MDA of Hexagonal Honeycomb Plates used for Space Applications

A. Boudjemai , M.H. Bouanane, Mankour, R. Amri, H. Salem, B. Chouchaoui

Abstract—The purpose of this paper is to perform a multidisciplinary design and analysis (MDA) of honeycomb panels used in the satellites structural design. All the analysis is based on clamped-free boundary conditions. In the present work, detailed finite element models for honeycomb panels are developed and analysed. Experimental tests were carried out on a honeycomb specimen of which the goal is to compare the previous modal analysis made by the finite element method as well as the existing equivalent approaches. The obtained results show a good agreement between the finite element analysis, equivalent and tests results; the difference in the first two frequencies is less than 4% and less than 10% for the third frequency. The results of the equivalent model presented in this analysis are obtained with a good accuracy. Moreover, investigations carried out in this research relate to the honeycomb plate modal analysis under several aspects including the structural geometrical variation by studying the various influences of the dimension parameters on the modal frequency, the variation of core and skin material of the honeycomb. The various results obtained in this paper are promising and show that the geometry parameters and the type of material have an effect on the value of the honeycomb plate modal frequency.

Keywords—Satellite; Honeycomb; finite element method, modal frequency; dynamic

I. INTRODUCTION

AN important problem facing design engineers in the aerospace industry is how to achieve better design concepts by considering structure performance and manufacturing cost in the early stages of product development. One of the most important considerations in designing a spacecraft is weight. By reducing the weight of a spacecraft, it is possible to increase the payload, which improves agility and also reduces the launch cost [1]. The structural and mechanical parts of a spacecraft generally represent a large percentage of its weight and, therefore, it is important to choose the proper material and structural configurations to minimize the weight [2].

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In many industrial applications, reducing the weight of a structure without compromising its strength and stiffness is considered as one of the most important design criteria. Today, the search of the best performance, quality, and cost for space vehicles became a complex process. The required optimum in a total way, on the level of the system, implies choices of compromise between the various elements which make it up in order to answer increasingly many and sometimes contradictory requirements.

Honeycomb sandwich structures have been widely used in the manufacture of the aerospace structures due to their lightweight, high specific bending stiffness and strength under distributed loads in addition to their good energy-absorbing capacity [3, 4, 5 and 6].

As in many areas of engineering generic applications are based on analytical methods and with the increasing complexity of the geometries, boundary conditions and material, in almost every case, the use of analytical methods become very tedious if not impossible. At this point, the use of computational methods comes into picture. With the help of computational methods, namely finite element method (FEM) for structural analyses, highly complicated problems can be handled with great accuracy. The disadvantage of using computational methods is that, in order to get accurate results, too much computational time is needed, and this increases when the problem becomes more complex. In addition, FEM models require a detailed study before the model is sent to the solver.

Finite element analysis of honeycomb sandwich panel has been performed by modelling the structure through different approaches using Msc Patran/Natran in order to study the natural frequencies.

To date, many equivalent methods of honeycomb sandwich plate had been studied [7, 8, 9, 10, 11 and 12, and 13]. In 2003, XIA Li-juan and al. proposed three equivalent methods that are called the sandwich theory, the honeycomb-plate theory and the equivalent -plate theory. Through the three methods the natural frequencies of a honeycomb sandwich plate including two load cases had been calculated. The computational results show that the three equivalent methods are reliable and practical in the finite element analysis [13].

In this study, the analytical method by using the equivalent models approach was carried out.

To verify the feasibility and the accuracy of the proposed FEM models, the numerical results calculated from the proposed FEM model are compared with the experimental measured results. A study on the honeycomb panel geometry and materials effect was also carried out.

II. HONEYCOMB PLATE THEORY

A. Constituent of Honeycomb sandwich structure

The first step in designing a sandwich structure is the choice of the different constituents, depending on the application: the face, the core and the adhesive joint to bond the faces to the core. Different choice criteria are of course the mechanical properties of the constituents, but also the processing and the price which can vary over several orders of magnitude.

Sandwich structures are often used in skin-frame designs. A honeycomb sandwich structure consists of two thin face sheets attached to both sides of a lightweight core (see figure 1). The design of sandwich structures allows the outer face sheets to carry the axial loads, bending moments, and in- plane shears while the core carries the normal flexural shears. Sandwich structures are susceptible to failures due to large normal local stress concentrations because of the heterogeneous nature of the core/face sheet assembly. Component mounting must therefore use potted inserts to distribute the point loads from connections. Sandwich panel face sheets are commonly fabricated using aluminium or graphite/epoxy composite panels. The core is typically fabricated using a honeycomb or aluminium foam construction [14, 15, 16, and 17].
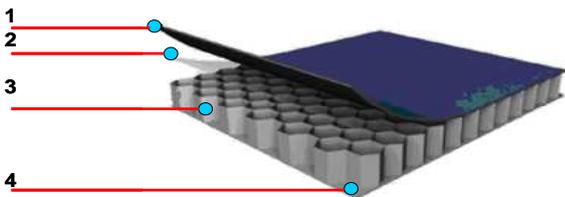

Fig. 1 Honeycomb sandwich structure
1. External Aluminium skin, 2. Adhesive, 3.Aluminium honeycomb core, 4. Internal Aluminium skin

B. Equivalent of Honeycomb Sandwich Plate

The generated equivalent model can be mostly used in the preliminary design stage of the design process. It can be used to reduce the time spent for the analysis of the honeycomb structure used in the satellite structural design and a great advantage to decrease in the pre-processor time and computation time.

The study of the mechanical behaviour of a composite material commonly uses the homogenisation concept. This concept makes it possible to avoid the problems involved in heterogeneities. One idealizes the real constitution of material by considering it continuous (see figure 2). The specific properties of material vary in discontinuous manners with the interfaces of passage between the various phases, a supposing, as clarified before each homogeneous and isotropic phase.
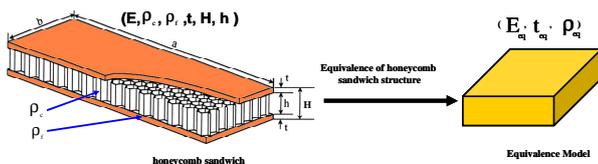

Fig. 2 Equivalent parameters of a Honeycomb Sandwich Plate

The equivalent characteristics of a honeycomb sandwich plate are determined by identifying its membrane and bending stiffness to those of an isotropic plate, as shown in the table I.

TABLE I
EQUIVALENT PARAMETERS OF SANDWICH STRUCTURE

	Honeycomb sandwich plate	Equivalente isotropic plate
Membrane stiffness	$\dfrac{2Et}{1-\nu^2}$	$\dfrac{E_{eq}\,t_{eq}}{1-\nu^2}$
Bending stiffness	$\dfrac{Eth^2}{2(1-\nu^2)}$	$\dfrac{E_{eq}\,t_{eq}^3}{12(1-\nu^2)}$

Where

$$t_{eq} = \left(3h^2\right)^{\frac{1}{2}} \qquad (1)$$

$$E_{eq} = \frac{2}{\sqrt{3}}.\frac{t}{h}.E \qquad (2)$$

h   height of honeycomb core or thickness of the plate.
E_{eq} equivalent elastic modulus.
E   Young modulus.
t   thickness of facing skin.
t_{eq} equivalent thickness.

In an anisotropic mechanical behaviour, all honeycombs are closed cell structures. By identifying a unit cell and deriving the volume fraction occupied by metal, the equivalent density is given by [13]:

$$\rho_{eq} = \frac{2\rho_f t + 2\rho_c\,(H-t)}{t_{eq}} \qquad (3)$$

Where
ρ_c density of honeycomb core material
ρ_f density of facing material,
ρ_{eq} equivalent density
H   height of sandwich panel including facing skins

For the analytical comparison of the first modal frequency of the equivalent model, we use in the analysis the theory applied in the case of a beam with clumped-free boundary conditions.

The fundamental frequency of a simple plate is given by:

$$f = \frac{1.01}{2\pi}\sqrt{\frac{E.h^2}{\rho a^4}} \qquad (4)$$

Where
ρ density of the plate
ν Poisson's ratio
E Young modulus
f fundamental frequency
a length of a sandwich panel or span of a sandwich beam

C. Dynamic response analysis of honeycomb plates

The vibration motion can be described mathematically by the general equation of motion (1) which is the basis for the system numerical analysis. In a finite-element model inertia (M), damping (D), stiffness (K) of the model are determined by the geometry and the material properties. The force F (t) defines the excitation, and u represents the displacement vector of all modes of the discretized model.

$$[M]\ddot{u}(t)+[C]\dot{u}(t)+[K]u(t)=F(t) \tag{5}$$

Where "[M], [C], [K]" stand for n*n order mass matrix, the damping matrix and the stiffness matrix, respectively; and "u" is n-dimensional vector displacement response.

The dynamic response characteristics of honeycomb plates under a single point loading are the concern of this paper, the effect of different geometrical parameter on the frequency and displacement are studied of "CFFF" plate. For the improvement of the perform an it of the structures in dynamics, structural damping and the reduction of dynamic response level of the structure, the variation of the damping ratio "C" and the thickness of both core and face sheet in honeycomb "HC " are highlighted which leads varied the mass of the structure, the study was done on a frequency bands from 0 to 2000 Hz.

III. MODAL ANALYSIS OF HONEYCOMB PANEL

In this section, we present a modal analysis made with honeycomb specimen by various methods: finite element analysis, analytical method using the theory of the equivalent model and the experimental test of the specimen. A comparison of the obtained results by these methods was also presented. The purpose of this study is to determine the Eigen frequencies and the modes shape of this plate under the Clumped-Free-Free-Free (C-F-F-F) boundary conditions. In this analysis the adhesive is not taken into consideration because its effect is negligible. Dimensions of the plate are given in table II.

TABLE II
DIMENSIONS OF THE HONEYCOMB PLATE

Length (a)	Width (b)	Thickness of the skin (t)	Cell seize (l)	Cell Thickness (t_{cell})	Core thickness (h)	Material
290 mm	40 mm	1mm	2 mm	0.2 mm	9 mm	Aluminium

A. Experimental testing

1. Experimental Measurement Setup

In this test, an instrumented hammer, B&K Type 4393-S accelerometers, a charge amplifier, a data acquisition board, clamping system, PC and a Frequency Response Function (FRF) analyser software have been used. The charge amplifier is connected to a data acquisition card and PC. The (FRF) analyser is installed on the PC that can be used for modal analysis. The honeycomb plate has been fixed at one end by means of a clamping system. The test setup is shown in figure 3.

By applying an impact force by the hammer to the plate, the hammer piezoelectric generates a corresponding voltage. The voltage is calibrated to force. Generally, an accelerometer consists of a frame, a mass and a piezoelectric element. Vibrating the mass in the accelerometer generates electrical current in the piezoelectric element. The corresponding voltage of the piezoelectric element is calibrated to acceleration, velocity and displacement. The signals from accelerometers and the impact hammer are transferred to a charge amplifier. The charge amplifier is connected to a data acquisition board and a PC. A Frequency Response Function (FRF) analyser is installed on the PC that can be used for modal analysis.

Two accelerometers have been used in this test. The first one is placed on the top of plate skin to measure the bending modes and the second accelerometer is placed on the core to measure the lateral modes. Figure 4 shows the accelerometers positions. The frequency span is set as 0 to 800 Hz.
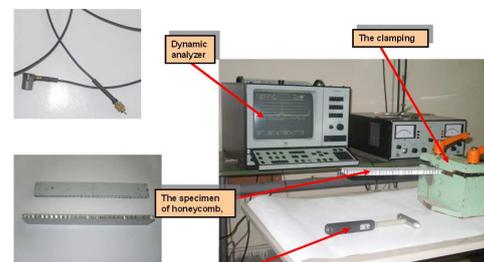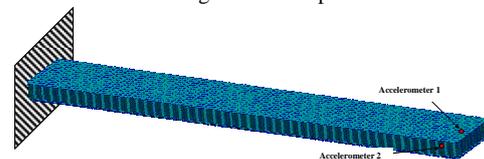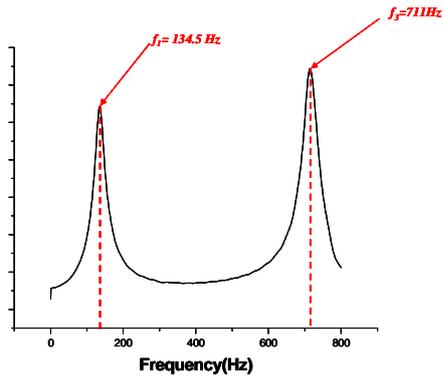

Fig. 3 Test setup
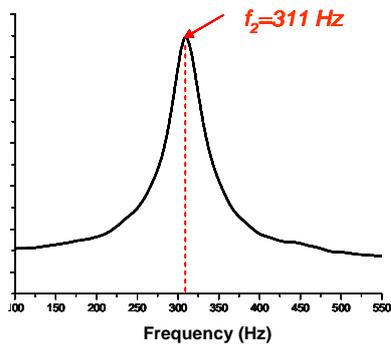

Fig. 4 Accelerometers positions

2. Experimental Results

After excitation using the shock hammer, the results measured are transformed from the analyzer into the computer. Figure 5 shows the first three frequencies measured of our specimen. These frequencies are:

- Mode 1, f1 = 134.5 Hz (1st bending mode)
- Mode 2, f2 = 311 Hz (lateral mode)
- Mode 3, f3 = 711 Hz (2nd bending mode)

Total elements and nodes of the FEM models are 23221 elements and 40426 nodes for a honeycomb sandwich plate

The boundary condition in the FEM simulation concerns the one edge of the short side which is constrained (displacement of x, y and z are zero, and rotation of x, y and z are zero) in the cantilevered honeycomb sandwich plate, which is shown in figure 7.

For the aluminium material, the elastic modulus is 72 GPa and the Poisson ratio is 0.33. The modal analysis results are shown in figure 8. These figures present the first four frequencies and the shape of each mode for our honeycomb sandwich plate.

The first bending frequency of the narrow honeycomb plate is found to be equal to 130.66 Hz, and then there is a lateral mode with 304.67 Hz. it can be found it easily this mode for a narrow honeycomb plate.

Generally, the case of torsional mode can be a second mode in the rectangular wide honeycomb plates, this is due to, when the width of the plate is large, and the rigidity of torsion will be weak, which involves a low value of frequency.
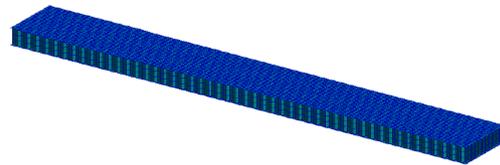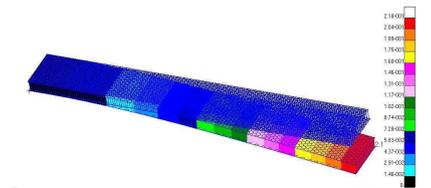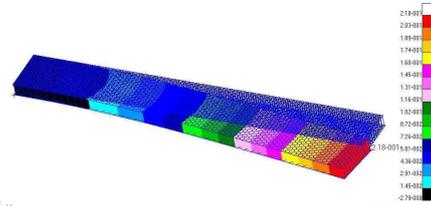

Fig. 6 FEM model of a honeycomb sandwich plate

Fig. 7 The displacement boundary of a honeycomb sandwich plate
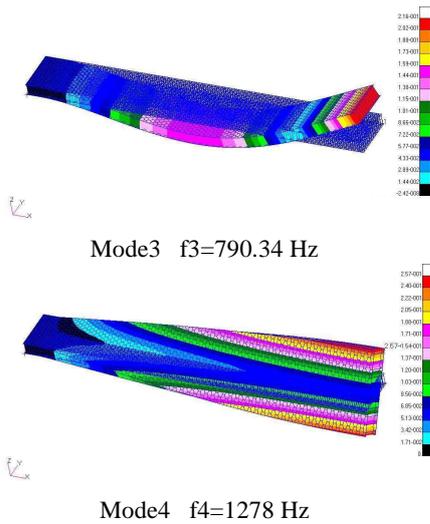

Mode1   f1= 130.66 Hz

Mode2   f2=304.67 Hz

(a)

(b)

Fig. 5 Measured frequencies of a honeycomb plate
(a) First and third frequency
(b) Second Frequency

Three sets of experiments were performed on the specimen in order to give more measurement about the third frequency of the specimen. The frequency span is set as 700 to 800 Hz, and the frequency resolution is 0.5 Hz.

Table III shows the third frequency (2nd bending mode) according to different measurement of the specimen.

We found that the clamping system has a huge effect on the measured frequencies and this is compared with similar results in the literature and good agreement is achieved [18].

TABLE III
THIRD FREQUENCY (2ND BENDING MODE) ACCORDING TO DIFFERENT MEASUREMENT OF THE SPECIMEN

	Measurement 1	Measurement 2	Measurement 3
Third frequency (Hz)	711	715.3	737.5

B. Finite Element Analysis of a honeycomb sandwich plate

The finite element model of a honeycomb sandwich plate, have been established by using MSC.Patran, shown in figure 6. The mesh of the skins and the core were made separately and the whole model of the honeycomb plate was assembled.

The elements employed in the finite element model are quad-4 element topology (four corner nodes) for honeycomb core and honeycomb faces.

Mode3   f3=790.34 Hz

Mode4   f4=1278 Hz

Fig. 8 Various shape modes of a honeycomb plate

Mode3   f3 = 807.69 Hz

Mode4   f4= 1449.9 Hz

Fig. 9 Various shape modes of equivalent specimen

C. Equivalent plate analysis

Using the equivalent model formulas given in section 2, the values of the equivalent stiffness (Eeq), the equivalent thickness (teq) and the equivalent density (ρeq) are:

Eeq = 6.29 GPa , teq = 17.16 mm   , ρeq = 428.12 kg/m3

For a simple plate, the first frequency can be calculated using equation 4, we obtain f1=134.91 Hz.

In order to give the first four frequencies and modes shapes, the equivalent plate has been analysed. The figure 9 shows the various modes shapes of the equivalent model given by the finite element method. The lowest frequency was in 1st mode. The frequency was increasing with each subsequent mode of vibration.
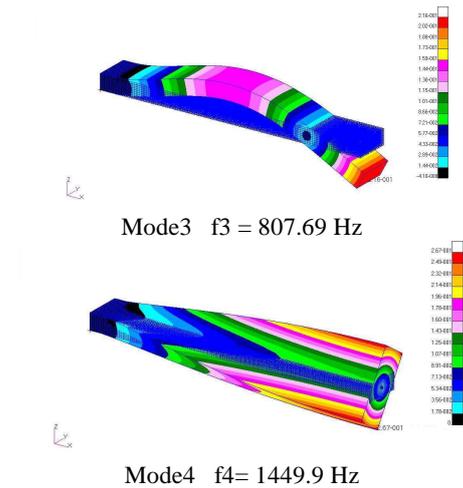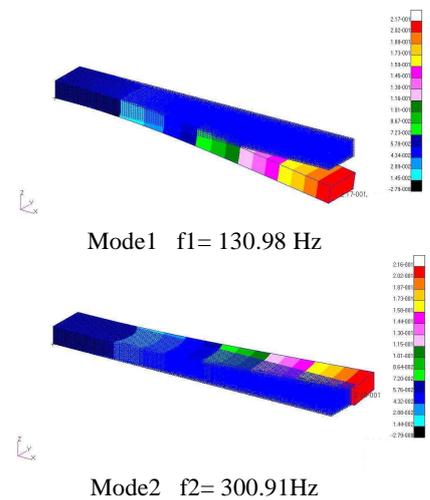

Mode1   f1= 130.98 Hz

Mode2   f2= 300.91Hz

D. Analysis and Comparison

Table 4 shows a comparison between the results of the three methods used for the modal analysis of the honeycomb plate, by the Finite Element Analysis (FEA), equivalent model theory and the experimental data. The first three frequencies have been taken for comparison.

TABLE IV
RESULTS COMPARISON

Frequency (Hz)	Finite Element Analysis	Equivalent Model	Experimental
f1	130.66	130.98	134.5
f2	304.67	300.91	311
f3	790.34	807.69	711

Table IV shows a good agreement between the results obtained for the first two frequencies, where the difference is less than 4%. The third frequency shows a significant difference in this analysis with less than 10%. As such, a plausible explanation is that the difference may be introduced by ignoring the effect of adhesive films between the faces and the honeycomb core.

Another parameter which can have an effect on the third frequency is the damping effect.   Furthermore, in high frequencies the errors are more significant. It can be also the boundary conditions using clamping system which cause the difference.

To give a good comparison between the results of FEA and equivalent methods at large frequency range [0 to 5000 Hz], figure 10 is given. It is clear that the equivalent model gives a good representation of the honeycomb plate for modal analysis, and can be used for complex and large honeycomb structures, which reduces the computing time of the analysis.

The natural frequencies and mode shapes for cantilever honeycomb plate are compared with similar results in the literature and good agreement is achieved [18, 19, 20, and 21].
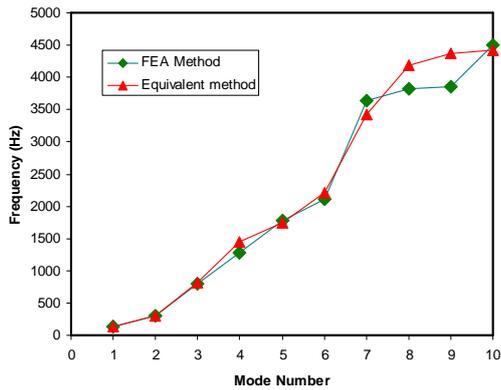
Fig. 10 Comparison between FEA and equivalent methods for large frequency range

IV. HONEYCOMB PANEL GEOMETRY AND MATERIALS EFFECT

A number of design parameters may affect the natural frequency of the honeycomb plate, e.g. the skins thickness of a honeycomb plate, the core thickness, and the cell seize.

The effect of the materials on the natural frequency of a honeycomb plate will be discussed in following sub-sections.

A. Effect of the core thickness

This section performs a geometrical analysis of the honeycomb panel vibration behaviour for the clumped free boundary conditions (CFFF). The three-dimensional geometrical model of the sandwich plate, given in figure 11 is analyzed by the finite element method using the Patran/Nastran software. The dimensions of the sandwich panel are following: the panel is 302 mm length and a 183 mm width, with h and t are the thicknesses of the core and the skins respectively, H is the total thickness of the plate, the size of the unit cells is l=2mm. Figure 10 presents the honeycomb finite element model mesh. The materials used are given in table V.

TABLE V
MATERIALS PROPERTIES

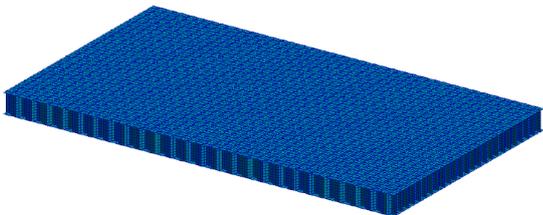| Core, Skins (Aluminium)	E (MPa)	ρ (g/mm3)	ν
	72000	0.0028	0.33
Titanium core	120000	0.0045	0.3
Epoxy carbon	143000	0.0016	0.33

Fig. 11 Mesh of a honeycomb panel

By changing the core/skins thickness ratio and keeping the same material properties. The numerical results obtained presented on figure 12 illustrate the variation of the frequencies versus thicknesses of the honeycomb core for the first four modes, the material is aluminium. It is clear that the frequencies increase with the increase thickness of the core (h) for bending and torsional modes, small decrease in lateral mode.
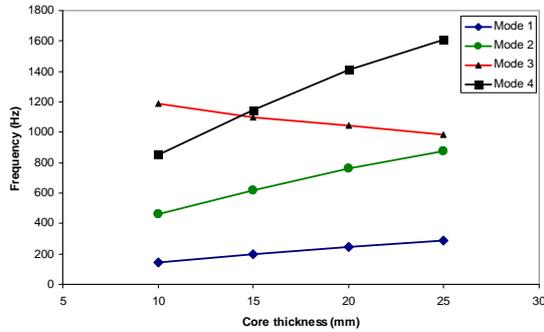

Fig. 12 Variation of frequency according to core thickness

B. Change of material

In this sub-section, considering the change of material, the analyses are discussed about the Titanium, Aluminum, and Epoxy-Carbon for honeycomb plate core materials.

Figure 13 shows the variation of the frequencies of the first four modes according to different materials core, which the following dimensions of the plate: h=15mm, t=1mm, l=2mm. It is clear that the change of material lead to a progressive increase in the frequencies, for these reasons the material selection plays a vital role in the total cost, weight, and lifetime of the spacecraft. Some important considerations while selecting a material are thermal conductivity, strength, stiffness, ductility, and corrosion resistance.
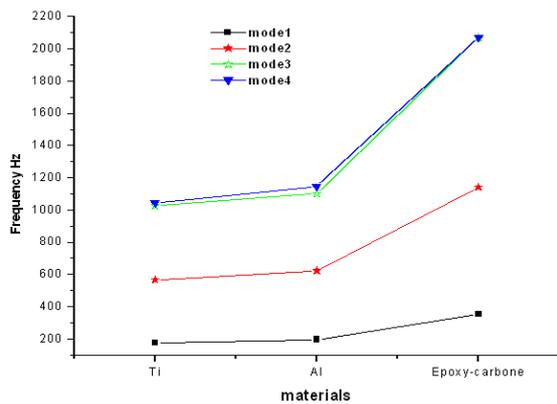

Fig. 13 Frequency variation according to a change of materials

C. Skins thickness effect

The influence of the skins thickness on the natural frequency is considered in this sub-section. The thicknesses of the skins are used as follows: 0.5mm, 1mm, 1.5mm, and 2mm.

Figure 14 shows the variation of the frequencies according to various thicknesses of the skins for the first four modes, we

used a plate with a thickness h=15mm and the cell size of l=2 mm. According to the results obtained, we note that the skins thickness affects on the Eigen frequencies values, so the thickness of the face layer is a key design parameter for the stiffness of the honeycomb plate.
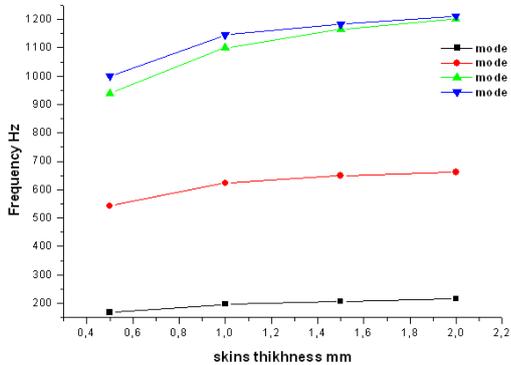

Fig. 14 Frequency variation according to skins thickness

D. Effect of the cell size

The influence of the cell size on the natural frequency is now discussed. The cells sizes are used as follows: 2mm, 3mm, 3.5mm, and 4mm.

Figure 15 illustrates the variation of the frequencies according to different cell size of the plate for the four first modes, the dimensions used in this section are: h=15mm, t = 1mm, according to the results, we note that the size of the cell does not have a major influence on the frequencies because the increase in the size of the cell will cause a reduction in the weight, and the density of the material which leave the results almost unchanged.
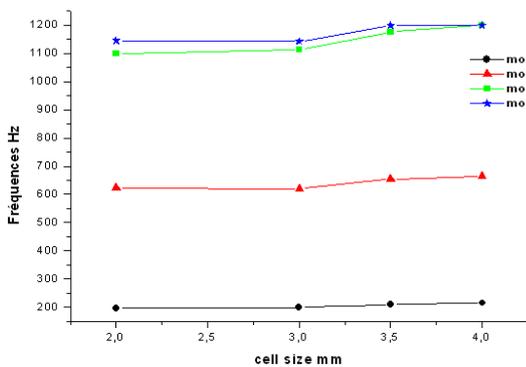

Fig. 15 Frequency variation according to cell size

E. Dynamic response results

A number of design parameters may affect the dynamic frequency response of the honeycomb plate, e.g. the skins thickness of a honeycomb plate, the core thickness, and the cell seize. The effect of the materials on the dynamic frequency response of a honeycomb plate will be discussed in this sub-section.

A three-dimensional geometrical model of the sandwich honeycomb plate is illustrated on the figure 16 and 17. The length plate is of a=302 mm and the width is about b=173 mm,

the upper and lower facing plates have the same thickness (HP=1 mm) while the core's thickness (Hc) may differ from the facing plates. l=2mm and t=0.1 are the cell size and wall cell thickness respectively, the model is under the nodal loading force F and its amplitude is about 10 N, the dumping factor is "c =0.05", it is located at the tip center of the plate.
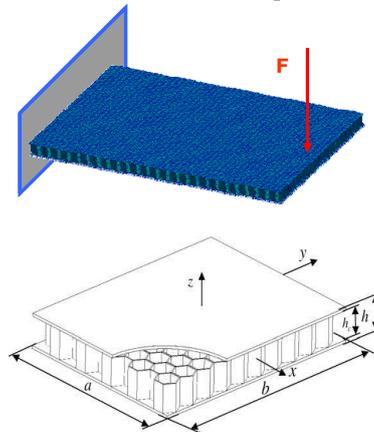

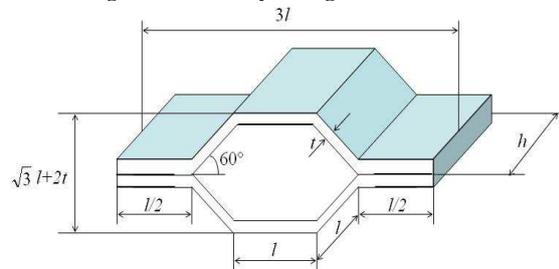Fig. 16 CFFF honeycomb geometrical model

Fig. 17 Unit cell geometry

The frequency range of investigation is here extended to the 0–2000 Hz.

Figure 18 represents the variation of the frequencies according to the honeycomb core thickness (Hc). We can see, that the increase of honeycomb core thickness Hc influences at the same time, on the values of displacements and the frequency response; which explains the major role of the Core for damping process of the honeycomb sandwich structures, by absorbing the vibration shocks.

Figure 19 show that the nature of materials is one element to control the frequency response of honeycomb plate, ant it indicate that materials proportion is one of the factors that dominates the vibration, as a consequent of its rigidity; The results shows a net changes of amplitude peak and frequency response ranges, the frequency response ratio arise by 51% for the first mode when the Young modulus ration E1/E2 = 50%, in addition, the displacement regress by 53%. We can say that, all vibration modes are influenced by the stiffness of honeycomb plate.

Figure 20 show that in the first mode (flexural mode) the effect cell size is not apparent but in the second and third mode the results shows the raising of the frequency response and decreasing of displacement, the difference of frequency of the

second mode is about 2.5% and 7.4 % when arising the cell size l=4mm and 8 mm respectively. Therefore, the binding and the transverse modes are the most affected by cell size difference.

It is clear in figure 21 that the amplitude of displacement decreases with the value of damping "c" becomes more significant without any change of the values of free modes from the plate; damping influences displacement decrease the risk of damage by it, which gives the importance of this factor for the optimization of vibrations.

In the figure 22 the first bending mode is not affected by the increasing of the thickness faces; however, there is a decreasing of displacement. However for the second and third mode the amount of frequency response rise when the faces thickness become thinner (2, 1.5, 1, 0.5 mm respectively), therefore the displacement increase by the decreasing of faces thickness.
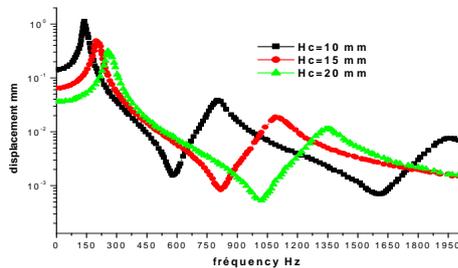

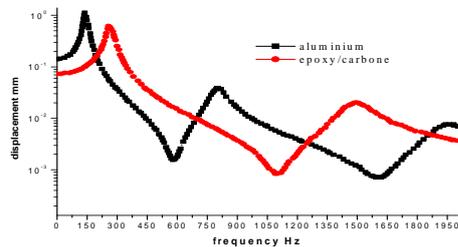Fig. 18 Frequency vs displacement for different core thickness (Hc)

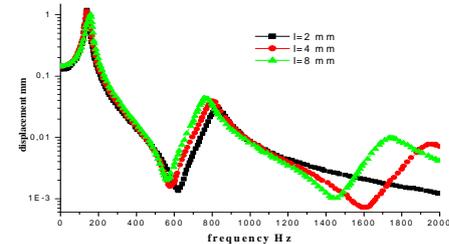Fig. 19 Frequency variation according to a change of materials
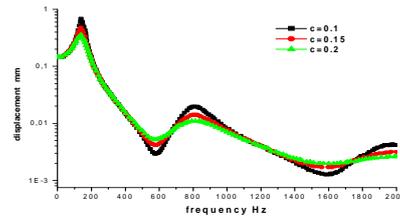

Fig. 20 Frequency responses according to cell size

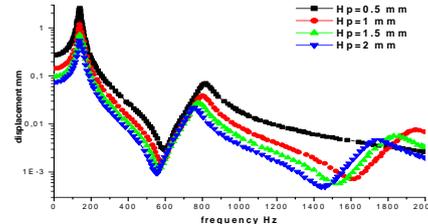Fig. 21 Frequency responses according to dumping coefficient

Fig. 22 Displacement response vs. frequency according to the face-sheet thickness (Hf)

F. Results and Discussion

The effects of design parameters, such as the skins thickness of a honeycomb plate, the core thickness, the cell seize, and the different core materials, have been discussed. Thus, from the results presented in Table 6, the following conclusions can be done:

- The frequencies are directly proportional to the thickness of the core for bending and torsional modes, slight decrease in lateral mode.
- The change of material lead to a progressive increase in the frequencies and the composite honeycomb plate has a stronger stiffness than the general honeycomb plate.
- The skins thickness impact the Eigen frequencies values, thus the thickness of the face layer is a key design parameter for the stiffness of the honeycomb plate.
- The size of the cell does not have a major influence on the frequencies.
- The increase of the cell's size causes a reduction in the weight, and consequently the density of the material making the results almost unchanged.
- On the dynamic frequency response, all geometrics parameters are crucial to optimize the vibration of sandwiches honeycomb structures.

V. CONCLUSION

The comparison between simulated and experimental results shows that FEM models are well suited for calculating the frequencies modes of different honeycombs plate designs. The maximum difference obtained is less than 4% in the first and the second frequency range and less than 10% for the third frequency. The results of the equivalent model presented in this analysis are obtained with good accuracy which presents an efficient approach during the development of the satellite structures leading into the reduction of the cost and the time of analysis.

The frequency dependence of damping was analyzed for honeycomb sandwich plate. The case of "CFFF" plates had been studied and the effects of thickness core, cell size, materials nature and dumping coefficient are highlighted.

The different results obtained in this paper are hopeful and show that the geometry and the type of material have an effect on the value of the honeycomb plate modal frequencies. This work can give a hint for the researchers in the design of the honeycomb sandwich structure.

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