Journal of Achievements in Materials and Manufacturing Engineering

Volume 66 • Issue 2 • October 2014

International Scientific Journal published monthly by the World Academy of Materials and Manufacturing Engineering

Modal analysis of honeycomb panels and the influence of boundary conditions on the calculation

J. Vlach, N. Pomp
VÚTS, a.s., Liberec, Czech Republic
Corresponding e-mail address: jarmil.vlach@vuts.cz; norbert.pomp@vuts.cz

Received 12.08.2014; published in revised form 01.10.2014

ABSTRACT

Purpose: In this work we investigate the elastic properties of sandwich beams manufactured by using the LF Technology.

Design/methodology/approach: The investigation of the behaviour of rectangular shaped sandwich specimens is focused on the modal analysis and the experimental determination of the samples damping properties. Panels are made by unique technique of dry lamination patented by Czech company 5M s.r.o. The Hexagonal cell honeycomb core is made of aluminium as well as the facesheets. The influence of the main directions of anisotropy and the different panel's thicknesses on the natural frequencies are investigated.

Findings: The results of experiments are compared with the theoretical calculations and finite element method(FEM)simulation results. Theories used for the calculations are the First-order shear deformation theory (FSDT) and the Reddy's third-order shear deformation theory (TSDT). FEM model had mapped mesh with 20-nodes brick elements.

Research limitations/implications: The results obtained from FEA were closest to the experimentally measured data, but still with a deviation. The main reason of different results are geometrical irregularities. While FEM model was too much idealistic, the specimens prepared for measurement were not precisely planar. The specimens with small thickness were more twisted and therefore we got bigger error in the measured data and consequently the bigger deviation in results. In the future, we would like to do further measurements to transfer the real specimen geometry with all irregularities to a FEM model and to do new computations.

Originality/value: Originality of this work is modal analysis of honeycomb panels and the influence of boundary conditions on the calculation.

Keywords: First Order Shear Deformation Theory (FSDT); Frequency Response Function (FRF); Honeycomb; Third Order Shear Deformation Theory (TSDT); Shear

Reference to this paper should be given in the following way:

J. Vlach, N. Pomp, Modal analysis of honeycomb panels and the influence of boundary conditions on the calculation, Journal of Achievements in Materials and Manufacturing Engineering 66/2 (2014) 81-86.

ANALYSIS AND MODELLING

1. Introduction

Composite structures such as honeycomb sandwich panels were invented in 1940's. Their use in Apollo project showed the high potential of sandwich structures in the aircraft and aerospace industry. Nowadays, they are more frequently used in everyday applications of civil and machinery engineering. The main advantage of the sandwich structure is its significantly smaller density while the mechanical properties such as stiffness and strength remain comparable with the solid body products made of the conventional materials such as steel.

There are higher requirements for the use of the sandwich materials. For example they have to be more resistant against the fire, they need to withstand bigger loads, and so on. Sandwich structures need to be able to absorb bigger vibrations, etc. Also the cost and the effectiveness are important aspects of the manufacturing process. These are some reasons why the new technological processes of manufacturing are still being developed.

Honeycomb panels from 5M s.r.o. are manufactured by unique patented technology of dry lamination called The LETOXIT® Foil Technology (LF Technology) 0. Epoxy resin in the form of flexible foil is applied between the core and the coating facesheet. Then the liquefaction and curing at elevated temperature occur in one step.

LF Technology was developed as a low-cost technology pursuant to disadvantages of Resin Film Infusion Technology (RFI). The main disadvantage of the RFI technology can be identified in the way the sandwich structure is manufactured. During the fabrication process honeycombs are filled with resin and the foaming of adhesive prevents from impregnation of the sandwich facesheet. We can say that the facesheets are not impregnated appropriately from the outer side of the sandwich. Obviously, this is not exactly the case of aluminium sandwich structures, but the point is that the distribution of resin between the core and the facesheets is better with use of LF than with RFI 0.

In this work we investigate the elastic properties of sandwich beams manufactured by using the LF Technology. We determine the shear modulus in the main directions [2,3]. Then we calculate the natural frequencies using the First-order shear deformation theory (FSDT), the Reddy's third-order shear deformation plate theory (TSDT) [4,5], and predict the frequencies by using the finite element analysis (FEA) software MSC.MARC 2010. The results are compared with those obtained from the experimental modal analysis [6].

2. General specifications on the specimens

Specimens are prepared from aluminium sandwich panels with hexagonal cell core OK6. The inner diameter (d) of the core cell is 6 mm. The thickness (t) of the cell wall is 0.06 mm. There are several core height modifications as reported in Table 1.

Table 1.
Specimens geometry and core Young's moduli in z-direction

#	Core height, mm	Length, mm	Width, mm	E^c_{xz}, MPa	E^c_{yz}, MPa
1	7.5	600	76	133	112
2	7.5	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	600	76	269	73
8	70	76	600	73	269

The panel's core density is 92 kg m^{-3}. The facesheet material density is 2770 kg m^{-3}. The rest of the elastic constants of sandwich core and facesheet are summarised in Table 2.

Table 2.
Core and facesheet properties

Constant	Value
E^f	70 332 MPa
ν^f	0.28
E^c_x	0.0013 MPa
E^c_y	0.0117 MPa
E^c_z	299.65 MPa
G^c_{xy}	87.5 MPa
ν^c_{xy}	0.33
ν^c_{yz}	0.000006
ν^c_{zx}	0.14

A sandwich structure is orthotropic as well as its core. However, the facesheets can be isotropic such as in the case considered here. That is why, there are only three-indexed elastic constants. But we need just two of them (E, ν) [7] for our purposes because thin facesheets do not transfer the shear stress (G) in case of sandwich structures. The shear stress is transferred only by the sandwich core.

3. Description of results

Experimental modal analysis is method used to obtain a mathematical model of the vibration properties and the behaviour of the structure. It is based on measurement of the response function of the system at the reference point. The frequency response function (FRF) is given as

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

where $X(\omega)$is the response function of the system and $F(\omega)$ is the excitation load, both expressed in the frequency domain [6]. The system is excited with defined size load in several points. To simplify the solution, the specimen is substituted by a uniaxial wired model evenly divided along the specimen's length to seven points.

In this case, the tested beam is freely suspended on rubber bands with low stiffness. The beam is excited with defined force of impact hammer and the acceleration at the endpoint of the beam is measured as a response function of the structure.

After the impact load is applied, time behaviour of both the excitation load and the response function at the reference point is recorded, and then the FRF is determined. FRF expresses the ratio of the frequency spectra of the output signals. We provide an example of FRF in Fig. 1. Five excitations are performed repetitively at each point and the resulting FRF is set as their average function.

Modal parameters of the system are determined from the measured response characteristics. Both natural frequencies and modal damping (Table 3) assignment is based on linear regression method of FRF displayed as a circle in the Nyquist's diagram. The coordinate system origin lies on that circle. Established modal parameters are linear approximation of the system behaviour.

4. Deformation plate theory

The classical thin plate theory neglects the influence of the transverse shear deformation effect. According to this theory, in fact, straight lines that are perpendicular to the mid-plane before the bending remain straight and perpendicular also after the bending. This seems to be the main reason for which the results do not match the values obtained by conducting the experiments. Therefore, for a better understanding and quantification of this difference, we compare the results obtained by using the FSDT, the TSDT, and the FEM analysis.

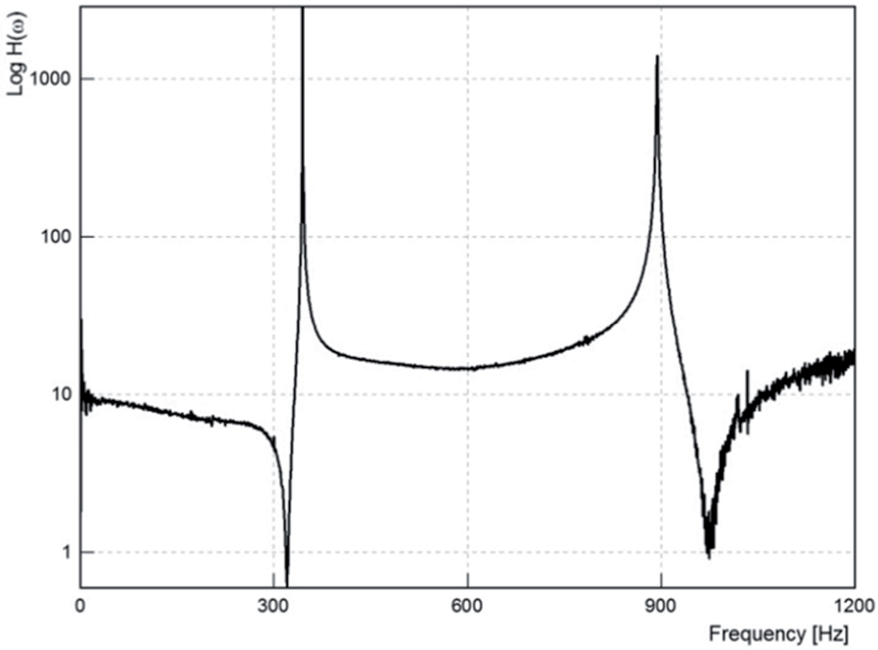

Fig. 1. Example of extracted frequency response function $H(\omega)$ of 3rd specimen

Analytical theories differ mainly in the distribution of the stress and the strain fields. The displacement field for time-dependent deformations is generally given by the following equations:

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

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

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

$$g = 1,3 \quad (5)$$

where $u_i^{(g)}$ is a component of the displacement field of TSDT; u_0, v_0 is a displacement of the transverse normal on plane $z = 0$; φ_x, φ_y is the rotation of the transverse normal on plane $z = 0$; P is a generalized term containing expressions of the extension and the higher-order rotation of the transverse normal.

4.1. First-order shear deformation theory

In this theory it is assumed that straight lines perpendicular to the mid-plane before the bending remain straight, but are no longer perpendicular to the mid-plane after the bending takes place. In other words, the distribution of the transverse shear stress is constant through the thickness.

Displacement of material point (x,y,z) caused by bending in the FSDT is given by four foregoing equations (2-5) and following conditions [2]:

$$P_1^{(1)} = P_2^{(1)} = P_3^{(1)} = 0 \quad (6)$$

4.2. Reddy's third-order shear deformation theory

Even if the FSDT allows to make more exact calculations, there is still a big difference between the calculated results and the reality represented by the measured data. Thus the introduction of a correction factor is required. By using TSDT, we assume parabolic distribution of transverse shear stress.

The displacement of material point (x,y,z) in TSDT is given by the same equations (2-5) but different conditions of the last terms on RHS of the equations [4].

$$P_1^{(3)} = -\frac{4z^3}{3h^2}\left(\varphi_x(x,y,t) + \frac{\partial w(x,y,t)}{\partial x}\right) \quad (7)$$

$$P_2^{(3)} = -\frac{4z^3}{3h^2}\left(\varphi_y(x,y,t) + \frac{\partial w(x,y,t)}{\partial y}\right) \quad (8)$$

$$P_3^{(3)} = 0 \quad (9)$$

The governing equation of the free flexural symmetric honeycomb panel is given by the following equation:

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

$$i,j = 1,2,3 \quad (11)$$

$$g = 1,3 \quad (12)$$

Term $L_{ij}^{(g)}$is a partial differential operator [4,5]. It was shown that the governing equation (10) is satisfied with functions w_0, φ_x, φ_y given bellow [2]:

$$w_0 = W_m e^{j\omega_m t} \sin\frac{m\pi x}{n} \quad (13)$$

$$\varphi_x = U_m e^{j\omega_m t} \cos\frac{m\pi x}{n} \quad (14)$$

$$\varphi_y = V_m e^{j\omega_m t} \cos\frac{m\pi x}{n} \quad (15)$$

$$n = a, b \text{ and } m = 1,2,\ldots \quad (16)$$

The solution for each natural frequency ω_m can be found. The governing equation becomes time-independent.

4.3. FEM model details

We prepared a fully mapped mesh with 20-nodes brick elements for each simulation in MSC.MARC MENTAT 2010. The material model for the sandwich facesheets is isotropic. The elastic-plastic model is used instead for the honeycomb core. Computations are realized by the MARC solver.

5. Results

Table 3 presents a summary of the results we have obtained from the calculations based on FSDT and TSDT, from the computations using FEA and from experimental data.

Based on the results, we can say that FSDT does not give satisfactory values at all. But we expected that. There is approximately 50% difference with the experimental data. The bigger the specimen height the bigger the difference.

While the result values given by FSDT are just smaller than the measured values, the resultant values given by TSDT are bigger for thin beams and smaller for thick beams than the values measured experimentally. We expected theoretical natural frequencies calculated according the TSDT to be much closer to the measured data. Best results are given by FEM for thick specimens. However, the variation in results is still perceptible. We will discuss the reason in detail in the conclusion.

Table 3.
Natural frequency

#	First-order, Hz	Reddy's, 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	1.20
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	933	0.05

6. Conclusions

Values obtained from TSDT and FEM should be much more accurate. However, the analysis results still differ from the measured data because the specimens are not precisely planar. The geometrical irregularities of the specimen cause measurements errors. That is also the main problem of the virtual models. They are too much idealistic and it is very difficult to include any irregularities to the model. Twisting is caused during the manufacturing by preparation technology and can be realized as nonlinearity in Fig. 2. The smaller the thickness of the specimen the bigger the twisting. As a consequence, we find that the biggest error in the prediction of the natural frequencies was in the thinnest beams which are the least stiff.

In the future, we would like to do further measurements to transfer the real specimen geometry with all irregularities to a FEM model and to do new computations.

Acknowledgements

The work was supported within the frame of the TIP Project under the FR-TI3/460 grant. Our acknowledgment belongs to all donors that helped us to reach the results of the work.

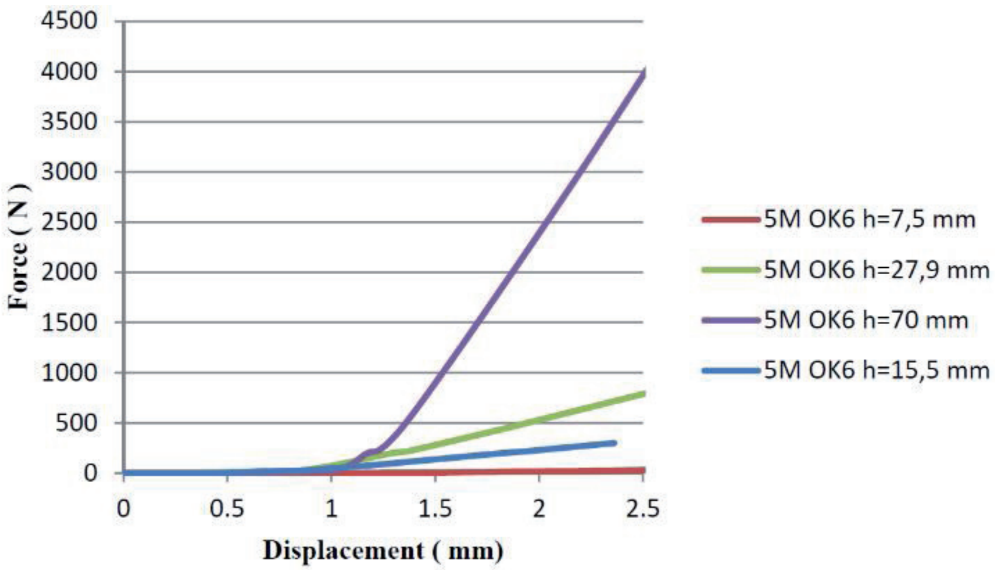

Fig. 2. Force-displacement tensile curve of 4-point bending (Specimens 1, 3, 5 and 7 from Table 1)

References

[1] 5M s.r.o., LETOXIT Technology, http://www.5m.cz/cz/lf-technology/

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

[3] International ASTM D7250/D7250M – 06, Standard practise for determining sandwich beam flexural and shear stiffness, ASTM Standards on disk, Vol. 15.03, 2011.

[4] L. Yongqiang, Z. 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.

[5] H. Kobayashi, K. Matsumura, Wave equations of Reddy plate based on higher-order shear deformable plate theory, Memoirs of the Faculty of Engineering Osaka City University 40 (1999) 37-44.

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

[7] EN ISO 6892-1, Kovové materiály – Zkoušení tahem – část 1: Zkušební metoda za pokojové teploty, 2010.