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Broadband damping of high-precision motion stages

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ABSTRACT

This paper presents a method to apply robust mass dampers on an industrial high-precision motion stage design and shows a substantial improvement of the modal damping of the stage's resonances. The robust mass damper is a passive add-on device with very high damping. The damper placement is based on mode shape analysis. Algorithms are used to optimize the damper parameters such that the suppression factor of resonances over a specified -control relevant- frequency range is maximized. Damper hardware is designed based on the optimization results and the approach is validated by experimental modal analysis. As a result, the equivalent modal damping of the resonances visible in the frequency response function is increased by at least 16 times between the relevant frequencies (between 1 and 4 kHz) and, in addition, by 10 times up to 6 kHz. The time response with robust mass dampers shows an amplitude decrease rate which is 20 times faster in comparison with the stage without dampers. The stage's mass increase amounts less than 2% to obtain this result. The ability to create broad-banded damping by RMDs with a limited amount of mass, in case of a realistic and complex motion stage designs is explicitly proven in this paper.

1. Introduction

1.1. The control problem of lightly damped systems

Motion systems are applied in many industries. One of these industries is the production of integrated circuits (ICs). The main step in IC production is the exposure of the IC pattern on the substrate's surface which is called a wafer [1]. For this purpose, the wafer is clamped to a motion stage which is positioned under an optical column. Currently, the exposure step is limiting the advancement in IC production for two reasons: 1) the minimum feature size is, among other things, limited by the positioning accuracy of the stage with respect to the optical column and 2) the throughput is limited by the achievable acceleration, velocity, and settling times [2]. These properties are largely determined by the motion control system [3,5] and, therefore, much research is executed in the field of motion control. Different control strategies are applied to deal with specific problems. A control system normally consists of a feedback loop and a reference feedforward path. One of the main functions of the feedback loop is suppression of disturbances and the creation of robustness against changes in dynamics [6]. The function of the feedforward path is application of acceleration forces to follow a setpoint trajectory by using an inverse plant model [7]. Together, the feedback controller and the feedforward controller determine the positioning accuracy of the stage.

A feedforward controller performs well only to the extent that the system's dynamical behavior is accurately known. Friction, nonlinear behavior, modeling errors and disturbances limit the effectiveness of feedforward control. A feedback controller counteracts the outgrowth of unmodeled system behavior and random disturbances acting on the motion stage. The suppression of disturbances is related to the controller gain, which is expressed with the term high-gain feedback.

Various control strategies have been developed to further increase the performance: H_∞-controller design for performance increase [8,9], and iterative learning control to track specific references and reject repeating disturbances [10,11]. The latter methods imply a systematic component in the error, which is not the case for random errors. These can only be counteracted by a high feedback gain.

The performance – and also the maximum controller gain – of a feedback control system is largely determined by the plant behavior [12]. Motion stages move predominantly as a rigid body at low frequencies, in contradiction to higher frequencies at which multiple flexible Eigen modes occur at the resonance frequencies or flexible modes. The resulting limit in servo control bandwidth is a function of: 1) the frequencies of these modes, 2) the actuator and

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sensor locations which determine which modes appear in the measurement and how they present themselves (phase behavior), and 3) the modal damping of the resonances. The displacements due to the Eigen-modes appear as vibrations which persist in the stage as function of the corresponding damping value. High-precision motion stages typically contain low modal damping values. This results in large amplification factors at the resonance frequencies and also transient responses that hardly damp out. These low damping values result in large amplification factors at the resonance frequencies, which, subsequently, lead to large circles in the Nyquist diagram. These circles limit the amount of controller gain that can be applied, and, therefore, in general it limits the positioning accuracy of the stage in case of disturbances. Compensation of these modes by the use of notch filters in the feedback controller might improve the bandwidth, but this can endanger the robustness of the closed-loop system for damping and/or frequency variations.

Many methods are presented in literature to increase modal damping. Different damping models for mechanical systems, with attention to viscously and non-viscously damped multiple degree-of-freedom (DoF) vibrating systems, are presented in [13,14]. One solution is to use materials with a high structural damping value [15,16]. However, ceramics are often used in motion stage design because of their high specific stiffness and possibilities to create low thermal expansion coefficients [17,18]. These materials usually lack damping [19,20]. Various approaches are available to increase the modal damping of those structures. Constrained layers can be designed for vibration and sound control for specific frequency regions, in both passive and active variants [21,22]. Squeeze film dampers can result in very high damping values [23]. The Tuned Mass Damper (TMD) is a solution used to suppress the amplification factor of a single resonance frequency [24]. These devices range from a few grams to devices with a few hundred tons of moving mass for tall buildings and towers [25]. Especially in structural engineering such devices have proven to be very useful and have been developed for decades [26,27,35], with extensions to semi-active [28] and active devices [29] as well as multi DoF versions [30,31]. Multiple TMDs at different resonance frequencies can be added to obtain a more broad-banded damping behavior [32].

To improve the modal damping over a broad frequency range, Robust Mass Dampers (RMDs) are proposed in [33,36]. It was shown that the bandwidth of a motion system can be increased by adding dampers to a motion stage. For good results with a minimum number of dampers added, the dampers have to have a broad-banded damping effect, which is created by relatively high modal damping values. This is the main difference with tuned mass dampers, which contain relatively low modal damping values (10–25%). Robust mass dampers typically show modal damping values larger than 50%. In the paper [33], broad-banded damping was shown for a square plate stage model together with corresponding bandwidth improvements.

1.2. Contributions and approach

This paper presents the results of a robust mass damper (RMD) implementation on a complex motion stage with realistic natural frequencies, to increase the modal damping of the out-of-plane modes. An existing motion stage design is used as a starting point. The placement of the dampers is solved and the most effective damper locations are determined, based on the frequency range of interest and the corresponding mode shapes. A design approach is presented which results in parameter values for the dampers to improve the modal damping over a specified frequency range. RMDs are designed based on a shear loaded high-viscosity viscoelastic fluid to obtain a linear time-invariant damper. The RMDs are realized and characterized experimentally to validate their behavior. Experimental modal analyses are performed at stage level with the RMDs added. The theoretical advantages of the implementation of RMDs are proven in practice, in particular for motion stages, and limitations and further challenges are observed and discussed.

Fig. 1. A photograph of an experimental stage design made of ceramics.

The main question to be answered is how to determine the damper parameters for a complex motion stage with realistic dampers included. For this purpose, a complex motion stage with its dynamic model is presented in Section 2. In addition, the dampers are introduced to increase the stage's damping. Section 3 shows the approach to determine the damper parameter values for optimal modal damping over a specified frequency range. Damper parameters are presented and the damper design is elaborated in Section 4. In addition, it shows the characterization of the dampers after manufacturing. Section 5 presents the experimental validation of the stage without RMDs and with RMDs added. For both cases, frequency domain as well as time domain results are given. Section 6 concludes the paper by summarizing the results and Section 7 discusses the interpretation of the results.

2. Stage and damper dynamic models

To increase the modal damping of a motion stage in full complexity, hardware is necessary as a starting point. In this paper, existing hardware is investigated to show that the approach is not design specific. This section describes the hardware and the problems that have to be solved.

2.1. Motion stage hardware

Fig. 1 shows a photograph of an experimental motion stage design, to be used in six actively controlled degrees of freedom. The stage is a monolithic ceramic structure with a complex geometry. The design is a square from top view and the ratio between edge length and stage height is roughly 4.8:1. The top plate is supported by vertical ribs to stiffen the structure. To increase realism, pockets are designed in the top plate in which measurement devices can be mounted. In addition, the corners show protrusions to accommodate for additional sensors. These features add a substantial amount of complexity to the stage design, which is necessary for considering this design as complex enough to validate the theory. An unassembled stage is analyzed in this approach. A computer aided design (CAD) model of this stage, corresponding to Fig. 1, has been made and is presented in Fig. 2.

A finite element model (FEM) is created and an undamped modal analysis is performed. Table 1 lists the properties of the model, the material, and the related software. The lack of material damping, the absence of part interconnections and the monolithic design justify an undamped modal analysis to calculate the stage's natural frequencies and corresponding mode shapes. The stage FEM model is unsuspended: the free natural frequencies are calculated. The natural frequencies of the model are calculated up to 6 kHz (we expect the closed loop bandwidth to stay below