Passive damping to enhance active positioning of a prototype lithography platen

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ABSTRACT

A viscoelastic tuned-mass damper was used to suppress specific structural modes of a prototype lithography platen. The platen is magnetically levitated and it is repositioned and held in position by a closed-loop feedback control system. Important capabilities of the platen control system are precise positioning and rapid repositioning, which tend to require high frequency bandwidth. The high bandwidth excites structural vibration modes which are disruptive to the control system. The present work was to develop and demonstrate a means to suppress these modes using passive vibration damping techniques. The motivation is to increase the robustness of the platen positioning and control system by reducing unwanted modal accelerations excited by high control system bandwidth. Activities performed and discussed in this paper include the analytical design of viscoelastic tuned-mass dampers and the demonstration / testing of their effectiveness on the platen while levitated and controlled.

2. INTRODUCTION

The robustness of control systems often degrade as gains are set so as to enhance authority. In the case at hand, position control for a GCA prototype platen, resulting instability is prohibitive to the function of the lithography device at all. The prototype platen, made to fit into a lithography device made by GCA, is a monolithic aluminum structure approximately 11.2" x 11.2" x 3" in dimension and with an 8.3" diameter hole in the center (Figure 1). When operating, the lithography device levitates the platen magnetically. It is repositioned and held in position by a closed-loop feedback control system developed at Sandia. Important capabilities of the platen control system are precise positioning and rapid repositioning, which tend to require high frequency bandwidth. As the bandwidth is extended upward, it includes resonance frequencies of structural vibration, and the control system senses and excites those modes, creating a regenerative instability. Activities performed and discussed in this memo include the analytical design of viscoelastic tuned-mass dampers and the demonstration / testing of their effectiveness on the platen while levitated and controlled.

Figure (1): Finite element mesh of prototype platen.

Though notch filters may mitigate the problem for isolated resonant frequencies, such an approach is inadequate as more resonant frequencies are encountered, and as those frequencies become more dense. An alternative approach, discussed at length by von Flotow⁵, is that of adding mechanical damping to the system being controlled. Such an approach was employed for the platen control problem, using viscoelastic tuned-mass dampers and testing of their effectiveness on the platen subject to magnetic levitation and position control.

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3. PLATEN CONTROL AND LITHOGRAPHY

The prototype platen is repositioned and held in position by a closed-loop feedback control system. Important capabilities of the platen control system are precise positioning and rapid repositioning, which tend to require high frequency bandwidth.

The present work was motivated by vibration problems resulting from excitation by the control system. The control system is intended to command rigid body motion and the sensors of the control system are filtered to begin to "roll off" below the natural frequencies of the platen. Unfortunately, there is still enough "spill over" into those frequencies to excite resonance. Drawing back on the control frequency will mitigate that resonance, but it also reduces the responsiveness of the system. Further, either slop in the control system or excited resonances will undermine the necessary precision of the lithography process. The introduction of passive damping is done to manage the resonance problem without unreasonable compromise of system authority.

Of primary interest in the present work is the first nature mode of the platen, occurring at 1,250 Hz and shown in Figure 2. This is a saddle shape having most deflection at the edges and very little at the mid-sides. The second mode occurs

Figure (2): First deformation mode of the platen.

at 1,970 Hz and has maximum displacements at mid-sides.

Because the platen had already been machined by the time that the existence of a vibration problem was identified, there was no opportunity to design damping into the structure through lamination. Because of the constraints of the problem, it was decided to mitigate the resonance through the use of viscoelastic tuned mass dampers.

4. VISCOELASTIC TUNED MASS DAMPERS

The strategy of tuned mass dampers (also known as vibration absorbers or frequency splitters) is to connect an additional mass to a structure using a spring whose stiffness is such that the frequency of that mass-spring system is identical to that of the original structure. For viscoelastic vibration absorbers, the connection between the added mass and the original structure is contrived to involve some damping when there is relative motion between the two. When the base structure resonates, the added mass will be excited to relative motion and not only will the structure be detuned to the original frequency, but some mechanical energy will be bled off through viscous dissipation.

The viscoelastic dampers employed in this project were small masses sitting on thin pads of very lossy viscoelastic solid. The damped vibration absorber was chosen to function in an extensional mode, as shown in Figure 3. The

Figure (3): Configuration for viscoelastic vibration absorber exploiting shear deformations in viscoelastic layer

effective extensional stiffness of such a layer depends on its aspect ratio $\beta = R/H$ since extensional motion of the mass requires a lateral flow of the viscoelastic material and the flow impedance depends on the aspect ratio. Semi-analytic calculations give that effective stiffness

$$K^*(\omega) = \frac{\pi R^2}{2H} E^*(\omega) \left(1 + \frac{\beta^2}{8} \right) \quad (1)$$

where $K^*(\omega)$ is the complex stiffness of the pad at the angular frequency ω and $E^* = 3G^*$ is the complex Young's modulus of the material.

Fraction of critical damping is estimated using what is usually referred to as the "modal strain energy (MSE) method" and is now the conventional one for analysis of most viscoelastic damping treatments¹. Though it has many serious limitations and assumptions built into it, it is generally a successful design tool. An effort is made in this section to enumerate the limitations and assumptions. (This approach is distinct from that usually attributed to Ross, Kerwin, and Unger³. The RKU analysis was a more rigorous treatment than that presented here, but it applied only to the constrained layer damping of plates and beams.)

The steps in using this approach are:

1. A deformation mode of the structure is presumed. These modes are usually the distinct eigenmodes of the undamped system. This first step is dependent on the assumption that the viscoelastic response does not significantly change the modal response of the structure. This assumption precludes the calculation of complex modes.
2. The deformations of the viscoelastic materials consistent with the above deformation mode are calculated. Usually this is done with a static elastic analysis, using the real pairs of the complex moduli evaluated at the frequency of interest. This also is an ad hoc approximation. Such a static analysis is inappropriate for the case of a viscoelastic vibration absorber, since its deformations are driven by inertial forces. A more appropriate method of deformation analysis for such devices is presented in Section 5.
3. The energy dissipation in each viscoelastic material is estimated. This is generally done presuming the dissipation to be some scalar, η , times the maximum strain energy stored in that material during a deformation cycle. This presumes that the real part of the matrix of complex material properties is proportional to the imaginary part. This assumption is true where the material is isotropic and there is only one mode of deformation evident, such as shear or bulk deformations.
4. A damping term is added to the second order system describing the original structure so that the energy dissipation per cycle for the whole structure is that calculated in the above step.

These steps are realized for the case of the tuned viscoelastic damper:

1. *Postulation of deformation mode:* Here we presume that the deformation mode of the platen itself is not significantly changed by the presence of the vibration absorber. If the mode of the platen is $U^*(x)$, and the vibration absorber is placed at location $\hat{x}$ the displacement of the platen at that location will be $\hat{u}(t) = \text{Im}\left(U^*(\hat{x}) e^{i\omega t}\right)$.
2. *Calculation of Deformation in Viscoelastic Material:* Where the relative motion between the mass, M_D , and the base of the vibration absorber is $\Delta u(t) = \text{Im}\left(\Delta u^* e^{i\omega t}\right)$, one solves the equation of motion of M_D and finds²:

$$\Delta u^* = \frac{\omega^2 M_D}{K^* - \omega^2 M_D} U^* \quad (2)$$

3. *Calculation of Energy Dissipation in Damper:* Integrating the force $\text{Im}\left(K^* \Delta u^* e^{i\omega t}\right)$ exerted by the viscoelastic pad times the relative velocity $\text{Im}\left(i\omega \Delta u^* e^{i\omega t}\right)$ across the pad, one finds the dissipation over a complete cycle:

$$\text{Diss}_{VE} = \pi \text{Im}\left(K^*\right) \left| \frac{\omega^2 M_D}{K^* - \omega^2 M_D} \right|^2 |U^*|^2 \quad (3)$$

4. *Evaluate of fraction Critical Damping:* The fraction of critical damping is expressed in terms of the ratio of energy dissipated over a cycle to the maximum energy stored in a cycle

$$\zeta = \text{Diss}_{VE} / 4\pi SE. \quad (4)$$

The strain energy SE associated with the normalized eigenmode is made available by the finite element code.

5. DAMPER DESIGN AND MATERIAL SELECTION

The masses used were brass (fairly dense, non-magnetic, and readily available) cylinders 1.25" diameter and 0.6" thick, weighing 0.23 lb. each.

The viscoelastic material used, Scotch-Seal synthetic putty 1167, was selected primarily for reasons of convenience:

1. readily available;
 2. stable and non-toxic;
 3. easily handled and formed;
- and 4. its viscoelastic material properties are known.

The viscometric functions of a similar material, Scotch Seal 1279, were measured at Sandia several years ago (Figure 4). The properties were consolidated into approximate formulas which become in this context:

$$\log_{10} E' = 5.78 + 0.84 \log_{10} \omega - 0.034 (\log_{10} \omega)^2 \quad (5)$$

and

$$\log_{10} E'' = 5.58 + 0.86 \log_{10} \omega - 0.041 (\log_{10} \omega)^2 \quad (6)$$

(Though complex shear moduli were measured, these quantities are converted to complex Young's moduli through the relation $E^* = 3G^*$, which is almost exact for rubbery polymers.)

Figure (4): Shear storage and loss moduli of Scotch Seal 1279 at 30° C, as measured by Doug Adolf at Sandia National Laboratories. These properties are converted to complex Young's moduli with the approximate relation: $E^* = 3G^*$.

Evaluation of the fraction of critical damping now requires calculation of the ratio of damping per cycle to the maximum strain energy per cycle. The strain energy could be obtained through an analysis approximating the platen as a uniform plate and using the formal expressions for plate bending energy⁴. Such an analysis yielded a strain energy of 52.3×10^{10} erg/cm². Because this ignores the existence of the circular depression and notches in the platen, it severely overestimates the strain energy and will result in an underestimate for the fraction of critical damping. A far better value is achieved by referring to the strain energy and displacement fields resulting from the finite element calculations: 9.28×10^{10} erg/cm². This is the value that was used in the calculations described below.

A MATLAB .m file was written to evaluate the series of equations presented above for the case of the tuned mass damper placed on a corner of the platen. Calculations were done for various thicknesses of scotch seal. The calculations of Figure 5 show the anticipated damping associated with various thicknesses of the viscoelastic layer. As expected, as the layer becomes thinner, the peak damping occurs at higher frequencies. Further, as the damper is tuned to lower frequencies the effectiveness of the damper decreases. These curves suggest that to damp a resonance at 1350 Hz, one would design a damper with a viscoelastic layer about 1.25 cm thick and would achieve about 2.5% of critical damping.

6. DAMPING OF THE SUSPENDED PLATEN

A series of modal tests showed the untreated platen to resonate at approximately 1250 Hz with almost no damping (0.08%). Modal tests were also performed on the platen for several configurations of tuned mass damper. All of these configurations involved a brass mass such as was used in the above calculations and a Scotch Seal synthetic putty similar to that of the simulations (Scotch Seal 1167 rather than Scotch Seal 1279). Frequency response curves for two cases are shown in Figure 6:

Figure 7 shows a comparison of measured fractions of critical damping for several thicknesses of viscoelastic material. (Measurements of the thicknesses were done by comparison to a metric ruler and are correct to approximately ± 0.05 cm.) Also shown are values of the curves of Figure 5 evaluated at a frequency of 1250 Hz. Though the calculations predict up to 2.5% of critical damping and at most 1.7% of critical damping was observed for each damper, agreement was extremely good given the uncertainties in both material and dimension. Particularly important is that the experiments show the maximum damping to occur when the viscoelastic pad is approximately 1.3 cm thick, to within measurable accuracy of that which was suggested by the simulations.

Figure (5): Predicted values of fraction of critical damping for different thicknesses of Scotch Seal over a range of excitation frequencies.

Figure (6): Frequency response curves for the suspended platen, undamped and heavily damped cases.

The experiments on the suspended platen demonstrate:

- Each small viscoelastic tuned-mass damper is capable of providing substantial damping, almost 2%, to even as massive a structure as the lithography platen.
- Straightforward application of analytic damping principles are sufficient to provide both good prediction and design guidance for the fabrication of viscoelastic tuned mass dampers.
- Stiffening effects of aspect ratios, β , are correctly calculated.

Figure (7): Comparison of predicted and measured damping values at the experimental resonance frequency, 1250 Hz.

7. DAMPING VIBRATIONS ASSOCIATED WITH THE CONTROL SYSTEM

The platen was mounted in the lithography device and levitated and centered under the authority of the control system. The role of the viscoelastic tuned mass dampers is best illuminated by experiments in which the platen was deliberately excited by white noise injected into the control system. A single 3-axis accelerometer mounted on a platen corner was used to obtain platen acceleration. The platen is shown in the lithography device in Figure 8.

Figure (8): The platen mounted on the lithography device. Viscoelastic vibration absorbers are situated on four sites on the platen.

The lowest resonant frequency was of greatest initial interest, and experiments of placing various numbers of viscoelastic vibration absorbers at the anti-nodes of the platen (the corners) were performed. Each vibration absorber was tuned to that first resonant frequency. Figure 9 shows the acceleration spectral response versus frequency for those various cases. One sees substantial suppression of that first mode: the case of four dampers achieves 28 dB reduction. As expected, there is almost no effect on the second mode.

Figure (9): Acceleration spectral response to low amplitude white noise. Cases of undamped platen and of platen with one, two, and three dampers situated to suppress the first resonant mode.

With the suppression of the first resonant mode, the second takes on added significance. An experiment was performed where two vibration absorbers tuned to the first natural frequency were put at antinodes of the first mode and two vibration absorbers tuned to the second natural frequency were placed at antinodes of the second mode. The results are indicated in Figure 10. We see that substantial vibration suppression - on the order of 20 db - are achieved for each mode.

8. CONCLUSIONS

The results show that the tuned-mass dampers were successful in significantly reducing acceleration at the specific structural modes (first and second) of interest. This can extend the robustness of the control system by reducing structural response to energies input by higher control system bandwidths.

9. ACKNOWLEDGEMENTS

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10. REFERENCES

1. C. D. Johnson, D.A. Keinholz, and L.C. Rogers, "Finite element prediction damping in beams with constrained viscoelastic layers", *Shock Vib. Bull.* 51, Pt 1, pp 71-81, May 1981.
2. A. H. Nashif, D. I. Jones, and J. P. Henderson, *Vibration Damping*, John Wiley and Sons, New York, 1985.
3. D. Ross, E. Unger, and E. M. Kerwin, "Damping of plate flexural vibrations by means of viscoelastic laminate", *Structural Damping*, ASME, New York, 1959, pp. 49-88.
4. S. Timoshenko and S. Woinowsky-Krieger, *Theory of Plates and Shells*, 2nd Edition, McGraw-Hill, New York, 1959, pp 46-47.
5. A. H. von Flotow and D. W. Vos, "The need for passive damping in feedback controlled flexible structures", *Proceedings of the VPI&SU Conference on Recent Advances in Active Control of Sound and Vibration*, April, 1991

Figure (10): Acceleration spectral response to low amplitude white noise. Cases of undamped platen and of two dampers placed at the antinodes of each of the first two modes.

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