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Mechatronic Design of an Active Two-body Vibration Isolation System

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Abstract:
Structural modes as for example decoupling of a mechanical subsystem are in general unwanted effects in high precision positioning systems. This paper proposes a well designed decoupling mechanism as a design choice to improve the energy efficiency of an active vibration isolation system that needs to position a structure comprising a high and low precision subsystem. Experimental setups of a single DoF system with a rigid and a decoupling mechanical system structure are developed and analyzed. PD and a PID controllers are designed for the rigid and the equivalent decoupling structure, respectively, resulting in the same disturbance rejection performance for the high precision subsystem. Experiments demonstrate that disturbances of 12.4 μm rms amplitude are reduced to below 118 nm rms for both systems and that the rms energy consumption of the decoupling structure can be reduced by 68% as compared to the rigid system structure.

Keywords: Vibration isolation, Systems design, Mechanical decoupling, PID controllers, Reduced energy consumption

1. INTRODUCTION

High precision positioning systems in production and metrology often require high control bandwidths to ensure the required levels of precision (Munnig Schmidt et al. (2014)). External vibrations are a common problem for these tasks, as they are in general sensitive to disturbances (Amick et al. (2005)). In literature thus numerous passive (Carrella et al. (2007)) and active vibration isolation systems (Kim et al. (2009)), equipped with sensors and actuators to actively reject external vibrations, are commonly proposed countermeasures in these fields. Active concepts that maintain constant proximity between a probe and a sample by means of closed loop control are also reported (Thier et al. (2015), Ito et al. (2015)).

Structural modes of the positioned structure represent a challenge for such active closed loop controlled concepts, as they in general may limit the achievable closed loop bandwidth. Structural modes originate either from modes of the individual components of the positioned structure or their interconnections, as in the case of a decoupling sub-mass (Munnig Schmidt et al. (2014)). Almost every structure that can be considered rigid at low frequencies shows internal structural modes, meaning additional system dynamics, at higher frequencies, depending on its shape and density. There are different strategies proposed in literature to cope with structural modes ranging from active damping (Babakhani and De Vries (2010)) and over-actuation (Schneiders et al. (2003), Falangas et al. (1994)), to the appropriate placement of actuators (Nestorovic and Trajkov (2013)). All of them, however, require an increased system complexity or a high system analysis effort.

In contrast to active concepts structural modes, such as decoupling sub-systems, are intentionally used in passive vibration isolation systems. Applications proposed in Giaime et al. (1996) and Pirro (2006) use passive vibration isolation stacks to reduce the transmissibilty of the resulting structure and to isolate sensitive equipment from ground vibrations. In Csencsics et al. (2016) it is shown that a well designed decoupling mechanism can also be introduced in an active vibration isolation system to improve versatility, without impairing the controllability of the system.

This paper presents the advantage of a designed decoupling mechanism introduced in an active vibration isolation system that positions a structure comprising a high and a low precision subsystem in constant distance to a reference. The proposed approach offers the design freedom to reduce the overall energy consumption of the system for high bandwidth positioning of the high precision subsystem. To demonstrate this targeted property the system design and the experimental setup of a rigid system concept with a high and a low precision sub-metrology-system is presented and analyzed in Section 2. Section 3 introduces the developed single axis prototype of a two-body system design with decoupling mechanism and provides a thorough system analysis. Based on the identification data PD and PID controllers are designed in Section 4 for the rigid and the decoupling system, respectively. In Section 5 it is demonstrated that the control effort for high bandwidth control of the high precision subsystem can be significantly reduced in the system with decoupling mechanism, while maintaining an equal disturbance rejection performance as in the rigid system. Section 6 concludes the paper.

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2. RIGID SYSTEM DESIGN AND IDENTIFICATION

2.1 System description and modeling

In Fig. 1 the concept of a high precision metrology system which is positioned in constant distance Δz to a measurement sample is shown. It comprises a lightweight sub-metrology-system MS1 which requires high precision and bandwidth and a heavy sub-metrology-system MS2 requiring only low precision and bandwdith. Both sub-metrology-systems are mounted on a rigid metrology platform with an entire mass $m_{1,2}$ (platform plus sub-metrology-systems). The platform is connected to mechanical ground via the actuator suspension (k_1 and d_1) and actuated via the force F. It is subject to external ground vibrations, which disturb the constant distance Δz to the sample. The coordinate $z_{1,2}$ represents the vertical position of the platform body.

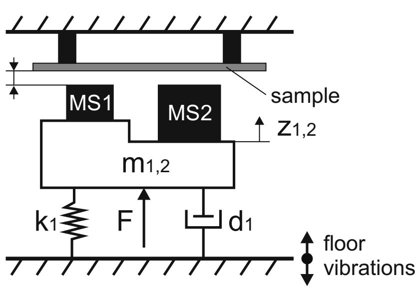

Fig. 1. Mechanical model of the rigid system design concept of a high precision metrology system that is positioned in constant distance to a sample. The high precision subsystem MS1 and the low precision subsystem MS2 are stiffly mounted to the metrology platform (rigid body). $m_{1,2}$ represents the mass of the entire metrology platform and is connected to mechanical ground via the actuator suspension (k_1 and d_1) and actuated by force F.

To model the dynamics of the rigid system along a single degree of freedom the lumped mass models in Fig. 1 is considered. The differential equation, describing the motion of the body of the rigid system, is

$$m_{1,2}\ddot{z}_{1,2}(t) = F - k_1 z_{1,2}(t) - d_1 \dot{z}_{1,2}(t). \tag{1}$$

The transfer function (TF) from the applied force F to the vertical position $z_{1,2}$ can be obtained by reordering (1) and applying the Laplace transformation. This results in the second order TF:

$$G_R(s) = \frac{Z_{1,2}(s)}{F(s)} = \frac{1}{m_{1,2}s^2 + d_1 s + k_1}. \tag{2}$$

To fulfill the requirements on disturbance rejection for both subsystems in closed loop control, the entire platform mass $m_{1,2}$ needs to be actuated over the entire frequency range, that is required to reach the high precision requirements of MS1. Especially for platforms with large mass and target cross over frequencies of several hundred Hertz this might quickly exceed the capabilities of the power amplifier or the actuator in terms of maximum rms current. This can impose stringent requirements of the platform weight, require unnecessary large actuators or high current power amplifiers, or even trouble the feasibility of the targeted system bandwidth and precision requirements.

2.2 Experimental setup

The rigid system structure in the experimental setup is composed of a solid aluminium block ($m_{1,2}$=5.7 kg). It is placed directly on the mover of a voice coil actuator (Shaker S51110, TIRA GmbH, Germany) that is placed on mechanical ground and used for vertical actuation. The actuator is driven by a custom made current amplifier (Amplifier type MP38CL, Apex Microtechnology, Tucson, AZ, USA). The amplifier is controlled by a current controller with a bandwidth of 10 kHz, implemented on the FPGA of a dSpace-platform (Type: DS1005, dSPACE GmbH, Germany). The controller implementation is done on the processor of the dSpace-platform running at a sampling frequency of 20 kHz.

For measuring the position $z_{1,2}$ of the mass $m_{1,2}$ an eddy current sensor S_E (eddyNCDT DT3702-U1-A-C3, Micro-Epsilon GmbH, Germany) is used. The input of the system is the input of the current amplifier, which via the motor constant exerts a force on the moving mass. The signal of the position sensor is considered as the system output.

2.3 System identification

The frequency response is measured by applying a sine sweep and measuring magnitude and phase response by the Lock-In principle with the dSPACE-platfrom (Masciotti et al. (2008)). The measured frequency responses of the rigid system is depicted in Fig. 2. The 2nd order mass-

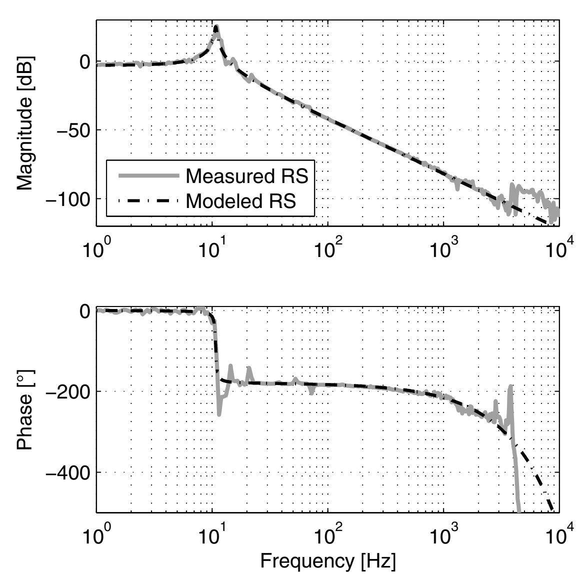

Fig. 2. Measured and modeled frequency response of the rigid system (RS). The frequency response shows mass-spring dynamics resulting from the rigid mass and the actuator suspension with a suspension mode at 11 Hz.

spring characteristic with a suspension mode at 11 Hz and a -40 dB slope after this frequency is clearly visible. Fig. 2 additionally shows the system model

$$G_{RS}(s) = K \cdot G_R(s), \tag{3}$$

with K=1.686e4, $G_R(s)$ according to (2) and coefficients according to Table 1, which is fitted to the measured data.