COMPORTAMENTUL VIBRATORIU AL PLATBANDELOR SANDWICH COMPOZITE RANFORSATE CU FIBRĂ DE STICLĂ
THE VIBRATION BEHAVIOUR OF COMPOSITE SANDWICH PLATBANDS REINFORCED WITH GLASS FIBER

CRISTIAN OLIVIU BURADA[1*] , COSMIN MIHAI MIRIȚOIU[2], MARIUS MARINEL STĂNESCU[3], DUMITRU BOLCU [4]

[1,4] Universitatea din Craiova, Facultatea de Mecanică, Dep. de Mecanică Aplicată şi Construcții civile, Strada Calea Bucureşti, Nr. 107, Cod 200512, Craiova, România,
[2] Universitatea din Craiova, Facultatea de Mecanică, Dep. de Autovehicule, Transporturi şi Inginerie Industrială, Strada Calea Bucureşti, Nr. 107, Cod 200512, Craiova, România
[3]Universitatea din Craiova, Dep. de Matematică Aplicată, Strada A.I. Cuza, Nr. 13, Cod 200396, Craiova, România

In this paper, we have built some new composite sandwich platbands with polypropylene honeycomb core with the thickness of 10, 15 and 20 mm reinforced with two layers of glass fiber. We have chosen for the platbands the next width values: 45 and 60 mm. Starting from the dynamic response of these sandwich platbands with damping, that are in free vibration, we have established an experimental procedure to determine the damping factor per unit mass and unit length, loss factor, stiffness and the dynamic elasticity modulus.

Pentru această lucrare, am construit câteva platbande sandwich cu miez din fagure de polipropilenă cu grosimea de 10, 15 şi 20 mm ranforsate cu două straturi din fibră de sticlă. Am ales pentru platbande următoarele valori ale lățimilor: 45 şi 60 de mm. Pornind de la răspunsul dinamic al acestor platbande cu amortizare, care se află în vibrație liberă, am stabilit o procedură experimentală pentru a determina factorul de amortizare pe unitatea de masă şi lungime, factorul de pierdere a energiei şi modulul de elasticitate dinamic.

Keywords: *damping factor, loss factor, dynamic elasticity modulus, glass-fiber*

1. Introduction

The vibration behaviour study, especially the damping and loss factors, of platbands and plates has developed since 1950 when the flexural vibrations of plates, walls or platband-like structures were damped by polymeric damping materials [1]. In [1] there is made a review and a comparison regarding the methods of measuring the loss factors of heavily damped platbands and plates damped by uniform layers of visco-elastic damping. In [2] the external damping losses in measuring the vibration damping properties in lightly damped specimens using transient time-domain methods was investigated. There were determined free vibrations of steel plate specimens to list up several causes of external damping sources. As a general conclusion, it was shown that the damping capacity of one type of steel alloy cannot be generalized to other steel-alloys structures.

The flexural vibrations damping in glass fiber composite plates containing one and two dimensional acoustic black holes were determined in [3]. There were studied sheets of composite with samples tested, the addition of two-dimensional acoustic black holes resulted in further increase in damping of resonant vibrations. Because the used composite materials had large values of the loss factor, there was no need in adding small pieces of absorbing layers to the indentations to achieve desirable levels of damping. In [4] there was used the Rayleigh-Ritz model and the finite element method to study the vibration and damping characteristics of free-free composite sandwich cylindrical shell with pyramidal truss-like cores manufactured using a hot press molding method. There were used modal tests to obtain the modal characteristics. As a general conclusion, there was obtained that the natural frequencies of the composite cylindrical shell increased with the ply angle increase from the inner and outer curve face sheets. Also, the loss factor for the studied shells did not increase monotonically.

A combination between Rayleigh-Ritz method and Reissner-Mindlin first-order shear deformation theory was used in [5] for prediction of the vibration and damping properties of composite honeycomb sandwich panels. A method for

* Autor corespondent/*Corresponding author,*
E-mail: cristian.burada@yahoo.com

resonance frequencies prediction and modal loss factors of platbands and damped samples was presented in [6]. The novelty inserted in [6] was characterized by the consistent modeling of the suspension with spring-damper elements defined with stiffness and damping factors with fixed values over the whole considered frequency range. There were obtained similar results by using experimental and finite element analysis methods.

In [7], by solving the Lagrange equation, there were obtained the free vibration frequencies and model loss factors of a plate. Then, when the concentrated force is suddenly withdrawn, the transient response of constrained damping is obtained. The next general conclusions were obtained: in both the two methods used (experimental and theoretical) the frequencies and the response time of theoretical calculation and the variation law with the parameters of the damping layer are identical.

In the work [8], the viscoelastic properties (the storage modulus and loss modulus) responses were obtained for aluminum 2024 alloy, carbon/fiber epoxy, glass fiber/epoxy and aluminum 2024/carbon fiber/epoxy and aluminum 2024 alloy/glass fiber epoxy composites. The experimental results were compared to calculated *E* modulus values by using micromechanics approach. The experimental values have shown that the experimental values are very close to the theoretical ones.

In the [9] study, there were studied the damping properties of laminated structures made of steel, rubber or epoxy adhesive and glass fiber reinforced composite. The loss factors of the hybrid structures were determined by frequency and time domain methods. The loss factors results of the constituent materials were used to estimate the loss factors of the hybrid structures by the rule of mixtures and both the theoretical results and experimental ones have shown small errors. In [10] there is made an asymptotic analysis of flexural dynamic characteristics for a sandwich plate with periodically perforated viscoelastic damping material core.

The vibration damping characteristics of nanocomposites and carbon fiber reinforced polymer composites containing multi-wall carbon nanotubes have been studied by using the free and forced vibration tests [11]. After the study, the dynamic loss modulus and loss factor of the nanocomposites and the corresponding carbon fiber reinforced polymer composites show appreciable increase with the addition of multi-wall carbon nanotubes, fact that is an indication of enhanced damping performance. Nonlinear vibrations of thin rectangular plates were considered in [12] by using the von Karman equations in order to take into account the effect of geometric nonlinearities. There was made an investigation in these plates of modal interaction in free and forced vibrations. There were compared the nonlinear normal modes of the undamped system with the observable periodic orbits of the forced and damped structure. Composite materials reinforced with glass fibers were used in many experimental investigations and tests. For example, glass fibers reinforced composites with polyester matrix used in construction building were studied in [13]. There was made a fatigue stand to test these composites. The method described in this paper also allows the testing of joints between parts on composites.

The novelty inserted by the authors in this paper is characterized by the sandwich platbands: made from classical parts (with polypropylene honeycomb core reinforced with two layers of glass fiber) combined in an original way. The studies performed in this paper are a continuation of the ones made by authors in [14].

2. General considerations regarding the transversal vibrations of platbands

The movement equations for the viscoelastic platbands transversal vibrations with external damping and constant section are presented in [15]. The researches from [15] have shown that if the ratio between the length and width is above 10, the differences between Timoshenko and Euler-Bernoulli theories for the bending moment, shear force and the medium fiber are not above 5%. The damping influence over the platband section rotational movement can be neglected for the first eigen modes.

For a platband with constant section, the motion equation for transversal vibrations can be written as [14-16]:

$$\ddot{v}+\dot{v}\cdot 2\cdot\mu+\frac{\langle E\cdot I\rangle}{\langle\rho\cdot A\rangle}\cdot\frac{\partial^4 v}{\partial x^4}=\frac{q(x,t)}{\langle\rho\cdot A\rangle} \quad (1)$$

In (1) we have marked with: *v(x,t)*- the transversal displacement of the platband section elastic center; $\dot{v}(x,t)$- the transversal speed of the platband section elastic center; $\ddot{v}(x,t)$- the transversal acceleration of the platband section elastic center; *ρ(x,t)*- is the platband material density; *E(x,y)*- is the Young modulus of the platband material; *q(x,t)*- is the transversal force that loads the platband; *μ*- is half of the damping factor per unit mass; *A*- the transversal section.

The damping calculus can be made by studying the free vibrations (q(x,t)=0) produced by an initial platband deformation [14-16]:

$$v(x,0)=h(x);\ \dot{v}(x,0)=0 \quad (2)$$

where h(x) is the initial deformation.

In this case, the free vibration of the platband can be determined with (3) [15-16]:

$$v(x,t)=\sum_{\xi=1}\int_0^l h(x)\cdot H_n(x)dx\cdot e^{-\mu t}\cdot\left(\Omega_\xi+\Psi_\xi\right)\cdot H_n(x) \quad (3)$$

$$\Omega_\xi=\frac{\mu}{\sqrt{\alpha_\xi^4\cdot\frac{\langle EI\rangle}{\langle\rho A\rangle}-\mu^2}}\sin\left(\sqrt{\alpha_\xi^4\cdot\frac{\langle EI\rangle}{\langle\rho A\rangle}-\mu^2}\cdot t\right),\ \Psi_\xi=\cos\left(\sqrt{\alpha_\xi^4\cdot\frac{\langle EI\rangle}{\langle\rho A\rangle}-\mu^2}\cdot t\right) \quad (4).$$

$$\nu_\xi=C_\xi\cdot\left[U(\alpha_\xi x)-\frac{S(\alpha_\xi\cdot l)}{T(\alpha_\xi\cdot l)}\cdot V_\xi(\alpha_\xi\cdot x)\right] \quad (5)$$

$$\nu_\xi\approx 0{,}1591549\frac{(EI)^{0,5}\cdot\alpha_\xi}{(\rho A)^{0,5}\cdot l^2} \quad (6)$$

In (3) and (4) we have marked with: μ- half of the damping factor per unit mass of the platband; $H_n(x)$- are the eigen functions that depend on the platband ends conditions; α_ξ - is determined from the platband ends conditions; l- the free length of the platband.

If the platband is clamped in x=0 and free in x=l, the eigen functions can be determined with (5) according to [15] and [16] or with (6) according to [14].

In (5) the parameter C_ξ can be determined with (7), from the orthogonally conditions [15], [16]. In (5) and (6) we have marked with: *S,U,T,V*- the Krîlov functions; *EI*- the platband stiffness; *l*- the platband length; ρ- the material density of the platband; *A*- the transversal area of the platband.

The free vibrations experimental recording gives the possibility of damping calculus in this way:

- there are determined the values where the displacement is zero;
- there is determined the cancellation movement period (more precisely *T* is the double time gap between two consecutive cancellations);
- the frequency ν and the pulsation ω are determined with (7)[14];
- the damping factor per unit mass is determined with (8)[14].

$$\nu=T^{-1};\ \omega=2\pi\cdot T^{-1}\approx 6{,}2831853\cdot T^{-1} \quad (7)$$

$$\mu=T^{-1}\cdot\ln\frac{\beta_i}{\beta_{i+1}} \quad (8)$$

- the damping factor per unit length is determined with (9)[14]:

$$C=2\cdot\mu\cdot\langle\rho A\rangle \quad (9)$$

In (6) we have marked with β_i, β_{i+1} the maximums separated by periods.

The most common procedure used to determine the loss factor is The Half-Power Bandwidth Method presented in [17-19]. According to this theory, the loss factor can be determined with (10).

$$\eta=(\nu_2-\nu_1)\cdot\nu_{max}^{-1} \quad (10)$$

In Figure 1 there was marked with Θ_{max} the maximum amplitude, ν_1 and ν_2 the frequencies at the $\Theta_{max}/\sqrt{2}$ value and ν_{max} the maximum peak frequency. If the eigen modes are studied, in [20] was presented a formula (given as the (11) formula in this paper) for the loss factor calculus which depends on the eigen frequencies values.

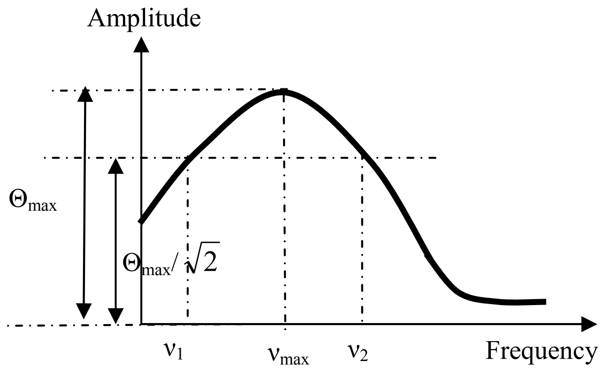

Fig. 1. - The Half-Power Bandwidth Method/ *Metoda Half-Power Bandwidth*

$$\eta_\xi=\mathrm{Im}(\nu_\xi^2)\cdot\mathrm{Re}(\nu_\xi^2)^{-1} \quad (11)$$

In (10) there were marked with $\mathrm{Im}(\nu_\xi^2)$ and $\mathrm{Re}(\nu_\xi^2)$ the imaginary and real parts of the eigen frequency. In [1], for platbands with damping, the loss factor can be determined with (12). In [16], there is presented the basic formula for a vibration. Starting from this formula, we can establish a correlation between the loss factor, half of the damping factor per unit mass and the eigen frequency(marked in this paper as relation (13)).

η= (1/2·π)·(Total energy dissipated in the system per harmonic cycle/ maximum potential energy stored during the cycle) (12)

In the relation (13) we have marked with: ζ- critical damping; ω- the pulsation; ω_ξ- eigen pulsation; μ- damping factor per unit mass; η- loss factor; ν_ξ- the eigen frequency; *t*- time.

$$x(t) = e^{-\mu t} \cdot \sin(\omega \cdot t + \varphi) = e^{-\zeta \cdot \omega_\xi \cdot t} \cdot \sin(\omega \cdot t + \varphi) [1]$$

$$\omega = \omega_\xi \text{ (small damping) } \Rightarrow \mu = \zeta \cdot \omega_\xi = \zeta \cdot 2 \cdot \pi \cdot \nu_\xi \Rightarrow \zeta = \frac{\mu}{2 \cdot \pi \cdot \nu_\xi}, \eta = 2 \cdot \zeta \ [18] \Rightarrow$$

$$\eta = \frac{\mu}{\pi \cdot \nu_\xi} \tag{13}$$

In [19] there is also presented a formula, for a platband with clamped-free boundary conditions, to determine the dynamic Young modulus (marked in this paper as relation (14)).

$$E_{din} = \frac{12 \cdot \rho \cdot l^4 \cdot \nu_\xi^2}{\lambda_\xi^2 \cdot g^2} \tag{14}$$

In (14) we have marked with: ρ- the material density; l- the free length of the platband; ν- the eigen frequency; λ= 0,5596 for the first eigen mode [21]; g- the platband thickness.

3. Experimental setup, recordings and results

We have built some new original platbands with polypropylene honeycomb core reinforced with two layers of glass fiber. We have marked the samples in this way:
- set 1: width= 45 mm; thickness= 10 mm;
- set 2: width= 45 mm; thickness= 15 mm;
- set 3: width= 45 mm; thickness= 20 mm;
- set 4: width= 60 mm; thickness= 10 mm;
- set 5: width= 60 mm; thickness= 15 mm;
- set 6: width= 60 mm; thickness= 20 mm.

We have clamped the platband at one end and we have placed an accelerometer at the free end (10 mm distance from the edge, according to Fig. 2). The bar is rigidly fixed in a massive press (with the mass of 80 kg) and has no freedom degree (is immovable).

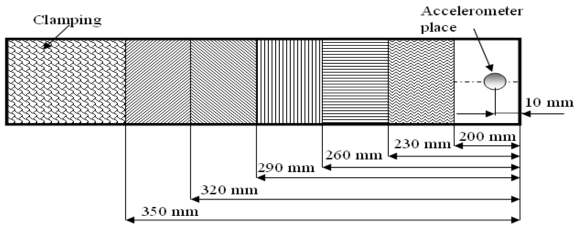

Fig. 2 - Experimental setup/ *Montaj experimental*.

We have considered more variants of the platbands free length: 350 mm, 320 mm, 290 mm, 260 mm, 230 mm and 200 mm. The clamping method is similar to the one presented by the authors in [14] ([14], Fig. 7,8). A general view with the tested samples is presented in Figure 3.

The geometrical characteristics of the platbands are written in Table 1.

Table 1

Samples geometrical characteristics/ *Caracteristici geometrice ale epruvetelor*

Width *Lățimea* (mm)	Thickness *Grosimea* (mm)	Free length *Lungimea liberă* (mm)	Samples set *Setul de epruvete*	Width *Lățimea* (mm)	Thickness *Grosimea* (mm)	Free length *Lungimea liberă* (mm)	Samples set *Setul de epruvete*
45	10	200	1	60	10	200	4
45	10	230	1	60	10	230	4
45	10	260	1	60	10	260	4
45	10	290	1	60	10	290	4
45	10	320	1	60	10	320	4
45	10	350	1	60	10	350	4
45	15	200	2	60	15	200	5
45	15	230	2	60	15	230	5
45	15	260	2	60	15	260	5
45	15	290	2	60	15	290	5
45	15	320	2	60	15	320	5
45	15	350	2	60	15	350	5
45	20	200	3	60	20	200	6
45	20	230	3	60	20	230	6
45	20	260	3	60	20	260	6
45	20	290	3	60	20	290	6
45	20	320	3	60	20	320	6
45	20	350	3	60	20	350	6

Fig. 3. General view with the samples/ *Vedere generală cu epruvetele.*

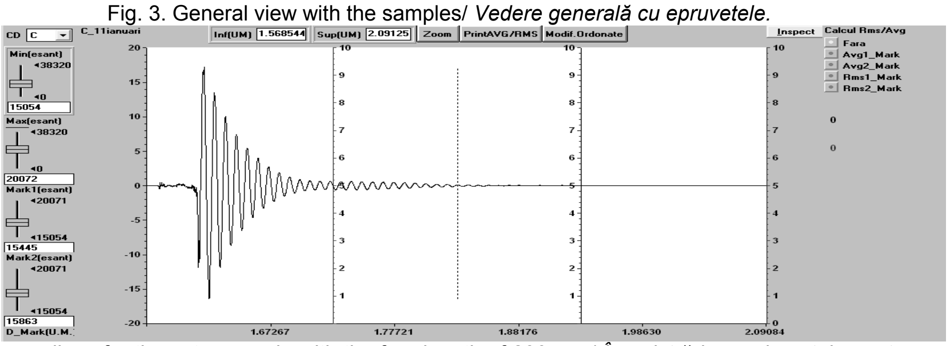

Fig. 4- Experimental recordings for the set 5 sample with the free length of 230 mm/ *Înregistrări experimentale pentru epruveta din setul 5 corespunzătoare lungimii libere de 230 mm.*

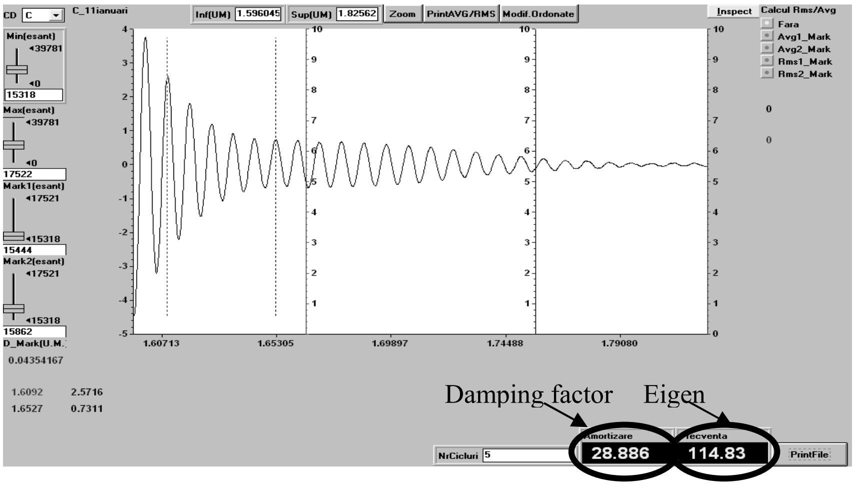

Fig. 5- Damping factor calculus for the set 2 sample with the free length of 230 mm/ *Calculul factorului de amortizare pentru epruveta din setul 2 corespunzătoare lungimii libere de 230 mm.*

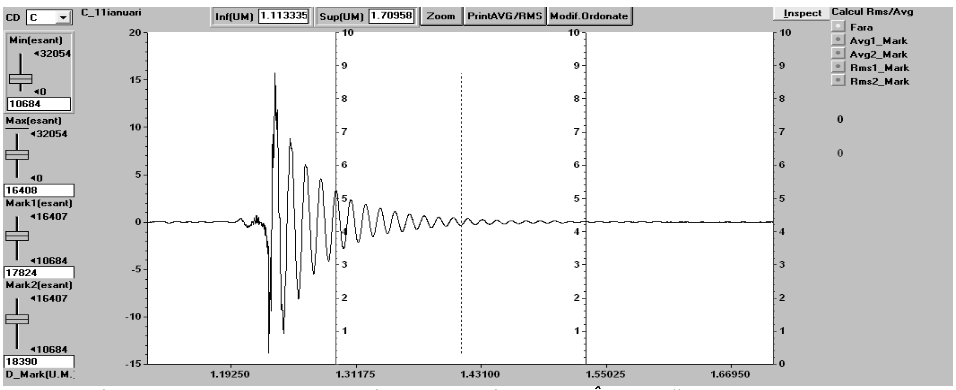

Fig. 6- Experimental recordings for the set 2 sample with the free length of 290 mm/ *Înregistrări experimentale pentru epruveta din setul 2 corespunzătoare lungimii libere de 290 mm.*

In the Figures 4 and 5 the experimental recordings and the damping factor calculus with the eigen frequency for the set 2 sample with the free length of 230 mm are presented. Other examples for the experimental recordings and the damping factor calculus in different variants are presented in Figures 6 - 9.

All the experimental data are written in Table 2. We have determined the stiffness with (6), damping factor per unit length with (9), the loss factor with (13) and the dynamic elasticity modulus with (14). All the results are given in table 3. The variation of the damping factor with the free length of the platband for all the set of samples is presented in Figures 10 - 12.

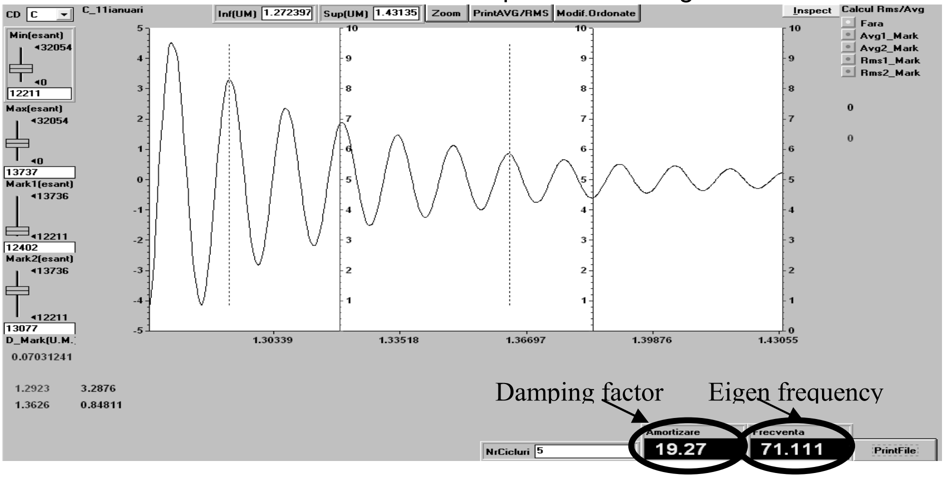

Fig. 7- Damping factor calculus for the set 2 sample with the free length of 290 mm/ *Calculul factorului de amortizare pentru epruveta din setul 2 corespunzătoare lungimii libere de 290 mm.*

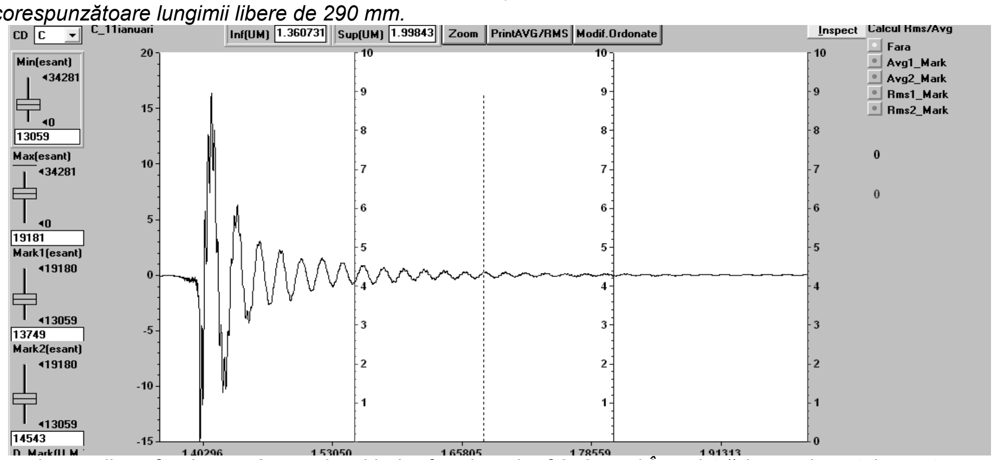

Fig. 8- Experimental recordings for the set 2 sample with the free length of 350 mm/ *Înregistrări experimentale pentru epruveta din setul 2 corespunzătoare lungimii libere de 350 mm.*

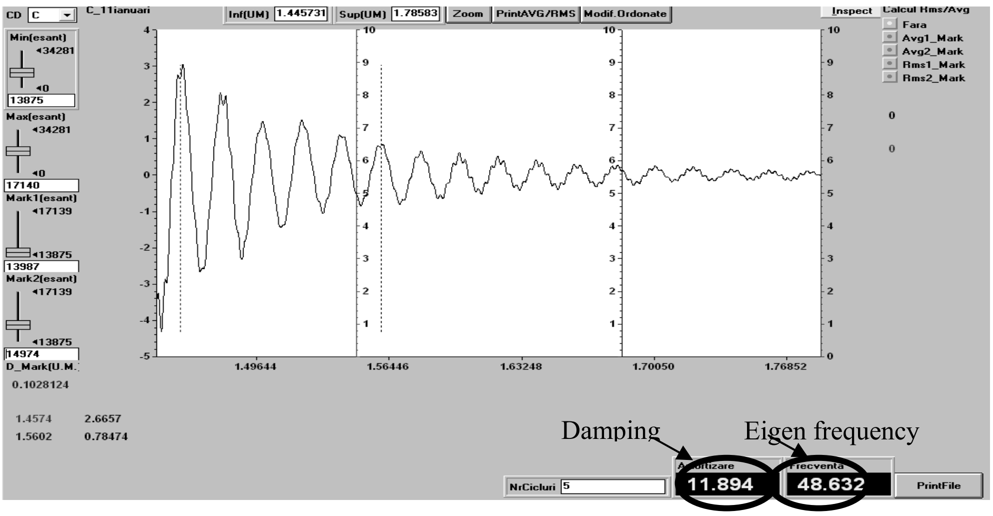

Fig. 9 - Damping factor calculus for the set 2 sample with the free length of 350 mm/ *Calculul factorului de amortizare pentru epruveta din setul 2 corespunzătoare lungimii libere de 350 mm.*

Table 2

Experimental data/ *Date experimentale*

Samples Set *Setul de epruvete*	Eigen frequency *Frecvența proprie* [1/s]	Damping Factor *Factor de amortizare* [(Ns/m)/kg]	Mass *Masa* [kg]	Samples Set *Setul de epruvete*	Eigen frequency *Frecvența proprie* [1/s]	Damping Factor *Factor de amortizare* [(Ns/m)/kg]	Mass *Masa* [kg]
1	122.22	24.868	0.078	4	122.3	25.246	0.078
1	96.386	19.464	0.078	4	93.204	19.15	0.078
1	70.796	16.463	0.078	4	72.948	15.34	0.078
1	57.831	13.603	0.078	4	60.302	12.765	0.078
1	47.104	10.556	0.078	4	48.747	10.992	0.078
1	39.604	9.2173	0.078	4	39.967	8.5774	0.078
2	144.14	34.434	0.088	5	159.13	29.556	0.088
2	114.83	28.886	0.088	5	115.585	24.535	0.088
2	85.359	23.591	0.088	5	91.603	19.738	0.088
2	71.111	19.27	0.088	5	76.555	16.952	0.088
2	60	16.399	0.088	5	63.482	13.258	0.088
2	48.632	11.894	0.088	5	50.42	11.755	0.088
3	179.61	36.658	0.096	6	182.16	37.968	0.096
3	142.99	30.274	0.096	6	134.83	31.336	0.096
3	108.38	25.934	0.096	6	109.84	26.755	0.096
3	87.912	22.209	0.096	6	86.486	21.706	0.096
3	73.283	18.945	0.096	6	73.282	17.876	0.096
3	60.206	14.508	0.096	6	59.685	13.63	0.096

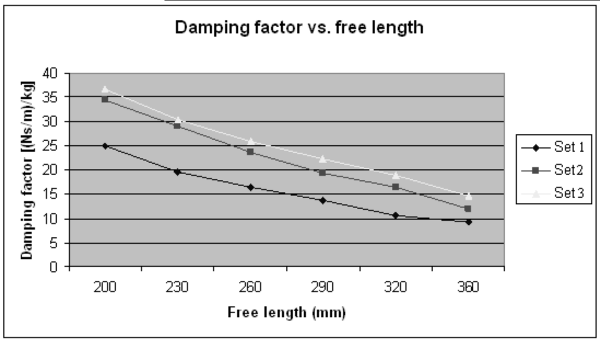

Fig. 10. - The damping factor vs. the free length of the samples for the 1,2,3 sets/ *Variația factorului de amortizare în funcție de lungimea liberă a platbandelor pentru seturile 1,2,3.*

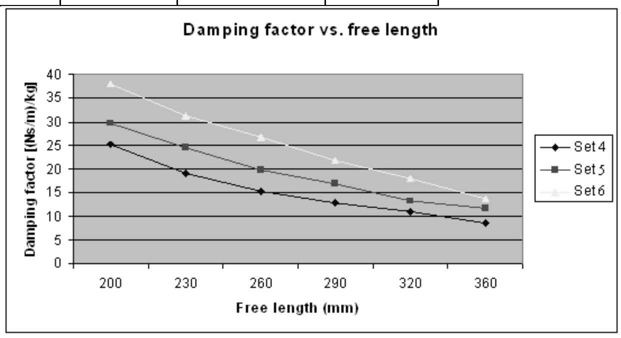

Fig. 11. - The damping factor vs. the free length of the samples for the 4,5,6 sets/. *Variația factorului de amortizare în funcție de lungimea liberă a platbandelor pentru seturile 4,5,6*

Table 3

Damping factor per unit length, stiffness, loss factor and dynamic elasticity modulus

Factorul de amortizare pe unitatea de lungime, rigiditatea, factorul de pierdere a energiei şi modulul de elasticitate dinamic

Sample set	<ρA> [kg/m]	Stiffness [Nm^2]	C [(Ns/m)/m]	Lossfactor	E_{din} [MPa]	Sample set	<ρA> [kg/m]	Stiffness [Nm^2]	C [(Ns/m)/m]	Lossfactor	E_{din} [MPa]
1	0.195	14.887	9.699	0.065	3969	4	0.26	19.875	13.128	0.066	3974
1	0.195	15.201	7.591	0066	4052	4	0.26	20.189	9.958	0.065	4037
1	0.195	14.266	6.421	0.074	3803	4	0.26	20.195	7.977	0.067	4038
1	0.195	14.733	5.305	0.075	3928	4	0.26	21.359	6.638	0.067	4271
1	0.195	14.491	4.117	0.071	3863	4	0.26	20.608	5.716	0.072	4121
1	0.195	14.66	3.595	0.074	3908	4	0.26	19.907	4.46	0.068	3980
2	0.22	23.36	15.151	0.076	1845	5	0.285	36.883	16.847	0.059	2185
2	0.22	25.93	12.71	0.08	2048	5	0.285	35.222	13.985	0.078	2087
2	0.22	23.397	10.38	0.088	1848	5	0.285	34.907	11.251	0.069	2068
2	0.22	25.133	8.479	0.086	1985	5	0.285	37.735	9.663	0.07	2236
2	0.22	26.526	7.216	0.087	2095	5	0.285	38.468	7.557	0.066	2279
2	0.22	24.94	5.233	0.078	1970	5	0.285	34.728	6.7	0.074	2057

Table 3 continues on next page

Sample set	<ρA> [kg/m]	Stiffness [Nm^2]	C [(Ns/m)/m]	Lossfactor	E_{din} [MPa]	Sample set	<ρA> [kg/m]	Stiffness [Nm^2]	C [(Ns/m)/m]	Lossfactor	E_{din} [MPa]
3	0.24	39.568	17.596	0.065	1319	6	0.31	52.571	23.54	0.066	1314
3	0.24	43.862	14.532	0.067	1462	6	0.31	50.373	19.428	0.074	1259
3	0.24	41.149	12.448	0.076	1371	6	0.31	54.592	16.588	0.078	1364
3	0.24	41.904	10.66	0.08	1396	6	0.31	52.384	13.458	0.08	1309
3	0.24	43.169	9.094	0.082	1439	6	0.31	55.758	11.803	0.078	1394
3	0.24	41.698	6.964	0.077	1390	6	0.31	52.932	8.451	0.073	1323

Using the same methodology from [22] we have determined a correlation between the damping factor and the platbands free length. The formula is given in (15) and the factors in Table 4.

$$\mu = \theta \cdot e^{\psi \cdot l} \qquad (15)$$

4. Stiffness and eigen frequency validation

The next step is to validate some of the experimental or calculated results. We will use the same experimental montage for stiffness calculus presented by us in [14], given in Figure 12. For stiffness results we will use the formula (16) (presented before in [14]).

Table 4

Calculus formulae for the damping factor per unit mass
Formule de calcul pentru factorul de amortizare pe unitatea de masă

Samples set	Θ, ψ	Correlation factor R^2
1	Θ= 29.7883032306323 Ψ= -0.199684001697239	0,995
2	Θ= 43.4128823152631 Ψ= -0.206165078864023	0,99
3	Θ= 43.9754635491223 Ψ= -0.177027491407201	0,9908
4	Θ= 29.7280563253825 Ψ= -0.207052746741988	0,9919
5	Θ= 35.4776536309762 Ψ= -0.18881996864639	0,9949
6	Θ= 47.3926272107 Ψ= -0.200440303358321	0,9935

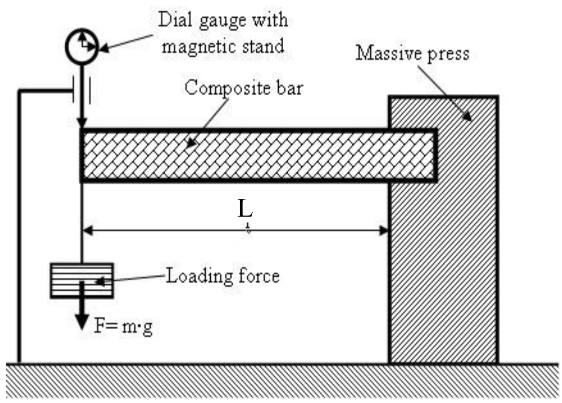

Fig. 12. Experimental montage for stiffness calculus/ *Montaj experimental pentru calculul rigidităţii.*

Table 5

Date experimentale/ *Experimental data*

Samples set	v [mm]	F [N]	EI [Nm^2]	v_{the} [s^{-1}]	l [mm]	Samples set	v [mm]	F [N]	EI [Nm^2]	v_{the} [s^{-1}]	L [mm]	l [mm]
1	2	2	14.29	119.746	200	3	0.7	2	40.83	182.451	350	200
1	2	2	14.29	90.545	230	3	0.7	2	40.83	137.959	350	230
1	2	2	14.29	70.856	260	3	0.7	2	40.83	107.959	350	260
1	2	2	14.29	56.954	290	3	0.7	2	40.83	86.778	350	290
1	2	2	14.29	46.776	320	3	0.7	2	40.83	71.270	350	320
1	2	2	14.29	39.101	350	3	0.7	2	40.83	59.576	350	350
2	1.2	2	23.82	145.553	200	4	1.5	2	19.06	119.767	350	200
2	1.2	2	23.82	110.059	230	4	1.5	2	19.06	90.561	350	230
2	1.2	2	23.82	86.126	260	4	1.5	2	19.06	70.868	350	260
2	1.2	2	23.82	69.229	290	4	1.5	2	19.06	56.964	350	290
2	1.2	2	23.82	56.857	320	4	1.5	2	19.06	46.784	350	320
2	1.2	2	23.82	47.528	350	4	1.5	2	19.06	39.108	350	350

Table 5 continues on next page

Date experimentale/ *Experimental data*

Samples set	v [mm]	F [N]	EI [Nm^2]	v_{the} [s^{-1}]	l [mm]	Samples set	v [mm]	F [N]	EI [Nm^2]	v_{the} [s^{-1}]	L [mm]	l [mm]
5	0.8	2	35.73	156.623	200	6	0.5	2	57.17	189.962	350	200
5	0.8	2	35.73	118.43	230	6	0.5	2	57.17	143.638	350	230
5	0.8	2	35.73	92.677	260	6	0.5	2	57.17	112.403	350	260
5	0.8	2	35.73	74.494	290	6	0.5	2	57.17	90.350	350	290
5	0.8	2	35.73	61.181	320	6	0.5	2	57.17	74.204	350	320
5	0.8	2	35.73	51.142	350	6	0.5	2	57.17	62.028	350	350

Table 6

Stiffness and eigen frequency validation/ *Validarea rigidităţii şi a frecvenţei*

Sample set	Stiffness error [%]	Frequency error [%]	Sample set	Stiffness error [%]	Frequency error [%]
1	4.178	2.066	4	4.276	2.115
1	6.365	3.138	4	5.923	2.918
1	0.168	0.085	4	5.955	2.935
1	3.1	1.54	4	12.062	5.86
1	1.407	0.701	4	8.122	4.196
1	2.589	1.286	4	4.444	2.196
2	1.931	0.971	5	3.227	1.601
2	8.858	4.335	5	1.422	2.402
2	1.776	0.891	5	2.303	1.159
2	5.512	2.719	5	5.612	2.767
2	11.36	5.528	5	7.663	3.761
2	4.702	2.323	5	2.804	1.412
3	3.091	1.557	6	8.044	4.107
3	7.426	3.647	6	11.889	6.132
3	0.781	0.39	6	4.509	2.28
3	2.63	1.307	6	8.372	4.277
3	5.729	2.824	6	2.47	1.243
3	2.126	1.057	6	7.413	3.777

In (16) we have marked with: *EI* – the platband stiffness; *F* – the force that loads the platband at its free end; *L* – the free length of the platband (for this case will be equal to 350 mm); *v* – the platband displacement measured with the comparative device. The stiffness values have been inserted in (6) and the theoretical eigen frequency has been determined. The errors obtained from the stiffness and eigen frequency have been written in Table 6.

$$EI = 0,(3) \cdot F \cdot \left(L \cdot \sqrt[3]{v}\right)^3 \qquad (16)$$

This is an approximate method because of the errors that may appear at the dial gauge displacement reading.

5. Conclusions

In this paper we have built some new original composite platbands (with classical parts but combined in an original way) made of: polypropylene honeycomb core (with the thickness of 10, 15 and 20 mm) reinforced with two layers of glass fiber. The bonding between the glass fiber and the polypropylene honeycomb is made by using the epoxy resin.

The damping factors analysis show that these factors must be experimentally determined for each type of material and sample, being difficult to deduce a quantitative correspondence with the parameters which influence the damping directly or indirectly. 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. The sample mass or specific linear mass influence the damping factor by the fact that the samples with higher mass and width, the deformation energy which is stored in sample through the initial deformation, is dissipated in a larger quantity of material. An influence may occur due to the sample rigidity, explained by the fact that a force initially applied on the sample produces a smaller deformation if the stiffness is higher. Good vibrations damping results in the case where the composite materials of the reinforcing layers have higher damping capacity and elastic properties. An influence to the damping factor may occur due to the sample stiffness, explained by the fact that a force initially applied on the sample produces a smaller deformation if the stiffness is higher.

In addiction to these general conclusions we can add the next particular ones:

- the damping factor per unit mass and per unit length is increased if the platbands have a thickness of 20 mm in comparison with the platbands that have a thickness of 15 and 10 mm; so the damping factor increases at higher values of thickness;
- the damping factor per unit mass of per unit length decrease exponentially with the free length of the platband; for the damping factor per unit mass this thing was shown in Figures 10 and 11, but also from the coefficients written in Table 4;
- the stiffness increase with the thickness and the width of the material (is higher for the platbands with the 20 mm thickness and 60 mm width in comparison to other samples);
- the dynamic elasticity modulus decrease with the increase of the platbands thickness;
- the dynamic elasticity modulus is not very high influenced by the samples width; for example, the samples with 10 mm thickness have a similar dynamic elasticity value (around 4000 MPa) no matter how is the value of their width;
- good results and small errors for the dynamic elasticity modulus have been obtained for all the samples sets for the free lengths of 200, 230, 260, 290, 320 and 360 mm (errors bellow 15%).

From the Tables 5 and 6 we can extract the following conclusions:

- the errors between the stiffness values and the eigen frequencies values are bellow 15% (12,062% for the set 4), accepted in practical engineering; the of 12,062% error can be explained by the fact that this method is an approximate one and it may appear because of the dial gauge reading value for displacement.

We consider that, the added values of this study can be:

- building some new composite platbands made by classical materials (such as glass-fiber, polypropylene honeycomb) combined in an original way;
- the experimental setup: the platbands are free at one end and clamped at the other where it is measured the vibratory response applied with a known initial force;
- the values of the damping factor (per unit mass and per unit length) for the built composite platbands;
- the eigen frequency determination of the new composite studied platbands and its validation by an experimental method;
- the stiffness determination and validation by an approximate experimental method;
- the dynamic elasticity modulus calculus;
- the loss factor calculus.

If we compare the set 3 sample with the free length of 260 mm with the sample 2 studied in [14] (sandwich platband with polypropylene honeycomb core reinforced with one layer of glass fiber in the upper and lower sides marked as sample no. 2 in Table 2), we can see that if the platband has two layers instead of one of glass fiber (with the same thickness of the core and the same length) the damping factor per unit mass will be 1,09 times higher, the damping factor per unit length will be 1,02 times higher and the stiffness will be 1,33 times higher. The increase appears because of the extra glass fiber layer.

These types of composite platbands can be used in practical engineering for: ship floor building, plane floor building, bus floor building, the construction of concrete forming, planes or vehicles bodies building, low weight frames that must have high vibration damping capacity and so on, to strengthen the thin reinforced concrete slabs (an example of strengthening reinforced concrete slabs with composite strips is presented in [23]).

From the results obtained in this research, and taking into account also the results from [14], we recommend for the construction of concrete forming platbands made from polypropylene honeycomb core reinforced with fiber-glass instead of polystyrene reinforced with fiber-glass because they have higher stiffness and smaller deformation. Taking into account both the experimental values obtained in this paper and in [14], to strengthen the thin reinforced concrete slabs we recommend platbands made with polypropylene honeycomb core (with two layers of fiber glass in the upper and lower sides) instead of the ones made by polystyrene core, no matter what their reinforcement is. From this type of platband, sandwich floors can be built to be placed under machines (example turning, milling, boring machines and so on), if vibrations appear during the manufacturing process. Because we have obtained higher values for the damping factor per unit mass and per unit length, we suggest the usage of polypropylene honeycomb core with the 20 mm thickness, instead of the ones with 10 and 15 mm thickness. This type of sandwich structure can be used to replace walls from gypsum board because it does not necessitate other repairs if, for example, a hole is made into the wall to attach house decorations.

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