Journal of Sound and Vibration 415 (2017) 78–90

Contents lists available at ScienceDirect

Journal of Sound and Vibration

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

The elastic properties of an actively controlled piezoelectric transducer: Measurement, analysis and tuning

Pavel Marton *, Jakub Nečásek, Jan Václavík, Pavel Mokrý

Faculty of Mechatronics, Informatics and Interdisciplinary Studies, Technical University of Liberec, Studentská 2, CZ-46117, Liberec, Czech Republic

ARTICLE INFO

Article history:
Received 1 February 2017
Revised 30 October 2017
Accepted 9 November 2017
Available online 23 November 2017

Keywords:
Piezoelectric transducer
Active elasticity control
Negative elasticity
Vibration control

ABSTRACT

We report on the development of a method for the measurement of the effective stiffness of a piezoelectric transducer which is bonded to a vibration control system and operated using an approach known as active elasticity control (AEC). Using high-accuracy measurement techniques it is shown that the effective stiffness of the transducer can reach negative values in a certain frequency range, when a suitable active shunt circuit is connected to its electrodes. Extensive analysis proves the consistency of mechanical and electrical measurements of the dynamically loaded piezoelectric transducer compared with the AEC model. A method which allows the computation of adjustments to the active shunt circuit based on the vibration control device's measured mechanical transfer function is developed. The applicability of the method to a real vibration control system is also demonstrated.

1. Introduction

It is known that the role of vibration control is becoming increasingly important in state-of-the-art measurement techniques. The ultimate example of the state-of-the-art in vibration control is the system developed for the Laser Interferometer Gravitational-wave Observatory (LIGO) which consists of an active vibration control layer, followed by a system of passive suspensions in a large vacuum chamber. Using this combined approach it was possible to reduce the level of vibrations to a desired detection sensitivity of 10^{-19} m [1]. Another field where vibration control is becoming increasingly important, is the construction of large ground-based telescopes. This is due to the increase in their dimensions and complexity, and actuation systems generating wideband vibrations [2]. And last but not least, active damping systems have been recently used to attenuate cantilever vibrations in an atomic-force microscope for achieving faster scanning rates [3]. It can be said that across many fields of science from nanotechnology to large-scale devices there is a large demand for advanced noise and vibration control methods.

The adjective “advanced” does not necessarily mean “complicated”. A promising approach in noise and vibration suppression focused on in this work is based on the use of piezoelectric transducers (PT) connected to active electronic shunt circuits [4]. The method is based on three key principles: Firstly, the transmission of vibrations through the PT is proportional to its mechanical impedance. Secondly, the mechanical impedance of a PT is predominantly controlled by its stiffness. Thirdly, the stiffness of the PT can be controlled to a large extent by an active shunt circuit connected to its electrodes. The suppression of vibration propagating through an interface can be achieved by means of creating an interface with extremely low stiffness (softening mode) [5,6]. On the other hand, high stiffness (hardening mode) can be desirable for the attenuation of noise propagating through planar structures (such as windows or the covers of vibration sources) [7–9]. Such an approach is called Active Elasticity Control (AEC) and has several advantages: simplicity of the device, low construction expenses, simple design rules, and excellent noise

* Corresponding author.
E-mail address: pavel.marton@tul.cz (P. Marton).

https://doi.org/10.1016/j.jsv.2017.11.017
0022-460X/

and vibration suppression efficiency in a possibly broad frequency range. Thanks to these advantages, the AEC method has been applied to several noise and vibration control devices [6,7,9,10].

It has been demonstrated by Fukada et al. [10] that noise and vibration suppression devices can operate in a *negative elasticity* mode. Such a feature opens up several interesting possibilities for the future research of new devices which employ the PT's negative elasticity mode. On the other hand, the negative elasticity mode might be a source of stability issues. This issue is even more important as a PT embedded in a vibration control device based on AEC frequently operates at the edge of its stability region [5,11]. The efficiency of AEC therefore strongly depends on very accurate tuning of the damping unit consisting of a PT shunted by an active circuit. As the electromechanical properties of a PT can change significantly during operation (e.g. due to the dissipation of energy), the shunt circuit needs to be controlled in order to adapt to the changes in the PT and keep the system tuned to its optimum operational point.

Looking at the noise or vibration control devices based on AEC from the perspective of the above paragraph, it is possible to identify two challenges for further research. The first challenge is the accurate measurement of the elastic properties of the PT, which is embedded in the vibration control device connected to the active shunt circuit. The second challenge is the control of the electrical parameters of the shunt circuit, which takes into account the mechanical boundary conditions of the PT.

The aforementioned issues have motivated this work, i.e. (i) conducting accurate measurements of the effective elastic stiffness of the PT bonded to the vibration control system and (ii) demonstrating that the effective stiffness of the PT is negative. (iii) using a comprehensive analysis of the PT's measured effective stiffness shunted by an active circuit, and using an analysis of the shunt circuit parameters the consistency of the analytical model of our vibration control system is directly proven with experimental data. (iv) Finally, with the use of the three above results, an alternative method is proposed for tuning the electrical parameters of the active shunt circuit. In Section 2 we present the principles of the method used to arrive at the announced results. These are in particular the method of elasticity control (Sec. 2.1), the principle of the vibration isolation device (Sec. 2.2), the high-accuracy measurements of the spring constant of the PT (Sec. 2.3), and finally the method for the direct verification of active elasticity control in the vibration control device (Sec. 2.4). The experimental results are presented in Sec. 3. Their discussion and conclusions are presented in Secs. 4 and 5.

2. Methods

2.1. Active elasticity control

It is well known that the mechanical properties of a PT depend on the termination of its electrodes. A PT with open-circuited electrodes is stiffer than one with short-circuited electrodes due to the additional electrostatic energy stored in the electric field inside the PT. The effective spring constant K^{eff} of an axially-operated piezoelectric element connected to a shunt electric circuit can be derived from the electro-mechanical constitutive equations for a piezoelectric material along with the Ohm's law for the attached shunt impedance as [12,13]:

$$K^{\mathrm{eff}} = K^{\mathrm{E}} \left(1 + \frac{k^2}{1 - k^2 + \alpha} \right), \tag{1}$$

where

$$\alpha = \mathcal{Z}_{\mathrm{PT}} / \mathcal{Z}_{\mathrm{shunt}} \tag{2}$$

is the ratio of the electrical impedances $\mathcal{Z}_{\mathrm{PT}}$ and $\mathcal{Z}_{\mathrm{shunt}}$ of the PT and the shunt circuit, respectively. Symbols k and K^{E} stand for the electromechanical coupling factor and spring constant of the PT with the short-circuited electrodes, respectively.

Eq. (1) expresses the possibility of tuning the spring constant of the PT using the AEC approach. The analysis of Eq. (1) for the real values of α, performed in Ref. [12], shows that it is theoretically possible to modify the effective spring constant K^{eff} within a broad range. Under the simplified assumption of purely real values of α, there are three qualitatively different PT operational modes. These are characterized by the value of $K^{\mathrm{eff}}/K^{\mathrm{E}}$: First, the softening mode, which is achieved when $\alpha < -1$. This mode is the most pronounced for $\alpha \to -1$ from below. Second, the negative elasticity mode for $-1 \le \alpha < -1 + k^2$, where the PT expands under the applied compressive force. Finally, the hardening mode, when $\alpha > -1 + k^2$. Again the hardening is the most pronounced when $\alpha \to (-1 + k^2)_+$ as it is reported in Ref. [10]. In the extreme case characterized by $\alpha = -1$, the effective value of K^{eff} equals zero and the PT is perfectly soft. In other words, the PT produces zero reaction force with the applied deformation. Although such an extreme condition is difficult to reach in real systems, the experimentally achieved softening can be very large. The three aforementioned PT modes are depicted in Fig. 1.

2.2. Vibration isolation device

The AEC method presented above is implemented in the vibration isolation device shown in Fig. 2. Its schematic in Fig. 2(a) is a modification of the setup used in our previous studies [5,6] which was also used in the present form in Ref. [14]. It consists of a source of vibrations (piezoelectric shaker) at the bottom and a damped mass at the top, which should be isolated from the vibrations.

The system is designed to represent a one-dimensional propagation path for the vibrations from the source to the damped mass. The overall height of the setup including the base under the shaker is 18 cm. The distance between the damped mass and