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Identification of nonlinear anti-vibration isolator properties

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A B S T R A C T

Vibrations are classified among the major problems for engineering structures. Anti-vibration isolators are used to absorb vibration energy and minimise transmitted force which can cause damage. The isolator is modelled as a parallel combination of stiffness and damping elements. The main purpose of the model is to enable designers to predict the dynamic response of systems under different structural excitations and boundary conditions. A nonlinear identification method, discussed in this paper, aims to provide a tool for engineers to extract information about the nonlinear dynamic behaviour using measured data from experiments. The proposed method is demonstrated and validated with numerical simulations. Thus, this technique is applied to determine the nonlinear parameters of a commercial metal mesh isolator. Nonlinear stiffness and nonlinear damping can decrease with the increase in the amplitude of the base excitation. The softening behaviour of the mesh isolator is clearly visible.

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1. Introduction

In many engineering applications, it is required to minimise the transfer of vibrations from the source to the receiver. In order to solve this problem and reduce the transmitted vibration, a vibration isolator should be added. From several isolation techniques, the passive isolator has been widely applied in engineering due to its simple design and high reliability. Different kinds of passive isolators are applied in many fields. For instance, typical vibration isolators employ metal coil spring to store the energy due to resilience and to maintain the force between contacting surfaces. Elastomeric shock mounts, such as rubber isolators that absorb mechanical energy by deforming, play an important role in noise and vibration control. They are widely used in automotive engines [1], aircraft components, industrial machinery, and building foundations. In practice, air spring, pneumatic and elastomeric vibration mount, are also commonly used as an important fundamental part of mechanical equipment requiring low natural frequency isolation and automobile suspension system [2]. Viscoelastic material isolators are considered as a relatively new damping material and have been extensively used in aerospace applications [3]. There are various types of this kind [4–6], such as the vibration isolator using solid and liquid mixtures (SALiM) [7], which was inspired by Yamamoto [8]. Further to that, Courtney carried out some experiments on a shock-absorbing liquid

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absorber to validate its basic properties, and referred to as the SALiM liquid [9,10]. Another kind of passive isolator is the passive negative stiffness isolator [11,12], which is a revolutionary concept in low-frequency vibration isolation. This isolator is provided by a spring that supports a load, combined with two springs, which are called corrector or auxiliary springs, acting as a negative stiffness mechanism. The metal mesh isolator, which is essentially comprised of stainless steel wires crimped, rolled or compressed into any geometric shape that is required, is one of the important passive vibration isolation products stop-shock. It can provide a solution for many engineering applications, for example, engines and gearboxes supports, railway lines and suspension bump stops. It has not only higher stiffness than the elastomeric materials, but also offers larger hysteresis loops and provides excellent isolation performance [13].

In order to design a nonlinear system and predict the dynamic behaviour, the modal analysis method, based on mathematical models of a single-degree-of-freedom system, is used. The modal quantities depend on several variables: amplitude of vibration, frequency of excitation, stiffness, and damping parameters. The main purpose to use nonlinear modal analysis methods is to allow engineers to identify and quantify the nonlinearity in a standard testing environment. The most significant application of modal testing is to compare the numerical analysis with experimental data and to apply the necessary changes on the model, in order to obtain satisfactory results.

The identification and quantification of nonlinearity has drawn much attention. There are many techniques currently available, presented in [14,15]. Worden and Tomlinson [16] summarised the background of harmonic balance method and the Hilbert transform. The latter was used by Feldman to propose a method that allows one to study the dynamic system for: free vibration analysis “FREEVIB” [17] and forced vibration analysis “FORCEVIB” [18]. Kerschen et al. [19] classified the identification methods according to seven categories. Some cited methods are: the restoring force durface (RFS) [20], the inverse method [21] and the linearity plots [22]. The RFS works in the time domain and the starting point is the application of Newton’s second law. Moreover, Rice [23] identified the nonlinear parameters using equivalent linearisation and determined the optimum one by minimising the average of the least square of the error. Guo [24] evaluated the transmissibility of a nonlinear viscously damped vibration system under harmonic excitation using a new method, based on the Ritz–Galerkin method, to investigate the effect of the damping characterisation parameters on this system. A. Carrella [25–27] has recently presented a new approach, COde for Nonlinear Characterisation from mEasured Response To vibratiOn, to identify and quantify the dynamic behaviour of vibration isolators, based on the analysis of experimental data. CONCERTO is applied to a single degree of freedom (SDOF) system which is subjected to harmonic base excitation or harmonic force excitation. The principle, upon which the approach is based, is effectively a linearisation; at a given response amplitude, the stiffness and the damping are considered constant. It is also assumed that the system responds at the same frequency as the excitation.

The main novelty of this work is the employment of the identification method mentioned previously to reconstruct the nonlinear stiffness and damping functions of a metal mesh isolator. This paper aims at investigating the dynamic properties of the examined isolator under different levels of excitation in order to improve the reduction of the transmitted vibrations. This paper is organised as follows: the following section introduces the procedure proposed in this work; in the third section, a comparison is performed with an existing nonlinear identification method based on the measured transmissibility [27] in order to validate the numerical model qualitatively and quantitatively; the transmissibility measured data are analysed to characterise and identify the nonlinear stiffness and damping of the investigated isolator in the fourth section.

2. Theoretical study

In this section, the used methodology is presented and discussed. It consists of the measurement of the transmissibility (displacement) from appropriate responses, on the one hand and on the extracting frequency (stiffness) and damping functions, on the other hand.

2.1. Overview of CONCERTO: COde for Nonlinear Characterisation from mEasured Response To vibratiOn

CONCERTO, presented in [25–27], is a frequency-domain method, whose aim is the identification and quantification of nonlinear parameters [25] from measured FRF [26] and transmissibility data [27]. This method is used to analyse numerical and experimental data [27].

The proposed SDOF (see Fig. 1) identification method, based on the assumption that the studied system with nonlinear stiffness and damping subjected to harmonic base excitation, can be depicted through the equation of motion as follows:

$$m\ddot{z} + k(1 + \mathrm{j}\eta)z = m\omega^2 Y \sin(\omega t) \tag{1}$$

where $z = x - y$ represents the deformation of the mount, ω the excitation frequency, k and η are the stiffness and damping loss factor respectively.

The absolute transmissibility is defined as the non-dimensional quantity that tells how the motion is transmitted from the base to the mass at various frequencies. It is measured as the ratio between the output and the input displacements.

$$|T| = \left|\frac{X}{Y}\right| = \left|\frac{k(1 + \mathrm{j}\eta)}{k(1 + \mathrm{j}\eta) - m\omega^2}\right| \tag{2}$$

This can be rewritten in terms of modal quantities as:

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