11th International Conference on Vibration Problems
Z. Dimitrovová *et al.* (eds.)
Lisbon, Portugal, 9-12 September 2013

MODAL ANALYSIS AND DAMPING PROPERTIES OF HONEYCOMB PANELS

Jarmil Vlach*[1], Norbert Pomp[2], Radek Zbončák[2]

[1]VÚTS, a. s., Liberec, Czech Republic
jarmil.vlach@vuts.cz

[2] VÚTS, a. s., Liberec, Czech Republic
norbert.pomp@vuts.cz, radek.zboncak@vuts.cz

Keywords: Honeycomb, Natural, Frequency, Shear, Thickness, Damping.

Abstract. *This paper focuses on modal analysis and experimental determination of honeycomb panels damping properties. The rectangular shaped panels with different core thickness which is composed of hexagonal aluminum honeycomb core and aluminum alloy face sheets were under investigation. Group of specimens was supplied by 5M Technologies, a reputable manufacturer, which mainly deals with machinery applications. The panels are made by a unique technology of dry lamination where epoxy resin in the form of foil is applied between the coating sheet and the aluminum core. Then, its liquefaction and curing at elevated temperature in one step will occur. The influence of main directions of anisotropy and panel thickness to the natural frequency was investigated. We were also interested in the relationship between theoretical computed natural frequencies and the results obtained by the experiment. Damping properties and a series of tests are carried out on specimen in bending.*

1 INTRODUCTION

However the composite structures and honeycomb panels are still primarily used in the field of aircraft and aerospace industry, they get increasingly it`s own place in everyday`s applications of machinery engineering. In this article we focus on modal analysis and experimental determination of damping properties of rectangular shaped honeycomb beams. Panels with different core thickness, which are made by technology of dry lamination, were supplied by 5M Technologies, a reputable manufacturer. Specimens consists of aluminum hexagonal honeycomb core OK6 and thin aluminum skins bonded to each of the surfaces with epoxy resin. Thickness of panel is denoted by h, thickness of core is denoted by h_c.

In the first step, we investigated elastic properties of beams, to determinate it`s shear modulus in main directions. Longitudinal direction is coincident with x-axis, W-direction is given by y-axis. After the determination of its properties we computed beams natural frequencies using three different methods. To predict the frequencies we used FEM MARC 2010, the first-order and Reddy`s third-order shear deformation plate theory. The predicted frequencies we compared with that obtained by experimental modal analysis.

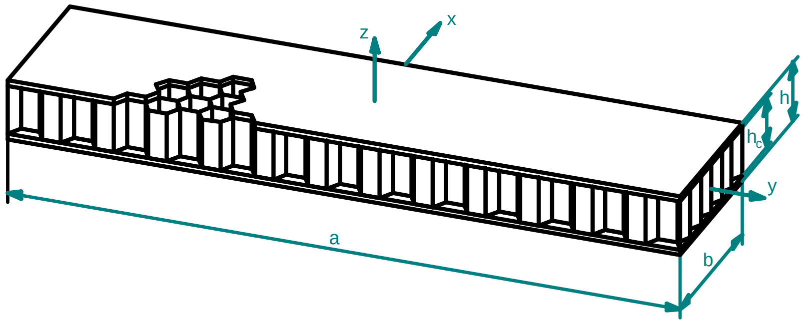

Figure 1.: Rectangular honeycomb panel

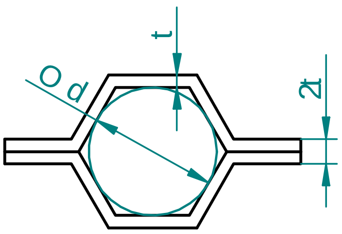

Figure 2.: Honeycomb cell unit

2 GENERAL SPECIFICATIONS OF SPECIMENS

The same hexagonal honeycomb core with different height was used for all specimens. It is described by inner diameter of hexagonal unit cell d=6 mm. Thickness of cell`s side is denoted by t and it is constant for all the specimens (t=0.06 mm). Specimens with it`s core heights is shown the Tab. 1. Core density was 92 $kg.m^{-3}$, density of facesheet material was 2770 $kg.m^{-3}$.

Beam flexural and shear stiffness was determined in accordance with the ASTM D7250/D7250M – 06. Young modulus of face sheet was determined by EN ISO 6892. Engineering constants of core and face sheet are denoted in Tab. 1 and 2.

Specimen No.	Core height (mm)	Length (mm)	Width (mm)	E_z^c (MPa)	E_z^c (MPa)
1	7,50	600	76	133	112
2	7,50	76	600	112	133
3	15,5	600	76	484	164
4	15,5	76	600	164	484
5	27,9	600	76	551	338
6	27,9	76	600	338	551
7	70,0	600	76	269	73
8	70,0	76	600	73	269

Table 1: Specimens description

Property	Value
E^f	70332 MPa
v^f	0,28 (-)
E_x^c	0,0013 MPa
E_y^c	0,0117 MPa
E_z^c	299,65 MPa
G_{xy}^c	87,5 MPa
v_{xy}^c	0,33 (-)
v_{yz}^c	0,000006 (-)
v_{zx}^c	0,14 (-)

Table 2: Core shear and facing properties

3 EXPERIMENTAL MODAL ANALYSIS

Experimental modal analysis is the method which allows to obtain mathematical model of the vibration properties and behaviour of the structure by experimental means. The modal parameters are established by measuring and processing of frequency response function $H(\omega)$, given as

$$H(\omega) = \frac{X(\omega)}{F(\omega)} \qquad (1)$$

where $F(\omega)$ is excitation and $X(\omega)$ is response of the system, both in frequency domain [4].

In our case we tested beam freely suspended on rubber bands with low stiffness. Beam was excited by measured force impulse and as response was measured acceleration at given point. One example of extracted frequency response function $H(\omega)$ is shown in Fig. 3. Founded 1-st natural frequency and modal damping are presented in Tab. 3.

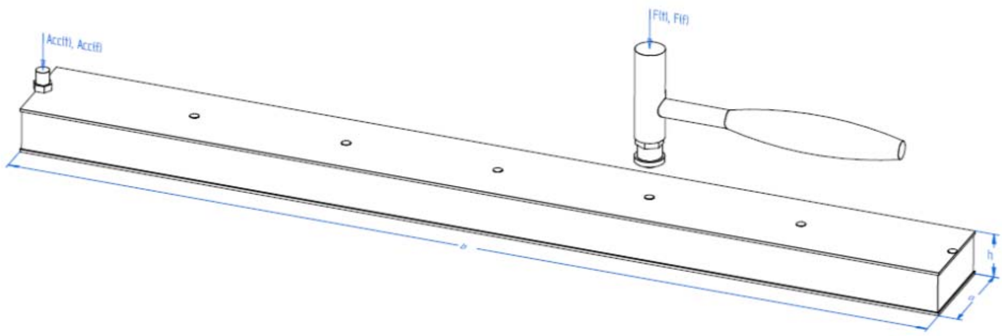

Figure 3: Schematic model of measuring frequency response function *H(ω)* with force hammer and sensor of acceleration.

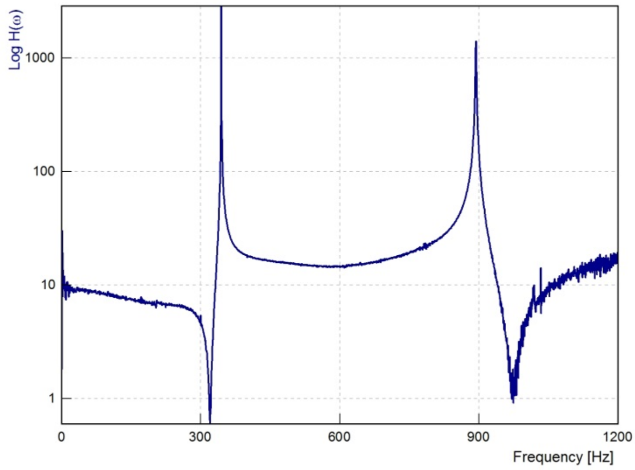

Figure 4: Frequency response function *H(ω)* of specimen 3.

4 DEFORMATION PLATE THEORY

There are few theories which we can use for mathematical modelling of plate cylindrical bending. It is known, that classical thin plate theory neglects the influence of shear stress and does not fit to values obtained by experiments. In this work we did a comparison of results obtained by using first-order, Reddy`s third-order shear deformation plate theory and FEM. Analytical theories mainly differ in the distribution of stress and strain field. The displacement field is given in general by Eqs. 2, 3 and 4.

$$u_1^{(g)}(x,y,z,t) = u_0(x,y,t) + z\varphi_x(x,y,t) + P_1^{(g)} \quad (2)$$

$$u_2^{(g)}(x,y,z,t) = v_0(x,y,t) + z\varphi_y(x,y,t) + P_2^{(g)} \quad (3)$$

$$u_3^{(g)}(x,y,z,t) = w_0(x,y,t) + P_3^{(g)} \quad (4)$$

$$(g = 1{,}3)$$

4.1 First-order shear deformation plate theory

The displacements of material point (x,y,z) caused by bending is given by Eqs. 2, 3, 4 and 5 [1].

$$P_1^{(1)} = P_2^{(1)} = P_3^{(1)} = 0 \quad (5)$$

4.2 Reddy`s third-order shear deformation plate theory

The displacements of material point (x,y,z) may be written by using Eqs. (2-4) and (6-8) [2].

$$P_1^{(3)} = -\frac{4z^3}{3h^2}\left(\varphi_x(x,y,t) + \frac{\partial w(x,y,t)}{\partial x}\right) \quad (6)$$

$$P_2^{(3)} = -\frac{4z^3}{3h^2}\left(\varphi_y(x,y,t) + \frac{\partial w(x,y,t)}{\partial x}\right) \quad (7)$$

$$P_3^{(3)} = 0 \quad (8)$$

The governing equation of free flexural symmetric honeycomb panel may be given as

$$L_{11}^{(g)}\, w(x,y,t) + L_{12}^{(g)}\varphi_x(x,y,t) + L_{13}^{(g)}\varphi_y(x,y,t) = 0 \quad (9)$$

$$L_{21}^{(g)}\, w(x,y,t) + L_{22}^{(g)}\varphi_x(x,y,t) + L_{23}^{(g)}\varphi_y(x,y,t) = 0 \quad (10)$$

$$L_{31}^{(g)}\, w(x,y,t) + L_{32}^{(g)}\varphi_x(x,y,t) + L_{33}^{(g)}\varphi_y(x,y,t) = 0 \quad (11)$$

$$L_{ij}^{(g)}(i,j = 1,2,3, g = 1,3)$$

where L_{ij} are the partial differential operators [2], [3].

It was showed [1], that governing equations is satisfied with functions $w_0, \varphi_x, \varphi_y$, given bellow.

$$w_0 = W_m e^{j\omega_m t} \sin\frac{m\pi x}{n}$$

$$\varphi_x = U_m e^{j\omega_m t} \cos\frac{m\pi x}{n}$$

$$\varphi_y = V_m e^{j\omega_m t} \cos\frac{m\pi x}{n}$$

$$(n = a,b;\ m = 1,2,\dots)$$

Then we can find the solution for every natural frequency ω_m, Eqs. (9-11) becomes time independent.

4.3 FEM model

FEM mesh was prepared by using MSC.MARC MENTAT 2010. For each simulation we prepared full mapped mesh. 20 nodes brick elements were used with every simulation. For computations we used MARC solver. Material model used was isotropic for face sheets and elastic-plastic orthotropic for honeycomb core.

5 RESULTS

Specimen No.	1^{st}-order (Hz)	Reddy (Hz)	FEM (Hz)	Experiment (Hz)	Damping (%)
1	67	250	317	140	1.35
2	67	230	316	137	0.14
3	143	625	420	344	1.2
4	139	335	409	337	0.05
5	244	829	563	514	0.05
6	240	650	559	504	0.07
7	467	733	1044	1009	0.06
8	376	403	705	833	0.05

Table 3: Comparison of natural frequencies

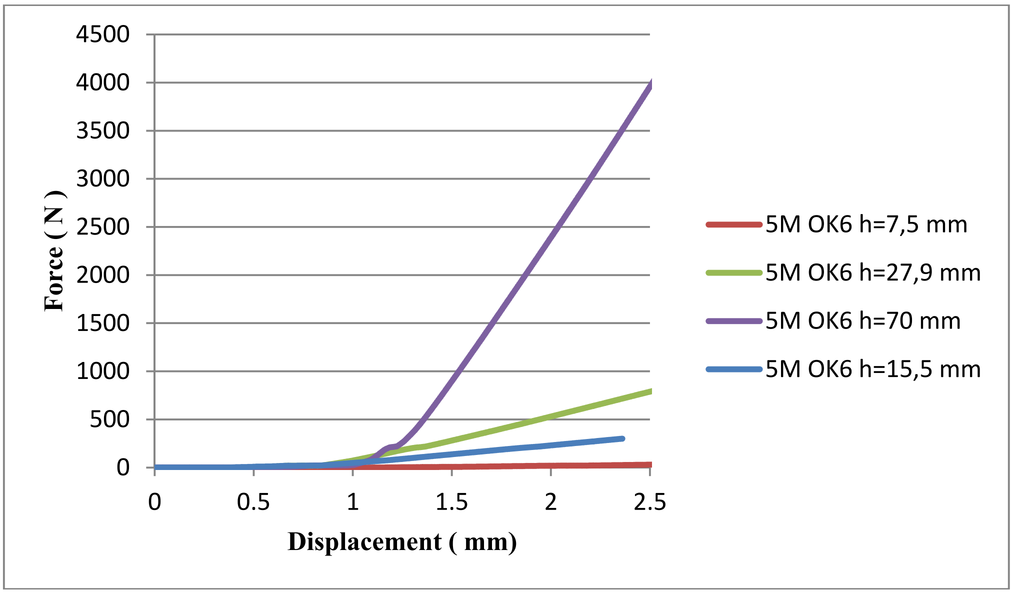

Figure 5: Force displacement dependence, 4 point bending test ASTM (specimens no. 1, 3, 5, 7).

6 CONCLUSIONS

- First order theory does not give satisfactory values.
- Values obtained by Reddy`s third-order theory and FEM analysis differ to measured data, because specimens which we tested were not straight. We could see that samples were little twisted. This effect is caused by the preparation technology and it manifests as non-

linearity in Fig. 3. So the geometry was not the same as we used in computations. It is denoted in ASTM, that dimensions must fit in defined tolerance.

- Highest error was obtained with prediction of natural frequency of the thinnest beams. It is caused also by preparation technology. Core of the panel is not stiff and plate becomes most twisted. These specimens varies also in modal damping.
- In near future, we are going to do new measurements and transfer the real panel geometry in fem model and do new computations.

REFERENCES

[1] J. M. Berthelot, Composite materials, Mechanical behavior and structural analysis. New York, Springer, 1999.

[2] Li Yongqiang, Zhu Dawei, "Free flexural vibration analysis of symmetric rectangular honeycomb panels using the improved Reddy`s third-order plate theory ", Composite structures 88 (2009) 33-39.

[3] H. Kobayashi and K. Matsumura, Wave equations of Reddy plate based on higher-order shear deformable plate theory, Mem. Fac. Eng. Osaka City Univ, pp. 37-44 (1999).

[4] S. Miláček, Modální analýza mechanických kmitů, Vydavatelství ČVUT 2001.