Damping analysis in flexural vibration of sandwich beams with debonding

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The paper presents, an analysis of the damping of sandwich composites with debonding. These sandwich composites consist of two thin faces composed of glass fibres and epoxy resin bonded to light weight and weaker core material of PVC foams. Natural frequencies and damping parameters are investigated using beam test specimens and an impulse technique. Modelling of the damping of a sandwich composite with debonding is established considering finite element analysis which evaluated the different energies dissipated in the material directions of the core and the skins. The effects of debonding variable lengths on the natural frequencies and the damping were studied numerically and compared with experimental results. The results show that the natural frequencies and damping are sensible to debonding length. These properties offer the sensitive indicators of damage of sandwich materials during lifetime.

1 Introduction

A sandwich structure consists of two thin, stiff and strong faces bonded to thick, lightweight and weaker core material using some adhesive. Sandwich composites show excellent mechanical properties due to their high specific stiffness and strength to weight ratios. The viscoelastic core has a high inherent damping capacity. In the flexural vibration of sandwich, the damped core is constrained to shear. The effects transverse shear causes the flexural motion to be damped and energy to be dissipated. In sandwich materials a high part of the energy is dissipated by the transverse shear effects induced in the sandwich core. Assarar et *al* [1] used the different energies dissipated in the material directions of the core and the layers of the skins for damping modeling of sandwich beam. For the fibre-reinforced composite, damping depends on fibre, layer orientation, stacking sequence [2]. Many researchers used the concept of a complex modulus. In this concept, the imaginary part is associated with the energy dissipation. Damage is mechanism which causes increased damping. Zenkert et *al* [3] used the finite element method and experiment for analyse the impact damage of sandwich. Recently Burlayenko et *al* [4] investigated the influence of size, location and number of debonded zones on dynamic characteristics of sandwich plates. The debonding results adhesive degradation, defects of manufacturing, impact by foreign objects, or stress concentration due to localized loading [5-8]. Debonding introduces friction in the unbounded region of the interface, the damping increases with the size of the debonding. Buket et *al* [9] studied the vibration behaviour of flat and curved sandwich (with 15°, 30°, 45° and 60° open angle) composite beams with face/core debond. They noted that with presence of debond, the damping variation for sandwich beam with curvature angle of 30° and 60° shows more sensitive results than that of sandwich beam with curvature angle of 15° and 45°. Suzuki et *al* [10] presented a finite element analysis of the damping of laminate composites with delamination. The modal strain energy method is also implemented into this finite element analysis. The results obtained show that this analysis describes fairly well the experimental results.

The purpose of this article is to develop modelling of damping of sandwich materials with debonding based on a finite element analysis. The effects of variable length debonding on natural frequencies and damping parameters were studied. The results derived from this analysis are compared to the experimental results.

2 Finite elements modeling

2.1 Strain energy

Finite elements based on the theory of plane stress are used. Finite element analysis gives the values of the stresses σ_{xx}, σ_{yy}, σ_{xy} and the strains ε_{xx}, ε_{yy}, ε_{xy} in each skins lower (*l*), upper (*u*) and core (*c*) for each finite element (*e*). The energy U_d^e stored in a given finite element *e* can be expressed as a function the strain energies related to the different directions as:

$$U_d^e = U_{xx}^e + U_{yy}^e + U_{xy}^e \tag{1}$$

With

$$\begin{aligned} U_{xx}^e &= \frac{1}{2}\iint_e \sigma_{xx}\varepsilon_{xx}dxdy, \\ U_{yy}^e &= \frac{1}{2}\iint_e \sigma_{yy}\varepsilon_{yy}dxdy, \\ U_{xy}^e &= \frac{1}{2}\iint_e \sigma_{xy}\varepsilon_{xy}dxdy, \end{aligned} \tag{2}$$

Where the integrations are extended over the area of the finite element *e*. The energy U_{xx}^e is the strain energy stored in tension–compression along the *x*-direction of the materials, U_{yy}^e is the strain energy stored in tension–compression along the *y*-direction of the materials and U_{xy}^e is the strain energy stored in-plane (plane (*x, y*)) shear. The in-plane strain energy stored in element *e* is given by Expressions (3) and the total in-plane strain energies stored in the finite element assemblage is then obtained by summation on the elements as:

$$\begin{aligned} U_{xx} &= \sum_{elements} U_{xx}^e , \\ U_{yy} &= \sum_{elements} U_{yy}^e , \\ U_{xy} &= \sum_{elements} U_{xy}^e . \end{aligned} \tag{3}$$

The strain energy U_d stored in the sandwich composite structure can be expressed as a function of the in-plane strain energies related to the material directions as:

$$U_d = U_{xx} + U_{yy} + U_{xy} \qquad 4)$$

2.2 Estimation of sandwich damping

The energy dissipated by damping in the skins or in the core of the sandwich material of the element *e* is derived from the strain energy stored in the skins or in the core as [2]:

$$\Delta U_p^e = \psi_{xxp}^e U_{xxp}^e + \psi_{yyp}^e U_{yyp}^e + \psi_{xyp}^e U_{xyp}^e \tag{5}$$

Introducing the specific damping coefficients of the skins or of the core. These coefficients are related to the material directions of the skins ($p=l$ or u) or of the core ($p=c$): ψ^{e}_{xxp} and ψ^{e}_{yyp} are the damping coefficients in tension-compression in the x-direction and y-direction of the skins or of the core, respectively; ψ^{e}_{xyp} is the in-plane coupling coefficient.

The damping energy dissipated in the element e is next obtained by summation on the core and on the skins of element e as:

$$\Delta U^{e} = \Delta U^{e}_{c} + \Delta U^{e}_{s} \tag{6}$$

Where s is skin and c core, the total energy ΔU dissipated in the finite element assemblage is then obtained by summation on the elements:

$$\Delta U = \sum_{elements} \Delta U^{e} \tag{7}$$

Finally, the damping of the finite element assemblage is characterized by the damping coefficient of the assemblage derived from relation [11]:

$$\psi = \frac{\Delta U}{U_d} \tag{8}$$

3 Materials

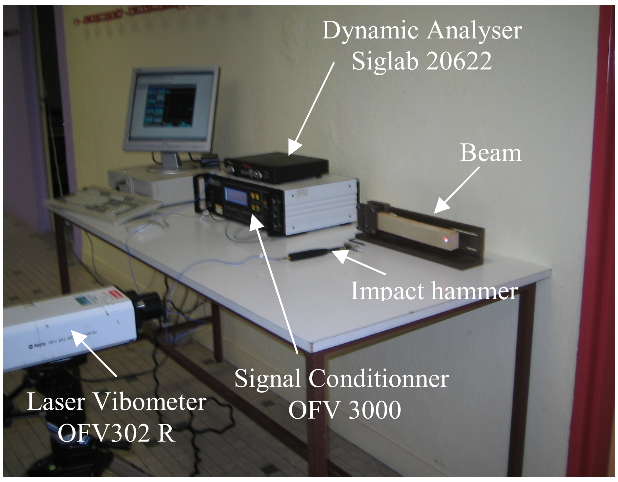
Sandwich materials were constructed with glass fibre laminates as skins and with PVC closed-cell foams as core. The glass fibre laminates of the skins are cross-ply laminates consisting of unidirectional layers of E-glass fibres in an epoxy matrix, arranged in the sequence $[0_2/90_2]_s$. The unidirectional layers were fabricated with unidirectional cloth of weight 300 g m^{-2} with glass fibres aligned in a single direction. The PVC closed-cell foams were supplied in panels of thickness 20 mm. Two foams were considered differing in their densities: 60 kg m^{-3} and 100 kg m^{-3}. Sandwich materials were constructed with these foams and with cross-ply glass-fibre laminates prepared by hand lay-up process that leads to a nominal thickness of 3 mm for the sandwich skins. Sandwich plates of different dimensions were cured at room temperature with pressure using vacuum moulding process, and then post-cured in an oven. The specimens have been cut up with a diamond disc, from sandwich plates. Dimensions of the specimens are shown in table 1.

Table 1: Dimensions of sandwich specimen

Core thickness e_a (mm)	Skin thickness e_p (mm)	Length of specimen L (mm)	Width of specimen b (mm)
20	3	250	40

4 Experimental procedure

The damping characteristics of the sandwich beam without or with debonding obtained by subjecting beams to flexural vibrations. The equipment used is shown in Figure 1.a. The test specimen is supported horizontally as a cantilever beam in a clamping block. An impulse hammer is used to induce the excitation of the flexural vibrations of the beam near the clamping block (Figure 1.b) and the beam response is detected near the free end of the beam by using a laser vibrometer. Next, the excitation and the response signals are digitalized and processed by a dynamic analyzer of signals. This analyzer associated with a PC computer performs the acquisition of signals, controls the acquisition conditions and next performs the analysis of the signals acquired (Fourier transform, frequency response, mode shapes, etc.). The flexural beam responses were identified in the frequency domain using analysis and fitting the experimental frequency responses using Matlab toolbox. Then, the identification procedure allows us to obtain the values of the natural frequencies f_i and the modal loss factors η_i, related to the specific damping coefficient by the relation $\psi_i = 2\pi\eta_i$.



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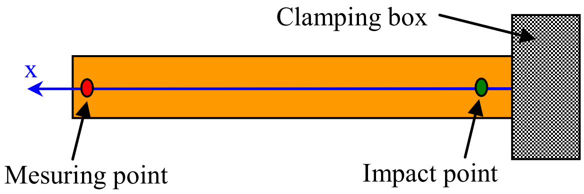


(b)

Figure 1: Analysis of beam vibrations; a) Experimental equipment, b) Impact and measuring points

5 Finite elements procedure

The finite element method is used to compute modal natural frequencies, stress and strain for sandwich beam without or with debonding. The debonding is between the skin and the core of sandwich beam, it is modelled as a void between them. The debonded surfaces are not in contact. The propagation of debonding was not allowed during vibration. The finite element simulations were conducted using ABAQUS 6.10. The reduced integrated continuum plane stress element (CPS4R) is used for mesh, CPS4R is a four node element. This element based on the theory of plane stress. The material for the skin of a sandwich is considered anisotropic and the cores were considered as homogeneous isotropic materials. The material properties used for numerical simulation are shown in Table 2 and 3. The sandwich beam with debonding in modelling finite element is shown in the figure 2.

Table 2: Mechanical characteristics of the PVC Foam

Density of the foam ($kg\ m^{-3}$)	Young modulus (MPa)	Poisson's ration	Shear modulus (MPa)
60	60	0.42	22
100	85	0.43	38

Table 3: Engineering constants of unidirectional glass fibre layers

E_L (GPa)	E_T (GPa)	ν_{LT}	G_{LT} (GPa)
29.9	7.5	0.24	2.25

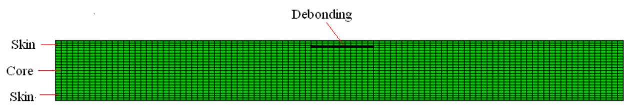


Figure 2: Mesh of sandwich beam with debonding

6 Results and discussion

6.1 Energy stored in the constituent of sandwich

The beam had one central debonding between the core and the face sheet of varying length (l) located at the mid-plane, symmetrically about the mid-span of the beam. For example, the figure 3 shows the different strain energies in the skins of the sandwich beams for the first mode of the bending vibrations. These results were obtained using a finite elements analysis for different normalized debonding length and for sandwich with foam of 60 kg m^{-3}. The results obtained show that the energies stored decrease when the debonding length increases. The results so show that the longitudinal strain energy is higher than the other energies. From this results it is possible to consider only the longitudinal energy stored (U_{xx}) in the skin.

The figure 4 shows the evolution of the energies stored in the foam core as functions of the debonding length for the first mode and for sandwich with foam of 60 kg m^{-3}. The shear energy stored (U_{xy}) decrease and the longitudinal energy stored (U_{xx}) increase when the debonding length increases. The energy U_{yy} is much lower and can be neglected.

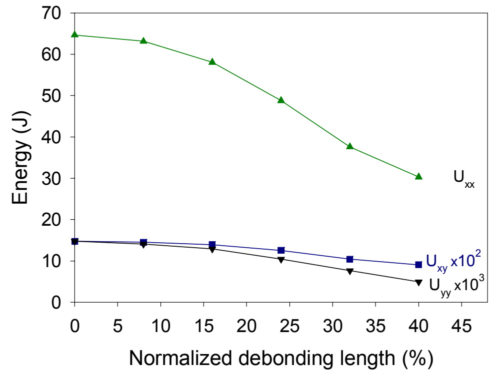
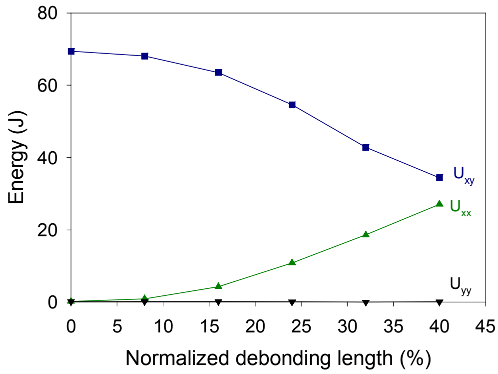


Figure 3: Energies stored as function debonding length in the skin of sandwich beam with foam of 60 kg m^{-3} of density

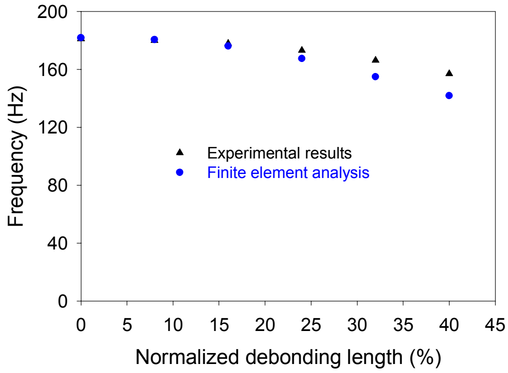


(b)

Figure 4: Energies stored as function debonding length in the core of sandwich beam with foam of 60 kg m^{-3} of density

6.2 Effect of the debonding length on the natural frequency

The figure 5 shows the effects of the variable debonding length, in debonded sandwich beams on natural frequency of the sandwich beam. The finite element results and experimental results are compared. The measured fundamental natural frequency of the undamaged sandwich beam is 180 Hz for core density of 60 kg m^{-3} and is 190 Hz for core density of 100 kg m^{-3}. The finite element natural frequency is about 182 Hz for core density of 60 kg m^{-3} and is 190 Hz for core density of 100 kg m^{-3}. For two types of sandwich the natural frequency of the debonded sandwich beam decrease drastically at 25% of normalised length of debonding for two types of sandwich. An increase in the extent of debonding gives a greater reduction in the natural frequencies of the sandwich beams. For two types of sandwich there is a critical debonding (25%) extent beyond which the reduction of the natural frequencies increases disproportionately. The comparison shows a good agreement between the experimental results and the results deduced from modeling until 25% of normalised debonding length.



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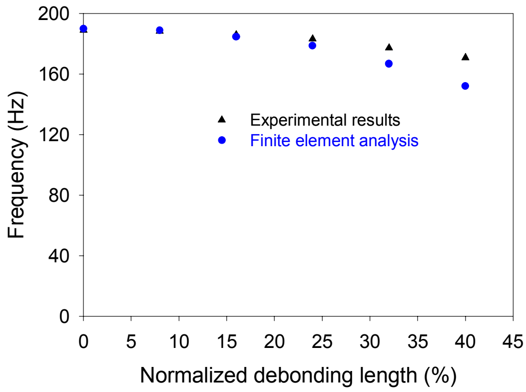

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Figure 5: Comparison between results deduced from experiment and modeling for the natural frequency versus the debonding length for the composite sandwich beams with two different foam densities: a) 60 kg m^{-3} and b) 100 kg m^{-3}

6.3 Loss factor damping of sandwich

Modeling performed taking into account the variation of the damping coefficient (or loss factor) of the various constituents of the sandwich materials as a function of frequency. The loss factor of the various constituents of the sandwich materials (skin and core) is investigated by El Mahi et *al*. [2] and Assarar [13], the loss factor as function the frequency are shown in figure 6 and 7. The figure 8 shows the loss factor damping evolution versus the normalized debonding length. These results are deduced from the experimental investigation and finite element analysis. The damping loss factor values increase with increasing debonding, because debond introduces more friction in the sandwich beam. The variation of damping ratios is considerably higher in case of the specimens with foam density of 60 kg m^{-3} (figure 8.a). For two different sandwich, the comparison shows a good agreement between the experimental results and the results deduced from modelling until 25% normalized debonding length. The analysis finite element tends to give higher loss factor damping compared to by the experimental investigation.

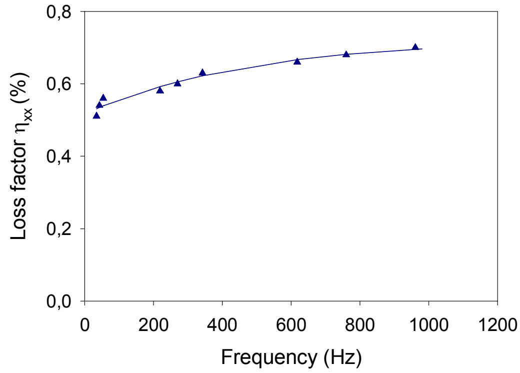


Figure 6: The loss factors as function of the frequency for longitudinal direction of the skin [2]

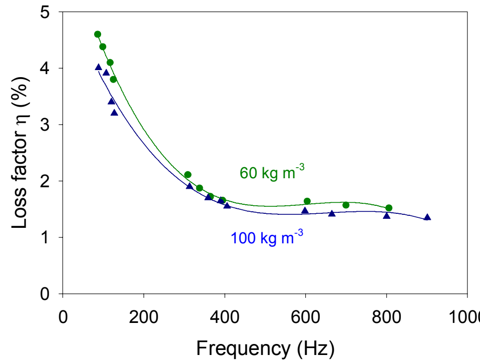
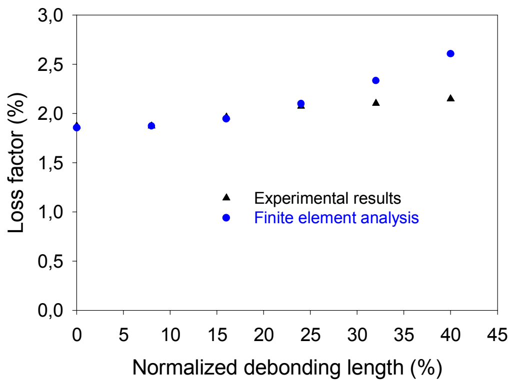
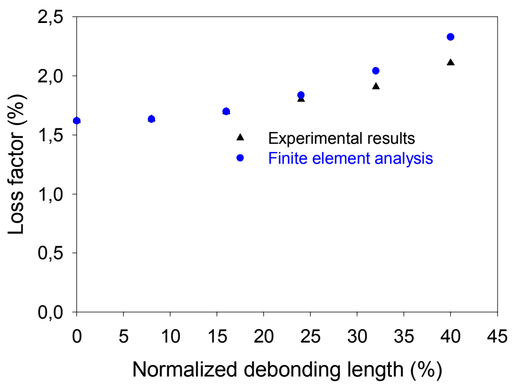


Figure 7: The loss factors as functions of the frequency for the two densities of the foams: 60 and 100 kg m^{-3} [13]



(a)



(b)

Figure 8: The effect of debonding length on the loss factor damping of the sandwich beam with foam core of density: a) 60 kg m^{-3} and b) 100 kg m^{-3}

7 Conclusions

An evaluation of the damping of sandwich materials fabricated with a foam core and laminated skins was presented based on the theory of plane stress and a finite element analysis in vibrations of a composite sandwich. The analysis derives the strain energies stored in the material directions of the foam core and in the material directions of the skins. Damping characteristics of sandwich for different debonding length were evaluated experimentally using beam specimens subjected to an impulse input. Damping modeling is developed using a finite element analysis which evaluated the different energies dissipated in the material directions of the core and the skins. The comparison shows a good agreement between the experimental results and the results deduced from modelling until 25% of level debonding. This agreement underlines that the modeling considered is well suited to describe the damping of sandwich materials with variable length debonding.

References

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