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Flexure design for precision positioning using low-stiffness actuators

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Abstract: This paper analytically investigates flexures to mechanically guide the moving part of a precision positioning system actuated by Lorentz actuators, where the control bandwidth is typically restricted by the second or higher mechanical resonances that can include internal modes of the positioning mass. Based on analytical models, mechanical resonant frequencies are derived for a given set of parameters to determine flexure dimensions and material. As a result of the derivation and analysis, a model is proposed to predict this second resonant frequency for a given first resonant frequency to achieve better control design and performance. As the verification, the effectiveness of the proposed model is confirmed by using Finite Element Analysis (FEA), as well as an experimental setup for frequency response.

Keywords: Flexure, Lorentz actuator, voice coil motor, precision positioning

1. INTRODUCTION

For positioning systems, a variety of components are available to guide motion, such as roller bearings [Fujii et al. (2010)], air bearings [Hou et al. (2012)] and magnetic bearings [Choi and Gweon (2011)]. Among them, flexures [Yong et al. (2012)], elastic components supporting the moving parts, are commonly used in compact precision positioning systems for their relatively simple mechanism, without nonlinear disturbances such as backlash and friction present in bearings. Particularly by using piezoelectric or Lorentz actuators (voice coil actuators), flexure-guided systems can even achieve positioning with nanometer resolution within a short stroke [Schitter et al. (2007); Tuma et al. (2014)]. Dependent on applications and actuator types, many methods are available to design flexures.

In the case of flexure-guided systems with piezoelectric actuators, the first mechanical resonance that deforms them typically limits the the control bandwidth of the resulting closed-loop system [Schitter et al. (2007)]. For this reason, the mechanical structures are designed to have the resonance at a high frequency for a high control bandwidth [Fantner et al. (2006)]. In the mechanical design, the first resonant frequency can be approximated by the undamped natural frequency $\omega_n = \sqrt{k/m}$ using the moving mass m and the stiffness k [Kenton and Leang (2012)]. In this calculation, it is typically assumed that the moving mass is sufficiently solid. The stiffness k is influenced by the combined stiffness of the piezos and the flexures. The stiffness of the flexures can be obtained from their dimension and material properties, for example by using the Castigliano's second theorem [Kenton and Leang (2012); Lin et al. (2013)]. For complicated mechanical structures, stiffness or compliance matrices are used with transformation matrices to calculate the natural frequency for each rotational and translational axis, treating the moving masses as a solid body [Lai et al. (2011); Xiao and Li (2013)]. To achieve a high control bandwidth, the structures are tuned, such that the natural frequencies of uncontrollable axes including the rotational modes are higher than those of controllable actuation axes [Yong et al. (2012)]. In addition to these well-studied flexure design strategies, there is a clear criterion to select the flexure materials that the ratio of the Young's modulus E and the density ρ (i.e. E/ρ) is desired to be maximized for high resonant frequencies [Kenton and Leang (2012)].

In contrast to the piezo-actuation systems, using Lorentz actuators can have a closed-loop control bandwidth that is much higher than the first resonant frequency, dependent on the mechatronic system design. These systems can be categorized as low-stiffness actuators [Ito and Schitter (2016)] and are applied to optical disk drives (ODDs) [Chaghajerdi (2008); Heertjes and Leenknegt (2010)] or an atomic force microscope [Ito et al. (2015a)]. The mechanical structures of these systems are typically designed, such that the first resonance is the suspension mode along the actuation axis. Its resonant frequency is set to a relatively low frequency by designing flexures for a large actuation range or for better disturbance rejection [Ito et al. (2015b)]. The low stiffness is also beneficial to decrease the power consumption when positioning with an offset is required. In contrast to the first, the second or higher resonances, including structural internal modes of the positioning mass, may limit the achievable closed-loop control bandwidth [Munnig Schmidt et al. (2014)]. Therefore, the second and higher resonant frequencies are typically targeted for high frequencies in the structure design [Lee et al. (2002, 2007)]. Particularly for ODDs to satisfy such requirements, finite element analysis (FEA) is often utilized in the design [Zhang et al. (2008)]. In such

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design, FEA can be applied iteratively to achieve desired performances by varying design parameters [Song et al. (2009)].

So far no clear guideline exits to design flexures for low-stiffness actuators in contrast to the well-studied mechanical design strategies of flexures for piezo-actuation systems. However, design guidelines of a low-stiffness-actuation system may allow an initial design that is already close to an optimum, which might be further improved by applying FEA. Therefore, this paper analytically investigates mechanical resonances that may limit the achievable closed-loop bandwidth. As a result, the paper proposes a simple model to predict the second mechanical resonant frequency, yielding design guidelines to achieve a high control bandwidth.

This paper is organized as follows. Section 2 introduces a flexure-guided positioning system actuated by Lorentz actuators. In Section 3 the resonant frequencies of the flexures are analytically derived. Using the results, design guidelines of low-stiffness actuators are discussed in Section 4. Section 5 and Section 6 present FEA and experimental results for verification. Section 7 concludes this paper.

2. SYSTEM DESCRIPTION

A positioning system to be considered in this paper is illustrated in Fig. 1, where the moving mass in the center moves along the Z axis. In the same manner as piezo-actuation systems, it is assumed that the moving mass itself is sufficiently rigid. (i.e. the structural internal modes occur at high frequencies.) For a large actuation range and high disturbance rejection of the system, the stiffness between the moving mass and the fixed frame is desired to be sufficiently low. To satisfy the requirements, Lorentz actuators (i.e. voice coil actuators) are supposed to be used to generate the force F_z for the Z actuation. Since these actuators utilize the Lorentz force, they have no mechanical stiffness between the moving mass and the fixed frame [Munnig Schmidt et al. (2014)], which is ideal to construct a low-stiffness actuator, unlike piezoelectric actuators.

To mechanically guide the moving mass along the Z axis, several types of flexures are available, such as hinges [Lin et al. (2013)]. Among them, leaf-spring flexures are selected in this paper because they are ideal to realize a low stiffness [Yong et al. (2012)]. For simplicity of implementation, all the leaf-spring flexures are identical, having dimensions of length L, width w and height h, as shown in Fig. 1. In the next sections, the influences of these dimensions and the flexure material on the mechanical resonances are discussed, in order to increase the second mechanical resonance with respect to the first. This is desirable to achieve a high closed-loop control bandwidth.

3. MECHANICAL ANALYSIS

3.1 First resonant frequency

In the case of a low-stiffness actuator, the structure is designed to have the suspension mode at a relatively low frequency as the first mechanical resonance. To derive its frequency in the flexure design, a single leaf-spring flexure can be modeled as shown in Fig. 2, where a coordinate frame is attached to the clamped end. On the other end, the force F_{flex} represents the actuation force distributed to each flexure. The bending moment $M(y)$ of the flexure at y can be given as

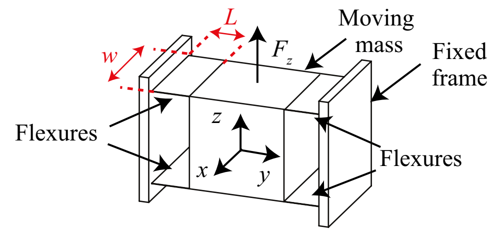

Fig. 1. Illustration of positioning system to be considered.

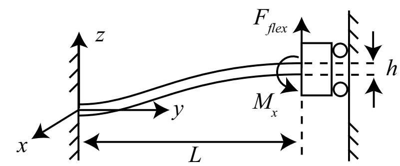

Fig. 2. Model of a single leaf-spring flexure to derive first resonant frequency of positioning system.

$$M(y) = -F_{flex}(L-y) - M_x, \tag{1}$$

where M_x is the moment resulting from the guide due to the layout of the multiple flexures. Using (1), the deflection $z(y)$ of the flexure at y can be derived by solving the following equation [da Silva (2006)]

$$\frac{d^2 z(y)}{dy^2} = -\frac{M(y)}{EI_x}, \tag{2}$$

under the following conditions

$$z(0) = 0, \qquad \left.\frac{dz(y)}{dy}\right|_{y=0} = 0, \qquad \left.\frac{dz(y)}{dy}\right|_{y=L} = 0, \tag{3}$$

where E and I_x are the Young's modulus and the second moment of inertia (i.e. $I_x = wh^3/12$), respectively. The solutions are given as

$$M_x = -F_{flex}L/2, \tag{4}$$

$$z(y) = \frac{F_{flex}y^2}{6EI_x}\left(\frac{3}{2}L - y\right). \tag{5}$$

Since the deflection at $y = L$ is the resulting displacement of the moving mass, the stiffness of a single flexure k_{flex} can be obtained as

$$k_{flex} = F_{flex}/z(L) = 12EI_x/L^3 = Ewh^3/L^3. \tag{6}$$

By assuming that the flexure damping and weight are sufficiently small, the undamped natural frequency approximates the first resonant frequency ω_1 of the low-stiffness actuator as follows

$$\omega_1 = \sqrt{\frac{nk_{flex}}{m_m}} = \sqrt{\frac{n12EI_x}{m_m L^3}}, \tag{7}$$

where m_m is the weight of the moving mass and n is the number of the flexures.

3.2 High resonant frequencies

While the first resonance of the positioning system corresponds to the suspension mode along the actuation axis (Z axis), the second resonance might result from a resonance along uncontrollable axes, such as rotational modes