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A time-domain modeling of systems containing viscoelastic materials and shape memory alloys as applied to the problem of vibration attenuation

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ABSTRACT

It is widely known that traditional damping materials such as elastomers present a number of interesting characteristics when applied for vibration mitigation, such as inherent stability and good damping performance in relatively broad frequency bands, besides cost effectiveness. However, the behavior of those materials is highly dependent upon environmental and operational parameters such as excitation frequency and temperature. Another typical drawback is the added weight entailed by viscoelastic treatments. Especially regarding environmental influences, uncontrolled temperature variations and moisture can jeopardize the damping capacity and endurance of viscoelastic dampers. On the other hand, shape memory alloys present potential advantages in vibration damping due to their large pseudoelastic hysteresis loop in stress–strain relationship and can be used both as a damping material and structural elements in various engineering applications. Thus, it becomes apparent the convenience of combining both types of materials in such a way to explore the advantageous features of each of them. In this paper, a time-domain modeling procedure of structures containing both viscoelastic materials and shape memory alloys is addressed. The main goal is the development of a finite-element-based methodology intended to perform the analysis of engineering structures treated by passive constraining layer damping and pseudoelastic shape memory alloy wires for vibration mitigation. The viscoelastic behavior is modeled by using a four parameter fractional derivative model. To model the hysteresis response of the shape memory alloy, a phenomenological simplified model suitable for performing the parametric study of such dynamic system is used. After the discussion of various theoretical aspects, the time-domain responses are calculated for a three-layer sandwich beam containing viscoelastic materials and shape memory alloy wires and the main features of the modeling methodology are highlighted.

1. Introduction

In the context of passive control of mechanical vibrations, elastomers have been commonly used, since they present great efficiency in mitigating vibrations levels at moderate application and maintenance costs [1,2]. As a result, some viscoelastic polymers have been incorporated in a large number of engineering applications such as aircraft and spacecraft structures. Moreover, many interesting other alternatives have been proposed in the field of passive control of vibrations due to their potential advantages over the traditional damping materials. Gaul and Becker [3] have investigated numerically and experimentally the efficiency of using passive and semi-actively controlled friction dampers in joints for vibration damping by a normal force control. Rongong and Abbas [4] have replaced the traditional polymeric materials by metal swarf in applications in which the classical surface viscoelastic damping treatments by constrained layers are usually difficult to be constructed and making the performance of the damping material less sensitive to temperature variations. Also, the use of shape memory alloys (SMAs) appears to be an interesting strategy in the field of passive control of vibrations due to their potential advantages over the viscoelastic materials, as they can overcome some performance trade-offs. For example, the damping capacity of the SMAs is much higher than that provided by the most commercially available viscoelastic materials [5]. Moreover, the damping factor of the viscoelastic materials can be reduced significantly at higher temperatures, compromising significantly their damping performance in critical engineering systems [6–8]. Since in applications of SMAs used for vibration attenuation requires an SMA that

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displays pseudoelasticity at constant temperature higher than the Austenite finish temperature, to ensure that the temperature of the material remains constant, a heating and cooling system by using a resistive Joule heating should be installed to negate the effects of the self-heating and self-cooling [9,10]. As a result, degradation in performance due to temperature variations could be reduced. In the present study, it has been assumed that the SMA wires are exchanging the phase transformation induced latent heat with the ambient in such way that the SMA wires are always isothermal and in an operating temperature identical with the Austenite finish temperature during the loading and unloading phases. However, Mirzeifar et al. [11] have demonstrated that the accuracy of assuming isothermal conditions in the pseudoelastic responses of SMAs is significantly dependent on the size of the SMAs devices and the ambient conditions in which the SMAs operate in addition to the rates of loading dependency. Hence, the nature of the pseudoelastic behavior exhibited when a SMA is subjected to cyclic loadings at temperature greater than the Austenitic finish temperature, has provided an opportunity to be used for passive damping augmentation in many engineering applications where the restrictions of the viscoelastic materials should be minimized.

More recently, much effort has been devoted to the development of constitutive mathematical models for pseudoelastic effects of SMAs, particularly adapted to be used in combination with finite element (FE) discretization procedure, making them very useful for many vibration damping applications. Feng and Li [12] have presented a modified plasticity model to represent the hysteresis behavior of SMAs for passive vibration damping. Wolons et al. [10] have experimentally examined the effects of loading rates, excitation frequency, strain amplitudes, temperature and static strain offset on the hysteresis damping characteristics of the SMAs. Gandhi and Chapuis [5] have proposed a FE model procedure of the pseudoelastic behavior in SMA materials with experimental validation in order to investigate the effectiveness of SMAs by assuming isothermal conditions for passive damping applications. Other works devoted to the modeling procedure of pseudoelastic SMAs as applied to the problem of vibration attenuation are reported in references [13–16]. However, most of the proposed models are computationally intensive and hard to implement in real-world engineering systems. Thus, in order to incorporate a physically based constitutive model for pseudoelastic hysteresis loop of SMAs in systems containing viscoelastic materials and to predict the dynamic responses, the phenomenological simplified and computationally less intensive SMA material model initially proposed by Lagoudas et al. [13] has been adopted herein.

Several approaches have been developed for performing dynamic responses of simple and more complex engineering structures containing viscoelastic damping devices, as reported in [1,2,17,18]. Among the widely used mathematical representations accounting for the typical dependence of the viscoelastic properties with respect to frequency and temperature, in this paper the four-parameter fractional derivative model (FDM) originally proposed by Bagley and Torvik [18] and modified by Galucio et al. [19] for the transient analysis of viscoelastically damped systems, has been retained.

Although the analytical methods of the considered damping materials are well known numerical techniques, the main contribution intended for this study is their extension to the time-domain modeling of engineering structures incorporating viscoelastic and SMA materials simultaneously for vibration mitigation, which are characterized by frequency- and temperature-dependent material properties and, thus, require particularly adapted analytical and numerical resolution procedures.

In the numerical study, it is considered a three-layer sandwich beam with embedded viscoelastic materials and discrete SMA wires. The numerical simulations are confined to harmonic and transient loadings where the influence of some operational and environmental conditions such as the excitation frequency and the temperature of the viscoelastic material, are also investigated. The obtained results enable to evaluate the improvement and limitations entailed by each damping material and to demonstrate the capability of the modeling procedure to accommodate such design variants.

2. Review of the FE formulation of a three-layer sandwich beam

One type of structure which can be frequently found, for example, in aerospace and automotive vehicles, is a thin or moderately thin sandwich beam structure. In this section the formulation of a three-layer sandwich beam FE is summarized based on the developments made by Galucio et al. [19]. Fig. 1 illustrates the two-node sandwich beam element of length L and width b (not indicated in the figure), which is formed by the base-beam (b), the viscoelastic core (v) and the constraining layer (c). In the same figure the transverse, $w(x,t)$, and the longitudinal, $u_b(x,t)$ and $u_c(x,t)$, displacements, and the cross-section rotations, $\theta_b(x,t) = \theta_c(x,t) = \partial w(x,t)/\partial x$, are also indicated.

In the development of the theory, the following assumptions are adopted: (i) all the materials involved are homogenous and isotropic and present linear mechanical behavior. Moreover, all layers are assumed to be perfectly bonded; (ii) normal stresses and strains in direction z are neglected for all the three layers; (iii) the elastic layers are modeled according to the Euler-Bernoulli beam theory, and for the viscoelastic core, Timoshenko's theory is adopted; (v) the cross-section rotations are the same for the elastic layers and the transverse displacements are the same for all the three layers. These assumptions have been considered by many authors as being adequate for the modeling of moderately thin sandwich structures, as it is the case of the structure addressed in the present paper, where it is assumed the facesheets are thin [20]. Moreover, previous studies carried-out by the authors demonstrated satisfactory correlation between model predictions and experimental results [21].

Thus, taking into account these assumptions and the kinematic relations $u_B(x,t) = u_c(x,t) + (h_c/2)\,\partial w(x,t)/\partial x$ and $u_C(x,t) = u_b(x,t) - (h_b/2)\,\partial w(x,t)/\partial x$, the mean axial displacement and the rotation of the viscoelastic core are obtained through the relations, $u_v(x,t) = (1/2)[u_B(x,t) + u_C(x,t)]$ and $\theta_v(x,t) = (1/h_v)[u_B(x,t) - u_C(x,t)]$. Also, the axial and transversal displacements fields within the element for each layer can be computed from the expressions:

$$u_{xk}(x,z,t) = u_k(x,t) + (z - z_k)\theta_k(x,t), \quad w_k(x,z,t) = w(x,t) \quad k = b, v, c \tag{1}$$

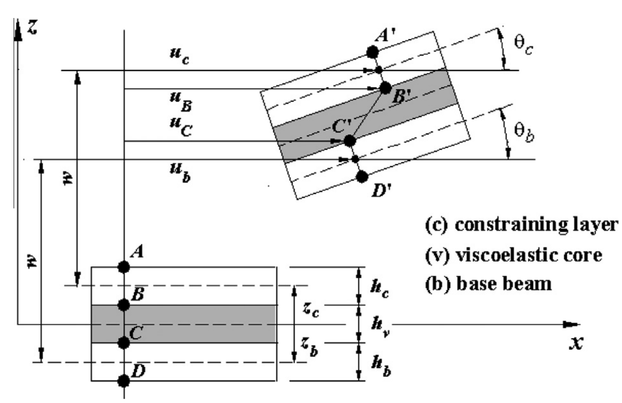

Fig. 1. Illustration of the three-layer sandwich beam finite element.