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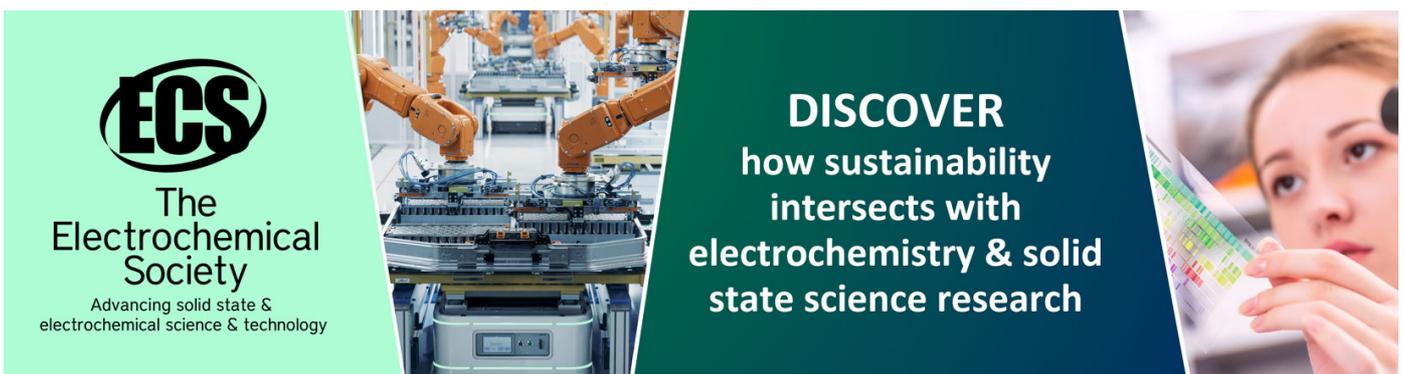
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Mechanical properties determination for a hybrid sandwich bar reinforced with steel wire mesh

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Abstract. In this paper, some new original hybrid sandwich beams were built in this way: the core is made with polypropylene honeycomb and the reinforcement is made with steel wire mesh. The connection between the reinforcement and the core was made with epoxy resin. The aim of this study is to create a beam from classic materials but combined in a way to obtain a new and original structure. The static experimental conditions are characterized by bending loading of the beams and the determination of statically stiffness. The dynamic parameters were determined from the beams free vibrations. The next experimental montage was used: the beams were clamped at one end and were left free at the other end. At the free end, a Bruel&Kjaer accelerometer with 0.04 pC/ms^{-2} sensitivity was placed. A force was applied at the free end to bend the beams and after bending, the force was cancelled and the beams were left to freely vibrate. From the free vibrations recording, the next mechanical parameters were determined: the eigenfrequency of the first eigenmode, the damping factors per unit mass and per unit length, the loss factor, the dynamic stiffness and the dynamic Young modulus.

1. Introduction

A so-called sandwich composite beam is a multy-layered beam (usually it has three layers) and it is fabricated by attaching two thin and stiff layers to a lightweight thick core. Usually, the core is low on strength but has high thickness that provides high bending stiffness with overall low density. These kind of beams are widely used in the construction of aerospace, civil, marine, automotive and other high performance structures due to their high specific stiffness and strength, excellent fatigue resistance, long durability and many other superior properties compared to the conventional metallic materials [1]. The breaking strength of composite beams is dependent on the outer skins materials type and on the interface between the core and the skins.

Recent applications have shown that the honeycomb – panels type reinforced with fibers can be used for new constructions or for reconditioning the existent ones. So, in [2], the vibrations of the structures with honeycomb are studied. The core has a sinusoidal geometry. There was made a higher – order vibration model to study the vibrations with energy methods.

In [3] there were studied the free curved sandwich bars vibrations, with flexible core, in different temperature conditions. The core and the exterior faces were considered being made with temperature – dependant materials. It was shown that the free bars vibrations frequency decreases with the temperature increase.

Studies regarding the movement and constraint conditions equations for asymmetric composite plates are given in [4]. There are studied the effects of direct and inverse piezoelectric influences on the free vibrations frequencies.



For this study, some sandwich strips with polypropylene honeycomb core reinforced with two layers of steel (the layers distribution can be observed in figure 6) wire mesh have been build, with the next mechanical properties:

- honeycomb core: density without facing: 40 kg/m³; compressive strength: ½ MPa; compressive modulus: 0,01 MPa; behaviour with fire: standard quality inflammable; possibility of M1/F0 classification for finished sandwich panels, depending on sandwich skin [5];
- stainless steel wire mesh: same mechanical properties as classical steel [6].

2. Samples used for study

Some samples have been built for this study with the geometry given in table 1. A general view with the built sample is given in figure 1.

Table 1. Samples geometry.

Sample	1	2	3	4	5	6
Length [mm]	420	420	420	420	420	420
Core thickness [mm]	10	10	15	15	20	20
Width [mm]	40	50	40	50	40	50
Specific mass [kg/m]	0.19	0.24	0.21	0.26	0.22	0.28

2.1. Dynamic study

The samples free vibrations were studied in this paper. The samples were clamped at one side, and a Bruel&Kjaer accelerometer was placed at the free end, at a distance of 100 mm from the free edge. A force was applied at the free end and the bars were left to freely vibrate. Two free length values were considered: 300 and 350 mm. The scheme with the vibration test is given in figure 2. All the experimental results and the dynamic stiffness (marked as EI), loss factor (marked as η) and the dynamic Young (marked as E_{dyn}) are given in table 2.

Table 2. Dynamic mechanical characteristics.

Sample	1		2		3		4		5		6	
	300	350	300	350	300	350	300	350	300	350	300	350
Free length [mm]												
μ [(Ns/m)/kg]	8.5	6.9	8.57	6.88	13.42	9.06	12.6	8.8	16.5	11.9	12.533	10.91
ν [1/s]	45.9	34.25	46.1	33.85	63.2	46.8	62.5	46.2	79.9	59.12	82.61	60.45
EI [Nm ²]	10.357	10.683	13.196	13.181	21.702	22.046	26.277	26.6	36.33	36.857	49.438	49.048
η	0.059	0.064	0.059	0.065	0.068	0.062	0.064	0.061	0.066	0.064	0.048	0.057
E_{dyn} [MPa]	3106	3204	3166	3163	1929	1959	1868	1891	1362	1382	1483	1471
C [(Ns/m)/m]	3.23	2.622	4.11	3.30	5.636	3.805	6.552	4.576	7.26	5.236	7.018	6.113

The dynamic stiffness is determined from the relation (1). In (1), the next parameters are marked: ρA - the specific mass; ρ - material density; A - the bar transversal section; E - the Young modulus; I - the section moment of inertia; l - the bar length, β - a parameter that depends on the bar restraints.

$$\nu = \left(\frac{\beta}{l}\right)^2 \cdot \left(\frac{1}{2 \cdot \pi}\right) \cdot \sqrt{\frac{E \cdot I}{\rho \cdot A}} \quad (1)$$

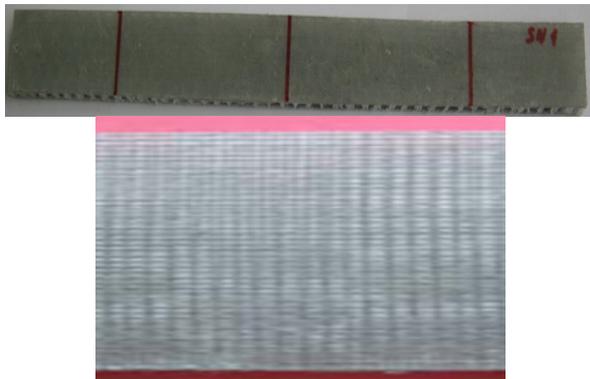


Figure 1. The built samples.

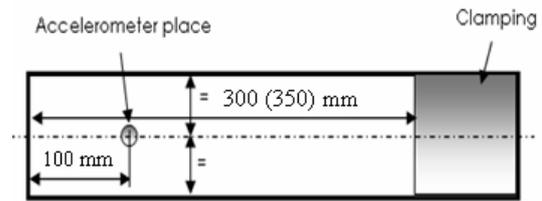


Figure 2. The vibrations scheme.

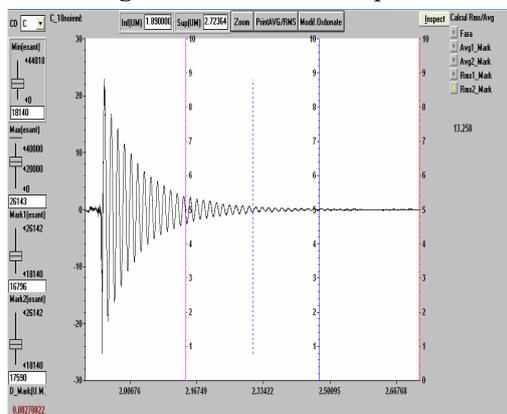


Figure 3. The vibration recording (sample 6, 350 mm free length).

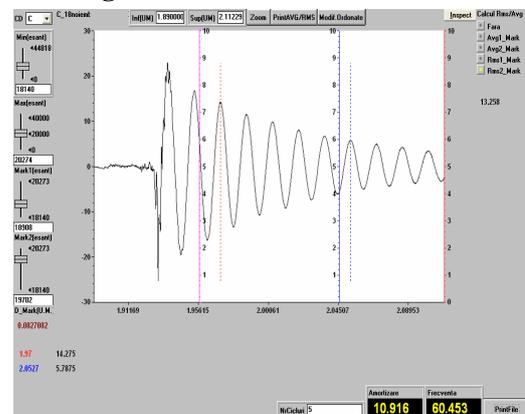


Figure 4. The eigen frequency and damping factor per unit mass determination (sample 6, 350 mm free length).

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 (2) and (3) [7];
- the damping factor per unit mass is determined with (4) [7].

$$\nu = \frac{1}{T} \tag{2}$$

$$\omega = \frac{2 \cdot \pi}{T} \tag{3}$$

$$\mu = \frac{\ln\left(\frac{\Delta_j}{\Delta_{j+1}}\right)}{T} \tag{4}$$

- the damping factor per unit length is determined with (5) [7]:

$$C = 2 \cdot \mu \cdot \rho \cdot A \tag{5}$$

In equation (4) we have marked with Δ_j, Δ_{j+1} the maximums separated by periods. The loss factor can be determined with (6) according to [7].

$$\eta = \frac{\mu}{\pi \cdot v_{\xi}} \quad (6)$$

In the relationship (6), the next parameters are marked: μ - damping factor per unit mass; η - loss factor; v_{ξ} - the eigen frequency; ξ - the number of the eigenmode.

From (1) the dynamic stiffness EI and Young modulus E values can be established with (7) and (8).

$$EI = \left(\frac{l}{\beta}\right)^4 \left(v \cdot 2 \cdot \pi \cdot \sqrt{\rho \cdot A}\right)^2 \quad (7)$$

$$E = \left(\frac{l}{\beta}\right)^4 \left(v \cdot 2 \cdot \pi \cdot \sqrt{\frac{\rho \cdot A}{I}}\right)^2 \quad (8)$$

2.2. Static tests

The samples were bending loaded, in three points. The bending loading device and scheme are presented in figure 5. The breakage of a sample, from the set 3, is presented in figure 6.

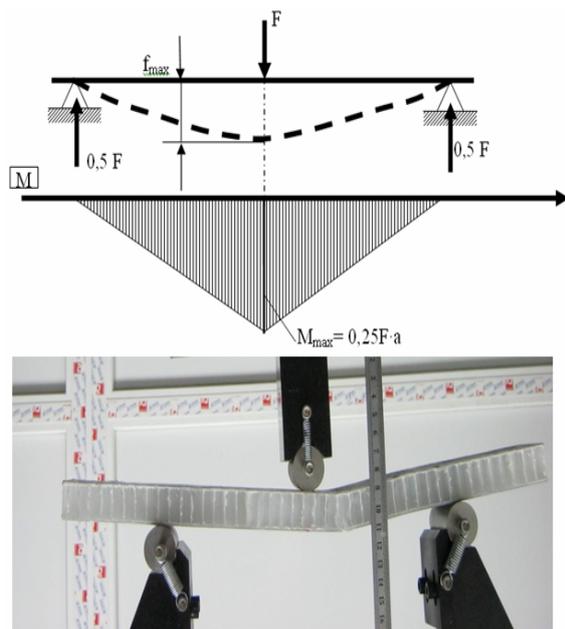


Figure 5. The Bending device and scheme.

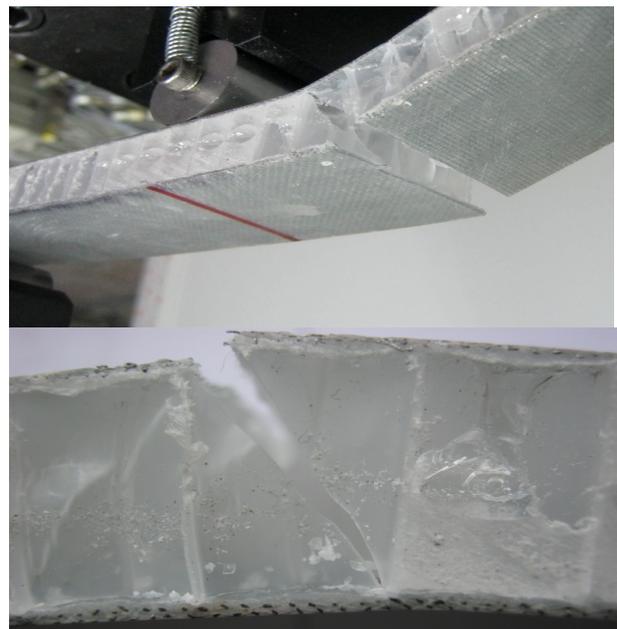


Figure 6. The fracture of sample 3.

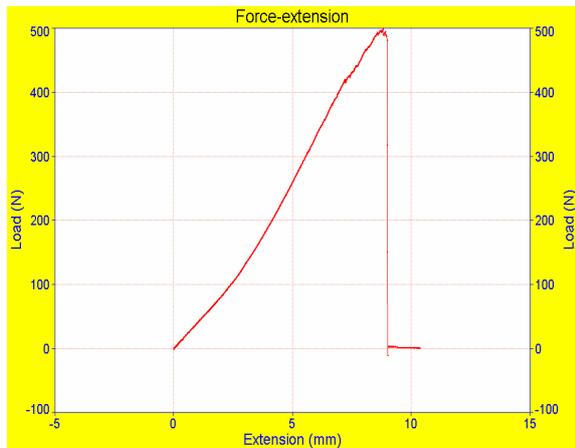


Figure 7. The force-extension curve for sample 6.

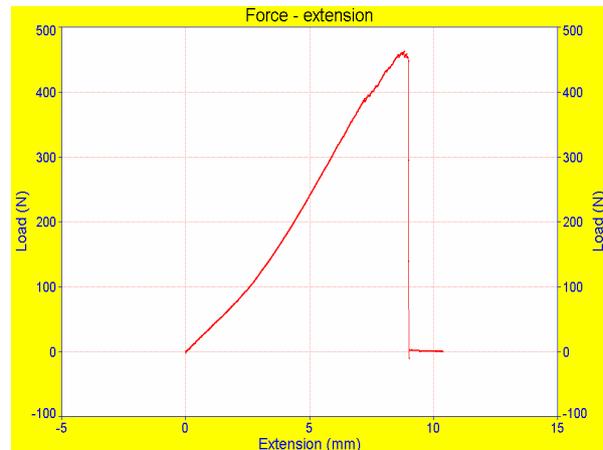


Figure 8. The force-extension curve for sample 5.

The test was made on a Walter Bai testing machine, with the maximum force of 30 kN according to ASTM D790-02 [8].

From the static tests, the stiffness was determined with the relationship (9).

$$f = \frac{F \cdot a^3}{48 \cdot \langle EI \rangle} \Rightarrow \langle EI \rangle = \frac{F \cdot a^3}{48 \cdot f} \tag{9}$$

In (9) the next parameters were written: f - linear displacement; F - the force; a - the distance between the indenters (240 mm); EI - the beams stiffness. In the figures 7 and 8 are drawn the characteristic curves (force-extension) for the samples 5 and 6. The static experimental results are written in table 3.

Table 3. The static mechanical characteristics.

Sample	1	2	3	4	5	6
EI [$N \cdot m^2$]	10.81	13.22	22.42	27.25	37.21	49.78

2.3. Experimental validation of the static stiffness results

Because of the samples complex geometry, the most rapid and effective validation for the experimental results is to use another experimental setup.

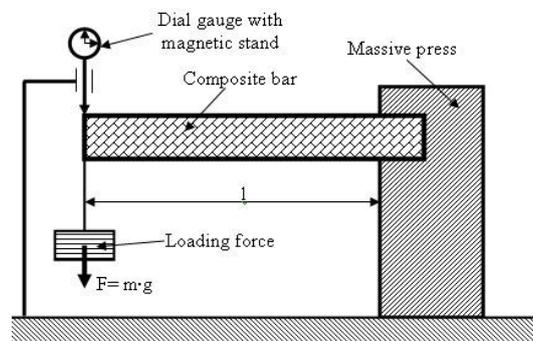


Figure 9. Experimental setup for stiffness validation.

In order to determine the stiffness, the relationship (10) can be used.

$$EI = 0,3 \cdot (\sqrt[3]{Fl^3v})^3 \quad (10)$$

In (10) the next parameters appear: EI - the strip 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 260 mm); v - the platband displacement measured with the comparative device. All the data are written in table 4.

Table 4. Stiffness validation.

Sample no.	v [mm]	F [N]	EI [N·m ²]	ε_m , %
1	0.2	0.2	10.67	1.326
2	0.15	0.2	14.22	7.047
3	0.1	0.2	21.33	4.84
4	0.08	0.2	26.67	15.92
5	0.06	0.2	35.56	4.44
6	0.05	0.2	42.67	14.29

This is an approximate method because of the errors that may appear at the dial gauge displacement reading. With ε_m it was marked in table 3 the errors for the stiffness values that appear between the two experimental methods (all the errors appear because the validation method is an approximate one, but are quite small - under 16%).

2.4. Results and discussions

The values analysis of damping factors indicates 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 factors 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 is acting on the sample. The air force that acts upon the bar is proportional with the bar area, so if the width increases the force that acts upon the bar increases too. The sample mass or specific linear mass has an influence on the damping coefficient by that, for the samples with higher mass and width, the deformation energy which is stored in the sample through the initial deformation, is dissipated in a larger quantity of material. An influence may occur due to the sample stiffness, explained by the fact that a force initially applied on the sample produces a less deformation if the stiffness is higher.

A good damping of vibrations is achieved in the case in which the composite materials of the external layers have the damping capacity and elastic properties which are superior. But the influence of these layers is dependent on the interaction with the middle layer and, for this reason, it is difficult to be analytically analyzed.

In addition to these general conclusions, by studying the tables 1 and 2, we can add new particular ones:

- the damping factors per unit mass and per unit length increase with the thickness increase;
- from the dynamic and static tests of the stiffness, we can see small differences for the obtained experimental values;
- the eigen frequency increases with the free length decrease; the dynamic Young modulus is not influenced by the samples free length.

3. Conclusions

The added value of this study is:

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

In addition to the dynamic parameters determination, the static stiffness was also determined by bending loading of the samples.

4. References

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