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The influence of viscoelastic film thickness on the dynamic characteristics of thin sandwich structures

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ABSTRACT

Passive damping techniques by means of viscoelastic materials are widely used for structural vibrations control. Sandwich structures composed of viscoelastic adhesive films and metallic constraining layers result in thin composite structures with improved dynamic capabilities. The nature and the small thickness of these sandwich structures enable them to be processed in conventional metal sheet transformation techniques to obtain components of complex geometries. In this work the influence of the viscoelastic film thickness on the dynamic properties, stiffness and damping, of thin sandwich structures is analysed from experimental and numerical results. Sandwich structures composed of same viscoelastic material but three different core thicknesses are tested and the dynamic properties of the viscoelastic film are obtained. From the experimental results a material model with fractional derivatives is proposed for the shear complex modulus of the viscoelastic adhesive film. The results show the viscoelastic film thickness has a greater influence in the loss factor than in the storage modulus, being this effect more pronounced at high frequencies. The bending stiffness of the sandwich structure is increased with core thickness even if the storage modulus is decreased. Therefore, the viscoelastic film thickness determines the vibrational response of thin sandwich structures.

1. Introduction

The reduction of structural vibration amplitudes in mechanical systems has currently become a key objective in many industrial sectors in order to extend the life of components, reduce acoustic radiation or increase comfort and security. There are different techniques aimed at the control of structural vibration, classified into active, semiactive, adaptive and passive techniques [1]. Today, passive surface treatments by means of viscoelastic materials are the most common damping technique, because of their simplicity in implementation and lower cost. Viscoelastic materials can be used in three different configurations: free layer damping treatment (FLD), constrained layer damping treatment (CLD) and tuned viscoelastic damper (TVD). In the first two treatments, the energy dissipated depends on the superficial deformations and so they are effective in vibrations of plates and beam structures, while in the TVD treatment it depends on the local displacements. In the FLD treatment, the viscoelastic material is subjected to low tension/compression strains, whereas adding a constraining elastic layer, CLD treatment, the core undergoes high shear deformations that enables high energy losses [2,3]. Thus, the CLD treatment is more effective than FLD treatment for a given added weight.

Kervin [4] was one of the first to analyse the damping effectiveness of the CLD treatment or sandwich structure and since then, numerous studies have been carried out to understand and model the damping mechanism and characteristics of sandwich structures [5–12]. Besides, there are many studies that analyse the influence of the design parameters on the damping of sandwich structures, as well as, optimal design procedures to attain the maximum damping in the sandwich beams [13–20]. However, the influence of the design parameters on the stiffness of sandwich structures must be known too; since for a given application's design, equilibrium between damping and stiffness must be found to ensure proper operation. In addition, a gap concerning thin sandwich structures has been found.

The use of viscoelastic adhesive films and metallic constraining layers enables the production of thin sandwich structures. These sandwich structures provide higher damping than their metallic base layers for a low added weight and moreover they can be processed in conventional metal sheet transformation techniques to obtain pieces of complex geometries. The main aim of this paper is to analyse the influence of the viscoelastic film thickness on

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the dynamic properties, stiffness and damping, of thin sandwich structures.

Sandwich structures composed of same viscoelastic material and three core thicknesses are characterised by experimental dynamic tests according to the ASTM E 756-05 standard [21] with the modification proposed by Cortés and Elejabarrieta [22]. From these results the shear complex modulus of the viscoelastic film is determined and a material model based on the four parameter fractional derivative model with an empirical modification is proposed fitting it to the experimental data. The influence of the viscoelastic film and base layer thicknesses is analysed from the experimental results and numerical ones obtained from the proposed viscoelastic material model and RKU equations [23]. Lastly, this analysis allows proposing a methodology to design symmetric thin sandwich structures according to their final application's dynamic requirements.

2. Experimental characterisation

The frequency domain technique established by the ASTM E 756-05 standard [21] with the modification proposed by Cortés and Elejabarrieta [22] is applied to characterise the different sandwich structures and the viscoelastic film in a bandwidth from 0 to 1 kHz.

2.1. Specimens

The sandwich structures analysed in this work are provided by Replasa S.A. and they are manufactured in a continuous coating process called coil coating. All of them are symmetric and composed of two AISI 316 steel base layers and different thicknesses of a polyester-based adhesive core. The core thicknesses studied in this work are 40 μm, 80 μm and 100 μm in wet. In Fig. 1, the configuration of the analysed sandwich structures can be seen, where L is the free length, b is the width and H, H_e and H_v are the thickness of the sandwich, base layer and core. Note that $(\bullet)_e$ and $(\bullet)_v$ refer to the elastic and viscoelastic materials, respectively.

Three sandwich specimens from each core thickness are tested, each of them with free lengths of 140 mm, 150 mm and 160 mm. Tables 1 and 2 show the geometrical and physical properties of the analysed sandwich structures and their components; where $H_{v,dry}$ and $H_{v,wet}$, are the thickness of the core in dry and in wet, and ρ, ρ_e and $\rho_{v,dry}$ are the density of the sandwich, base layer and core ones in dry.

The nomenclature used is the following: the letter S refers to sandwich specimens and then the thickness of the core in wet is specified, 40 μm, 80 μm or 100 μm. The thickness of the viscoelastic core in wet is provided since it is the parameter controlled in the fabrication process of sandwich structures; nevertheless in the material properties extraction procedure, the geometrical and physical properties of the core once it is cured is used.

Fig. 1. Sandwich configuration.

Table 1
Geometrical and physical properties of the analysed sandwich structures.

	b (±0.002 mm)	H (±0.002 mm)	ρ (±0.05 g/cm^3)
S100	9.900	0.552	7.51
S80	9.900	0.521	7.70
S40	9.900	0.523	7.86

Table 2
Geometrical and physical properties of sandwich structure's components.

	H_e (±0.002 mm)	$H_{v,dry}$ (±2 μm)	$H_{v,wet}$* (μm)	ρ_e* (g/cm^3)	$\rho_{v,dry}$* (g/cm^3)
S100	0.258	36	100	7.95	1.13
S80	0.251	19	80	7.95	1.13
S40	0.258	7	40	7.95	1.13

* manufacturer's data, Replasa S.A..

2.2. Experimental technique

The experimental characterisation is carried by a forced vibration test with resonance according to the ASTM E 756-05 standard [21] and the modification proposed by Cortés and Elejabarrieta [22].The dynamic properties of the sandwich structures and their cores are obtained in a two-step process. First, the sandwich beam is tested and the homogenised complex modulus, E^*, of the sandwich is obtained from the transmissibility function. Which assuming a linear viscoelastic behaviour is given by

$$E^*(f) = E(f) + iE'(f) = E(f)(1 + i\eta(f)), \quad (1)$$

where $E(f)$, $E'(f)$ and $\eta(f)$ are the storage modulus, the loss modulus and the loss factor of the sandwich structure, respectively [24]. After determining the homogenised complex modulus and the geometrical and physical properties of the sandwich components, the dynamic properties of the viscoelastic film can be identified [21]. Assuming the Poisson ratio is constant in frequency [25,26], the shear complex modulus, $G_v^*(f)$, of the viscoelastic film yields

$$G_v^*(f) = G_v(f) + iG_v'(f) = G_v(f)(1 + i\eta_v(f)), \quad (2)$$

where $G_v(f)$ is the shear modulus, $G_v'(f)$ is the loss modulus and $\eta_v(f)$ is the shear loss factor of the core.

In the extraction of the dynamic properties of the sandwich structure and core, the classical analysis based on the Euler–Bernoulli beam theory is used, so it must be ensured that the beam's cross section is much less than its length [21]. Moreover, the extensional terms for the core are not included since the storage modulus of the viscoelastic core is much lower than the base metal ones and the loss factor of the base metal is assumed to be zero. Furthermore, the results obtained from the first vibration mode are ignored as established by the ASTM E 756-05 standard [21,23].

Fig. 2 shows the experimental set-up used for measuring the transmissibility functions of the cantilever sandwich beams. The specimens are excited from the base by an electrodynamic shaker (Ling Dynamic Systems Vibrator, Model 406) and the base acceleration consists of a white noise in a bandwidth from 0 to 1 kHz generated by a vibration controller (LDS Dactron LASER Shaker Control System). The acceleration of the base, $\ddot{s}(t)$, is measured by a piezoelectric accelerometer (B&K, Type 4371) with a charge conditioning amplifier (B&K, Type 2635) and the velocity of the free end of the specimens, $\dot{u}(t)$, is measured by a laser vibrometer (Polytec OFV 505 LR100). Data acquisition and signal processing are performed with the OROS (OR763) analyser of four channels connected to a PC. The obtained transmissibility functions are the derivative of the velocity output of the beam's free end divided by the acceleration of the base.

In the experimental tests, first the transmissibility functions of each specimen in the bandwidth from 0 to 1 kHz are obtained and the resonant frequencies are identified. Then, the modal transmissibility functions are measured in order to obtain a better resolution. Finally, the analysed bandwidth is extended to 4 kHz in