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EXPERIMENTAL DETERMINATIONS OF SOME MECHANICAL PROPERTIES FOR NEW TYPES OF COMPOSITE BARS WITH POLYPROPYLENE HONEYCOMB CORE

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In this paper we have experimentally determined, using some known methods, the equivalent elasticity modulus for new types of sandwich bars characterized by: polypropylene honeycomb core, with a thickness of 20 mm, having the exterior layers made of epoxy resin reinforced (on the upper and lower sides) with 5 and 10 layers of steel wire mesh. We have considered as a reference method for elasticity modulus determination the modal analysis. We have determined the first three eigenmodes and used them to determine the elasticity modulus. The obtained results were checked with another experimental method characterized by bending loading of the bars in three points. In the last part of the paper, we have made a comparative study between the flexural rigidity values obtained from different sandwich structures with polypropylene honeycomb core that have thickness of 20 mm. The bars have different face sheets: bar 1 has one layer of fiber-glass with epoxy resin, bar 2 has two layers of fiber-glass with epoxy resin, bar 3 has five layers of steel wire mesh with epoxy resin and bar 10 has ten layers of steel wire mesh with epoxy resin. Starting from the first two eigenmodes of the bars, it is established a method used to determine the bars flexural rigidity. The results are validated by using an approximate experimental method.

Key words: sandwich bars; eigenmodes; elasticity modulus; flexural rigidity; eigenfrequencies.

1. INTRODUCTION

The sandwich bars and plates can be studied by various methods that mostly differ by the inclusions or neglecting the effects of angular deformation and respectively, the rotational inertia. A first theory, named as FSDT (First-Order Shear Deformation Theory), was presented in [1] and further developed in [2]. According to this theory, a straight line normal on the median plane before deformation remains straight without keeping the perpendicularity during deformation (on the median surface). The refined theories rely on a non-linear distribution of shear stresses along the thickness of the plate or bar. The inclusion of high order terms implies the inclusion of supplementary unknowns. Moreover, when fulfilling both the distribution of shear stresses in thickness are parabolic and if the limit conditions are accomplished on external surfaces, a correction factor is not necessary. Based on these assumptions, a HSDT (High-Order Shear Deformation Theory) theory was presented in [3]. According to this theory, the stresses and strains normal to the median plane are null. In [4] it was developed a theory that considers the stresses normal to the median plane. This theory removes a series of contradictions that appear in previous theories by accepting non linear factors of shear stresses in thickness. It is not neglected a part of the normal stresses obtained by the loading of the composite structure. A better characterization of sandwich bars can be obtained by using LWM (Layer-Wise Models) models. In [5, 6], it is considered each layer in a sandwich structure as a separate bar. The damped, forced and nonlinear vibrations for simply supported rectangular plates with viscoelastic core were studied in [7]. The core was modelled like a Voigt-Kelvin solid. There was also studied the influence of the layers thicknesses and material properties over the nonlinear plates response. There have also been made some studies on the damped vibrations of Euler-Bernoulli and Timoshenko beams. The material was assumed to be incompressible, whereby the same viscoelastic operators could be both used for the flexural and shear deformations. This permitted the use of the normal modes and their orthogonally conditions to solve this viscoelastic forced vibration problem. Relevant to these studies are the papers [8, 9]. In [10] the linear vibrations of Timoshenko beams are studied. It is detailed that if the ratio of length and thickness of a bar is greater than ten, then the difference between Timoshenko and Euler-Bernoulli theories for the bending moment, shear force and the medium fiber deformation, are smaller than 5%. There is shown that the damping influence of rotational motion of the bar section can be neglected (for the first eigenmodes of vibration). In [11] it was investigated the mechanical behaviour and failure mechanism (such as compressive and shear deformation or strengths) of honeycomb composite consisting of Nomex honeycomb and 2024Al alloy face sheets, at different temperatures ranged between 25–300^{0}C. The average elastic and strength characteristics of Nomex honeycomb were also investigated in [12]. The static and fatigue behaviour of aluminum hexagonal honeycomb cores were analyzed in [13]. The out-of-plane compressive properties of thermoplastic hexagonal honeycombs using the finite element analysis were investigated in [14]. The equivalent transverse shear and in-plane moduli of honeycomb cellular structures were evaluated in [15] and it was discussed about the structural efficiency of honeycombs. Using honeycomb test specimens made of Nomex, aluminum alloy and paper, in [16] there were explored the crushing phenomena of the cells. In [17] there was made an experimental investigation on low-velocity impact responses and damage modes of sandwich composites (aluminum honeycomb core and glass/epoxy face sheets) because of the impact loading changing location and wall partition angle of the honeycomb core. Numerical and experimental methods were used in [18] to examine the crashworthiness and rollover characteristics of a low- floor bus vehicle made of sandwich composites (aluminum honeycomb core and WR580/NF 4000 glass-fabric/ epoxy laminate face sheets). Further studies regarding the mechanical parameters determination (like free vibrations, natural frequencies, elasticity modulus, decoupling effects and so on) for various composite bars are presented in [19, 20, 21,22, 23].

In this paper, some mechanical properties for new types of composite sandwich bars, with the core made of polypropylene honeycomb, were determined. The materials used for the sandwich bars (polypropylene honeycomb, steel wire mesh, fiber-glass and epoxy resin) are classical, but their combination is original.

2. THE EQUIVALENT ELASTICITY MODULUS DETERMINATION

2.1 Equivalent elasticity modulus determination using the modal analysis

There are analyzed new types of composite bars marked in this way: model 1 – sandwich bar with rectangular section (25×23,4 mm) with polypropylene honeycomb core (with a thickness of 20 mm), having the exterior layers made up with epoxy resin reinforced with each **5** layers (upper and lower) of steel wire mesh; model 2 – sandwich bar with rectangular section (25×26 mm) with polypropylene honeycomb core (with a thickness of 20 mm), having the exterior layers made up with epoxy resin reinforced with each **10** layers (upper and lower) of steel wire mesh. The epoxy resin is RESOLTECH 1050 type and its hardener is RESOLTECH 1058 type (having a mixed density of 1.11 g/cm^3, mixed viscosity of 633 MPa·s at 23^{0}C, 5% elongation to break and 6% flexion to break). The polypropylene honeycomb is NIDATECH 20 made by NIDAPLAST firm (having a cell size of 20 mm, compressive strength of 0.5 MPa and compressive modulus of 0.01 MPa). The model 1 of sandwich bar type is presented in Fig. 1. For similar structures, in [24] the damping factor was determined. The methodology for the equivalent elasticity modulus (between the transversal and longitudinal one) calculus is according to [25].

The experimental montage is similar to the one presented in [25]. The bars are rigidly fixed at one end and at the other end, at a 10 mm distance it is placed an accelerometer Bruel&Kjaer with 0.004 pC/ms^{-2} sensitivity. The excitation of the bar was made by using an impact Bruel&Kjaer hammer with 1.020 pC/ms^{-2} sensitivity. The scheme of the experimental montage is presented in Fig. 2a.

We will consider this method as *the reference one* for the equivalent elasticity modulus value. We will obtain the vibration eigenmodes according to the considered experimental montage and to the procedure from [25]. The steps for the modal identification are according to the study from [26]. We determine the vibratory response (with the accelerometer) and the excitation force (with the impact hammer) for both models. These experimental recordings for model 1 are presented in Fig. 2b. As in the paper [26], we first determine the frequency response function in cartesian and polar coordinates and then, by the usage of this function, we determine the modal parameters (Fig. 3 and Fig. 4).

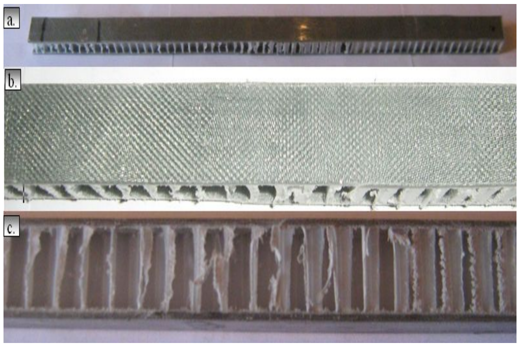

Fig. 1 – a. General view; b. a detail with the steel wire mesh distribution; c. A detail with the polypropylene honeycomb.

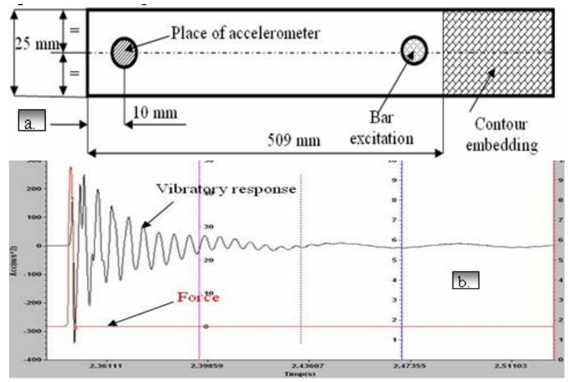

Fig. 2 – a. The experimental scheme; b. experimental recordings (model 1).

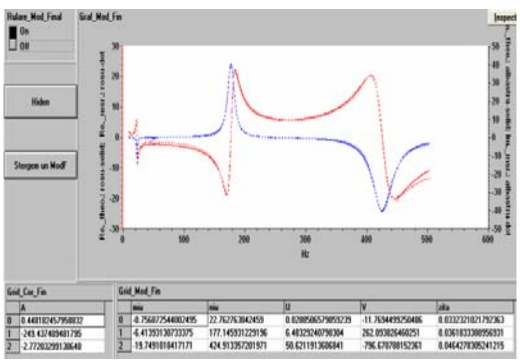

Fig. 3 – The final panel of modal parameters (model 1).

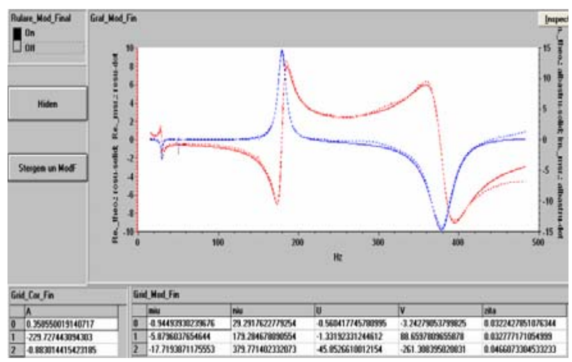

Fig. 4 – The final panel of modal parameters (model 2).

From Figs. 4 and 5 we withhold the values from the *niu* column (*niu*-**ν** is the eigenfrequency of each modal parameters). We have for model 1: ν_1 = 22.762; ν_2 = 177.145; ν_3 = 424.913; and for model 2: ν_1 = 29.291; ν_2 = 179.284; ν_3 = 379.771. According to [25], the elasticity modulus for each eigenmode is:

$$E_k = \frac{\left(2\cdot\pi\cdot L^2\right)\cdot\rho\cdot A}{I_y}\cdot\left(\frac{\nu_k}{\beta_k^2}\right)^2 . \tag{1}$$

In relation (1) we have the next parameters: k – is the number of the eigenmode; E – the elasticity modulus (MPa); $L = 509\cdot10^{-3}$m – the free length of the bar; ρ is the material density and is 424.861 for model 1 and 654.22 kg/m^3 for model 2; A – the transversal section area and is $5.85\cdot10^{-4}$ for model 1 and $6.5\cdot10^{-4}$ m^2 for model 2; I_y – the inertia moment of the bar and is $2.669\cdot10^{-8}$ for model 1 and $3.662\cdot10^{-8}$ m^4 for model 2; β – the solutions of the equation chβ·cosβ + 1 = 0; ν – the eigenfrequency of each eigenmode [Hz]. We made the next simplifying assumptions: we have considered the section of the bars being rectangular; the solutions β_k were approximated. For the models 1 and 2, we have obtained three equivalent elasticity moduli (between the longitudinal and transversal ones) that correspond to each eigenmode:

$$E_1 = 1\,232 \text{ MPa},\quad E_2 = 1\,616 \text{ MPa},\quad E_3 = 1\,217 \text{ MPa (model 1)}, \tag{2}$$

E_1= 2 161 MPa, E_2= 2 062 MPa, E_3= 1 870 MPa (model 2).

According to [25], in order to obtain the final value of the equivalent elasticity modulus, we make the arithmetic mean of the data obtained for each model. According to [25], the equivalent elasticity modulus is the same across the bar length. So, we have obtained the relations (3)

$$E_{\text{model 1}} = 1\ 355 \text{ MPa (model 1)}, \quad E_{\text{model 2}} = 2\ 031 \text{ MPa (model 2)}. \tag{3}$$

2.1. Results validation for the equivalent elasticity modulus

We shall validate the elasticity modulus by using the three points bending of the bar. We will determine the pairs stress and strain obtained from this loading. The accuracy of the results obtained with the used device for bending is presented in [27]. The bending loading scheme is presented in Fig. 5. According to the loading scheme, the stress is (the value of M is in N·mm and the value of W is in mm^3):

$$\sigma_{mark} = \frac{M}{W} = \frac{V_1 \cdot 260}{W} = \frac{\frac{90 \cdot 260}{440} \cdot F}{2281.5} = 0.02331 \cdot F. \tag{4}$$

In (4) we have marked with M [N·mm] the bending moment obtained from the loading scheme, with W [mm^3] the axial strength modulus and with V_1 the reaction force from the left simple support. To obtain the stress, the signal of the force (recorded with the force transducer S9 –Fig. 4) will be amplified with 0.02331 (the stress from relation (4) is determined in the point where the active strain gauge is glued, in a point at 260 mm from the left simple support – Fig. 5, depending on the force value measured with the force transducer). We have determined the stress value from the force signal in order to obtain the elasticity modulus as a ratio between the stress and the strain. In order to obtain the strain (marked as $\varepsilon_{\text{mesured}}$ in the next relations), we have bounded four strain gauges on the two models in a half-bridge connection. The strain values are obtained directly with the strain gauges. In order to obtain the elasticity modulus, we have used the Hooke law, by considering that the loadings are very small and are produced in the elastic domain.

$$E_{mesured} \cdot 10^{-6} = \frac{\sigma_{mesured}}{\varepsilon_{mesured}}. \tag{5}$$

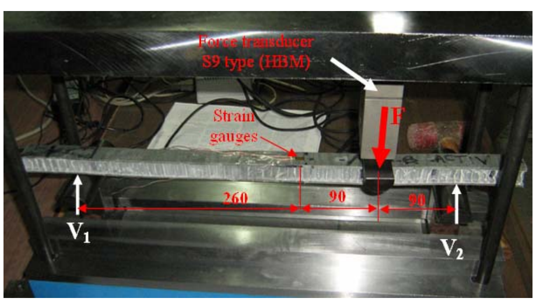

Fig. 5 – The bending loading scheme and device (model 1).

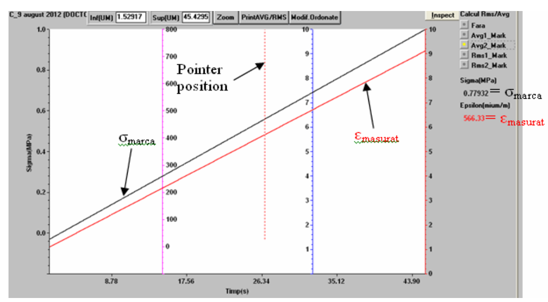

Fig. 6 – The stress and strain at a chosen point from the experimental data.

The experiment is characterized by loading the bars with a force (with a small value) and by recording the obtained experimental data. From the experimental graphic, we have chosen some points for which we have determined the stress and strain values. In Fig. 6 it is presented a chosen point from the experimental graphic (that corresponds to the point 4 from the Table 1) given by the pointer position (represented with dash line), for which the parameters σ_{mesured} and $\varepsilon_{\text{mesured}}$ were determined. All the chosen points and the experimental data for the stress and strain are written in Table 1. Also we have determined *the errors that appear compared to the values obtained with the modal analysis* and we have written them in Table 1. We have done the same thing for the model 2, and we have written the results in Table 2.

From the two tables, we can see that the errors compared with the modal analysis method are small (under 10 %). The modal identification method, used in this research, is an accurate one, fact proven by the small errors obtained between the two experimental methods. This method was also used in [25] for elasticity modulus determination for composite materials with random distribution of reinforcement, and the results were compared to the ones obtained from the tensile test on an universal testing machine and the errors were under 4%.

Table 1

Stress, strain, elasticity modulus and errors obtained in each chosen point (model 1)

Point	σ_{mark} [MPa]	$\varepsilon_{mesured}$ [μm/m]	E [MPa]	Error [%]
1	0.27246	210.87	1292	4.644
2	0.17187	140.33	1225	9.612
3	0.71466	520.99	1372	1.22
4	0.77932	566.33	1376	1.532
5	0.99009	714.15	1382	2.264

Table 2

Stress, strain, elasticity modulus and errors obtained in each chosen point (model 2)

Point	σ_{mark} [MPa]	$\varepsilon_{mesured}$ [μm/m]	E [MPa]	Error [%]
1	0.76527	345.14	2217	8.401
2	0.83563	376.49	2220	8.494
3	0.92944	418.29	2222	8.596
4	1.0584	475.77	2225	8.703
5	1.3516	606.39	2229	8.88

3. EXPERIMENTAL DETERMINATION OF THE FLEXURAL RIGIDITY

3.1. The usage of modal analysis for flexural rigidity calculus

In [21] the equations of motion for transversal vibrations of viscoelastic bars that have constant section and external damping were presented. It was proved that, if the ratio of length and thickness of a bar is greater than ten, the difference between Timoshenko and Euler-Bernoulli theories for the bending moment, shear force and the deformation of medium fiber is smaller than 5%. For the first vibration eigenmodes, the damping influence of rotational motion of the bar section can be neglected. In order to determine the bars flexural rigidity (the product between the axial moment of inertia and the equivalent elasticity modulus), the relation (1) can be used. We have made bars from composite materials with polypropylene honeycomb core NIDATECH 20, having a 20 mm thickness. These bars have different exterior upper and lower layers: **bar 1 –** 1 layer of glass fabric with epoxy resin RESOLTECH 1050; **bar 2 –** 2 layers of glass fabric with epoxy resin RESOLTECH 1050; **bar 3 –** 5 layers of steel wire mesh with epoxy resin RESOLTECH 1050; **bar 4 –** 10 layers of steel wire mesh with epoxy resin RESOLTECH 1050. The fiber-glass fabric is E-GLASS type with the next basic characteristics: elastic limit – 2750 MPa, Young modulus – 72 GPa, tensile strength – 1950 MPa, Poisson ratio – 0.21. The width of all the bars is 30 mm. A general view with the bar samples is presented in Fig. 7.

The bars are rigidly fixed at one end and at the other end, at a 10 mm distance is placed and accelerometer Bruel&Kjaer with 0.004 pc/ms^{-2} sensitivity. The excitation of the bars was made by using an impact hammer Bruel&Kjaer with 1.020 pc/ms^{-2} sensitivity. We have also considered two variants of the bar embedding, in this way (we will refer to the free parts of the bars – namely the parts where the accelerometer is located and where the measurements will be made): **Variant 1**: the free length is 350 mm; **Variant 2**: the free length is 320 mm. We have chosen these values for the free length of the bars to apply Bernoulli theory (where the ratio between the bar length and thickness must be higher than 15) valid in the case of this bars. The experimental scheme is presented in Fig. 8.

There have been made experimental measurements, determining the eigenmodes in the considered measuring point. In order to avoid any possible errors, we have made each measurement twice. After obtaining the eigenfrequencies, we have determined the bars flexural rigidity with relation (1). We have presented in Figs. 9 and 10 the experimental recordings for the bar 2 in both 1 and 2 embedding considered variants. In Figs. 11 and 12 we present the final panel with the modal parameters determined from the experimental scheme.

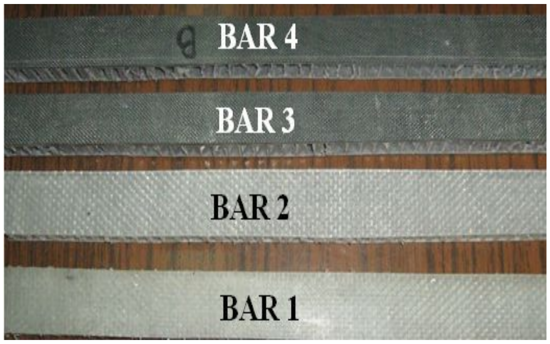

Fig. 7 – A general view with the used samples.

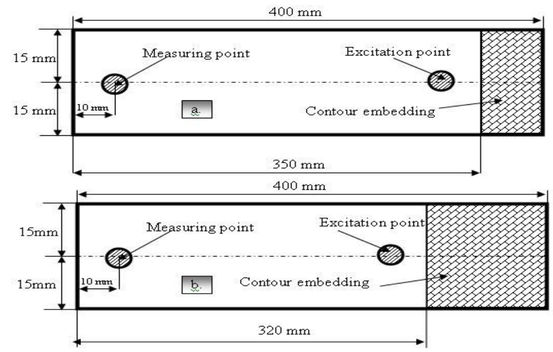

Fig. 8 – Experimental scheme: a. variant 1; b. variant 2.

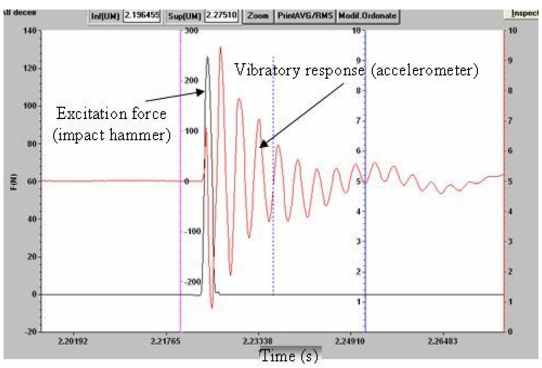

Fig. 9 – Experimental recordings (bar 2 – Variant 1).

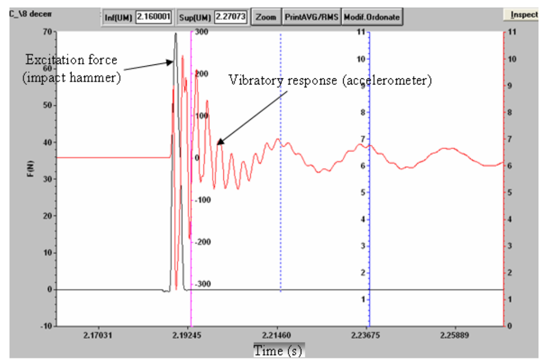

Fig. 10 – Experimental recordings (bar 2 – Variant 2).

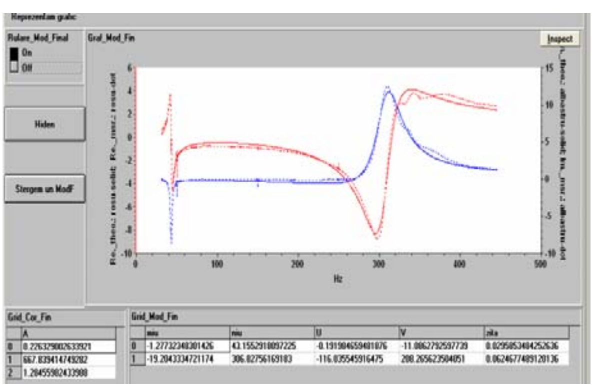

Fig. 11 – Experimental recordings (bar 2 – Variant 1).

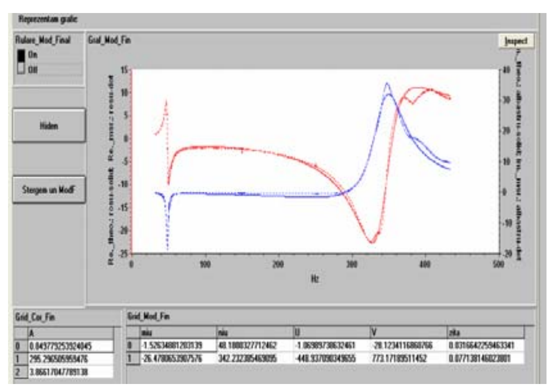

Fig. 12 – Experimental recordings (bar 2 – Variant 2).

Important mark. The eigenfrequency for each eigenmode can be found in the Figs. 11 and 12 on the column that corresponds to the *niu (ν)* notation. For example, in Fig. 12, from the *niu* (ν) column we can find the next values: $\nu_1 = 48.18$ Hz for the first eigenmode and $\nu_2 = 342.232$ Hz for the second eigenmode. All the results obtained from the experimental data and the bars flexural rigidity are written in Table 3, where we have marked with k the number of the eigenmode.

In order to obtain the bars flexural rigidity we shall make the arithmetic mean of the EI_k values obtained for each eigenmode. We shall determine the errors between the flexural rigidity values obtained for each eigenmode (the ε_s values from Table 4).

Table 3

The results from the experimental data

Bar No.	*m* [kg]	ν_k [Hz]	Embedding variant	*k*	EI_k [N·m^2]	EI_k/m [N·m^2/ kg]
1	0,050	35,05	1	1	9,904	198,08
1	0,050	260,7	1	2	11,788	235,76
1	0,050	41,854	2	1	10,794	215,88
1	0,050	307,834	2	2	12,561	251,22
2	0,060	43,155	1	1	18,017	300,283
2	0,060	306,827	1	2	19,593	326,55
2	0,060	48,18	2	1	17,163	286,05
2	0,060	342,232	2	2	18,63	310,5
3	0,102	43,561	1	1	31,208	305,961
3	0,102	286,048	1	2	28,95	283,824
3	0,102	48,067	2	1	29,041	284,716
3	0,102	325,282	2	2	28,611	280,5
4	0,167	42,060	1	1	47,635	285,24
4	0,167	276,180	1	2	44,185	264,581
4	0,167	45,567	2	1	42,73	255,868
4	0,167	318,290	2	2	44,852	268,575

Table 4

The bars flexural rigidity and errors between eigenmodes

Bar No.	Embedding variant	*EI* [N·m^2]	*EI*/m [N·m^2/ kg]	ε_s [%]
1	1	10,846	216,92	15,982
1	2	11,678	233,55	14,067
2	1	18,805	313,417	8,044
2	2	17,896	298,275	7,874
3	1	30,079	294,893	7,235
3	2	28,826	282,608	1,481
4	1	45,91	274,911	7,243
4	2	43,791	262,221	4,731

3.2. Experimental validation of the flexural rigidity values

The flexural rigidity of a bar which is embedded at one end and loaded at the other end with a force *F* can be calculated with (6). In (6) we have marked with: *EI* – the bar flexural rigidity; *F* – the force that loads the bar; *l* – the free length of the bar; *v* – the bar displacement. We shall load the bars with a force $F = 200$ gf $= 200\cdot10^{-3}$ kgf ≈ 0.2 daN ≈ 2 N. The free lengths of the bars *l* correspond to the variants of the bar embedding presented at the modal analysis experiment (*l*= 350 and 320 mm).

$$E\cdot I = \frac{F\cdot l^3}{3\cdot v}. \tag{6}$$

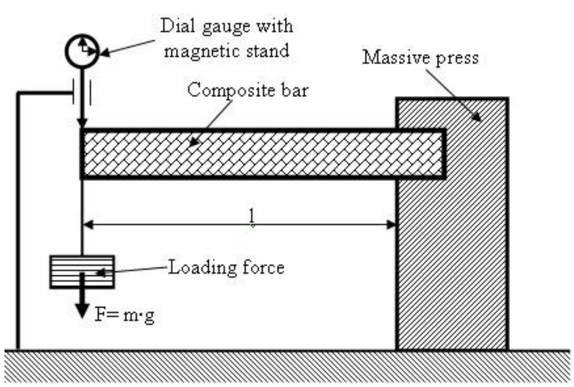

Fig. 13 – Experimental assemblage.

The bars are loaded to bending, so the flexural rigidity will be determined in order to validate the results obtained at the Chaper 3.1. The displacement will be measured with a comparative device – a dial gauge with magnetic stand having a precision of 0.01 mm. The scheme of the experimental assemblage is presented in Fig. 13. All the experimental data is written in table 5. This is an approximate method because of the errors that may appear at the dial gauge displacement reading. We have marked with ε_m in Table 5 the errors for the flexural rigidity values that appear between the two nondestructive experimental methods.

From Table 5 we can see that the errors between the experimental methods are quite small, bellow 10%.

Table 5

Experimental data

Bar No.	Embedding variant	*EI* [N·m^2]	*EI*/m [N·m^2/ kg]	ε_m [%]	*v* [mm]
1	1	11.036	220.721	1.722	2.59
1	2	11.209	224.183	4.016	2.55
2	1	17.116	285.263	8.982	1.67
2	2	19.056	317.593	6.087	1.5
3	1	27.222	266.885	9.498	1.05
3	2	29.167	285.948	1.169	0.98
4	1	42.034	251.702	8.443	0.68
4	2	43.308	259.33	1.103	0.66

4. CONCLUSIONS

We consider that the added value of the study presented in this paper is:

– making some new original composites bars, with classical elements (like steel wire mesh, fiber-glass, epoxy resin, polypropylene honeycomb) combined in an original sandwich bar;

– the number of the reinforcing layers (5 and 10 steel wire mesh layers, respectively 1 and 2 fiber-glass layers);

– determining some mechanical characteristics for these composite bars, like: equivalent elasticity modulus, flexural rigidity, eigenmodes, eigenfrequencies (marked with *niu* – ν, in Figs. 3, 4, 11, 12), critical damping for each eigenmode (marked with *zita* – ζ, in Figs. 3, 4, 11, 12), damping factor for each eigenmode (marked with *miu* – μ, in Figs. 3, 4, 11, 12) – these values characterize the vibratory response of the studied sandwich bars;

– proposing new structures with applications in practical engineering (in fields like civil constructions, mechanical engineering or materials engineering) for: planes floor building, ships floor building, walls of civil constructions or concrete forming.

The hardening time of the epoxy resin used in this paper was of 24 hours at room temperature. The presented experimental methods have the advantage that are non-destructive and the samples can be used for further investigations. Also, the modal analysis has the advantage that it can be used in the case of complex systems built from composite materials. In the first part of the paper, the equivalent elasticity modulus was determined for two types of sandwich bars reinforced with different number of steel wire mesh layers. The elasticity modulus determination for this kind of sandwich bars is important in practical engineering for the next cases:

– when we want to make comparisons between the stresses obtained from different loadings for this kind of bars and other known sandwich bars;

– when we want to make a comparison between the stresses per unit mass between these sandwich bars and metallic beams of different sections;

– when we want to make a finite element modeling of an equivalent bar, by inserting the elasticity modulus value, and so on.

From Table 4, we can see the next tendencies:

– the bar with 10 steel wire mesh layers has an increased flexural rigidity (around 33%) in comparison with the sandwich bars with 5 layers of steel wire mesh, fact that can be explained by the 5 aditional layers between the two bars;

– the bars reinforced on the upper and lower face sheets with 1 layer of fiber glass and resin have the smallest flexural rigidity, therefore these composites can be used for structures building that are not heavily loaded;

– for heavily loaded structures, we recommend using elements made in the same way as the bar 4;

– we have observed little errors between the flexural rigidity values obtained for each eigenmode (under 16% – value accepted in practical engineering);

– we have observed little differences between the flexural rigidity values obtained at each embedding variant, for the same bar;

– we can see that the eigenfrequency value, for all the bars, is higher (for all the eigenmodes) for the variant 2 of embedding in comparison with the variant 1.

As a further research, we would consider the possibility of replacing the metallic beams that have a caisson section with these sandwich bars. Regarding this study, we aim to load the composite bars and the metallic beams in several variants and make comparisons from the mass per unit stress point of view. We wish to see if there are loading variants where the composite sandwich bars have an increased strength compared to the metallic ones. The results of the experimental determinations can be applied in the behaviour study of this composite materials type usable for structural and nonstructural walls at civil constructions, in order to improve their behaviour at extraordinary loadings like the seism but also at buildings shock loaded (for example: the explosions). In this sense, in a future paper there will be studied the stresses in composite planar lamellar and plates elements, including the application of these composite materials on the strength parts like beams, walls, platforms, columns, etc.

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