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STRUCTURAL DYNAMICS CHARACTERIZATION OF METALLIC HONEYCOMB SANDWICH PANELS FOR AEROSPACE APPLICATIONS

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Abstract. *This paper presents experimental modal analyses performed on rectangular panels made from a HexWeb HIII aluminum honeycomb core and 2024 T3 aluminum face plates, which will be used in the main structure of the Brazilian Geostationary Satellite. This type of structure is commonly used in the aerospace industry because of its low density and high mechanical strength. Tests were conducted using an impact hammer for exciting the structure and a Laser Vibrometer to measure the responses. Data acquisition was performed using the Labview / NI CompaqDAQ platform. Frequency Response Functions (FRFs) were generated and analyzed through routines elaborated in MATLAB software. Finite element models were also developed using the ANSYS APDL software. Based on the experimental data, numerical models were adjusted, improving the theoretical-experimental correlation.*

Keywords: *modal analysis, honeycomb, Frequency Response Functions, aerospace structures.*

1. INTRODUCTION

Honeycomb sandwich panels are composite structures consisting of two thin laminated plates and a core composed of a cellular set of structures in the hexagonal format and made of composite material. This configuration allows these structures to have reduced weight, high mechanical strength, high thermal and acoustic insulation capacity, fire resistance and high damping coefficient. (Portela *et. al.*, 2010). These properties made the use of these panels in the aerospace industry more and more common.

Because it is a material commonly used in aerospace engineering, understanding its dynamic properties is of great importance, since such properties are directly related to the integrity of the structure (Ewins, 2000). The dynamic study of a structure occurs through the identification of modal parameters such as natural frequencies, damping factor and natural modes of vibration (Soeiro, 2001). It can be realized by correlating experimental modal analysis and finite element numerical models.

The modal test is an experimental procedure which allows obtaining a mathematical description of the dynamic behavior of a structure. This test consists of exciting the structure and investigating its vibrations by means of velocity, acceleration or displacement measurements. Most of the methods include also measuring the force excitation. The procedure is usually called Experimental Modal Analysis (Gevinski, 2014). One of the most common application of this type of analysis is the validation of numerical models (Ewins, 2000). Also, it allows to obtain dynamic properties that many times the numerical model is not able to identify, such as damping and nonlinearity characteristics (Gevinski, 2014). It can also contribute to determine if there are possible structure failures, determine critical damping, and indicate the useful life of the material (Craig and Kurdila, 2006).

The numerical model can be obtained by the Finite Element Method (FEM). This method consists of dividing structures into small grid-like regions, transforming continuous structures into discrete ones (Assan, 2003). This allows the behavior of the structure to be analyzed for each region individually instead of solving the whole common problem. The final result will be the combination of the results of each of these small regions.

Due to the complexity of the calculations, numerical solutions of this method involve intensive use of computational tools. However, this computational approach can cause problems in a project. It is not unusual that, due to the high level of automation and quality graphical representation of results available in most modern FE software, the designer might accept the results without knowing what is behind the computational tool. This fact can generate failures due to some error in the input values (Azevedo, 2003). Therefore, it is necessary to seek a good correlation between the numerical and experimental results in order to validate the finite element model.

For the analysis of the dynamic properties of four honeycomb panels, modal tests were performed correlating the results with a finite element model. As it deals with structures with reduced weight, the use of sensors that are coupled to the structure, as is the case of acceleration, can generate different data than expected, since both the natural frequencies and the damping factors are related to the mass of the system. For this reason, a laser vibrometer that measures the response of the system without having to have direct contact with the structure was used for the modal test of these structures.

2. METODOLOGY

Investigations were performed on the dynamic properties of four honeycomb panels whose faces and core are composed of aluminum alloy and have dimensions and characteristics presented in the table 1 below.

Table 1. Characteristics of the panels.

Thickness T (mm)	Direction L (mm)	Direction W (mm)	Honeycomb	Faces
10	670	300	HexWeb CRIII – Al 5056 – 1/4” – 0.001P (10P) - (MIL- C - 7438G or AMS -C-7438) – 9.4 mm	Al 2024 T3 NON CLAD (AMS QQA 250/4 and AMS 4037) – 0.3 mm
15	280	300	HexWeb CRIII – Al 5056 – 1/4” – 0.001P (10P) - (MIL-C-7438G or AMS -C-7438) – 14.4 mm	Al 2024 T3 NON CLAD (AMS QQA 250/4 and AMS 4037) – 0.3 mm
30	240	340	HexWeb CRIII – Al 5056 – 1/4” – 0.001P (10P) - (MIL-C-7438G or AMS -C-7438) – 29.4 mm	Al 2024 T3 NON CLAD (AMS QQA 250/4 and AMS 4037) – 0.3 mm
39.5	620	260	HexWeb CRIII – Al 5056 – 1/4” – 0.0015P (15P) - (MIL-C-7438G or AMS -C-7438) – 38.7 mm	Al 2024 T3 NON CLAD (AMS QQA 250/4 and AMS 4037) – 0.4 mm

The finite element model was developed using the ANSYS APDL software. It consists of three integrated surfaces, and the two outer ones simulate the faces with isotropic properties of aluminum alloy 2024 that has density of 2780 kg / m3, Young modulus of 0.33 and modulus of elasticity of 70GPa. The middle surface was designed to simulate the honeycomb structure. Its properties are orthotropic and the values supplied by the manufacturer were used. For panels of 10, 15 and 30mm thickness as they have the same honeycomb’s configuration, their properties were the same with a modulus in the L direction of 220MPa, in the W direction of 103MPa and a compression factor of 1.8MPa. The panel of 39.5mm thickness has a slightly different configuration from the others. Its properties are 345MPa of modulus in the L direction, 152MPa in the W direction and compression factor of 3.3MPa (Hexweb, 1999).

Mesh convergence tests were performed using two elements SHELL181 and SOLID185. SHELL181 is an indicated element for analyzing fine structures. It has six degrees of freedom on each of the nodes. The composite shell modeling uses first-order shear-strain theory. SOLID185 is a more suitable element for thicker structures. It has three degrees of freedom at each node and has plasticity, hyperelasticity, stress stiffening, creeping, large deflection, and great deformation capabilities. In addition to the ability to simulate elastoplastic deformations of almost incompressible materials and fully incompressible hyperelastic materials (Ansys). This convergence analysis aims to identify which elements are most appropriate for the numerical model of each panel, in addition to determining the number of points necessary for the values of the first three natural frequencies to remain constant up to the third decimal place.

For experimental modal analysis, a stand was used where the panels were fixed with nylon wires, with the objective of simulating the boundary conditions of the free-free type. The Portable Digital Vibration Vibrometer - PDV100 -

Polytec Vibrometer was used to measure the response of the panels, the PCB 086C03 impact hammer with nylon tip to excite the structures, the National Instrument eDAQ 9172 acquisition board and the LabView software to generate the data Fig.1. After obtaining the data, the MATLAB software and the EasyMod toolbox (Kouroussis et al, 2012) were used to analyze the data, generating the frequency response functions (FRFs) and obtaining the natural frequencies and modes of the panels.

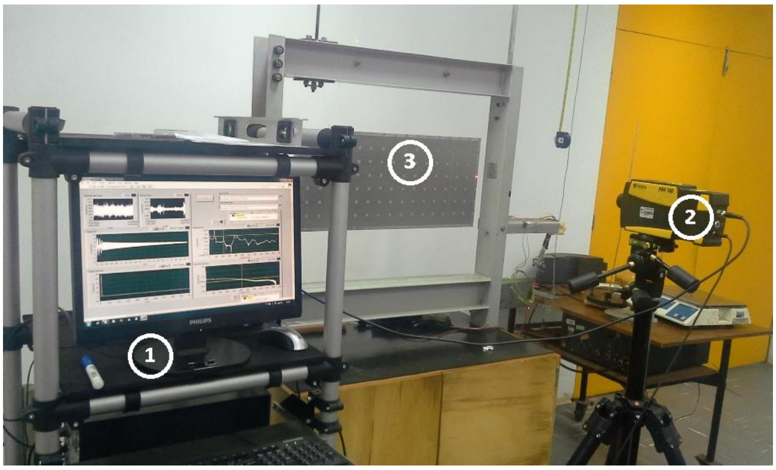
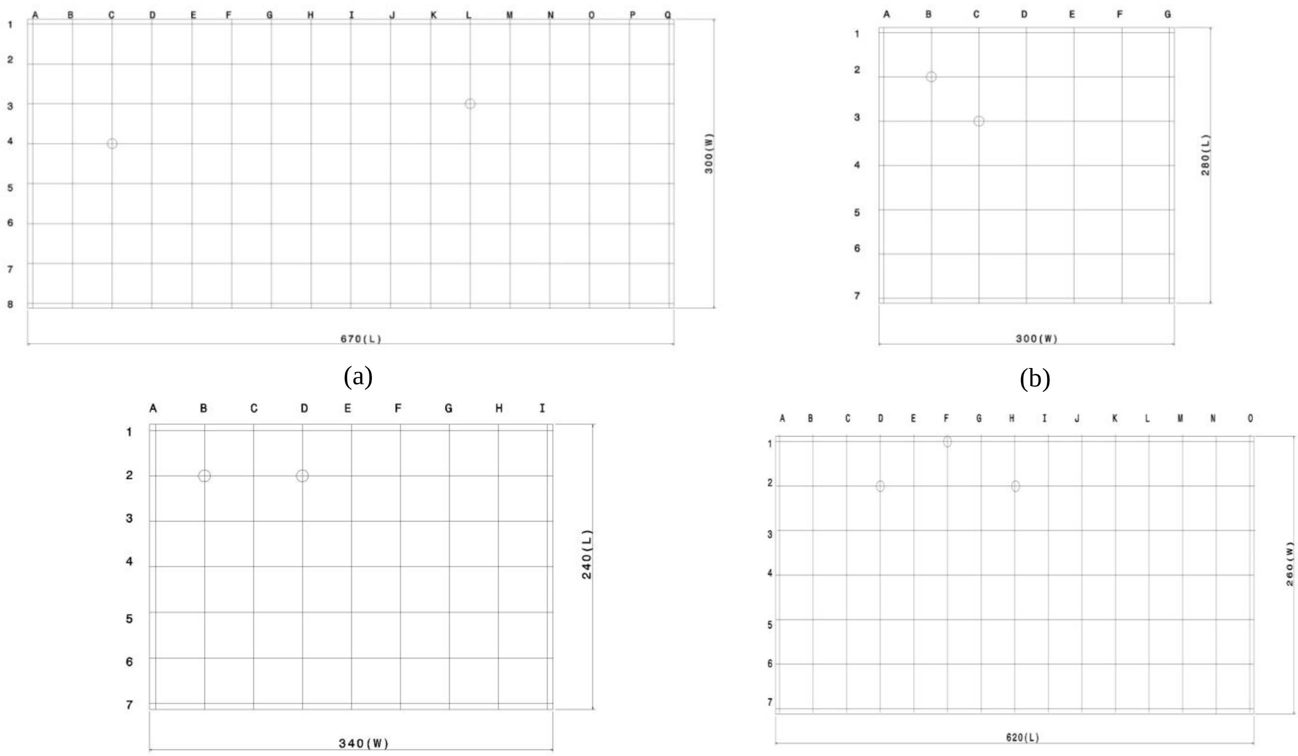


Figure 1. Modal test setup with (1) Labview data acquisition, (2) LDV Polytec 100 vibrometer, and (3) Honeycomb panel supported by nylon wires.

To generate the experimental data required for the extraction of modal parameters, each panel was discretized at measurement points in a mesh. A reflective adhesive tape was put at each point of this mesh to better reflect the laser and generate data with the least possible noise. To identify the points, it was standardized that each column would be named with letters of the alphabet following the order from left to right and each line was assigned a number in order from top to bottom Fig 2.



(a) (b) (c) (d)

Figure 2. Representation of the grid and excitation points (circle) in panels with thickness: (a) 10mm (b) 15mm (c) 30mm (d) 39.5mm

Three measurements were taken at each point. Each measurement generated a '.txt' file that contained information on strength, time, and speed. With these data, averages were performed to minimize the effects of eventual noise on the signals, and generated the FRF for each pair (excitation, response) point with MATLAB software. Then, with the help of the EasyMod toolbox, it was possible to analyze the data and obtain the modal characteristics of the panels.

3. RESULTS AND DISCUSSION

The convergence analysis contributed to analyze the best element to be used in each of the panels and the size of the mesh that can generate consistent data without spending a high processing time. It is expected that for the panels with smaller thickness the SHELL181 element is the most indicated and for the panels of greater thickness is the SOLID185.

3.1. Convergence Analysis

Convergence analysis was performed for the first three natural frequencies. When the values of the first three natural frequencies did not change until the third decimal place, it was considered that the mesh converged. However, for the presentation of the convergence, graphs were generated using the experimental values of SHELL181 and SOLID185 for the first natural frequency.

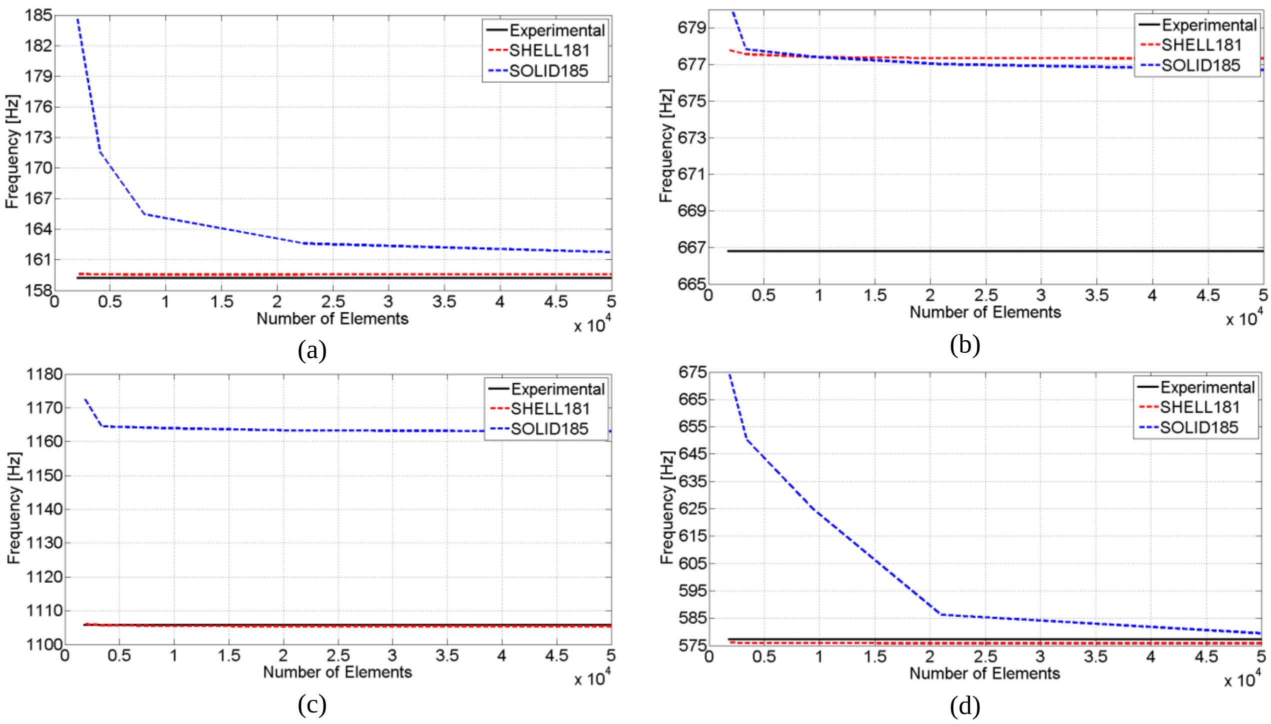


Figure 3. Convergence of mesh for the first natural frequency of the panel with thickness (a) 10mm, (b) 15mm, (c)30mm and (d) 39.5mm

In Figure 3 it is possible to observe that in all panels the SHELL181 showed better convergence than the SOLID185. Also, for the panels of 10mm Fig 3a, 30mm Fig 3b and 39.5mm Fig 3c of thickness SHELL181 presented results closer to the experimental. For the 15mm thickness, Fig. 3b, the SOLID185 presented results closer to the experimental, but it did not show convergence of results for a mesh of 50,000 elements

3.2. Experimental Modal Analysis

To identify the largest number of modes within the measuring range, more than one excitation point was chosen. With the help of the EasyMod toolbox it was possible to identify the natural frequencies, damping ratios and the natural modes of the panels.

For the panel with 10mm thickness, 6 modes were obtained up to 1kHz. Table 2 presents the results obtained in the experimental modal analysis of the 10mm panel.

Table 2. Experimental results of the panel with 10 mm thickness with excitation at points C4 and L3

Modes	Experimental Natural Frequency C4 Excitation (Hz)	Damping Ratio C4 (%)	Experimental Natural Frequency L3 Excitation (Hz)	Damping Ratio L3 (%)
1	159.2	1.34	159.8	1.39
2	201.6	1.84	201.3	1.65
3	430.4	2.32	433.4	0.65
4	728.0	0.78	727.9	0.76
5	759.0	0.71	757.9	0.92
6	854.2	1.70	852.4	1.08

For the panel of 15mm thickness the measuring range was extended to 2kHz in order to obtain a greater number of modes, because up to 1kHz it was only possible to obtain 1 mode. With this increase, 5 modes were identified for the 15mm panel. Table 3 presents the results of the 15 mm panel for 2-point excitation.

Table 3. Experimental results of the panel with 15 mm thickness with excitation at points C3 and B2

Modes	Experimental Natural Frequency C3 Excitation (Hz)	Damping Ratio C3 (%)	Experimental Natural Frequency B2 Excitation (Hz)	Damping Ratio B2 (%)
1	666.8	1.24	666.5	1.28
2	1031.7	0.45	1030.9	0.45
3	1335.1	1.35	1334.8	1.18
4	1604.7	1.77	1602.4	2.40
5	1694.7	2.15	1694.8	1.31

For the panel with 30mm thickness the measurement interval had to be applied to 4kHz which made it possible to identify 6 modes as shown in table 4. Two excitation points were chosen and this choice was necessary since for point B2 they were obtained 5 modes while for D2 6 modes were obtained. Also, for the fourth mode the point B2 has a lower damping ratio than the point D2, this may mean that the point B2 better excites this mode than the point D2.

Table 4. Experimental results of the panel with 30 mm thickness with excitation at points B2 and D2

Modes	Experimental Natural Frequency B2 Excitation (Hz)	Damping Ratio B2 (%)	Experimental Natural Frequency D2 Excitation (Hz)	Damping Ratio D2 (%)
1	1102.1	1.81	1105.7	1.55
2	1383.9	1.30	1384.2	1.64
3	2360.7	1.69	2341.4	1.90
4	2756.2	0.79	2723.4	4.47
5	3318.9	0.97	3321.9	0.26
6			3834.8	0.35

The panel with 39.5mm thickness was analyzed at a range of 2kHz. To analyze each mode adequately, excitations were performed at 3 different points. This allowed to obtain 4 modes as shown in table 5.

Table 5. Experimental results of the panel with 39.5 mm thickness with excitation at points F1, H2 and D2

Modes	Experimental Natural Frequency F1 Excitation (Hz)	Damping Ratio F1 (%)	Experimental Natural Frequency H2 Excitation (Hz)	Damping Ratio H2 (%)	Experimental Natural Frequency D2 Excitation (Hz)	Damping Ratio D2 (%)
1	576.9	0.68	577.2	0.66	577.3	0.73
2	725.8	0.71	727.2	1.10	725.0	0.69
3	1468.7	1.19	1484.5	1.92		
4	1506.3	0.96	1508.2	0.61	1505.3	1.02

3.3. Experimental – Numerical Correlation

To correlate numerical and experimental results, it is necessary to compare natural frequencies and the corresponding natural modes. The EasyMod toolbox contributed to identify most of the modes obtained experimentally. Due to the complexity of the code it was not possible to identify all the modes clearly in order to compare with the numerical results. However, considering the identified modes it was possible to obtain a good correlation with the numerical modes.

The numerical model of the panel with 10mm thickness was generated using the SHELL181 element that presented results closer to the experimental one than the SOLID185. A total of 22,400 elements were used, which already showed convergence. Also, it was necessary to insert a density of 77.64 kg/m^3 for the honeycomb layer so that the weight approached the real one, which allowed the results to have a good correlation as it is presented in Fig 4.

For the panel with 10mm thickness it was possible to clearly identify up to 4 modes that showed a good correlation with the numerical. Considering the experimental frequencies as reference, the largest relative difference was 1.46%.

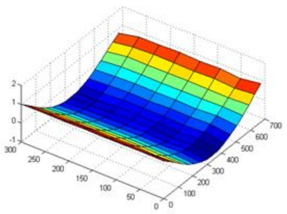
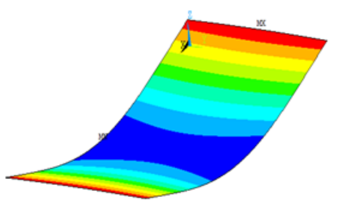
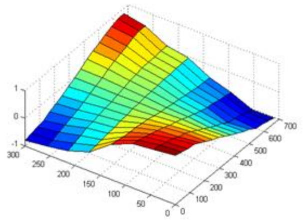
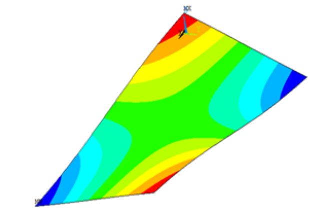
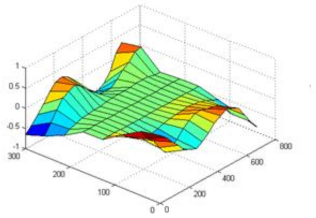
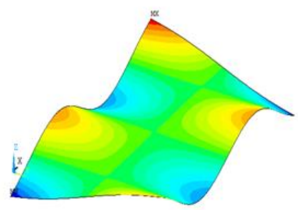
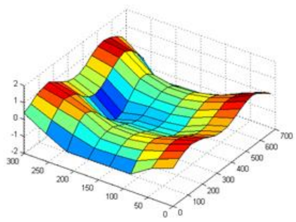
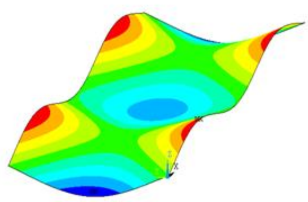
Experimental Frequency (Hz)	Experimental Mode	Numerical Mode	Numerical Frequency (Hz)
159.22			159.57
201.95			204.91
724.29			728.29
759.15			767.27

Figure 4. Numerical and experimental modes and frequencies to panel with 10mm thickness

The numerical model of the panel with 15mm thickness was generated using the SHELL181 element that presented convergence better than SOLID185. A total of 21,000 elements were used and the honeycomb's density of 70.18 kg/m^3. It was possible to identify 4 experimental modes Fig 5. These characteristics of the numerical model allowed to obtain results with relative differences of up to 1.77% considering the experimental frequency as reference.

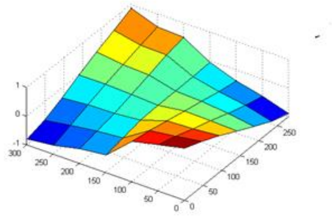
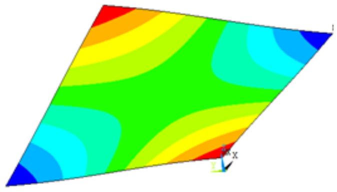
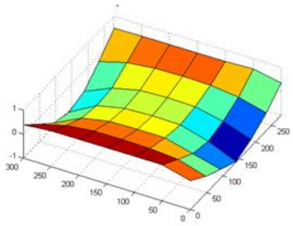
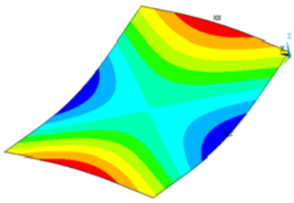
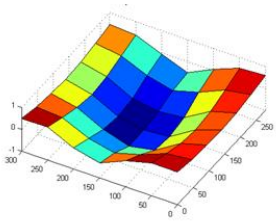
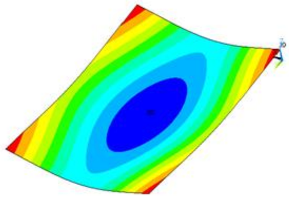
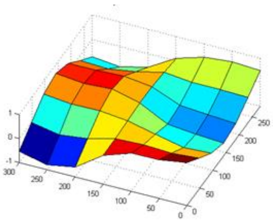
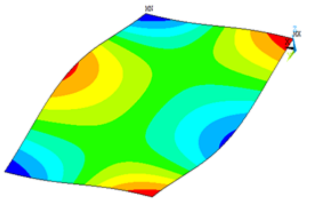
Experimental Frequency (Hz)	Experimental Mode	Numerical Mode	Numerical Frequency (Hz)
666.17			677.41
1031.50			1013.17
1335.90			1328.60
1604.57			1586.04

Figure 5. Numerical and experimental modes and frequencies to panel with 15mm thickness

It was expected that for the 30mm panel, using SOLID185 elements would present better results than the SHELL181 based model. However, SHELL181 proved to be more appropriate, with better results and convergence. A total of 20,400 elements were used, with the honeycomb's density of 72 kg / m^3. It was possible to identify 3 experimental modes that showed a good correlation between the numerical and experimental with difference of 2.24% Fig 6.

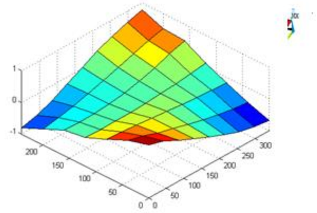
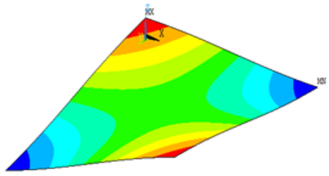
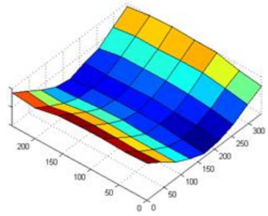
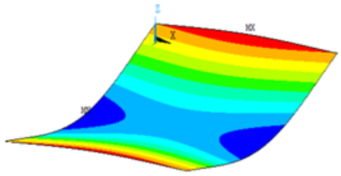
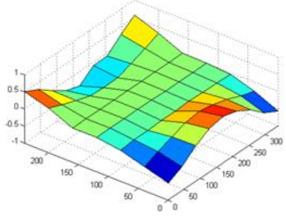
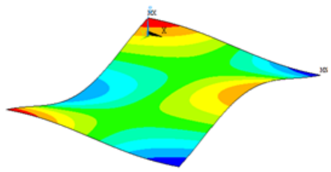
Experimental Frequency (Hz)	Experimental Mode	Numerical Mode	Numerical Frequency (Hz)
1099.74			1105.46
1381.33			1387.67
2351.78			2299.01

Figure 6. Numerical and experimental modes and frequencies to panel with 30mm thickness

For the panel with 39.5mm thickness the SHELL181 showed better results than the experimental data. A mesh with 18,000 elements was generated and the honeycomb density was 72.37kg / m. With this it was possible to correlate 3 experimental and numerical modes with a relative difference of 1.29% Fig 7.

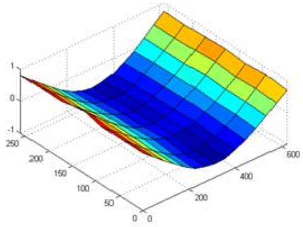
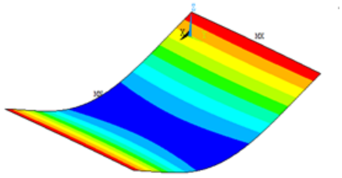
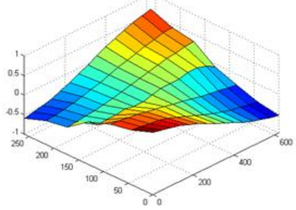
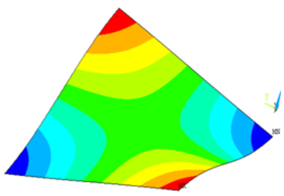
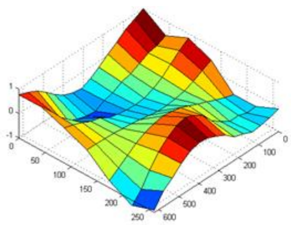
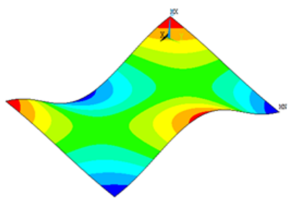
Experimental Frequency (Hz)	Experimental Mode	Numerical Mode	Numerical Frequency (Hz)
576.93			575.84
724.92			727.82
1504.16			1484.71

Figure 7. Numerical and experimental modes and frequencies to panel with 39.5mm thickness

4. CONCLUSION

Finite element analysis using the SHELL181 element proved to be more adequate than the SOLID185 for all the panels. Crucial to this was the convergence of the mesh. While the SHELL181 converged with up to 20,000 elements, SOLID185 showed convergence above 100,000 elements. This higher number of elements strongly influences the computational cost.

Numerical modeling using isotropic faces properties and orthotropic properties for honeycomb showed a good correlation with the experimental data. The properties that most affected the results were the modulus in the L and W direction and the compressive strength in the T direction. The modulus in the T direction and the compressive strength in the L and W directions did not significantly change the numerical results. Another important factor was the density of honeycomb. This contributed to the values of numerical frequencies approaching the experimental values. It was estimated using the mass of the panels and the approximate volume of honeycomb.

From the validation of the finite element model it will be possible to perform other analyses of these panels. Panels in aerospace engineering projects will have boundary conditions different from those applied in experimental modal analysis. Once the numerical model has been validated it will be possible to identify the dynamic behavior of the panels even when submitted to more realistic boundary conditions.

5. ACKNOWLEDGEMENTS

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