Enhancing the Dynamics of Flexure Mechanisms Using Low-Wave-Speed Media

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Abstract

Lightly damped poles and zeros in the response of flexure-based servomechanisms often limit their dynamic performance. In this paper, we present a simple technique for increasing the damping of such mechanisms by attaching a layer of low-wave-speed foam to the flexure blades. Measurements conducted on single- and a double-parallelogram flexure stages coupled to low-density, low-wave-speed foams in various configurations show that addition of the foam yields relatively high damping of in-plane, out-of-plane, and higher-order resonances. At frequencies high enough for waves to propagate into the foam, strong interactions between the foam and flexure structure occur, giving rise to a great deal of damping.

1 Introduction

Flexures are mechanical elements that utilize the inherent elasticity of a material to provide smooth and precise motion [1]. In contrast to conventional slideways, flexure mechanisms are free of wear and friction. Hence they are used in several precision engineering applications such as positioning stages, precision bearings, couplings, interferometers, optical scanners, and so on [2, 3]. Flexures can also be used as constraint devices [4]. They are often used to mount delicate objects (such as optical elements, precision instruments, and so on) onto machine bases. However, resonant modes in these mechanisms degrade performance: when incorporated in a motion-control application these lightly-damped modes can potentially destabilize the control system, whereas when used as constraint elements, unwanted disturbances can be amplified by such modes. Hence, designers have long focused on the manipulation of the stiffness and inertia to avoid resonance (by moving the resonant frequencies of the stage away from the forcing frequency).

But it is often impractical or even impossible to avoid resonance, and if resonance occurs, damping plays a critical role in maintaining the performance of the mechanism [5]. The energy dissipated in most structural materials (such as steel or aluminum) suitable for design of precision flexures is usually negligible in the elastic range [6]. Most of the damping in a typical machine arises from micro-slip at material interfaces, but such effects are usually minimized in the design of precision flexures, and therefore most resonances in such systems exhibit very little damping. In this paper, we present a low-cost method to introduce predictable broadband damping into flexure mechanisms using low-wave-speed foam.

The loss factor associated with a given resonance can be determined from a strain-energy weighting. Consider a system consisting of N structural elements, whose kth element has loss factor η_k. For a given mode shape, if the kth structural element stores strain energy V_k, the system will exhibit loss factor η given by

\[ η = \frac{\sum_{k=1}^{N} η_k V_k}{\sum_{k=1}^{N} V_k} \]  

(1)

Because most of the elements of a precision flexure mechanism have negligible loss factors, most damping treatments employ supplementary lossy elements such as squeeze films or viscoelastic or piezoelectric materials, configured so that significant strain energy is induced upon the lossy elements during vibration.

Non-resonant passive dampers couple lossy elements to the primary structure so that quasi-static deformation of the primary structure in the mode shapes of interest induces significant strain energy into the lossy element. For example, a constrained-layer damper formed by bonding to a machine a layer of viscoelastic material backed by a stiff layer channels strain energy into the lossy viscoelastic material when the element of the machine flexes. The damping attainable by such configurations is determined by the degree to which the constraining layer stiffens the structure (or equivalently, increases the strain-energy stored by the system) [7].

This necessary modification of the stiffness of a structure using a lossy element is usually an obstacle

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to the use of strain-based dampers in flexure mechanisms with significant ranges of motion or sensitivity to creep. It is also often difficult to design strain-based dampers (and especially constrained-layer dampers) to effectively damp vibration modes with widely different mode shapes, for example in-plane and out-of-plane modes [8] or low-frequency and high-frequency modes of a planar flexure mechanism.

The broad class of dynamic dampers employ dynamic responses in the added structure to induce large strain energy into lossy elements, as in tuned-mass dampers. These require very little mass and need not modify the elastic behavior of the primary structure, but usually provide significant damping over only a narrow frequency range, typically targeting a single mode [9].

In a series of papers, Varanasi and Nayfeh [10, 11] have explored the use of low-density, low-wave-speed media (such as granular materials or foams) as distributed dampers. At frequencies high enough for wave propagation to occur through the low-wave-speed medium, significant damping is induced into the structure. This is a form of dynamic damper that can very simply be employed to dissipate energy in a broad set of modes without significantly changing the static stiffness of a mechanism or limiting its range of motion.

2 Experiments

In this section, we detail measurements on the pair of flexure mechanisms sketched in Figure 1. The single-parallelogram stage sketched in Figure 1(a) consists of aluminum blades of length 183 mm, width 25.4 mm, and thickness 1 mm that move a mass \( m_1 \) of 0.24 kg. A photograph of the stage is shown in Figure 2. The double-parallelogram flexure stage of Figure 1(b) is manufactured by water-jet machining a one-inch thick aluminum plate. The length and thickness of the flexure blades are 40 mm and 0.6 mm, respectively. The dimensions of the main mass \( m_1 \) are 50\( \times \)58 mm and that of the intermediate masses \((m_2 \text{ and } m_3)\) are 12.7\( \times \)100 mm. A photograph of the stage is shown in Figure 3. Two types of foam layers are used in the experiments: EAR C3201 and EAR C3001 [12]. The density of the EAR C3201 foam is 104 kg/m\(^3\) and Varanasi [13] measures the extensional modulus of this foam to vary as the square root of frequency \((\text{Re}(E_f) = 2075\omega^{1/2}; \ 300 < \omega < 12600)\) and the loss factor and Poisson's ratio to be approximately 0.8 and 0.36, respectively.

For each of the two stages, we measure force-to-acceleration frequency responses by exciting the main mass \( m_1 \) using an impact hammer and measuring the response using an accelerometer. Typical frequency responses with and without foam layers attached to the flexure blades are plotted in Figures 4 and 5. In the absence of foam layers the res-
Figure 4: Measured force-to-acceleration frequency responses for single-parallelogram flexure: dotted (without foam), 0.5 inch thick C3201 foam (dash-dot), 0.5 inch thick C3001 foam (dashed), and 1 inch thick C3001 foam (solid)

Figure 5: Measured force-to-acceleration frequency response of the double-parallelogram stage when the excitation and response are in the compliant direction: without foam (dotted), with 0.5 inch EAR C3201 foam (dashed), and with 0.5 inch EAR C3001 foam (solid)

3 Discussion

Flexure mechanisms are often operated under closed-loop control for precision-positioning applications. For the flexure stages in which actuation and feedback sensor locations lie on the same rigid body, the control is collocated. Under this collocated control, primary resonances (such as the first resonance of Figure 1(a) and first and second resonances of Figure 1(b) in the compliant direction) can be compensated via closed-loop control. However, the higher modes (such as the out-of-plane and in-plane stiff-direction modes, flexible blade modes, and so on) can potentially destabilize and degrade the performance of the stage because it is difficult to robustly compensate for these modes using feedback control (e.g., [5]).

As we have seen in the previous section, damping by incorporation of low-wave-speed media is well suited to improve the system robustness by increasing the damping in the various types of in-plane and out-of-plane resonance observed in a typical flexure stage.

For non-collocated applications (actuation and feedback sensor are located on different masses), the achievable bandwidth is limited by the damping in the drive resonance (Varanasi [14]). Once again, the damping approach presented herein can be useful to increase the damping in such modes and significantly improve performance of the positioning stages. The higher-order flexible modes of the flexure blades typically manifest as lightly-damped zeros in the system’s transfer functions. These zeros can severely deteriorate performance as they can amplify disturbance forces and error motions. Moreover, it is almost impossible for the feedback system to compensate for these dynamic effects caused by the zeros. From the results of the previous section, we observe that the foam layers improve the damping associated with the zero dynamics of the system as well.

Modeling approaches for systems involving low-wave-speed dampers are studied by Varanasi [13], to which the reader is referred for details. For higher-order flexural, extensional, or torsional modes of a structure coupled to a low-wave-speed medium, good estimates of the damping over a given frequency range can be obtained by estimating the complex wavenumber associated with wave propagation. For lower-order modes, the mode shape of the structure and its coupling to the medium must be taken into account. Hence, for all the modes involving large motions of the main mass, we obtain compatible displacements in the foam, solve for wave propagation in foam and combine it with the dynamics of the stage to obtain estimates for damping.
4 Conclusions

In this paper, we show that coupling the flexure blades with low-wave-speed foam is an effective method for introduction of significant damping into flexure mechanisms. The damping method is tested on two flexure configurations and several experiments have been documented. The results show that relatively high and predictable damping can be introduced into several modes of the stage. Also depending on the thickness, density, and speed of sound of the medium, even the lowest modes can be damped without affecting the range of travel of the stage. For modes involving rigid-body motion of the main mass, good estimates of damping can be obtained by considering the mode shape of the stage and its coupling to the medium. For all the higher-order flexible modes of the blades, the associated complex wave number can be used to estimate damping.

When this damping approach is applied to flexure mechanisms operating under feedback control, the closed-loop performance can be robustly improved as significant and predictable damping can be introduced into high-frequency destabilizing resonances and lightly-damped, disturbance-amplifying anti-resonances (complex zeros). Likewise, the response of flexure-based constraint devices can be significantly attenuated using this damping approach.

References


