A NEW 3D FINITE ELEMENT FOR SANDWICH STRUCTURES WITH A VISCOELASTIC CORE

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1. INTRODUCTION

Nowadays, noise and vibrations control is a major concern in several industry fields such as aeronautics and automobile. The reduction of noise and vibrations is a major requirement for performance, sound quality and customer satisfaction. Passive damping technology using viscoelastic materials is classically used to control the vibration. The steel industry proposes damped sandwich panels with thin layer of viscoelastic core (Metal/Polymer/Metal). This type of structures has appeared recently as a viable alternative to classical add-on or spray-on treatments. It has been shown that this class of materials enables manufacturers to cut weight and cost while providing noise, vibration and harshness performance. This motivated the development of prediction methods for their vibration and acoustic indicators. Initially, analytical techniques were developed to predict the performance of damped sandwich panels with classical boundary conditions. The fundamental work in this field was pioneered by Ross, Kerwin and Ungar (RKU) [1] who used a three-layer model to predict damping in plates with constrained layer damping treatments. Kerwin [2] was the first to present a theoretical approach of damped thin structures with constrained viscoelastic layer. He presented the first analysis of the simply supported sandwich beam using a complex modulus to represent the viscoelastic core. Several authors (DiTaranto [3], Mead and Markus [4]) extended Kerwin's work using his same basic assumptions. Six-order equations of motion were developed in term of axial displacements by DiTaranto [3] for the unsymmetrical three-layer beam, and this was subsequently refined [4]. However, these analytical solutions are only appropriate for simple structures such as beams or plates with simple boundary conditions. In practice it is often necessary to design damped structures with complicated geometry, complex loadings and non-uniform features such as material discontinuities. Consequently, it is natural to consider the finite element method (FEM) to represent correctly the physics of such complicated problem. However existing finite elements methods necessitate the use of plate-solid-plate models which are computationally expensive.

In this paper a new sandwich finite element model has been developed. It allows for both symmetrical and unsymmetrical configurations. The rotational influence of the transversal shearing in the core on the skins behaviours, ensure a displacements consistency over the interfaces between the viscoelastic core and the elastic skins; thus resulting in an accurate representations of the physics. Validation examples, consisting on sandwich structure with various geometrical and mechanical behaviours, have been conducted to demonstrate the validity and accuracy of the developed element to (i) estimate the modal resonances; (ii) the frequency response functions and (iii) the damping loss factors. Validations were performed versus both analytical and classical Finite elements models using MSC/Nastran® (Nastran).

2. FINITE ELEMENT FORMULATION

The displacement field of the skins is based on the Love-Kirchhoff's assumptions but is corrected to account for the rotational influence of the transversal shearing in the core. The Mindlin model is used to describe the displacement field of the core. The rotation effects of the transversal shearing in the core as well as the bending of the panel are described by the rotations γ_x and γ_y angles and the transversal displacement w.
The displacements fields of each of the three layers are written as follows:

$$\begin{cases} U_1 = U_{20} - z\theta_x + z_2\psi_x \\ V_1 = V_{20} - z\theta_y + z_2\psi_y \\ W_1 = W \end{cases} \begin{cases} U_2 = U_{20} - z\theta_x + z\psi_x \\ V_2 = V_{20} - z\theta_y + z\psi_y \\ W_2 = W \end{cases} \begin{cases} U_3 = U_{20} - z\theta_x + z_3\psi_x \\ V_3 = V_{20} - z\theta_y + z_3\psi_y \\ W_3 = W \end{cases}$$

Where the following notations are used:
$\psi_x = \theta_x + \gamma_x$ and $\psi_y = \theta_y + \gamma_y$
z_2 and z_3 are the distance between the reference axis and the lower and upper faces of the core, respectively..

The generalized displacements u is related to an elementary degrees of freedom vector q_e witch contain four degrees of freedom per node in the case of the beam and seven for the plate. In the latter case, these are the transverse displacement w, the two rotations of the face sheets θ_x and θ_y, the two rotations related to the transversal shearing in the core ψ_x and ψ_y and the in-plane displacements u_{20} and v_{20} of the middle planes of these face sheets. To account for the curvature, rotational degrees of freedom around the normal to the plan of the beam or plate are added. This result in nine degrees of freedom per node for both cases: $u_{20,}$ $v_{20,}$ $w,$ $\theta_{x,}$ $\theta_{y,}$ $\theta_{z,}$ $\psi_{x,}$ $\psi_{y,}$ $\psi_{z.}$

3. RESULTS

3.1 Sandwich ring

A damped sandwich ring of axial length b and a single point load applied in the radial direction is investigated in this example. Figure 1 and table 1 gives the associated dimensions and material properties. In the following, indices 1and 3 refer to the skins and 2 to the core.

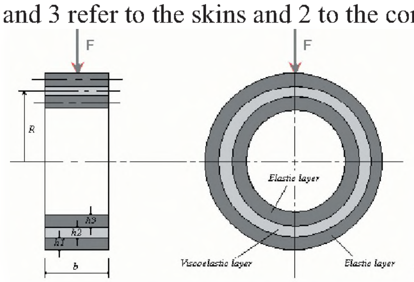

Fig. 1. Sandwich ring

Table 1. Ring's configurations and the material's properties used for the numerical validation

R=0.1015835m; b=0.01m; h1=h3=1.52mm; h2=0.127mm			
E1= E3 (Pa)	7.037 x 10^{10}	E2 = 7.037 x 10^{5} (Pa)	
nu1= nu3	0.3	nu2	0.49
ρ1= ρ3 (kg/m3)	2770	ρ2 (kg/m3)	970
η1=η3	.0001	η2	.3

Figure 2 resents the comparisons between the present finite element model and a classical finite elements model using Nastran. The Nastran model uses solid finite elements for the core and shell finite elements (with offset option) for the skins. Excellent agreement is observed. Both the resonance frequencies and the resonance amplitudes of the first six modes are accurately estimated.

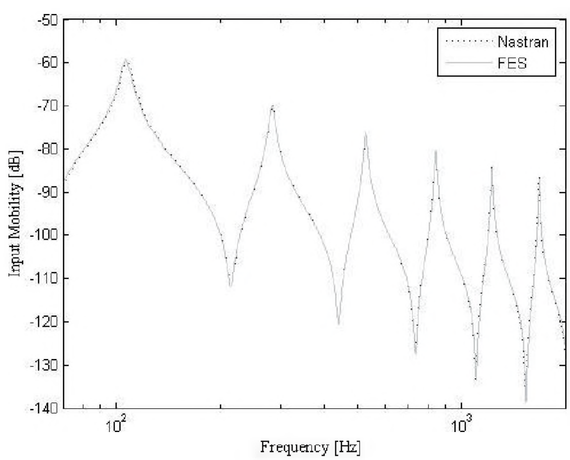

Fig. 2. Input mobility (dB) of a sandwich ring. Numerical validation: (—) finite element sandwich; (-----) Msc. Nastran

3.2 Simply supported sandwich plate

This section compares the modal frequencies of free vibration predicted by an existing analytical solution [5] and finite element method (Nastran) [6], to those predicted by the developed element for a simply supported sandwich plate with symmetric isotropic aluminium skins and a viscoelastic core. The complex shear modulus of the core is assumed constant over the frequency range. The geometrical and physical parameters of the plate are presented in Table2

Table 2. Plate's configurations and the material's properties used for the numerical validation.

L_x=304.8mm; L_y=348mm; h1=h3=0.762mm;h2=0.254 mm			
E1= E3 (Pa)	6.89 x 10^{10}	E2 = 2.67008 x 10^{6} (Pa)	
Nu1= nu3	0.3	nu2	0.49
ρ1= ρ3 (kg/m3)	2737	ρ2 (kg/m3)	999
η1=η3	.0	η2	.5

Table 3. Comparison of natural frequencies and loss factors of a symmetric sandwich with isotropic face-plates.

	Analytical		Nastran (10x12 elements)		FES (10x12 elements)	
	f (Hz)	Eta	f (Hz)	Eta	f (Hz)	Eta
1	60.3	0.190	57.4	0.176	58.24	0.171
2	115.4	0.203	113.2	0.188	114.44	0.191
3	130.6	0.199	129.3	0.188	130.44	0.189
4	178.6	0.181	179.3	0.153	176.96	0.168
5	195.7	0.174	196.0	0.153	196.59	0.165

Compared to the analytical results, it is observed that the present finite element (FES) is more accurate than NASTRAN for the same number of elements. This is corroborated by other tests. Moreover, a substantial savings in computation time is achieved. However, current challenges, fir using the new element to model real life applications, concerns in its interface with classical plate and solid elements.

4. CONCLUSION

A new sandwich finite element for laminated steels has been introduced. It allows for both symmetrical and unsymmetrical configurations. Validation comparisons of the presented approach versus analytical and numerical methods were presented. These studies show that the proposed element sandwich is accurate for the modeling of the studied laminated steels.

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