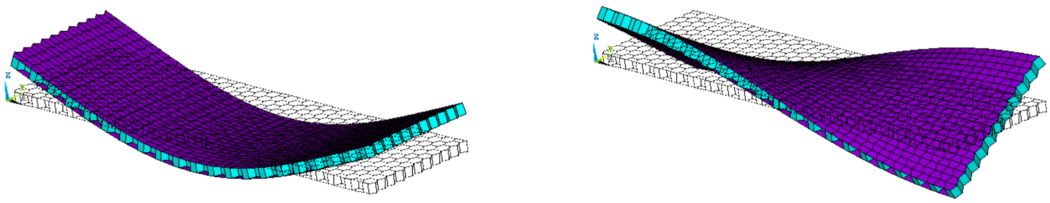
(a) ANSYS1 mesh basic cell (b) ANSYS2 mesh basic cell

Fig. 10 In-plane propagated basic cells for generating the honeycomb core meshes for ANSYS®

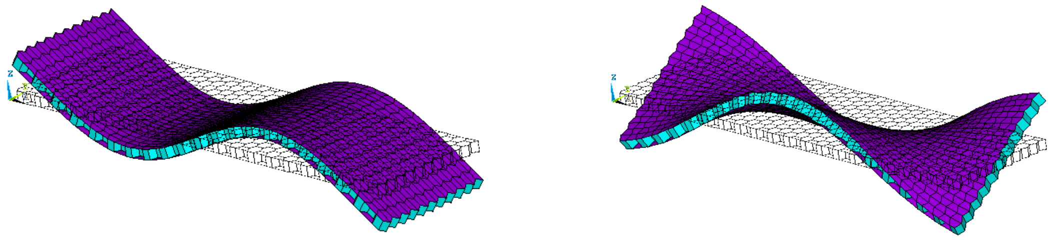


(a) SS BCs modelling (b) Cantilever BCs modelling

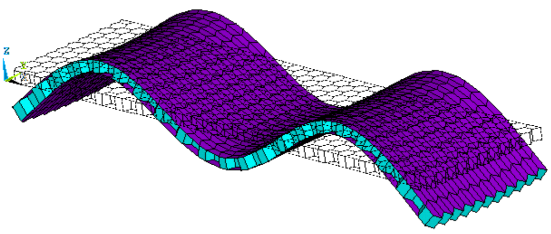
Fig. 11 Implementation for ANSYS® of the SS and cantilever BCs using the detailed FE 3D approach



(a) mode 1 (x-z bending) (b) mode 2 (torsion)



(c) mode 3 (x-z bending) (d) mode 4 (torsion)



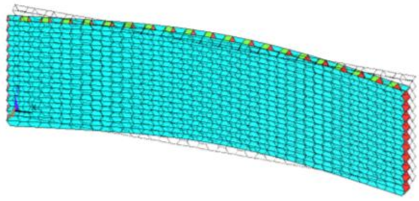
(e) mode 5 (x-z bending)

Fig. 12 Hexagonal honeycomb sandwich panel simulated first 5 modes under free BCs using ANSYS1

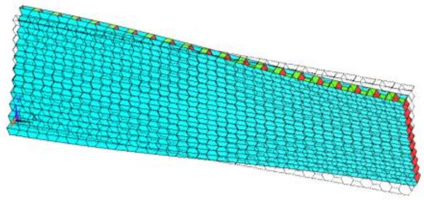
Table 3 Honeycomb sandwich panel numerical frequencies (Hz) for free BCs via the detailed FE approach

Mode	1	2	3	4	5	6	7	8
Type	x-z bending	torsion	x-z bending	torsion	x-z bending	**x-y** bending	torsion	x-z bending
ABAQUS	563.31	1182.9	1541.1	2438	2986.3	-	-	-
ANSYS1	585.02	1186.1	1587.9	2443.8	3035.8	-	-	-
Err (%) *	3.85 (4.47**)	**0.27**	3.04	**0.24**	1.66	-	-	-
ANSYS2	567.52	1138	1547.65	2351.1	2978.64	3612.37	3705.37	4790.65
Err (%) *	**0.75** (**1.34****)	-3.80	**0.43**	-3.70	**-0.26**	-	-	-
ANSYS3	560.4	1131.17	1528.73	2334.58	2943.61	3573.68	-	-
Err (%) *	*-0.52* (*0.07***)	-4.37	-0.80	-4.24	-1.43	-1.07***	-	-
ANSYS4	575.54	1140.1	1568.7	-	-	-	-	-
Err (%) *	2.17 (2.78**)	*-3.62*	1.79	-	-	-	-	-

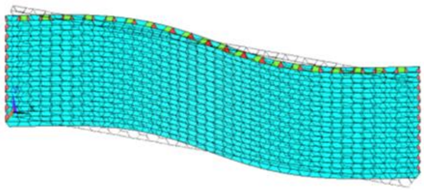
Err (%) = 100(ANSYS-ABAQUS)/ABAQUS. **Ref. to free fall test. ***Ref. to ANSYS2.



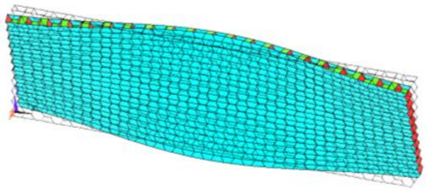
(a) mode 1 (x-z bending)



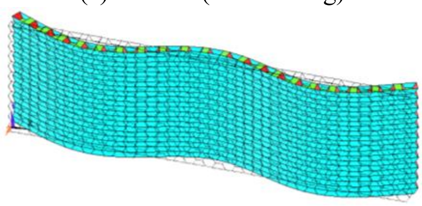
(b) mode 2 (torsion)



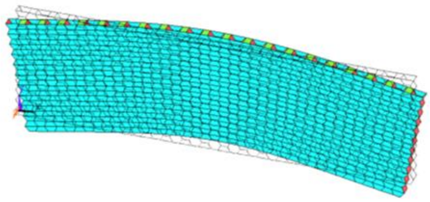
(c) mode 3 (x-z bending)



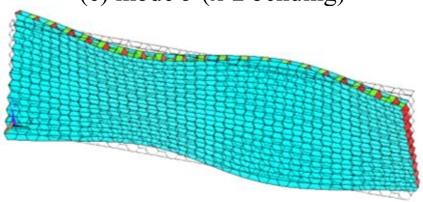
(d) mode 4 (torsion)



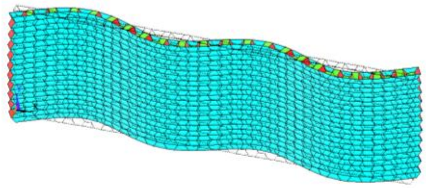
(e) mode 5 (x-z bending)



(f) mode 6 (x-y bending)



(g) mode 7 (torsion)



(h) mode 8 (x-z bending)

Fig. 13 Hexagonal honeycomb sandwich panel simulated first 8 modes under free BCs using ANSYS2

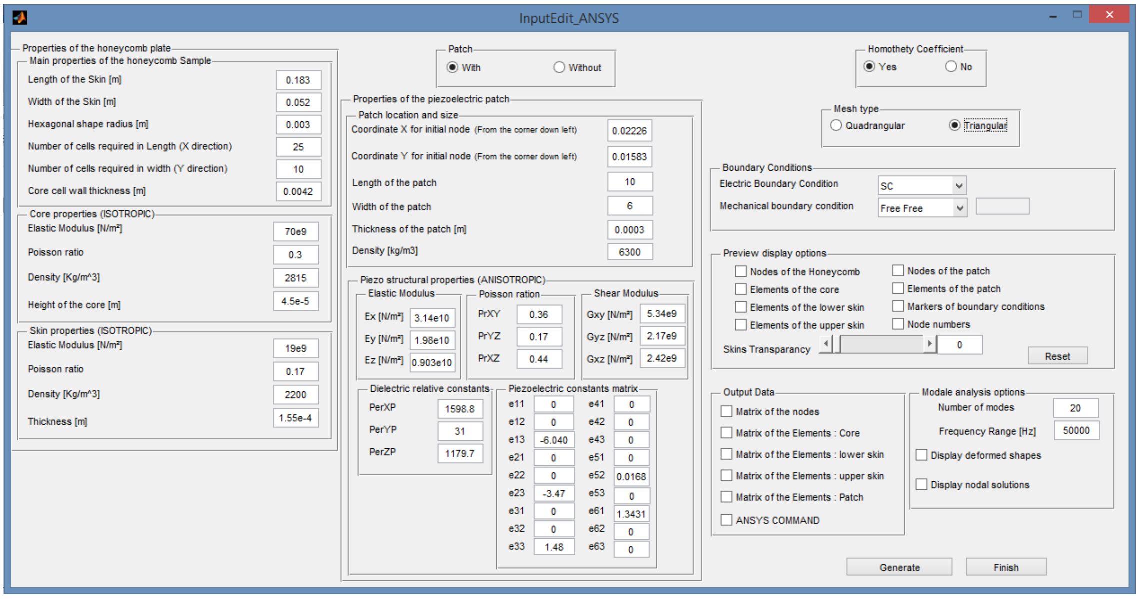
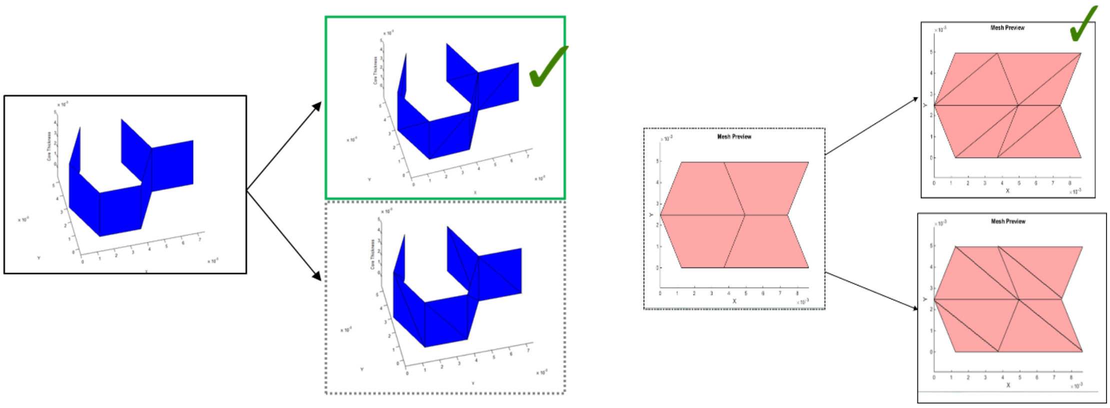


Fig. 14 The developed GUI for the free-vibration analysis of hexagonal honeycomb sandwich panels



(a) Core hexagonal cell mesh triangularization (b) Face cell mesh triangularization

Fig. 15 Triangularization of the honeycomb core and faces quadrangular meshes (✓ : implemented)

A second ANSYS® independent implementation was made using a new and simpler basic cell (Fig. 10b), than those for ABAQUS® (Fig. 7a) and ANSYS1 (Fig. 10a), for detailed meshing of the honeycomb core. This time, the latter's hexagonal cell wall in-plane bonding double thickness is considered as an additional property of the 4-noded shell FE modelling it. The frequencies of the first 8 modes (see Fig. 13) of the sandwich panel are also compared in Table 3 to the free fall test and ABAQUS® reference results. It appears that ANSYS2 frequencies are more accurate than those of ANSYS1 for the transverse (x-z) modes, but less accurate for the torsion ones. This may be due to the effect of considering the in-plane bonded walls double thickness. The good accuracy may be also due to privileging the *actual* dimensions of the panel instead of the number of cells.

An *interactive* GUI, as multiple *prompts* and *waiting bars*, is proposed for acquiring geometric and materials input parameters, BCs, display and output files options. However, the saw teeth-like edges effect is still present (see Fig. 13). The former is solved by developing an all-in-one user-friendly GUI (see Fig. 14), while the latter is avoided by adding *triangular* elements to the width sides of the faces so that they become straight as actually they are. The corresponding results are given as ANSYS3 in Table 3. They show enhanced accuracy only for the fundamental mode.

Triangular elements have been also implemented in the MATLAB mesh tool for ANSYS® and used for generating the detailed FE mesh of the sandwich panel hexagonal honeycomb core and faces. This was reached by dividing each wall quadrangular element in two triangular ones. Only one of the two possibilities of triangularization, shown in Fig. 15, has been implemented.

It is worth mentioning that the quadrangular and triangular mesh options, shown in the GUI of Fig. 14, can be used independently or in combination for the core and the faces. As an illustrating example, the first three frequencies resulting from the triangular meshed core and quadrangular meshed faces are listed, as ANSYS4, in Table 3. It appears that the core triangular mesh enhances the torsion modes' accuracy but decreases that of the transverse modes.

5. Conclusions and perspectives

The present work proposed a *practical*, *simplified* and *detailed* FE approach for the 3D free-vibration analysis of *actual* aircraft and spacecraft-type *lightweight* and *thin* hexagonal honeycomb sandwich panels. It proposes a user-friendly MATLAB® GUI, which handles geometric and materials inputs as well as display, preview, analysis and output files options, an automatic detailed quadrangular or triangular shell elements (sharing the same number of nodes and DOFs type and number per node, leading to the same total number of DOFs) meshing tool and a macro-command language translator for input files of ABAQUS® and ANSYS® commercial FE solvers. The methodology was successfully applied to the free-vibrations of an actual aircraft and spacecraft-type sandwich panel, made of *very thin* GFRP-Epoxy woven composite faces and a *thin* aluminum alloy *hexagonal* honeycomb core. For this purpose, experimental and numerical modal analyses were conducted under various BCs and different aspects, related to the detailed meshing of the hexagonal honeycomb sandwich components, like the effects on the panel frequencies of the saw teeth-like edges, core mesh repetitive basic cell shape and FE type (quadrangular or triangular), were investigated. It was found that this simplified detailed FE approach, which meshes each core cell wall by a single 4 nodes quadrangular shell element or two 3 nodes triangular ones, is very accurate for the first few vibration modes that are usually used for the pre-design of aircraft and spacecraft honeycomb sandwich structural components. Besides, representing the faces lateral actual straight edges, through additional triangular elements on the width sides, enhances the accuracy of the fundamental mode only, while the mesh propagating basic cell shape has a slight effect on the considered modes frequencies. Also, it was found that the triangular FE shape mesh of the core cells walls has a positive effect on the torsion modes frequencies accuracy but a negative effect on that of the transverse (x-z) modes frequencies. The here provided experimental and numerical frequencies and mode shapes under different BCs can be used for validating other approaches, such as those based on the core homogenization or the beam/plate/shell theories.

It is worth mentioning that the here proposed practical, simplified and detailed FE simulation procedure is extendable to other *global response* analyses, as for geometric or material parametric frequency analyses, modal experimental-numerical inverse identification of effective properties of other aircraft and spacecraft hexagonal honeycomb sandwich panels made of other lightweight, thin or *thick* and hard or *soft* (NOMEX®) core and multilayer (Carbon or Graphite reinforced) composite faces. For the latter purpose, more experimental modes under free BCs are necessary. Sandwich panels can be also functionalized through, for example, an extruded 3D FE patch made of viscoelastic (for damping) or piezoelectric (for sensing, actuation, damping or energy harvesting) material. The latter can be also integrated in laminated faces or at the sandwich interfaces. The case of the bonding of a piezoelectric macro-fiber composite (MFC) patch on the top surface of the presently investigated hexagonal honeycomb sandwich panel has been already implemented (Benjeddou 2018). A constrained viscoelastic patch can be also bonded on the opposite surface of the sandwich panel so that a (semi-)active-passive hybrid damping treatment can be realized or the MFC patch can be used as exciter for damping or noise transmission (Guerich and Assaf 2013).

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Appendix: List of abbreviations

The following abbreviations have been introduced at their first appearance in the text, then used throughout this document:

- 2D: Two-dimensional
- 3D: Three-dimensional
- APDL: ANSYS Parametric Design Language
- BCs: Boundary conditions
- ChomX: Coefficient of homothety along the x direction (length)
- ChomY: Coefficient of homothety along the y direction (width)
- CPT: Classical Plate Theory
- DOF: Degree Of Freedom
- ESL: Equivalent Single-Layer
- FE: Finite Element
- FSDT: First-order Shear Deformation Theory
- FTF: Fourier Transfer Function
- GFRP: Glass Fiber Reinforced Polymer
- GUI: Graphic User Interface
- LW: Layer-Wise
- MFC: Macro Fiber Composite
- SS: Simply-Supported
- TSDT: Third-order Shear Deformation Theory