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Development and testing of an active platen for IC manufacturing

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ABSTRACT

The conflicting demands for finer features and increased production rates in integrated circuit manufacturing have emphasized the need for improved wafer positioning technology. In this paper we present operational test results from a magnetically levitated platen with structurally integrated piezoelectric actuators. The strain based actuators provide active damping of the platen's flexible body modes, enabling increased bandwidth on the mag-lev positioning system. Test results reveal a dramatic reduction in steady state positioning error and settling time through implementation of active vibration control.

Keywords: integrated circuit, photolithography, magnetic levitation, vibration suppression, active control

INTRODUCTION

This paper summarizes the development and initial testing of an active vibration control system for improving the performance of a wafer platen used in the manufacture of integrated circuits (IC). The continual push to increase circuit densities while improving production levels has placed significant burdens on the wafer positioning system. Recent developments in magnetic levitation positioning are yielding unprecedented levels of precision,[1] but the conflicting demand for rapid wafer repositioning can lead to unacceptable levels of vibration in the platen containing the silicon wafer. Consequently, the positioning bandwidth for this prototype system must be reduced to minimize platen vibrations and achieve sufficient circuit resolution.

Passive damping techniques can offer limited relief to many vibration problems, but the use of viscoelastic materials characteristic of this approach is generally incompatible with vacuum requirements for future IC production. As an alternative, a strain based actuation methodology suitable for active control of bending vibrations in thick bars and plates has been developed. In this approach, piezoelectric stack actuators are integrated into the structure to induce dynamic bending moments that counter measured deformations.[2] Computer simulations demonstrated the viability of the technique for improving positioning performance,[3] and benchtop experiments on a cantilevered bar and a free-free platen validated the actuator integration approach.[4-5] For the benchtop platen, significant reductions in modal peak amplitudes resulted from acceleration feedback control, with first mode damping estimates increased from 0.2% to 8% critical.

In the present work, the previously developed techniques are extended to a magnetically levitated platen to better quantify system level performance enhancements. In the following section, the actuation methodology is summarized. To elucidate the significance of actuator location and placement on flexible mode controllability, an expression for the modal control forces is developed using the finite element method. After a brief description of the hardware testbed, operational test data is then provided, indicating that a significant reduction in ambient vibration results from application of closed loop control.

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Furthermore, a 40% reduction in settling time is achieved for a one micron step maneuver. The implications of these results are discussed further in the concluding section.

1. VIBRATION CONTROL OF THICK PLATES

In this section, the method of structurally integrated stack actuators is described for the problem of controlling bending vibrations in thick plates. A thin plate analogy is first considered to elucidate the coupling of the actuator deformation to the plate bending moments using a distributed formulation. An expression for the modal control forces generated by actuator excitation is developed to highlight the significance of actuator location and placement. Actuator positioning is restricted to the top surface of the plate, consistent with the geometric constraints imposed on the platen by the mag-lev system.[3] A discretized version of the formulation suitable for thick plates is also provided.

Thin plate formulation

To elucidate the interaction of the actuators with bending vibrations, we first consider a uniform thin rectangular plate of dimensions $a \times b$ and thickness h as shown in Figure 1. A distribution of PZT stack actuators are shown mounted in pockets flush with the top surface of the plate using a two-point preloaded contact. In general, the external influences acting on the plate can be described by a distributed force per unit area $f(x, y, t)$, and distributed moments per unit area $m_x(x, y, t)$ and $m_y(x, y, t)$. Using a normal mode expansion to describe the plate bending, the modal control forces are given by the inner product of the mode shapes and the external forces[2]

$$Q_j(t) = \int_0^b \int_0^a \phi_j(x, y)\left(f(x, y, t) + \frac{\partial m_x(x, y, t)}{\partial y} - \frac{\partial m_y(x, y, t)}{\partial x}\right)dxdy \quad (1)$$

In response to an applied voltage $V(t)$, each actuator produces opposing forces concentrated at the actuator/plate interfaces according to

$$F(t) = K_a p d_{33} V(t) \quad (2)$$

in which K_a is the stiffness of the actuator assembly, p is the number of wafers in the stack, and d_{33} is the piezoelectric constant. In equation 2, the passive elastic contribution of the actuator force resulting from deformation of the host structure has been omitted.

As a consequence of the offset distance $h/2$ from the mid-plane of the plate, the actuators yield discrete moments given by

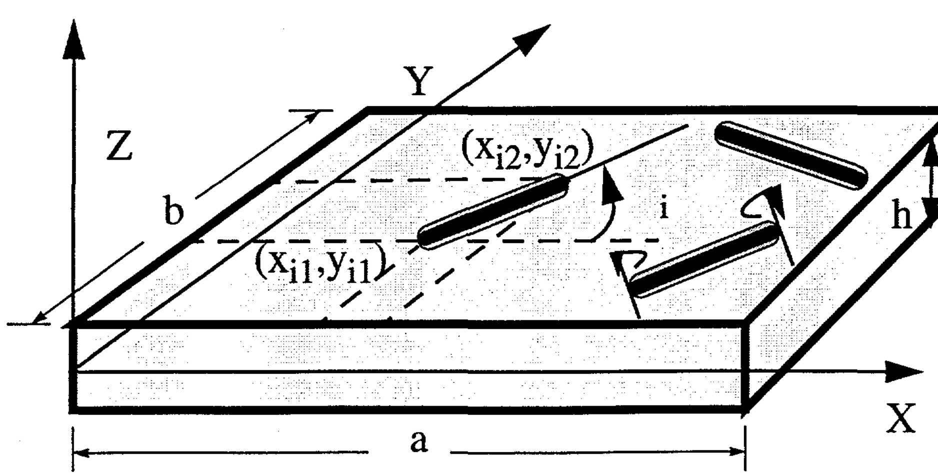

Figure 1. - Rectangular plate with structurally integrated PZT stacks at the top surface.

$$M_i(t) = \frac{h}{2}K_a p d_{33} V_i(t)\,,\, i=1,2,...n \tag{3}$$

where n is the number of actuators. For top surface actuators, the moment is considered positive when the actuator is in compression (applied forces pointed away from actuator tips). The discrete moments can be decomposed into their into their x and y components according to their mounting angles α_i. After substituting the result into equation 1 and integrating by parts, the modal control forces can be written as

$$Q_j(t) = \frac{h}{2}K_a p d_{33}\sum_{i=1}^{n} V_i(t)\left[\cos\alpha_i\left(\frac{\partial\phi_j(x_{i2},y_{i2})}{\partial x} - \frac{\partial\phi_j(x_{i1},y_{i1})}{\partial x}\right) + \sin\alpha_i\left(\frac{\partial\phi_j(x_{i2},y_{i2})}{\partial y} - \frac{\partial\phi_j(x_{i1},y_{i1})}{\partial y}\right)\right] \quad . \tag{4}$$

Examination of equation 4 reveals that the optimal location for controlling a given mode coincides with the locations of highest strains. However, proper orientation of the actuators require additional consideration. In particular, the directional dependence of the mode shape derivative dictates the most efficient orientation. In summary, actuators should be oriented along the directions of maximum curvature in the regions of highest strain to ensure good coupling with a particular mode.

Discretized Thick Plate Formulation

In the previous section, the actuator-plate coupling was elucidated under the assumption of thin plates dynamics, legitimizing the use of distributed mode shape expressions. Since most practical problems featuring thick plates with complex geometries and boundary conditions necessitate the use of a discretized dynamic model, this section uses the finite element method to briefly describe a comparable discrete coupling model.

For generality, the structure is modeled using solid elements with three displacement degrees of freedom at each node, but higher order elements could readily be considered without added complication. The nodal displacement vector is given by

$$\underset{\sim}{u}_k = \left[u_{kx}\ u_{ky}\ u_{kz}\right]^T \tag{5}$$

in which $\underset{\sim}{u}_k$ is the relative displacement of the k^{th} node expressed in global Cartesian coordinates. The corresponding nodal force vector is given by

$$\underset{\sim}{f}_k = \left[f_{kx}\ f_{ky}\ f_{kz}\right]^T \tag{6}$$

in which $\underset{\sim}{f}_k$ represents the applied forces at the k^{th} node. Although the form of the finite element model depends on nodal connectivity, without loss of generality the full system model with N nodes can be written as

$$M\ddot{\underset{\sim}{u}}(t) + K\underset{\sim}{u}(t) = \underset{\sim}{f}(t) \tag{7}$$

where M is the $3N$x$3N$ system mass matrix, K is the $3N$x$3N$ system stiffness matrix, $\underset{\sim}{u}(t) = \left[\underset{\sim}{u}_1^T\ \underset{\sim}{u}_2^T\ \underset{\sim}{u}_3^T\ \dots\ \underset{\sim}{u}_N^T\right]^T$ is the system displacement vector, and $\underset{\sim}{f}(t) = \left[\underset{\sim}{f}_1^T\ \underset{\sim}{f}_2^T\ \underset{\sim}{f}_3^T\ \dots\ \underset{\sim}{f}_N^T\right]^T$ is the system force vector.

Analogous to the distributed case, the mass and stiffness matrices can in general be decomposed to yield a set of normal modes and frequencies such that

$$\underset{\sim}{u}(t) = \Phi\underset{\sim}{q}(t) \tag{8}$$

in which Φ is the $3N$x$3N$ matrix of mode shape vector and $\underset{\sim}{q}(t)$ is the $3N$x1 vector of modal displacements. Substituting Equation 8 into Equation 7 and invoking the ortho-normality conditions yields the equivalent system

$$\ddot{q}_j(t) + \omega_j^2 q_j(t) = Q_j(t)\,,\, j=1,2,...3N \tag{9}$$

with the modal forces defined according to

$$Q_j(t) = \phi_j^T \underline{f}(t), j=1,2,...3N \tag{10}$$

For thick plate-like structures, we concern ourselves primarily with controlling bending vibrations about the X and Y axes. Consequently, stack actuators mounted in pockets on the top surface produce control forces along the X and Y directions only. Opposing forces are concentrated at the r^{th} and s^{th} nodes where the i^{th} actuator meets the host structure. Using the angle and sign convention previously described, the nodal control forces are given by

$$\underline{f}_r(t) = -K_a p d_{33} V_i(t) \begin{bmatrix} \cos\alpha_i & \sin\alpha_i & 0 \end{bmatrix}^T \text{, and} \tag{11}$$

$$\underline{f}_s(t) = K_a p d_{33} V_i(t) \begin{bmatrix} \cos\alpha_i & \sin\alpha_i & 0 \end{bmatrix}^T . \tag{12}$$

Substitution of equations 11 and 12 into equation 10 yields the modal control force

$$Q_j(t) = \sum_{i=1}^{n} K_a p d_{33} V_i(t) [\{\phi_j(is)_x - \phi_j(ir)_x\} \cos\alpha + \{\phi_j(is)_y - \phi_j(ir)_y\} \sin\alpha] , \tag{13}$$

in which $\phi_j(is)_x$ represents the j^{th} mode shape vector evaluated at the s^{th} node of the i^{th} actuator along the x direction.

Although the discretized version of the coupling model presented in this section is of greater practical significance, the distributed version associated with the thin plate dynamics shown in the previous section provides greater insight into the influence of the actuators on the plate dynamics. For instance, note that the actuator moment arms explicitly shown in the thin plate derivation are absent from the nodal forces given in Equations 11 and 12. Conversely, using the solid element formulation the actuator offset distance is implicitly expressed in the nodal connectivity. In addition, effective actuator placement strategies can readily be deduced for the distributed case by looking at the derivatives of the mode shapes. In the discretized case, node pairs exhibiting large differences in their modal deformations must be determined to ensure good coupling with a particular mode. Commercial finite element packages featuring strain energy contour plots can be helpful in determining actuator locations.[3]

2. TEST SET-UP

Mag-lev positioning system

In the late 1980s, a concept for a magnetically levitated fine positioning technique was developed at MIT by Dr. David Trumper. Refinements of the technique, and the design of a control system for its implementation have resulted in the fine stage positioning system featured in this study. The central component of the system is a levitated platen containing sixteen ferrous targets and interferometer mirrors needed for full six degree of freedom positioning control. Sixteen electromagnets and six capacitive position sensors are mounted in the frame. Gap measurements provided by the capacitive sensors are used to determine appropriate current levels for desired actuator forces. Custom current amplifiers with 1 amp and 100 volt output limits provide rapid actuator response, allowing for the possibility of high bandwidth positioning. Measurement of platen position (3 translation and 3 rotation DOFs) relative to the frame is accomplished using a Hewlett Packard laser interferometry system which provides resolution of 0.618 nm in X and Y (in plane) and 1.236 nm in Z.

Position control is provided by an embedded 486 PC for user interface, and an assemblage of seven DSPs for data acquisition, manipulation, and real time control. Six independent control loops govern the rigid body motion using critically damped PID algorithms with a user specified bandwidth. Requested forces and torques are transformed to the configuration space for implementation by the electromagnets. Since the resulting set of inputs are not orthogonal to the flexible body modes of the platen, some destabilizing spillover into the higher energy modes is evident. This phenomenon becomes problematic as the positioning bandwidth is increased, necessitating the use of vibration mitigation strategies.

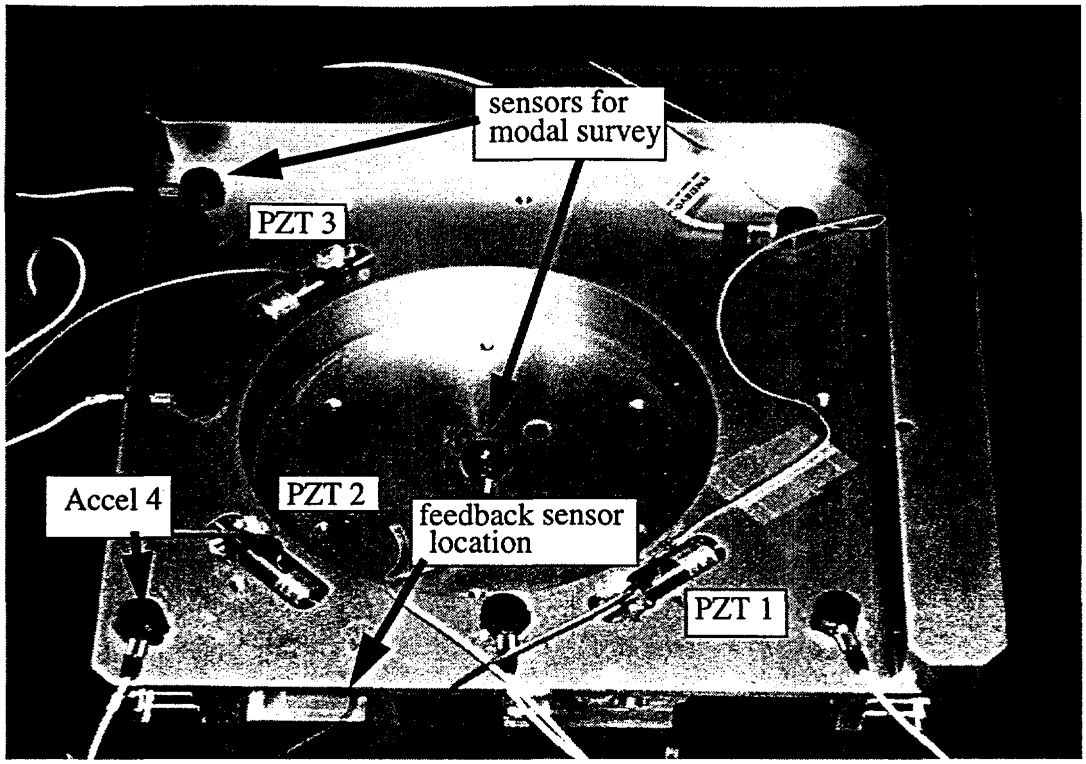

Figure 2. - Benchtop platen fitted with accelerometers and PZT stack actuators for feedback control of bending vibrations.

Platen structural control

A model platen fitted with sensors and actuators and resting on a styrofoam block is shown in Figure 2. The platen shown is geometrically identical to that used in the mag-lev system, although it lacks the mirrored surfaces required for interferometer based position sensing. A number of accelerometers used for modal characterization are evident in the figure. In addition to the modal test sensors, a feedback accelerometer was placed at the indicated location.

Three Physik Instrumente model 840.1 actuators are shown mechanically pre-loaded into the mounting pockets. The chosen actuators have an unloaded stroke of 15 μm, stiffness of 55 N/μm, and an voltage range of -20 to 120 V.[6] The actuator locations and orientations were optimized for controlling the first two modes of vibration. Actuator inclusion resulted in an estimated 3% reduction in static stiffness of the platen. This represents a marked improvement over the previous experiments,[4-5] and can be partially attributed to a reduction in the mounting pocket profiles as well as improvements in the actuator mounting hardware. The custom built interface hardware features rounded tips which were threaded into the ends of the actuators and extended to pre-load the actuators in the mounting pockets. The point contact at the interface eliminated the transmission of bending stress to the PZT stack, and thus, the actuators experience only axial deformation. A lock nut was included to minimize the possibility of pre-load loss during operation. In addition to the mechanical pre-load, a 20 Volt DC bias was applied to the actuators to allow for positive and negative relative control forces.

The active vibration control (AVC) system featured a linear phase low pass filter with a break frequency of 4 KHz. This filter was used to enhance the clarity of the sensor signal prior to it being sampled by the dSPACE controller operating at 17,000 samples per second. A second order positive position feedback[7] algorithm was used to augment the damping in the target modes. After passing through a smoothing filter, the control signals were augmented using Kron Hite amplifiers prior to being sent to the actuators.

3. OPERATIONAL TESTING RESULTS

In order to quantify the performance enhancements enabled through the implementation of AVC, two series of tests were conducted. First, in the absence of AVC, the bandwidth of the rigid body positioning system was increased until an instability was encountered. The second flexible mode of the platen with a natural frequency near 1600 Hz was driven unstable as the mag-lev bandwidth exceeded 90 Hz. After configuring the second order AVC compensator to augment the damping of the second mode, the test was repeated. This time, instability was not encountered until the mag-lev bandwidth exceeded 150 Hz. For comparison, some typical time histories of the measured X and Y position are shown in Figures 3 and 4 with the mag-lev bandwidth set at 150 Hz. As illustrated, the peak to peak positioning error in the absence of AVC was approximately 250 and 350 nm for the X and Y axes, respectively. These errors were reduced to less than 5 nm peak to peak subsequent to the activation of the AVC, indicating that much finer IC features are enabled through the PZT based active damping.

Next, the platen settling time was examined for a series of commanded step maneuvers. One such result is depicted in Figure 5 which shows the Y response to a one micron step maneuver commanded at T=5 milliseconds. With the mag-lev bandwidth set at 150 Hz and the AVC deactivated, the second mode instability persisted throughout the maneuver, essentially invalidating the measure of settling time. With the AVC turned on, the platen reached its steady state by 9 milliseconds with a residual vibration of approximately 4 nm peak to peak. It is significant to note that a comparable residual vibration level was

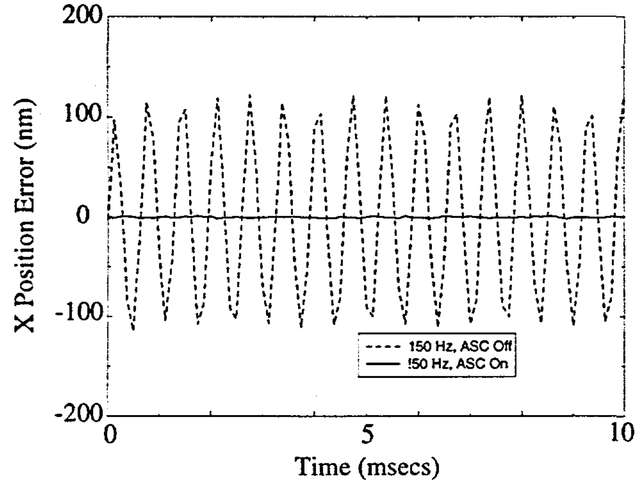

Figure 3. - X direction steady state error with open and closed loop structural control.

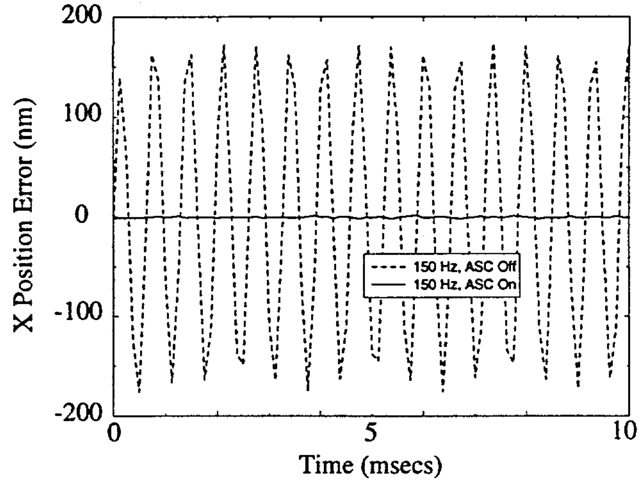

Figure 4. - Y direction steady state error with open and closed loop structural control.

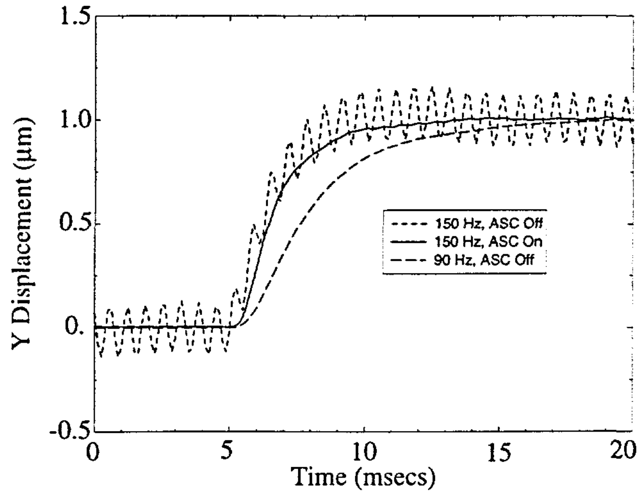

Figure 5. - Y response to a one micron step maneuver with open and closed-loop AVC.

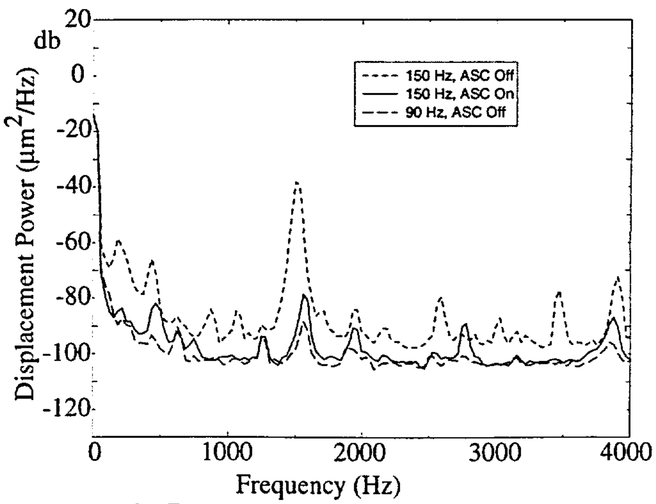

Figure 6. - Power spectrum of Y response to the step maneuver with open and closed-loop AVC.

achieved in the absence of AVC by reducing the mag-lev positioning bandwidth to 90 Hz as shown. However, the settling time for this case is approximately 15 milliseconds, or 60 percent higher than that obtained for the case with active damping. These results indicate that it may be possible to improve IC production rates by reducing time required to step and settle between exposures.

Figure 6 shows the displacement power spectrum for the three maneuver cases considered. For the 150 Hz mag-lev bandwidth, the effectiveness of the AVC is highlighted by the reduction in the peak near 1600 Hz as well as other peaks throughout the spectrum. However, some destabilizing spillover also resulted from this simple vibration control strategy, evident in the increased vibrational energy near 2800 Hz. Despite having a similar residual vibration amplitude, the power spectrum for the 90 Hz mag-lev case reveals a different character in comparison to the AVC case. In particular, the content of its noise is concentrated in the lower frequency range, a problem that is apparently minimized through application of the AVC.

4. CONCLUSIONS

The development and testing of a supplemental vibration control system for a magnetically levitated platen has been described. The mag-lev platen is an enabling technology for precision wafer positioning in integrated circuit fabrication. The vibration suppression system features structurally integrated piezoelectric stack actuators for suppressing platen flexible body modes. Active damping of platen vibrations enables expansion of the mag-lev positioning bandwidth, leading to reduced settling times and steady state errors. In particular, vibration mitigation targeting the second mode leads to a reduction in steady state vibration amplitude from 250 nm to 5 nm peak to peak. Furthermore, a 40% reduction in platen settling time was realized for a representative rest-to-rest maneuver. System performance, however, was limited by destabilizing spillover into the higher frequency modes. Therefore, a strong possibility for further improvements in system performance exists through a more judicious design of the vibration compensator. Still, available results do validate the proposed methodology for improving positioning accuracy and production rates.

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