Mechanical Systems and Signal Processing 118 (2019) 696–715

Contents lists available at ScienceDirect

Mechanical Systems and Signal Processing

journal homepage: www.elsevier.com/locate/ymssp

Identification of multi-dimensional elastic and dissipative properties of elastomeric vibration isolators

Ram S. Ramesh, Luke Fredette, Rajendra Singh *

Acoustics and Dynamics Laboratory, NSF Smart Vehicle Concepts Center, Department of Mechanical and Aerospace Engineering, The Ohio State University, Columbus, OH 43210, USA

ARTICLE INFO

Article history:
Received 9 January 2018
Received in revised form 1 June 2018
Accepted 3 September 2018

Keywords:
System identification
Experimental characterization
Modal methods
Stiffness coupling
Rigid body isolation

ABSTRACT

Experimental methods must be adopted to characterize the dynamic properties of elastomeric isolators, especially their multi-dimension elastic and dissipative properties. To facilitate a tractable problem statement a rigid body isolation system (under a weight-type preload) is proposed and reduced to a planar problem with 3 degrees of freedom that can be replicated in any vertical plane. First, this article employs modal methods in tandem with analytical, lumped-parameter models to experimentally characterize the dynamic stiffness matrix, including off-diagonal terms. Fundamental stiffness properties are identified about the elastic center, facilitating a clear relationship between component- and system-level dynamics. Dissipative properties are analyzed in terms of global, structural type and mode-dependent viscous type damping formulations. Modal decomposition is employed to demonstrate the effectiveness of the dissipative models for both coupled and uncoupled motions. The proposed characterization method is validated by comparing predicted dynamic properties of a multi-isolator system with measured responses in multiple directions. Finally, physical insight into the underlying behavior of elastomeric interfacial elements is sought by highlighting the role of the elastic center, comparing structural loss factor and viscous damping matrix models, identifying the correlation between surface hardness and Young's Modulus, and briefly comparing non-resonant and resonant methods. Several annular or cylindrical elastomeric devices with varying size and material demonstrate the proposed method's breadth of application; subsequently, two production mounts are utilized for validation purposes. Various identification issues such as uniqueness of the identified stiffness, damping and elastic center properties are discussed throughout the article.

1. Introduction

Natural and synthetic elastomeric materials are employed for a broad range of vibration isolation systems. Such devices exhibit amplitude and frequency dependent stiffness and damping properties which pose difficulties in the dynamic characterization. Furthermore, these materials are often sensitive to manufacturing processes, and may be non-homogenous and anisotropic; therefore, experimental methods must be adopted [1].

* Corresponding author.
E-mail address: singh.3@osu.edu (R. Singh).

https://doi.org/10.1016/j.ymssp.2018.09.009
0888-3270/

Methods of characterizing the dynamic properties of elastomeric isolators may be broadly categorized into resonant or non-resonant approaches. Non-resonant methods typically use uniaxial, sinusoidal excitation using an electro-hydraulic system [2–7] or electro-dynamic shaker [8–11]. This method applies the dynamic displacement while the specimen is under a mean load, and the force transmitted to the rigid base is analyzed at the excitation frequency to yield the dynamic stiffness of the device. With a visco-elastic model, the stiffness modulus and loss factor may be mapped over a range of frequencies, amplitudes, and mean loads. However, only diagonal stiffness matrix elements (such as compression and shear) may be measured in this manner, and isolator fixtures are often required which may introduce additional dynamics that distort some results.

Resonant (modal) methods generally rely on a system perspective and thus isolators are in a mechanical or structural system, and dynamic mobilities are measured using impulse hammer or shaker tests. The component-level property is then inferred from the system-level vibration response [10–13]. For example, Lin et al. [10] and Ooi et al. [11] characterized the vertical dynamic stiffness of an isolator in a simple uniaxial system using the frequency response from an impulse hammer test and validated their results through comparison with a non-resonant test. However, neither took into account additional degrees of freedom. Kim and Singh [12] placed the isolator between two compact, identical masses and conducted cross-point measurements, using a mobility method to indirectly identify the multi-axis stiffness of the interfacial element, but no mean load was applied in their experiments. Meggitt et al. [13] used a procedure similar to Kim and Singh's, and extended the mobility method to include an isolator placed between two flexible elements to simulate an *in situ* measurement. While both studies showed reasonable accuracy over a wide range of frequencies and considered several isolator geometries, the mobility method is inherently prone to the accumulation of numerical error since the solution is found through matrix inversion. Noll et al. [4] investigated the relationship between the dynamics of an isolated elastic beam structure and the isolator stiffness matrix of dimension 3, including limited off-diagonal terms. A modal approach (but without a mean load) was taken to analyze the system, and the inferred isolator stiffness values were validated by non-resonant measurements. The importance of system-component interactions is highlighted by [4], suggesting that proper characterization of boundary conditions is critical for any indirect characterization method. Further, Joodi et al. [5] have shown the numerical errors that can result from such an indirect method. Finally, while resonant and non-resonant methods are distinct, elements of each may be combined to form a hybrid procedure. For instance, Thompson et al. used a non-resonant measurement, but focused on the resonance behavior of the resulting frequency response functions to indirectly identify the uniaxial stiffness of a rail pad [9].

The concept of elastic center has occasionally been used in the study of mechanical systems [14,15], but to the authors' knowledge, the utility of this physical characteristic has not been applied in the identification of elastomeric interfacial devices. The center of elasticity is a location about which pure translational forces yield pure displacements and pure moments produce rotations only; in short, the component stiffness matrix will be diagonal if and only if the coordinate system's origin is coincident with the elastic center. Locating this point offers great clarity vis-à-vis off-diagonal stiffness terms since these depend on the coordinate system. This article seeks to develop a simplified, experimental identification procedure based on locating and using the elastic center for interfacial connections. The proposed modal analysis based approach, while relying on key knowledge elements from prior literature, aims to offer multi-axis dynamic stiffness (including off-diagonal terms) and damping characterization, as well as insights into component and isolation system level behavior.

2. Problem formulation

To facilitate a tractable problem statement, the full 6 degree of freedom isolation system is reduced to a planar, 3 degree of freedom problem that can be replicated in any vertical plane. A modal approach is employed with a frequency range of interest up to 200 Hz. Several annular or cylindrical elastomeric devices with geometry depicted in Fig. 1(a) and with different sizes and materials demonstrate the proposed method's breadth of application. Subsequently, several production mounts

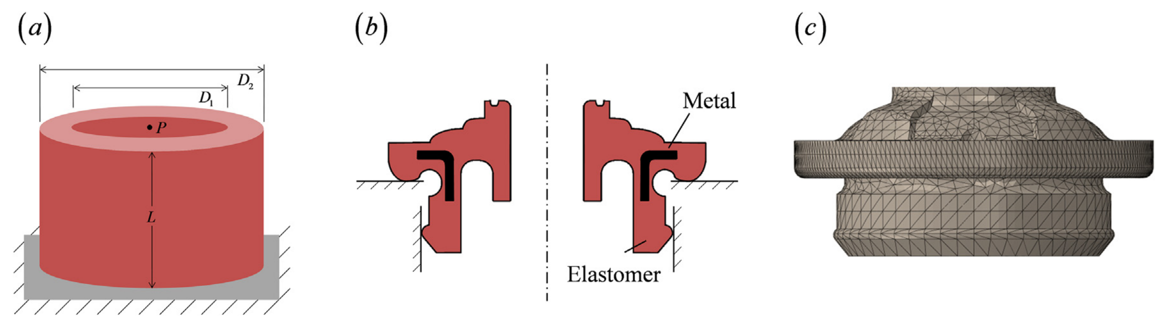

Fig. 1. Geometry of (*a*) laboratory isolators, (*b*) production isolator V, and (c) the finite element model of the isolator V. Here, L is the length, D_1 is the inner diameter, and D_2 is the outer diameter of the lab isolators, and P is the central point of the mating surface in (*a*), representing the location of physical connection to the supported structure.