American Journal of Mechanical Engineering, 2016, Vol. 4, No. 7, 236-240
Available online at http://pubs.sciepub.com/ajme/4/7/1

DOI:10.12691/ajme-4-7-1

A Sensitivity Analysis of the Dynamic Behavior of Aluminium Honeycomb Sandwich Panels

Róbert Huňady*

Department of Applied Mechanics and Mechanical Engineering, Technical University of Košice, Faculty of Mechanical Engineering, Košice, Slovakia
*Corresponding author: robert.hunady@tuke.sk

Abstract Paper deals with an investigation of the dynamic behavior of aluminium honeycomb sandwich panels. The presented study is based on numerical modal analysis where the free vibrations of nine panels are analyzed in order to examine the influence of geometric parameters, such as thickness and height of the core, on their modal parameters. The analyzed panels have a rectangular shape and consist of two aluminum alloy face sheets of thickness 1 mm and hexagonal aluminum honeycomb core, of which height and wall thickness are varying. Additionally, the modal parameters of one single core were analyzed in dependence of wall thickness. The all FEM models were built and the simulations were executed in Abaqus/CAE. The results (natural frequencies and mode shapes) are processed in tabular and graphical form.

Keywords: *FEA, modal analysis, aluminium honeycomb panel*

Cite This Article: Róbert Huňady, “A Sensitivity Analysis of the Dynamic Behavior of Aluminium Honeycomb Sandwich Panels.” *American Journal of Mechanical Engineering*, vol. 4, no. 7 (2016): 236-240. doi: 10.12691/ajme-4-7-1.

1. Introduction

The sandwich constructions are one of the most valued structural engineering innovations developed by the composites industry. They were invented in 1940’s. Used extensively in aerospace, automotive and many other industries, due to the various advantages it offers in terms of weight-savings and high stiffness [1]. They provide the following key benefits over conventional materials:

- very low weight,
- high stiffness,
- durability,
- production cost savings.

The sandwich construction generally consists of two laminated face sheets on both sides and a core in the middle. Core can be made of cellular foam, trusses or honeycombs. Conventional hexagonal honeycombs are typically employed for the cores of the sandwich construction. The honeycomb sandwich constructions, made as panels, can be considered as structural materials with minimal density and relative high out-of-plane compression properties and out-of-plane shear properties. An advantage for these panels is their fatigue resistance. The reason for the greater fatigue resistance is that the face sheets are continuously bonded to the core and therefore they do not have stress concentrations [2]. The base concept of sandwich construction is to use thick and light core bonded with thin, dense strong sheet materials. Each component is relatively weak and flexible but provides a stiff, strong and lightweight structure when working together as a composite structure.

There are several different materials that are commonly used in sandwich construction, but in quite recent times the interest in new materials such as fibre-glass laminate, carbon fibre composite, solid foams, etc., has increased [3]. The choice of face sheet and core materials depends heavily on the performance of the materials in the intended operational environment. For example they have to be more resistant against fire, they need to withstand bigger loads or to be able to absorb bigger vibrations, etc. Also the cost and the effectiveness are important aspects of the manufacturing process [4]. The most commonly used honeycomb cores are made of aluminium or impregnated glass or aramid fiber mats. The face sheets can be made of aluminium, steel, melamine, and other resin plates. Figure 1 shows the usual structure of an aluminium honeycomb panel. These panels possess the highest strength/weight ratio, unmatched by any other structural materials. Their mechanical properties are controlled by foil thickness, cell size and cell shape.

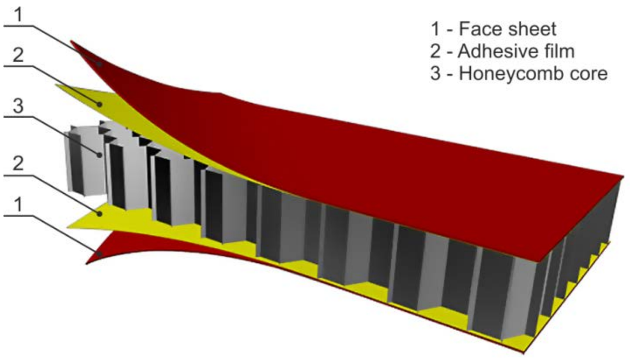

Figure 1. Structure of an aluminium honeycomb panel

2. Literature Review

The research of material properties and dynamic behavior of honeycomb sandwich structural is of crucial importance in efficient mechanical analysis. This is one of the actual topics in material science. There exist many works dealt with this issue.

An analytical solution for the dynamic response of honeycomb sandwich material is not available but equivalency in the form of a homogenous orthotropic thick plate can be formulated. Boudjemai et. al [5] studied modal properties of honeycomb panels used in the satellites structural design. They performed experimental modal analysis of the real panel and FEM analysis of an equivalent model of which properties were determined by identifying membrane and bending stiffness of the panel to those of an isotropic plate. The comparison between simulated and experimental results showed that equivalent FEM model is well suited for calculating the frequencies modes of different honeycombs plate designs.

Vlach and Pomp [4] analyzed the behaviour of aluminium sandwich specimens by using the modal analysis and the experimental determination of the samples damping properties. They investigated the influence of the main directions of anisotropy and the different panel's thicknesses on the natural frequencies. The results of experiments were compared with the theoretical calculations and finite element method (FEM) simulation results. The results obtained from FEA were close to the experimentally measured data, but still with a deviation due to geometrical irregularities (twisting caused during the manufacturing) between the real specimens and idealistic FEM model. The results showed the specimens with small thickness were more twisted and therefore they produce bigger error in the measured data.

Jiang et al. [6] proposed another approach on determining the equivalent parameters of honeycomb core. The method consists in determining the initial value of equivalent elastic parameters by using an analytical method; discretizing the elasto-dynamic problem by using the finite element method; selecting the sensitive and inaccurate parameters by analyzing the internal architecture and the relative sensitivity analysis of natural frequencies with respect to equivalent parameters; transforming the parameter identification into an optimization problem with constraints; and estimating the parameters by minimizing the objective function of vibration test and numerical modal data. Then, the inaccurate elastic constants of the honeycomb core will be accurately and efficiently determined.

3. Description of the Models

The model of an analyzed aluminium honeycomb panel consists of the hexagonal aluminum core and two aluminium sheet plates. The adhesive layers were not included in the analyses. The material properties of the individual parts are listed in Table 1.

The face sheet plates are rectangular shaped with dimensions of 1000 x 199.19 mm (See Figure 2). The thickness of the plates is 1 mm. The core consists of 2300 hexagonal cells. Their arrangement and dimensions are obvious from Figure 3. The height *h* of core, as well as the thickness *t*, depends on a given panel variant. All the variants and their weight are listed in Table 2.

Geometry of the core was generated in SolidWorks due to its complexity. Mesh of the core was created in NX Nastran and subsequently imported as orphan mesh into Abaqus/CAE where the rest of the models was built and solved. Quadrilateral shell elements S4R with characteristic length of 5 mm were used. As an example, the resultant finite element model of the variant P20 is shown in Figure 4.

Table 1. Material properties of the model parts

Part	Material	Young's modulus	Poisson ratio	Density
Core	Al 3003 Alloy	69 GPa	0.33	2.8 g/cm^3
Face sheets	Al 5050 Alloy	70 GPa	0.33	2.7 g/cm^3

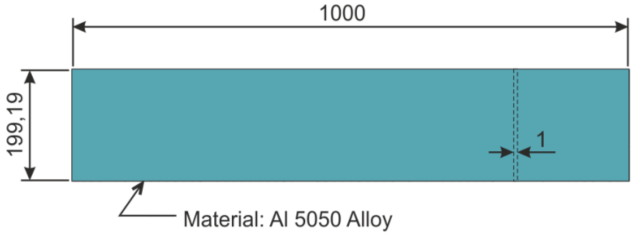

Figure 2. Dimensions of the face sheet plates

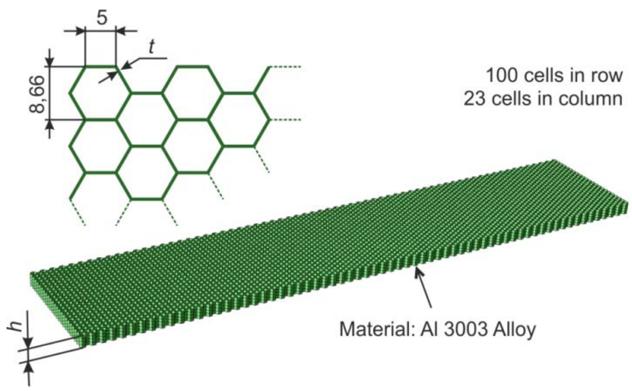

Figure 3. Dimensions and arrangement of the core

Table 2. The model variants

	Weight of the model [kg]		
Variant	*t* = 0.03 mm	*t* = 0.05 mm	*t* = 0.07 mm
P10	1.114	1. 141	1.167
P20	1. 154	1. 206	1. 258
P30	1.193	1.271	1.349
C30 core only	0.117	0.195	0.273

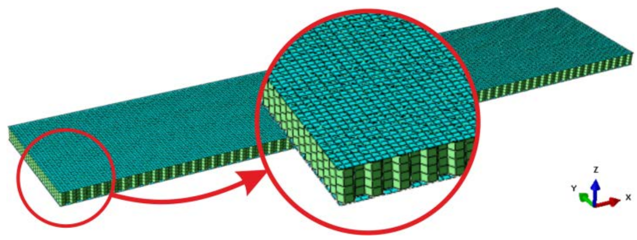

Figure 4. Finite element model of the variant P20

4. Finite Element Analysis

In Abaqus/CAE, the numerical modal analysis of twelve models was performed for the purpose of determination their natural frequencies and mode shapes. In the all cases, the Lanczos eigensolver was used. As all the models were considered as free-supported, the first 6 rigid-body modes were ignored, so only the first 15 flexible modes were taken into account.

4.1. Modal Analysis of the Core C30

In the first phase, the modal analysis of the single core C30 was performed for three different thicknesses of its wall. The computed natural frequencies are listed in Table 3. The shapes of vibration, corresponding to the individual modes, are shown in Figure 5.

In general, natural frequency values increase with the increasing thickness of the core wall. Simultaneously, the sequence of some higher mode shapes changes too. This change is caused by the change of stiffness in the individual directions. Figure 6 shows the frequency shift of the natural frequencies in dependence on the thickness *t*.

4.2. Modal Analysis of the Panels P10, P20 and P30

In the second phase, the modal parameters of the individual aluminium honeycomb panels were determined. The computed natural frequencies are listed in Table 4, Table 5 and Table 6. The vibration shapes, corresponding to the individual modes, are shown in Figure 7.

Diagrams in Figure 8, Figure 9 and Figure 10 show the shift of natural frequencies of the panel P10, P20 and P30, in dependence on the thickness of the core wall, respectively.

The shift of the natural frequencies depending on the height of the core is graphically presented in Figure 11, Figure 12 and Figure 13.

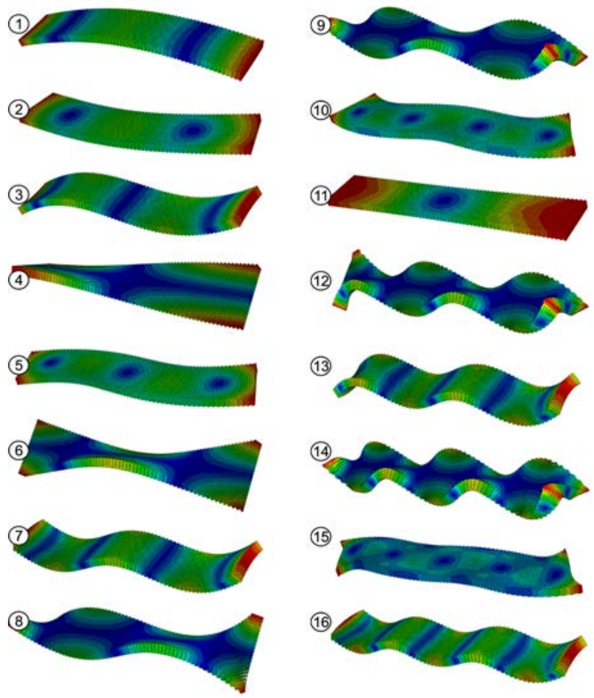

Figure 5. Core mode shapes

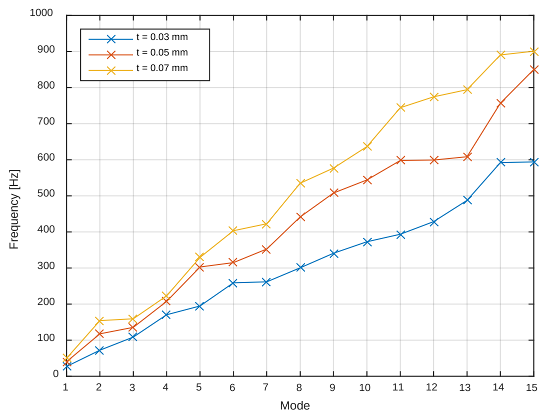

Figure 6. Natural frequency shift of the core

Table 3. Frequencies of the modes and corresponding shape number according to Figure 5

C30	Natural frequencies		
Mode	*t* = 0.03 mm	*t* = 0.05 mm	*t* = 0.07 mm
1.	27.125 Hz (1)	40.321 Hz (1)	49.796 Hz (1)
2.	72.313 Hz (2)	117.68 Hz (2)	154.26 Hz (3)
3.	108.59 Hz (3)	135.55 Hz (3)	159.33 Hz (2)
4.	170.50 Hz (4)	207.50 Hz (4)	222.86 Hz (4)
5.	194.80 Hz (5)	302.89 Hz (7)	329.47 Hz (7)
6.	259.02 Hz (6)	314.95 Hz (5)	403.35 Hz (6)
7.	261.72 Hz (7)	352.65 Hz (6)	422.68 Hz (5)
8.	300.55 Hz (8)	441.50 Hz (8)	534.95 Hz (8)
9.	340.93 Hz (9)	507.92 Hz (9)	577.02 Hz (13)
10.	372.80 Hz (10)	544.04 Hz (13)	636.43 Hz (9)
11.	393.57 Hz (11)	597.97 Hz (10)	744.21 Hz (12)
12.	428.56 Hz (12)	599.23 Hz (12)	774.46 Hz (11)
13.	487.39 Hz (13)	608.34 Hz (11)	794.18 Hz (10)
14.	592.10 Hz (14)	756.80 Hz (14)	890.54 Hz (16)
15.	594.03 Hz (15)	850.03 Hz (16)	900.83 Hz (14)

Table 4. Natural frequencies of the panel P10

P10	Frequencies of the modes and corresponding shape number according to Figure 7		
Mode	*t* = 0.03 mm	*t* = 0.05 mm	*t* = 0.07 mm
1.	95.343 Hz (1)	95.213 Hz (1)	94.618 Hz (1)
2.	230.77 Hz (2)	243.49 Hz (2)	248.85 Hz (2)
3.	248.37 Hz (3)	253.58 Hz (3)	254.57 Hz (3)
4.	450.62 Hz (4)	472.83 Hz (4)	481.11 Hz (4)
5.	458.33 Hz (5)	488.26 Hz (5)	501.65 Hz (5)
6.	680.60 Hz (6)	734.42 Hz (6)	758.55 Hz (6)
7.	681.02 Hz (7)	735.80 Hz (7)	761.94 Hz (7)
8.	899.30 Hz (8)	894.03 Hz (9)	885.55 Hz (9)
9.	903.00 Hz (9)	987.86 Hz (8)	1032.6 Hz (8)
10.	924.19 Hz (10)	1023.0 Hz (10)	1072.3 Hz (10)
11.	1115.2 Hz (11)	1246.1 Hz (11)	1315.8 Hz (11)
12.	1172.9 Hz (12)	1326.8 Hz (12)	1409.8 Hz (12)
13.	1330.8 Hz (13)	1511.5 Hz (13)	1612.3 Hz (13)
14.	1421.9 Hz (14)	1636.6 Hz (14)	1759.0 Hz (14)
15.	1547.8 Hz (15)	1784.1 Hz (15)	1921.6 Hz (15)

Table 5. Natural frequencies of the panel P20

P20	Frequencies of the modes and corresponding shape number according to Figure 7		
Mode	*t* = 0.03 mm	*t* = 0.05 mm	*t* = 0.07 mm
1.	174.56 Hz (1)	174.00 Hz (1)	171.96 Hz (1)
2.	376.52 Hz (2)	406.93 Hz (2)	419.84 Hz (2)
3.	432.73 Hz (3)	447.88 Hz (3)	450.67 Hz (3)
4.	738.04 Hz (4)	801.94 Hz (5)	823.97 Hz (5)
5.	745.30 Hz (5)	805.43 Hz (4)	836.21 Hz (4)
6.	888.83 Hz (6)	871.86 Hz (6)	856.35 Hz (6)
7.	1075.2 Hz (7)	1191.0 Hz (8)	1248.0 Hz (8)
8.	1075.2 Hz (8)	1196.4 Hz (7)	1254.3 Hz (7)
9.	1387.2 Hz (9)	1563.9 Hz (9)	1656.9 Hz (9)
10.	1407.6 Hz (10)	1607.9 Hz (10)	1715.2 Hz (10)
11.	1680.2 Hz (11)	1927.8 Hz (11)	2011.2 Hz (15)
12.	1736.7 Hz (12)	2023.5 Hz (12)	2065.7 Hz (11)
13.	1962.6 Hz (13)	2045.8 Hz (15)	2189.3 Hz (12)
14.	2060.8 Hz (14)	2287.4 Hz (13)	2361.7 Hz (16)
15.	2084.1 Hz (15)	2400.9 Hz (16)	2477.5 Hz (13)

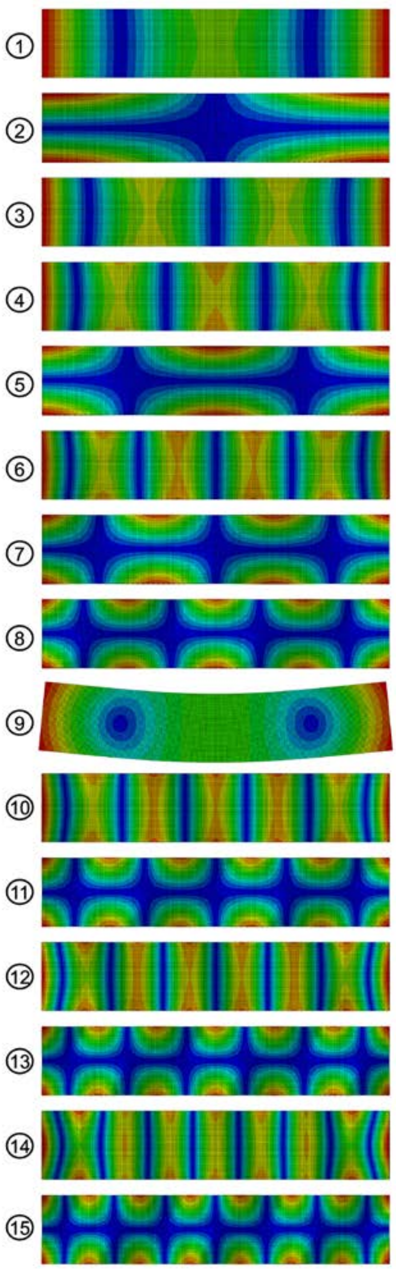

Figure 7. Panel mode shapes

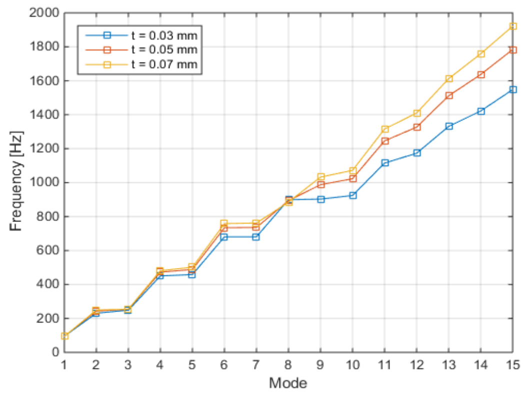

Figure 8. Natural frequency shift of the panel P10 depending up the thickness of the core wall

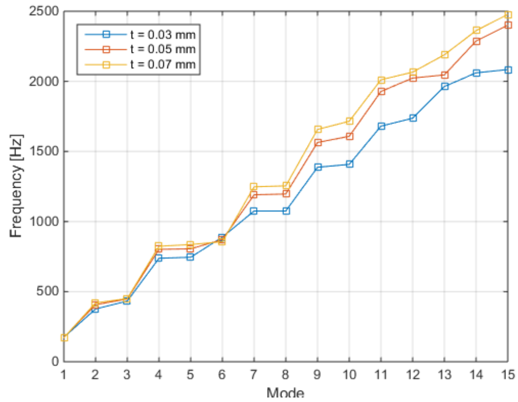

Figure 9. Natural frequency shift of the panel P20 depending up the thickness of the core wall

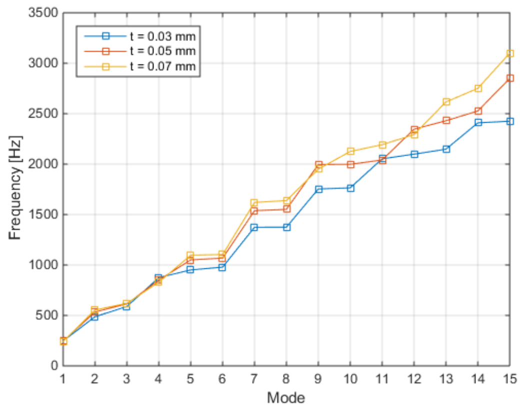

Figure 10. Natural frequency shift of the panel P30 depending up the thickness of the core wall

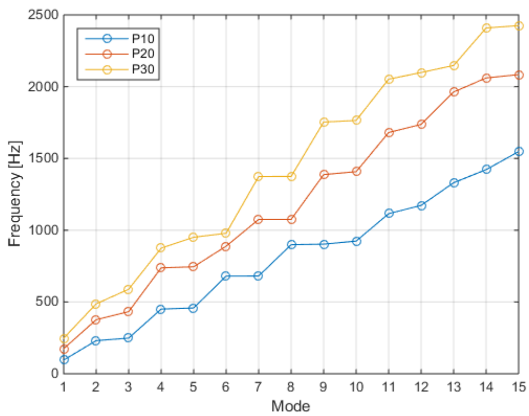

Figure 11. Natural frequency shift of the panels with the core wall thickness of 0.03 mm

Table 6. Natural frequencies of the panel P30

P30	Frequencies of the modes and corresponding shape number according to Figure 7		
Mode	*t* = 0.03 mm	*t* = 0.05 mm	*t* = 0.07 mm
1.	247.63 Hz (1)	246.47 Hz (1)	242.50 Hz (1)
2.	487.16 Hz (2)	534.19 Hz (2)	554.59 Hz (2)
3.	588.94 Hz (3)	615.43 Hz (3)	620.29 Hz (3)
4.	875.09 Hz (4)	851.04 Hz (4)	829.72 Hz (4)
5.	951.05 Hz (5)	1050.7 Hz (5)	1097.0 Hz (5)
6.	978.68 Hz (6)	1068.2 Hz (6)	1103.8 Hz (6)
7.	1374.0 Hz (7)	1538.5 Hz (8)	1620.6 Hz (8)
8.	1374.5 Hz (8)	1551.2 Hz (7)	1638.3 Hz (7)
9.	1753.5 Hz (9)	1996.0 Hz (9)	1950.5 Hz (11)
10.	1764.6 Hz (10)	1998.0 Hz (11)	2125.4 Hz (9)
11.	2052.2 Hz (11)	2040.8 Hz (10)	2192.7 Hz (10)
12.	2098.7 Hz (12)	2346.1 Hz (15)	2292.7 Hz (15)
13.	2147.6 Hz (13)	2429.8 Hz (13)	2616.3 Hz (13)
14.	2409.1 Hz (14)	2526.7 Hz (14)	2750.7 Hz (14)
15.	2425.4 Hz (15)	2849.3 Hz (16)	3099.8 Hz (16)

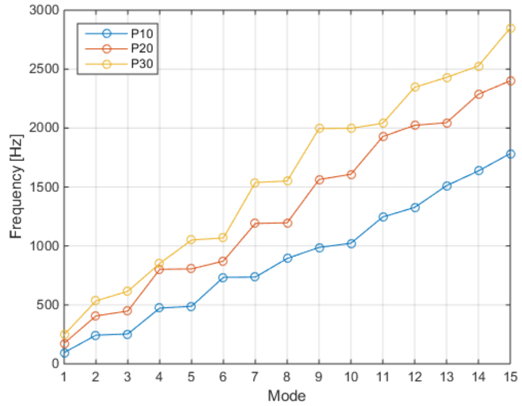

Figure 12. Natural frequency shift of the panels with the core wall thickness of 0.05 mm

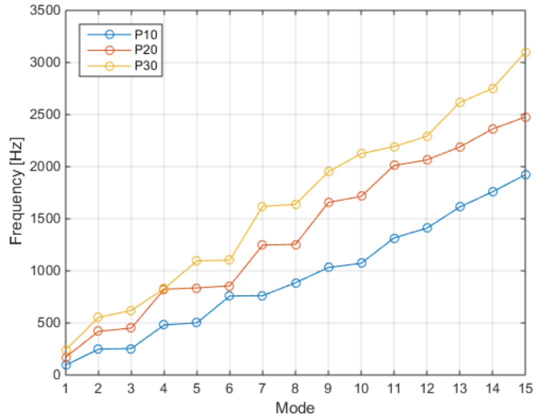

Figure 13. Natural frequency shift of the panels with the core wall thickness of 0.05 mm

5. Conclusion

In the paper, the numerical modal analysis was used for the investigation of dynamic behavior of aluminium honeycomb sandwich panels. The sensitivity analysis was focused to determine the influence of two geometric parameters, the thickness and height of the hexagonal core cell, on the natural frequencies and mode shapes of the panels. The results show the shift of natural frequency values with varying both parameters. In general, the frequency increases with the increasing thickness of the core wall as well as with its height. The height of core has a greater impact than the thickness. It has also been found that the order of some mode shapes can be changed due to change of stiffness in different directions.

Acknowledgements

The author gratefully acknowledge the support of the Slovak Grant Agency VEGA – Grant No. 1/0393/14.

References

[1] Sypeck, D. J. and Wadley, H. N. G., “Cellular Metal Truss Core Sandwich Structures”, Advanced Engineering Materials, 4 (10), pp. 759-764, 2002.

[2] Bitzer, T., Honeycomb Technology: Materials, Design, Manufacturing, Applications and Testing, Chapman & Hall, pp. 80-120, 1997.

[3] Lei, H., Composite Sandwich Structures with Honeycomb Core subjected to Impact. All Theses. Paper 1547, 2012.

[4] Vlach, J., Pomp, N., “Modal analysis of honeycomb panels and the influence of boundary conditions on the calculation”, Comat 2014, Pilsen, Czech Republic, 2014.

[5] Boudjemai, A., Amri, R., Mankour, A., Salem, H., Bouanane, M.H., Boutchicha, D., “Modal analysis and testing of hexagonal honeycomb plates used for satellite structural design”, Material design, 35, pp. 266–275, 2012.

[6] Dong J., Dahai Z., Qingguo F., Shaoqing W., “An approach on identification of equivalent properties of honeycomb core using experimental modal data”, Finite Elements in Analysis and Design, 90 (1), pp. 84-92, 2014.