COMPORTAMENTUL DINAMIC ŞI STATIC AL PLATBANDELOR COMPOZITE SANDWICH CU MIEZ DIN FAGURE DE KEVLAR

DYNAMIC AND STATIC BEHAVIOUR OF COMPOSITE SANDWICH PLATBANDS WITH KEVLAR HONEYCOMB CORE

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In this paper the authors have experimentally determined, using some known methods, the loss and damping factors, equivalent dynamic Young modulus and flexural rigidity for composite sandwich platbands with two types of kevlar honeycomb core (rectangular and hexagonal) reinforced with kevlar-carbon fiber. The experimental procedure is characterized by clamping the platbands at one end, placing an accelerometer at the free end for recording their free vibrations. The samples have a thickness of 4 mm, a length of 335 mm and 60 mm width. The connection between the kevlar-carbon fiber and kevlar honeycomb core is made by using epoxy resin with its hardener. The flexural rigidity values were validated by a static loading experimental montage: the platbands are clamped at one end and free at the other; at the free end a force has been applied and the displacement was measured with a displacement transducer. From the displacement value, the flexural rigidity was obtained. In the end, the authors have obtained that the sandwich platbands with hexagonal honeycomb core have better mechanical properties compared to the ones with rectangular honeycomb core.

În această lucrare, autorii au determinat experimental, folosind câteva metode deja cunoscute, factori de amortizare şi de pierdere a energiei, modulul lui Young dinamic echivalent şi rigiditatea la încovoiere pentru platbande compozite sandwich cu două tipuri de fagure din kevlar (dreptunghiular şi hexagonal) ranforsate cu fibră de carbon-kevlar. Procedura experimentală s-a caracterizat prin încastrarea platbandelor la un capăt, amplasarea unui accelerometru la capătul liber pentru înregistrarea vibraţiilor libere ale acestora. Epruvetele au o grosime de 4 mm, o lungime de 335 mm şi lăţime de 60 mm. Legătura dintre fibra de kevlar-carbon şi fagurele din kevlar este realizată folosind răşina epoxidică şi întăritorul său. Valorile rigidităţii la încovoiere au fost validate prin folosirea unui montaj experimental de încărcare statică: platbandele sunt încastrate la un capăt şi libere la celălalt; la capătul liber a fost aplicată o forţă şi săgeata a fost măsurată cu un traductor de deplasare. Din valoarea săgeţii s-a obţinut rigiditatea la încovoiere. În final, autorii au constatat că platbandele sandwich cu miez tip fagure hexagonal au proprietăţi mecanice mai bune în comparaţie cu cele care au miez tip fagure dreptunghiular.

Keywords: *damping factor, loss factor, Young modulus, kevlar honeycomb, kevlar-carbon fiber.*

1. Introduction

Most of the studies regarding kevlar (especially the kevlar 29 material), carbon (in different combinations such as carbon fibers with E-glass fibers) and kevlar-carbon refer to impact characteristics, especially the ballistic impact such as [1 - 5]. Other researches refer to mechanical characteristics determination for kevlar, carbon and kevlar-carbon composites. Interfacial bonding under bending and uniaxial compression for carbon-fiber and aluminum-honeycomb sandwich composites with and without kevlar-fiber interfacial toughening was investigated in [6]. A set of modal tests were performed in [7] to determine the vibration and damping factors for hybrid carbon fiber composite pyramidal truss sandwich pannels with viscoelastic layers. There were used several different fiber orientations. The shock absorption performance of a motorbike helmet with aluminum honeycomb reinforced liner was obtained in [8]. Some composite platbands with kevlar-carbon and carbon fibers were studied in [9]. There were determined half of the damping factor per unit mass and the natural frequencies. In [10], for unidirectional kevlar (abbreviated as KFRP) and carbon fibre (abbreviated as CFRP) reinforced epoxy resin composites the effects of interfacial coating and temperature on the facture behaviour were studied. The infleunce of fiber length and fiber orientation on damping and stiffness for polymer composite materials (such as: fiberite 934 resin casting, AVCO5505 resin casting, graphite/epoxy, aramid (kevlar)/epoxy, boron/epoxy) was described in [11]. The decomposition behaviour of kevlar 49 fibers at 500^0C and 550^0C has been elucidaded in [12]. The

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isothermal decomposition of the fibers is characterized by: progressive reduction and eventually total loss in fiber cristalinity, progressive loss in weight, surface damages, hollowness and deterioration in tensile properties. The non linear or linear visco-elastic response for aramid fibers was studied in [13]. There were made some tests to measure the creep and relaxation response. Other researches related to Kevlar material can be found in [14] and [15].

Researches from engineering literature, regarding composite sandwich platbands with honeycomb core, are presented in [16]. Most of them include Nomex, aluminium, paper, polypropylene or kevlar honeycomb combined mostly with carbon or glass fiber. In this paper, the authors study the loss and damping factors, Young modulus and flexural rigidity for new types of composite platbands, made from classical materials (Kevlar honeycomb core with hexagonal and rectangular holes reinforced with epoxy/Kevlar-carbon fabric) combined in order to obtain an original sandwich platband. There was used the same epoxy resin as in [16]. The hardening time of the epoxy resin used in this paper was of 24 hours at room temperature. The difference of the studied material and the materials from other papers is the combination of the sandwich platband parts.

2. Materials and methods

2.1 Loss and damping factors, young modulus and flexural rigidity calculus from the samples dynamic response

The authors have build some new composite sandwich platbands marked in this way: sample 1 – rectangular honeycomb core, thickness 4 mm, width 60 mm; sample 2 – hexagonal honeycomb core, thickness 4 mm, width 60 mm. These values are constants in the experimental tests. A general view with the used samples is presented in Figure 1. The authors have clamped the platbands at one end and these were left free at the other end. At the free end, the authors have placed an B&K 8309 accelerometer in order to record the free vibrations. Like in [17], the authors have considered several free lengths values for the samples (values in mm): 200, 210, 220, 230, 240, 250, 260, 270, 280, 290, 300. The accelerometer is connected to a measuring system which is compound by: signal conditioner B&K NEXUS 2692-A-0I4 and data acquisition system HBM SPIDER 8. The SPIDER 8 apparatus is connected to a notebook through USB port. A similar experimental setup was used before by in [17] where good results for other composite sandwich platbands were obtained.

From the free vibrations recording, the damping factor per unit mass can be determined in

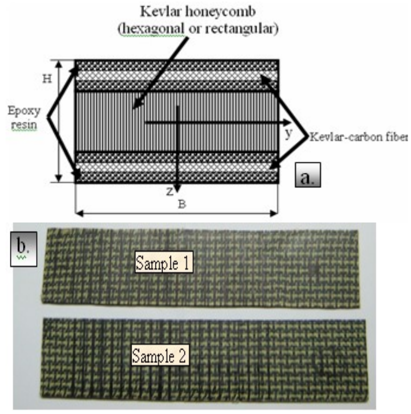

Fig. 1 - a. A scheme of the samples transversal section / *O schemă cu secţiunea transversală a epruvetelor* ; b. A general view with the used samples / *O vedere generală cu epruvetele utilizate.*

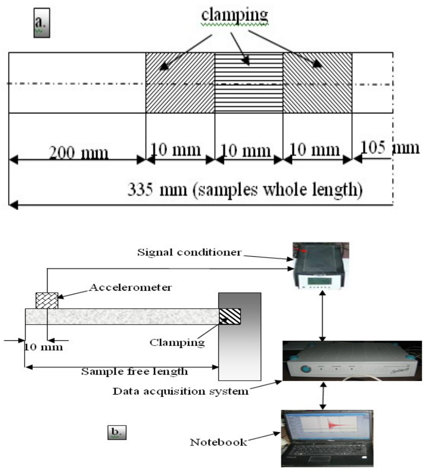

Fig. 2- a. The samples free lengths / *Lungimile libere ale epruvetelor*; b. Experimental setup scheme / *Schema montajului experimental.*

this way [17]: the values where the displacement is zero are determined (where the graph intersects the time axis); the period of movement cancellation *T* is determined (the time gap between two succesive cancellations); the eigenfrequency *v* and the eigenpulsation *ω* is determined with (1); the damping factor per unit mass μ is determined with (1) (where b_i and $b_{i+\Phi}$ are the graphs peaks separated by periods; Φ is the number of the cycles); the damping factor per unit length *c* is determined with (1) (where *ρ* is the material density and *A* is the section transversal area).

$$v = T^{-1};\ \omega \approx 6.2831853 \cdot T^{-1};\ \mu = \Phi^{-1} \cdot T^{-1} \cdot \ln(b_i \cdot b_{i+\Phi}^{-1});\ c = 2 \cdot \Phi^{-1} \cdot T^{-1} \cdot \ln(b_i \cdot b_{i+\Phi}^{-1}) \cdot \rho \cdot A \quad (1)$$

In order to determine the loss factor, the authors start from the basic formula of a vibration given in [18] and the relation (2) is obtained (where ω_1 is the pulsation; *ω*, *v* are the eigenpulsation and eigenfrequency; *t* is the time; *μ* is the damping factor per unit mass).

$$x(t)=e^{-\mu t}\cdot\sin(\omega_1 t+\varphi)=e^{-\frac{\eta}{2}\cdot\omega\cdot t}\cdot\sin(\omega_1 t+\varphi)\Rightarrow\mu=\frac{\eta}{2}\cdot\omega=\frac{\eta}{2}\cdot 2\cdot\pi\cdot\nu\Rightarrow\eta=\frac{1}{\pi}\cdot\frac{\mu}{\nu} \quad (2)$$

$$E_{dyn}\approx 38.32\cdot\rho\cdot\left(\frac{l^2\cdot\nu}{g}\right)^2 \quad (3)$$

$$\nu=22.4\cdot\sqrt{\frac{E\cdot I}{A\cdot\rho\cdot l^4}}\Rightarrow E=\frac{\nu^2\cdot\frac{\pi\cdot d^2}{16}\cdot l^4\cdot\rho}{\frac{\pi\cdot d^4}{64}\cdot 22.4^2}=\frac{1}{31.36}\cdot\rho\cdot\left(\frac{l^2\cdot\nu}{d}\right)^2 \quad (4)$$

According to [19] and [20], for the first eigenmode, the dynamic Young modulus can be determined with (3) for platbands clamped at one end and free at the other (where *38.32* is a constant obtained from the platbands end conditions, meaning the (*12/0.5596*^{2}) value; ρ is the material density; l is the free length; ν is the eigenfrequency and g is the platband thickness). According to [19] and [20], this method is standardised and accurate. A similar method, but with different supporting type was used in [21] for the dynamic Young modulus determination. In [21] the composite platbands were suspended by two nylon strings, not clamped like in this paper. An accelerometer was placed to record the vibratory response and the platband excitation was made with an impact hammer. The used specimen had a circular transversal section with a d diameter. For the elasticity modulus, the formula (4) was obtained, starting from the eigenfrequency calculus relation. The eigenfrequency was determined from the frequency response functions. A similar procedure was used by us in [16] for a platband clamped at one end and free at the other.

If the relations (3) and (4) are comapred, it can be seen that, for the elasticity modulus E determination, the material density ρ, the platband length l and the eigenfrequency ν values are used. The thickness g appears in (3) because the used samples have a rectangular section instead of diameter d which appears in (4) because the samples have a circular section. The only difference between the relations (3) and (4) is the constant value. This can be explained by the fact that the platbands have different boundary conditions: in our case these are clamped and in [21] these are simple supported. So it can be concluded that the method used in this paper was also used by other authors for composite materials, like the example in [21], but for different supporting conditions. In the vibrations theory [18], depending on the platband supporting conditions, for certain parameters like the natural frequency, different constants are used in the calculus relations.

For the flexural rigidity, according to [22], the formula (5) (where *3.194* is a constant obtained from platbands end conditions for the first eigenmode, meaning the ratio *[(2·π)/3.515625)]*^{2}*)* was used with good results for sandwich platbands with polystyrene and polypropylene honeycomb core.

$$EI\approx 3.194\cdot\rho\cdot A\cdot\nu^2\cdot l^4 \quad (5)$$

Important remark: in the relations (3) and (5) there were taken into account the first digits after the point that do not influence the flexural rigidity or the dynamic Young modulus values. Like in [9, 16, 17, 21 or 22], the authors have used the Euler-Bernoulli theory for obtaining the mechanical properties calculus relations.

In order to validate the flexural rigidity value obtained with the formula (5), the formula (6) from [23] is used to determine the flexural rigidity for various composite sandwich beams with balsa, plastic and foam materials used as cores.

$$EI=\frac{m\cdot l^4\cdot\nu^2}{C_1^2} \quad (6)$$

According to [23], in (6) m is the specific mass, l – is the platband free length, ν – is the frequency of the first eigenmode and C_1 is a constant of the first mode and has *0.56* value.

In order to avoid any possible errors inserted by the experimental equipment, a "high-pass" Butterworth filtration at 4 Hz was made. Some experimental recordings with the damping factor per unit mass for 5 cycles (the cycles are delimited by the two dotted vertical parallel lines) and eigenfrequency calculus are presented in Figures 3 and 4.

Important remark: because the form of the platbands deformed medium fiber is similar to their first vibration eigenmode, the measured frequency was considered as the first eigenfrequency.

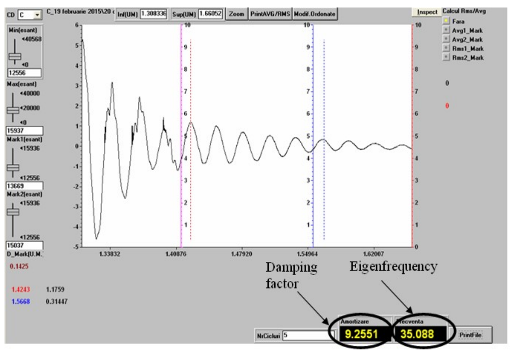

Fig. 3 -Damping factor and eigenfrequency for the sample 1, 250 mm free length / *Factorul de amortizare şi frecvenţa proprie pentru epruveta 1, lungimea liberă de 250 mm.*

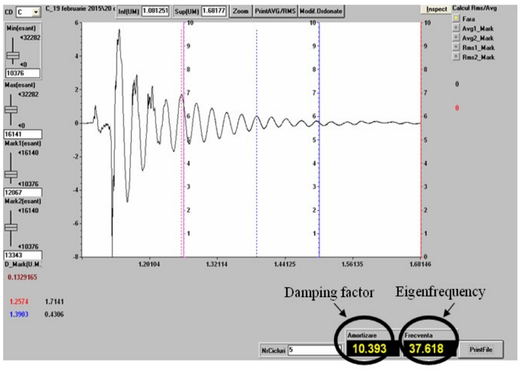

Fig. 4 - Damping factor and eigenfrequency for the sample 1, 240 mm free length/ *Factorul de amortizare şi frecvenţa proprie pentru epruveta 1, lungimea liberă de 240 mm.*

2.2 Experimental determination of the flexural rigidity from static loading

In order to validate the flexural rigidity from the samples dynamic response, the authors proposed an experimental montage for flexural rigidity calculus from the samples static loading (with a known force) and displacement (measured with a displacement transducer WA 50 mm (the displacement measuring domain is between 0..50 mm) made by HBM connected with the data acquisition system SPIDER 8, directly recognized by it. The experimental montage is presented in Figure 5 and the obtained displacement graphic in Figure 6. The flexural rigidity was obtained by considering the platband being clamped at one end and left free at the other. At 5 mm distance from the free end, a force is applied and the displacement of the point at 60 mm from the free end is measured.

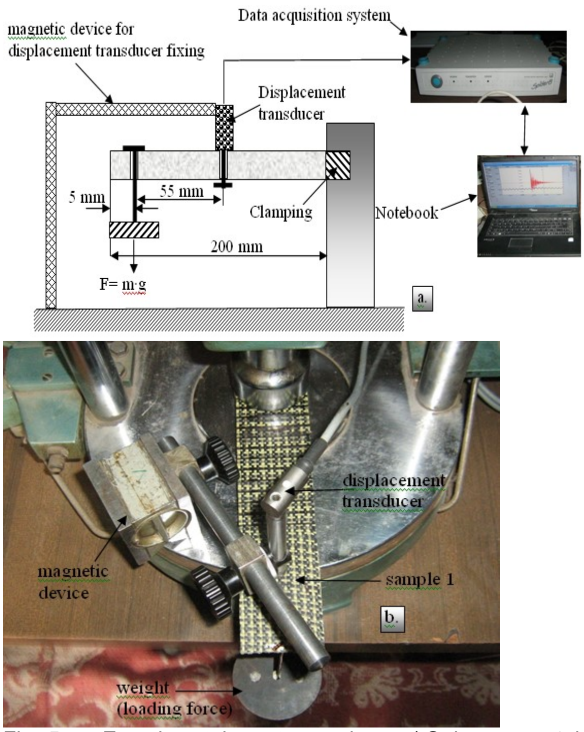

Fig. 5- a. Experimental montage scheme / *Schema montajului experimental*; b. Sample 1 connection / *Conectarea epruvetei 1.*

In this case, by using the Mohr-Maxwell method, the flexural rigidity at 60 mm from the free end can be determined with (8).

In (8), the value of force *F* is in *N* and *δ* must be multiplied with 10^{-3} because the value is obtained in *m* and the displacement transducer records values in *mm*.

$$EI = \delta^{-1} \cdot 10^{3} \cdot \int_{0}^{0,140} F \cdot (0.055 + x) \cdot x dx \approx C \cdot F \cdot \delta^{-1},$$

C= 1.453(6) (8)

3. Results and discussions

3.1. Dynamic loading

The other important characteristics for the samples characterization are: sample 1: mass 0.02 kg; density 248.756 kg/m^3; sample 2: mass 0.022 kg; density 273.632 kg/m^3. These values are considered constants for all the samples during the experiments. In the Figures 3 and 4 the damping factor per unit mass and eigenfrequency calculus was presented for the sample 1, with the free lengths of 250 and 240 mm. All the important geometrical characteristics, experimental results and mechanical characteristics (flexural rigidity, dynamic Young modulus, loss factor, damping factor per unit length) are written in Table 1. The average values for the flexural rigidity and dynamic Young modulus, for each samples are: sample 1: flexural rigidity EI ≈ 0.926 Nm^2; dynamic Young modulus= 2877 MPa; sample 2: flexural rigidity EI ≈ 1.128 Nm^2; dynamic Young modulus≈ 3507.273 MPa.

The flexural rigidity calculated with 5 and 6 is written in Table 2. Also, the errors between the calculus formulas are written in the same table. The flexural rigidity is given in *Nm^2*, the platbands free length in *mm* and the errors in *%*. From the Table 1 analysis, it can be seen that both (5) and (6) formulas can be succesfully applied for the flexural rigidity calculus from the first eigenmode frequency, because errors bellow 1% were obtained.

By using the same methodology from [17], direct calculus formulas were obtained for the damping factor per unit mass *μ* versus the platbands free length. The calculus formula is given in (7).

$$\mu(l) = a + b \cdot x + c \cdot x^{2} + d \cdot x^{3} + e \cdot x^{4} + f \cdot x^{5} + + g \cdot x^{6} + h \cdot x^{7} + i \cdot x^{8} \qquad (7)$$

Table 1

Experimental results, geometrical and mechanical characteristics/ *Rezultate experimentale, caracteristici geometrice şi mecanice*

Width [mm]	Specific mass [kg/m]	Free length [mm]	Eigenfrequency [1/s]	Damping factor per unit mass [(Ns/m)/kg]	Sample no.	Damping factor per unit length [(Ns/m)/m]	Loss factor	Dynamic Young modulus [MPa]
60	0.06	200	56.901	16.095	1	1.931	0.09	3086
60	0.06	210	49.627	13.311	1	1.597	0.085	2854
60	0.06	220	44.568	12.133	1	1.456	0.087	2772
60	0.06	230	42.248	10.944	1	1.313	0.089	2976
60	0.06	240	37.618	10.393	1	1.247	0.088	2797
60	0.06	250	35.088	9.2551	1	1.111	0.084	2865
60	0.06	260	32.884	9.1287	1	1.095	0.088	2944
60	0.06	270	29.575	8.4125	1	1.009	0.091	2769
60	0.06	280	28.436	7.3667	1	0.884	0.082	2961
60	0.06	290	25.451	6.8236	1	0.819	0.085	2729
60	0.06	300	24.49	5.9036	1	0.708	0.077	2894
60	0.066	200	58.182	21.839	2	2.883	0.119	3550
60	0.066	210	53.215	16.942	2	2.236	0.101	3609
60	0.066	220	47.584	14.739	2	1.946	0.099	3476
60	0.066	230	43.597	13.427	2	1.772	0.098	3486
60	0.066	240	40.713	11.160	2	1.473	0.087	3604
60	0.066	250	37.588	9.671	2	1.277	0.082	3617
60	0.066	260	33.97	8.111	2	1.071	0.076	3456
60	0.066	270	31.704	6.623	2	0.874	0.066	3501
60	0.066	280	29.358	6.349	2	0.838	0.069	3472
60	0.066	290	26.936	5.469	2	0.722	0.065	3363
60	0.066	300	25.478	4.823	2	0.637	0.06	3446

Table 2

The flexural rigidity determined with (5) and (6), according to [22] and [23]
Rigiditatea la încovoiere determinată cu (5) şi (6), în conformitate cu referinţele [22] şi [23]

Free length	200	210	220	230	240	250	260	270	280	290	300
EI [22] (sample 1)	0.993	0.918	0.892	0.957	0.9	0.922	0.947	0.891	0.953	0.878	0.931
EI [22] (sample 2)	1.142	1.161	1.118	1.121	1.159	1.163	1.112	1.126	1.117	1.082	1.108
EI [23] (sample 1)	0.991	0.916	0.89	0.956	0.898	0.92	0.945	0.889	0.951	0.878	0.93
EI [23] (sample 2)	1.1399	1.16	1.116	1.119	1.157	1.16	1.109	1.1242	1.1149	1.08	1.1066
Errors (sample 1)	0.201	0.218	0.224	0.104	0.222	0.217	0.211	0.224	0.21	0	0.107
Errors (sample 2)	0.184	0.086	0.179	0.178	0.173	0.258	0.27	0.16	0.188	0.185	0.126

Table 3

Calculus formulas for the damping factor per unit mass versus the platbands free length/ *Formule de calcul pentru factorul de amortizare pe unitatea de masă în funcţie de lungimea liberă a platbandelor*

Damping factor	Parameters	Correlation factor R^2
Per unit mass [(Ns/m)/kg] (sample 1)	a= -26805.083; b= 3827.4702; c= -95.745828; d=1.1051479; e=-0.0072161929; f= $2.8435148\cdot10^{-5}$; g= $-6.7327979\cdot10^{-8}$; h=$8.8528865\cdot10^{-11}$; i= $-4.9820297\cdot10^{-14}$	0.998
Per unit mass [(Ns/m)/kg] (sample 2)	a= 352586.52; b= -10793.663; c=145.91184; d= -1.383766; e=0.0056072112; f= $-1.784866\cdot10^{-5}$; g= $3.5823478\cdot10^{-8}$; h=$-4.1394704\cdot10^{-11}$; i= $2.1051141\cdot10^{-14}$	0.999

Important remark: all the decimals must be kept from the coefficients because the functions must approximate best the experimental results.

According to Table 1, the next general conclusions can be extracted:

-the damping factors per unit mass and per unit length decrease with the platbands free length; this phenomenon can be expained by the fact that the inserted deformation energy, at the initial moment (by applying a force at the free end) is dissipated in a larger material quantity;

-the elasticity modulus have small variations with the free length; these variations can be expained by the fact that, in the initial deformation which was necessary to determine the damping and loss factors, the proportionality area between the stress and strain was over-passed; the results can be also explained by the fact that the material is not homogenuous and isotropic, so small variations on the platband length should exist; the proposed method is an approximate but it has the advantage that can be quickly applied in order to estimate a material dynamic elasticity modulus;

-the dynamic elasticity modulus is mainly influenced by the eigenfrequency value, the other parameters (free length, density and thickness) do not influence its result or have small infleunce; the same conclusion was obtained, for example, in [21];

- there were obtained better mechanical properties for the sample with hexagonal honeycomb core compared to the one with rectangular honeycomb core;

-if there is taken into account the results from this paper and other researches (like the ones from [17] or [21]), the damping factor per unit mass and per unit length must be experimentally determined for each type of material and sample, being difficult to obtain a quantitative corespondence with the parameters which influence directly or indirectly the damping phenomenon; the values of damping factors may depend on several features such as: sample dimensions, specific mass or the quantity of material from sample, elastic and damping properties of component materials; the sample width can influence the damping coefficient, by the fact that it determines the surface in which the air friction acts on the sample.

3.2 Static loading

If the results from Figure 6 obtained with the displacement tranducer are inserted in relation (8), then the next stiffness values are obtained for sample 1: EI ≈ 0.928 N·m^2 (for F= 3N) and EI ≈ 0.929 N·m^2 (for F= 4.5N). Similar results are obtained for both loading variants. The sample 2 is loaded with the F= 3 N and the next value is obtained: sample 2: EI≈ 1.112 Nm^2.If the results are compared with the ones obtained from vibrations, small errors are obtained (under 7%).

The errors between the flexural rigidity values obtained from the first eigenmode frequency and the ones obtained from the static loading can be explained by the fact that, in the first case the flexural rigidity is influenced by the dynamic Young modulus and in the second case by the static one. It is known, from many researches that these moduli can have different values. For example, results of static and dynamic Young modulus between different materials, especially from the rocks study area, are presented in [24], with an increased valued for the dynamic one. The same conclusion regarding a higher value for the dynamic Young modulus can be extracted from [25]. Not for all materials the dynamic modulus is higher than the static one. For example, in [26] the static Young modulus is larger than the dynamic one by the factor 2.

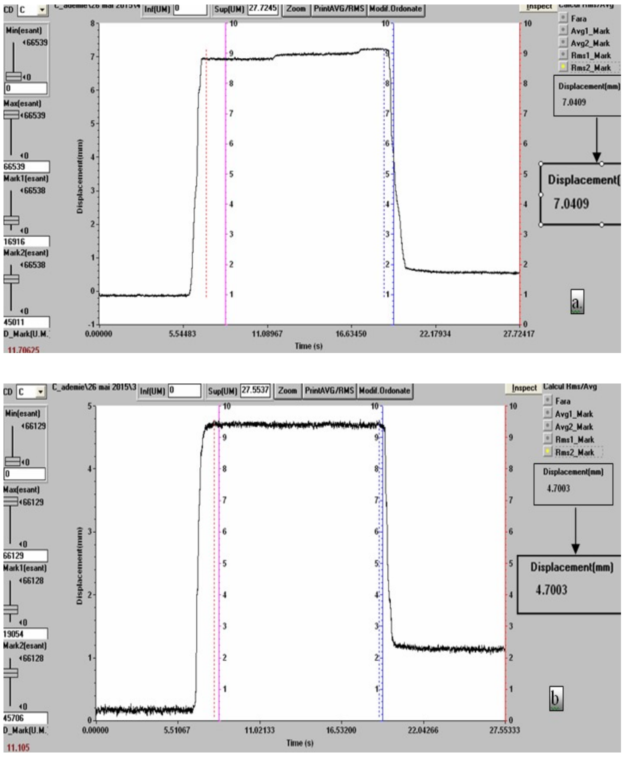

Fig. 6 - a. Displacement at F= 4,5 N / *Săgeata la F= 4,5 N*;
b. Displacement at F= 3N / *Săgeata la F= 3 N.*

This type o structure can be used to strengthen thin reinforced concrete slabs, for example at the study described in [27]. In a future research, the authors intend to apply the procedure of failure mode maps from [28] to sandwich platbands with kevlar honeycomb core with different reinforcements. Using different reinforcement distributions, their effect upon the composite platbands mechanical properties can be studied. And for the damping characteristics, the procedure from [29] can be applied and comparisons with the model presented in this study can be made.

4. Conclusions

The added value of the study presented in this paper is:

-making some new original composites platbands, with classical elements (hexagonal or rectangular kevlar honeycomb core reinforced with kevlar-carbon fiber);

-determining some mechanical characteristics for these sandwich platbands like: flexural rigidity (from the free vibrations and from the platband static loading by measuring its displacement), dynamic Young modulus, eigenfrequency, damping factor per unit mass and per unit length and the loss factor (which characterizes the material vibration damping capacity, without taking into account the sample dimensions);

- proposing new sandwich 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, concrete forming or to strengthen thin reinforced concrete slabs.

This kind of materials are usually used for structures that are under vibrations during their working time (like ships, planes, buses or cars floor building; to strengthen the walls for different civil constructions; bikes or motorbikes helmets), that is why it is important to know how the oscillations decay in these composite sandwich systems after an external disturbance (fact defined by the damping coefficient). These sandwich bars are also usually subjected to bending, so it is important to know how they resist against bending deformation, phenomenon characterized by the bending stiffness value determined in this research.

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