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IFAC-PapersOnLine 49-21 (2016) 222–227

Design of Tuneable Damping for Precision Positioning of a Two-Body System

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Abstract: Mechanical decoupling poses a limit to the achievable positioning precision of a two-body system actuated by a single Lorentz actuator. To control such a system, the damping between the two bodies needs to be adjusted to a trade-off value, which allows both high control bandwidth of the directly actuated body and good isolation from environmental vibration. In this paper, hydraulic shock absorbers are employed to tune the damping. An experimental setup of a two-body system is built, with the shock absorbers mounted between the bodies. A higher level of damping of the decoupling mode is observed by using fluids with higher dynamic viscosity. The effectiveness of the proposed solution is confirmed by comparing the theoretical and experimental values of the damping coefficient for different values of the dynamic viscosity.

Keywords: Two-body systems, precision positioning, damping

1. INTRODUCTION

In metrology, nanopositioning and nano-manufacturing, a high control bandwidth is typically desired to achieve high levels of precision (Munnig Schmidt et al., 2014). In most high-precision applications, an additional limit to the attainable system performance is caused by environmental vibrations (Amick et al., 2005). Because of their ability to attenuate such vibrations, low-stiffness actuators such as Lorentz actuators can be employed (Ito and Schitter, 2015). In Lorentz-actuated systems, the achievable control bandwidth is typically limited by higher structural modes of the positioned system (Lee et al., 2007). Such modes commonly arise either from the internal degrees of freedom of the mechanical structure or from the decoupling of its different components, as in the case of multi-body systems (Meirovitch, 2001).

Multi-body systems are used in several high-precision applications. For example, a metrology platform consisting of multiple components is proposed to measure nanoscale material properties in a production environment (Thier et al., 2015). This platform is directly actuated to maintain a constant distance between a sample and a measurement tool, thereby mitigating the effect of ground vibrations. Other examples of multi-body systems are wafer scanners, used for the manufacturing of integrated circuits. Because of the high precision requirement (Butler, 2011), they employ vibration isolation frames, which are typically actuated to counteract external disturbances. These systems consist of complex architectures, including several subsystems, sensors and actuators, which also cause structural resonance modes at a given frequency.

The simplest case of a multi-body system is given by a two-body system, shown in Fig. 1. The system consists of (i) the body S_1 that is directly actuated and (ii) the body S_2 that is mechanically connected to S_1. Such a model represents typical structural modes of the positioned structure, which is also referred to as a decoupling mass (body S2; see Munnig Schmidt et al. (2014)). This results in a combination of a resonance- and an anti-resonance in the system transfer function and typically is a limiting factor for the achievable control bandwidth for feedback operation. Typical countermeasures aim to suppress such unwanted structural modes, e.g. via active damping (Babakhani and Vries, 2010) and overactuation (Schneiders et al., 2003). In such cases additional actuators need to be placed into the structure, leading to higher system as well as control complexity.

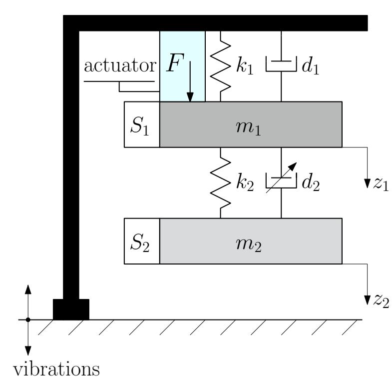

Fig. 1. Schematic model of a two-body system.

In contrast to such strategies, Csencsics et al. (2016) utilizes a resonance mode on purpose. In this approach, the position of the two-body system is controlled via a single Lorentz actuator. This method takes advantage of the second resonance mode of the system (known in this case as decoupling mode) to actively control S_1 with high bandwidth, while isolating S_2 from environmental

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10.1016/j.ifacol.2016.10.554

vibrations. To achieve this however, high damping of the antiresonance and of the second resonance peak is needed. An undamped antiresonance would in fact impair control at this frequency and lead to undesirable high control effort. Additionally, an undamped decoupling resonance may cause extensive excitation of S_2 at this frequency, e.g. due to external disturbances. This shows that the damping d_2 between S_1 and S_2 is an important design parameter, as by changing its value the damping of the antiresonance dip and of the resonance peak can be tuned.

To increase d_2, viscoelastic free-layer and constrained-layer damping can be directly applied to the mechanical connections between S_1 and S_2 (Jones, 2001). In this case, the models describing the damping coefficient are computational-intensive and often require the use of Finite Element Method (Vasques et al., 2010), leading to high design complexity. Eddy-current dampers (Wesselingh, 2015), as well as electrorheological (Holzmann et al., 2009) and magnetorheological dampers (Lai and Liao, 2002) allow to adjust the level of damping by regulating the intensity of an external electromagnetic field. In this case the drawback is the need for an external energy source, with the consequence of increased system complexity.

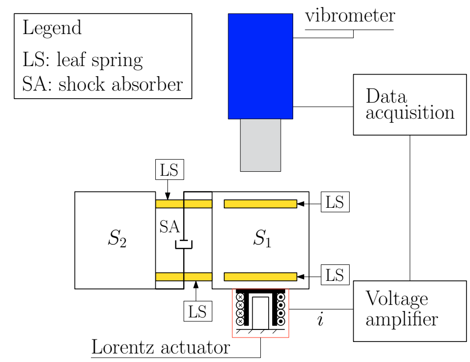
However, a suitable solution can be found in hydraulic shock absorbers (Dixon, 2008). In this case, the damping coefficient can be determined as a linear function of the viscosity of the fluid used (Rao, 2016). This allows to tune the level of damping by simply changing the fluid, without the need for an external energy source. Because of these advantages, hydraulic shock absorbers are proposed in this paper to introduce damping in a two-body system.

The paper is organized as follows. In Section 2, the experimental setup of the two-body system is presented. In Section 3 the dynamics of the setup are simulated. In Section 4, the design of the shock absorber is presented and the dependence of the damping coefficient on the dynamic viscosity coefficient is shown. In Section 5, the frequency response of the two-body system is measured for fluids of different viscosity and the tuneability of the damping is demonstrated. The paper is concluded in Section 6.

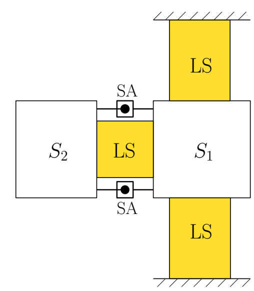
2. SYSTEM DESCRIPTION

In order to demonstrate the tuneability of the damping, an experimental setup of a two-body system is built (see Fig. 2 and Fig. 3). It consists of a body S_1, connected via leaf springs (CuZn37 brass, thickness 0.4 mm) to mechanical ground and to a body S_2. Two hydraulic shock absorbers are installed between the two bodies to introduce additional damping in the system.

The system is actuated by a Lorentz actuator (AVM12-6.4, Akribis Systems, Singapore), which is installed underneath S_1. The coil of the actuator is stiffly connected to S_1 (see Fig. 2(a)), while the permanent magnet is fixed to mechanical ground. The actuator is driven by a custom-made voltage amplifier. For measuring the position of the two bodies, a single-point vibrometer (OFV-534, Polytec GmbH, Hörsching, Germany) is focused on the center of S_1 and S_2, respectively. For data acquisition, a system analyzer (3563A, Hewlett-Packard, Palo Alto, USA) is used.



(a) Side view of the experimental setup.



(b) Top view of the experimental setup.

Fig. 2. Schematic (a) side and (b) top view of the experimental setup.

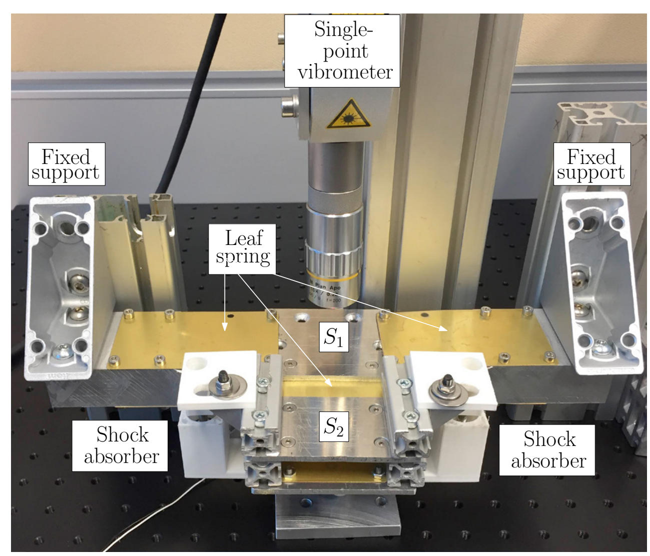


Fig. 3. Experimental setup of the two-body system.

3. SYSTEM ANALYSIS

3.1 Dynamic model

The lumped-mass model of the experimental setup is shown in Fig. 1. The masses m_1 and m_2 represent the mass of S_1 and S_2, respectively. The spring constants k_1 and k_2 represent the stiffness of the leaf springs connecting S_1 to mechanical ground and to S_2, respectively. The damping coefficient d_1 represent the internal damping of the leaf springs connecting S_1 to mechanical ground. The damping coefficient d_2 results from both the internal damping of the leaf springs connecting the two bodies and the tuneable