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The advantage of linear viscoelastic material behavior in passive damper design-with application in broad-banded resonance dampers for industrial high-precision motion stages

Cornelis A.M. Verbaan [a,*], Gerrit W.M. Peters [b], Maarten Steinbuch [a]

[a] *Control Systems Technology Group, Department of Mechanical Engineering, Eindhoven University of Technology, Den Dolech 2, 5612 AZ Eindhoven, The Netherlands*
[b] *Polymer Technology group, Department of Mechanical Engineering, Eindhoven University of Technology, Den Dolech 2, 5612 AZ Eindhoven, The Netherlands*

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ABSTRACT

In this paper we demonstrate the advantage of applying viscoelastic materials instead of purely viscous materials as damping medium in mechanical dampers. Although the loss modulus decreases as function of frequency in case of viscoelastic behavior, which can be interpreted as a decrease of damping, the viscoelastic behavior still leads to an increased modal damping for mechanical structures. This advantage holds for inertial-mass-type dampers that are tuned for broad-banded resonance damping. It turns out that an increase of the storage modulus as function of frequency contributes to the effectiveness of mechanical dampers with respect to energy dissipation at different mechanical resonance frequencies. It is shown that this phenomenon is medium specific and is independent of the amount of damper mass.

1. Introduction

Motion stages are mechanical structures that are position controlled. This concept finds a wide range of applications in production and measurement equipment [3]. The performance of a complete system depends on the properties of the position controlled closed-loop system, of which the performance is usually limited by the flexible (non-rigid) behavior of the mechanics [2]. This flexible behavior associated with the mechanical resonance frequencies, is excited during acceleration of the motion stage. One solution to improve the closed-loop performance is by increasing the modal damping at these resonance frequencies [17].

A Robust Mass Damper (RMD) is a passive device which has the ability to dissipate kinetic energy at a broad range of resonance frequencies [21], where the Tuned Mass Damper (TMD) acts at a specific single resonance frequency [5,12]. The dissipation of energy at a resonance frequency is called modal damping and is related to the amplification factor of the resonances of a mechanical system. In order to maximize the damping properties for a single resonance, i.e., reduce the amplification factor, active and semi-active variants of the TMD are developed [4,6], as well as strategies with multiple

* Corresponding author. Current address: NTS Systems Development, Dillenburgstraat 9, 5652 AM, Eindhoven.
E-mail address: Kees.Verbaan@nts-group.nl (C.A.M. Verbaan).

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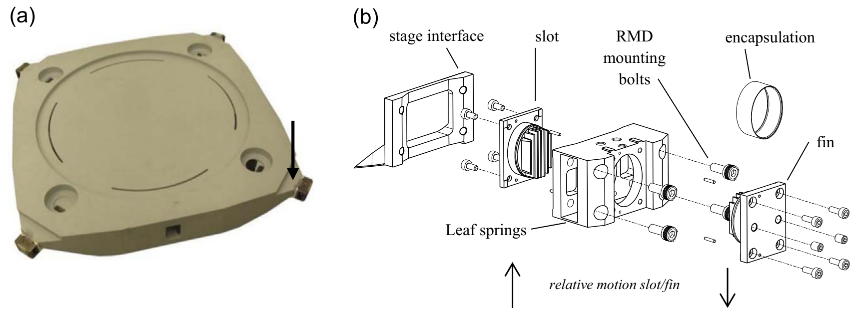

Fig. 1. (a) A motion stage with RMDs added on the corners. The arrow indicates the location of the force actuation and displacement sensor. (b) An exploded view of a Robust Mass Damper (RMD) as mounted on the stage's corners. The leaf springs determine the stiffness and the slot and the fin part act as a shear plate damper. The figure shows multiple fins between which the fluid is encapsulated. The relative motion between the slot and the fin is indicated by the two arrows in opposite direction. The geometrical damping factor (GDF) is determined by the total effective shear area (see slot part) divided by the gap width between those areas.

tuned dampers have been invented [7,8]. Other principles [16] and materials are also explored [1,9,14,23,24]. RMD is relatively new in the field of motion stages and differs from the TMD by its robustness against parameter variations and the frequency range in which the dampers are effective due to the high damping value. This large damping value implies that the damping becomes a more important parameter than in case of a TMD application.

This paper presents the influence of linear viscoelastic behavior on the amount of modal damping that can be obtained over a broad range of resonance frequencies of a mechanical structure. In high-precision motion stage design, the linearity of the damping material is an advantage with respect to the design of the motion controller [15], which is often based on a linear time-invariant (LTI) approach. This implies that nonlinear damping materials, although the damping performance might be good [11,20,25], are suited less for improving the damping properties without introducing side-effects.

In general, the existing idea is that the phase of a damping medium should show a constant 90 deg over a large frequency range, i.e. behave purely viscous, and so obtain large damping values in a mechanical structure. Linear viscoelastic material properties show a decreasing phase angle for increasing frequencies and this is thought to result in a loss of damping for higher frequencies [5]. This paper presents the advantageous application of linear viscoelastic behavior in inertial mass-based damper design, i.e., making use of the decrease of the loss modulus as function of frequency and turning it into an advantage for the mechanical system.

2. Motion stage model and RMD effect

Fig. 1a shows an example of a motion stage as applied in the lithographic industry for processing wafers. This photo shows a ceramic structure, which has a low material damping, and, therefore, a low modal damping. The new RMD devices on the four corners are applied to improve these damping values. The arrow in Fig. 1a depicts the location of the actuation force and position sensor. An exploded view of the Robust Mass Damper is shown in Fig. 1b. A dynamic model of these dampers is depicted in Fig. 2a. Parameter $x_f(t)$ represents the vibration displacement of the stage corner, which enters the damper base mass m_m with displacement $x_m(t)$ via the stiffness c_d and damping d_d of the stage-damper interface. The effective damper mass m_d with displacement $x_d(t)$ is connected to the damper base mass by stiffnesses c_d and damping d_d which represent the stiffness and damping of the leaf spring. Parallel to the leaf springs, the linear viscoelastic damping behavior is described by the spring-dashpot elements, indicated by the dashed line box, with parameters $(m_1 \cdots m_n, c_1 \cdots c_n)$.

These viscoelastic elements are also known as Maxwell elements [13]. The details of the modeling and design optimization are given in [21]. These elements enable to describe the damping behavior as function of frequency. The storage and loss moduli of a realistic fluid (Rocol Kilopoise 0868) are given in Fig. 2b [13]. The damper is of the shear plate type as depicted in Fig. 1c [19] and enables to define geometrical damper parameters that make sense in practical RMD design: the system damping parameter in Ns/m is calculated by the total effective area divided by the gap width between the shear plates, in combination with the fluid's storage and loss moduli. The ratio between total effective area and gap width is called geometrical damping factor (GDF) and is used in this paper as parameter to adjust the damping. A frequency response function (FRF) in z-direction (direction of the arrow in Fig. 1a) of the mechanical stage without dampers is shown in Fig. 3 by the dashed line for the input-output location indicated by the arrow in Fig. 1a. The solid curve presents the FRF of the stage with dampers added. The two discontinuous lines which connect the peaks between frequencies f_1 and f_2, of respectively the undamped and damped FRF's, show the achieved amplitude reduction at the resonance frequencies and, therefore, represent the modal damping improvement. The damped curve is calculated by using optimization techniques, which determine the optimal stiffness and damping values to obtain maximal suppression over the frequency interval f_1–f_2. This paper investigates how the linear viscoelastic fluid behavior as function of frequency influences the maximal achievable amount of resonance suppression.