

Reduced order models for dynamic behavior of elastomer damping devices

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ABSTRACT

In the context of passive damping, various mechanical systems from the space industry use elastomer components (shock absorbers, silent blocks, flexible joints,...). Among other specificities, elastomers have frequency dependent characteristics. The computational cost of the associated numerical models, using viscoelastic constitutive behavior, may become too expensive during a design process. To solve this problem, the aim of this work is to propose an efficient reduced viscoelastic model of rubber devices. The first step is to choose an accurate material model that represents the viscoelasticity. The second step is to reduce the rubber device finite element model to a super-element that keeps its frequency dependent properties. This reduced model is first built by taking into account the fact that the device's interfaces are much more rigid than the rubber core. Therefore, kinematic constraints enforce the rigid body motion of these interfaces, reducing them to twelve degrees of freedom only (three rotations and three translations per face). The super-element is then built by condensing the core of the finite element model on the interfaces via a Component Mode Synthesis (CMS) method, adapted here to viscoelastic damping. As an application, the dynamic behavior of a structure supported by four rubber devices under harmonic loads is analyzed to show the efficiency of the proposed approach.

1. Introduction

Due to their capacity to dissipate energy, elastomer materials are often used in damping devices like silent blocs or joints. That kind of device is for instance found in the space industry: during take-off and flight phases, launchers are subjected to significant vibrations coming either from the propulsion engine or the acoustical environment. Shocks may also occur during the pyrotechnic separation of the launcher's stages. All these vibration sources may damage the satellite or any other sensitive on-board equipment, therefore viscoelastic devices are commonly used as a passive damping solution. The design and optimization of these devices can be done using finite element codes but the computational cost of the associated material models may become prohibitive. Many solutions already exist to reduce numerical models of linear undamped structures, but only a few give access to reduced models with damping behavior, especially when it comes to viscoelasticity which may be seen as a strong form of damping.

A first reduction solution found in the literature is to replace the damping device model by an equivalent rheological model which is identified through experimental measurements. The main issue of this approach is that the rheological model may not fit the behavior of the

damper in each direction. Therefore the rheological model is not predictive and can not be used for design purposes [1].

Other reduction solutions belong to the Component Modes Synthesis (CMS) family introduced by Guyan [2] and Hurty [3]. They rely on substructures eigenproperties. For an overview of CMS foundations, with fixed or free interfaces, the reader is referred to Craig and Bampton [4], MacNeal [5] and Rubin [6]. The extension of the CMS methods to non-conservative systems is usually related to complex modes [7–11]. In most cases, one or more transformations are used to write the dynamic equations in a convenient state-space form [12–15]. Techniques to improve complex mode approaches are also proposed in the literature. For instance, in Ref. [16] a convenient formulation of the dynamic equations is introduced which leads to an easier computation of complex modes. Specific criteria for the vibration modes selection can also be used [17,18].

In the context of reduced order for highly damped structure, one of the main difficulty is to solve the non-linear eigenvalue problem induced by the frequency-dependence of the stiffness matrix. Following the review made by the authors in Ref. [19] which compares modal projection techniques for frequency-dependent problems, the multi-model approach [20] appears to be both accurate and easy to implement. It relies on the

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use of frequency-dependent component modes, computed at a few master frequency points. Examples of its application to viscoelastic structures [21–23] show the good predictability of the method. A similar approach is proposed in Ref. [24], which involves the use of an interpolation scheme to compute the frequency-dependent component modes.

The present work combines a multi-model approach and the classic Craig-Bampton technique to obtain a reduced model of an elastomer silent block. In order to achieve a greater reduction, the difference of stiffness between the elastomer core and the aluminum interfaces of the damper is taken into account and a kinematic constraint enforces rigid body motion at the interfaces. The combination of all these techniques leads to a 12-degrees of freedom (12-dofs) super-element instead of the full silent block FE model.

For clarity reasons, the used application (a structure supported by 4 rubber devices) is presented at first in the paper and the proposed method is applied to it. The method is more general and can be used to any equivalent system.

2. Overview of the studied case

A brief overview of the geometry, materials and boundary conditions applied to the studied case are given in this section. The used finite

element code in this work is an in-house program developed in both Python and Fortran. The Python part is in charge of the global calculation of the frequency response function while the Fortran part is used for the computation and the assembly of the different mass and stiffness matrices. Both meshing and post-processing are done through the free mesh generator Gmsh [25]. The finite element model presented in this section is referred as the reference model in this paper.

2.1. Geometry and mesh

The complete model is composed of an aluminum structure mounted on four hourglass shaped dampers, as seen in Fig. 1. Each damper is made of two aluminum thin interface plates and an elastomer core. The connection of the dampers to the support structure is done through their upper interfaces. The finite element mesh is composed of 8-node hexahedron elements. In order to choose the number of elements, five different meshes of the damper with 1,155/3,128/7,540/11,392 and 23,236 nodes have been compared by performing a modal analysis. Fig. 2 shows that the two finest meshes give almost the same 150 first eigen-frequencies. Therefore, the mesh made of 11,392 nodes is used in the following. As only the computational cost of the dampers is studied here, the mesh of the structure is arbitrarily chosen and contains 11,688 nodes.

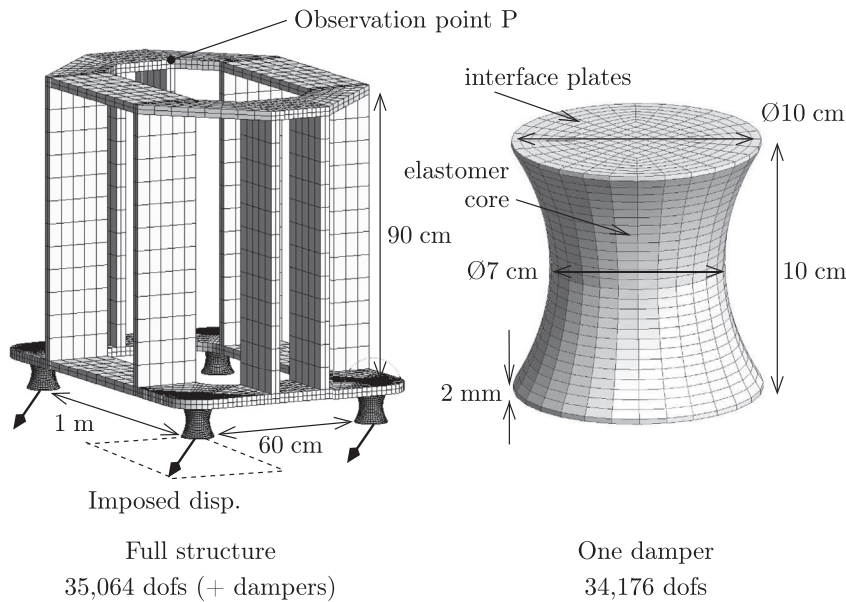


Fig. 1. The complete structure mounted on four dampers and details of a single damper.

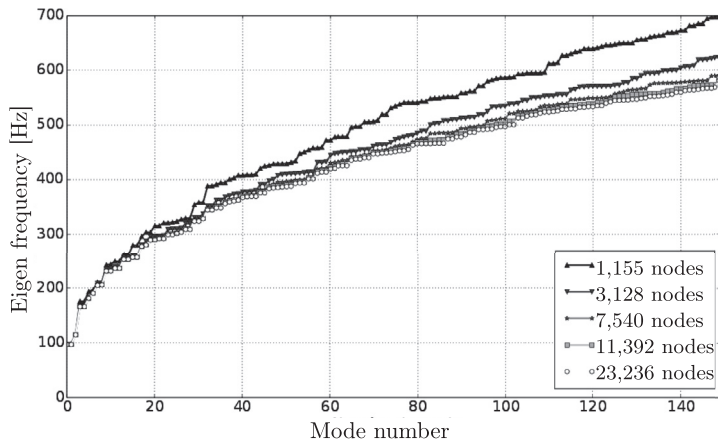


Fig. 2. Convergence study of the damper's mesh.

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