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ACTIVE CONTROL OF BENDING VIBRATIONS IN THICK BARS USING PZT STACK ACTUATORS

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ABSTRACT

An experimental investigation into active control of bending vibrations in thick bar and plate-like structural elements is described. This work is motivated by vibration problems in machine tools and photolithography machines that require greater control authority than is available from conventional surface mounted PZT patches or PVDF films. The focus of this experiment is a cantilevered circular steel bar in which PZT stacks are mounted in cutouts near the bar root. Axially aligned and offset from the neutral axis, these actuators control the bending vibrations by generating moments in the bar through their compressive loads. A Positive Position Feedback control law is used to significantly augment the damping in the first bending mode. The implications of the experimental results for machine tool stability enhancement are discussed.

INTRODUCTION

Although originally intended for space applications, much of the recently developed adaptive structures technology can provide benefits in other areas such as manufacturing. The performance of many manufacturing machines is limited by vibration phenomena that may be responsive to structural vibration control treatments. In many cases, augmenting the damping of a single vibrational mode can have a significant impact on a machine's performance. In the case of cutting tools such as drills, lathes, and milling machines, metal removal rates are often limited by regenerative vibration at the cutting tool-workpiece interface whose frequency is related to a particular vibrational mode (Tarng and Li, 1994, Jemielniak and Widota, 1989, Smith and Tlusty, 1991). In integrated circuit manufacturing, rigid body control spillover induces flexible body vibration in a photolithography platen, and can lead to instability. The platen's flexible body dynamics limit the minimum achievable feature size and the chip positioning speed during processing (Redmond et al, 1994). Vibration problems such as these are likely to become more prominent with the development of lighter and more flexible machines that can be quickly reconfigured to accommodate varying production requirements. Adaptive structures technology offers direct benefits for such machines.

Much of the recent research in adaptive structures has focused on the development of actuators and control methodologies for structural shape control and vibration suppression. Shape memory alloys have been widely used for shape control (Chaudhry and Rogers, 1991) but are not well suited for many vibration suppression applications due to the time required to induce phase transitions through heating and cooling. Lead zirconate titanate (PZT) patches and polyvinylidene fluoride (PVDF) films are suitable for vibration damping, but have been restricted to thin beam and plate-like structures because of their limited force outputs (Chaudhry and Rogers, 1993, Koconis et al, 1994). Controlling the low amplitude high frequency vibrations that are characteristic of machine tools requires greater control authority. PZT stack actuators can provide the required force levels, but have thus far been used primarily as active elements for truss damping (Preumont et al, 1992).

In this paper, a method suitable for damping bending vibrations in bars and plates utilizing PZT stack actuators is investigated. Unlike conventional d_{31} patch actuators, the d_{33} geometry and stacked configuration of these actuators provide large control forces in response to applied electric fields. To demonstrate their utility in damping high frequency low amplitude vibrations, two PZT actuators are mounted in cutouts near the root of a cantilevered steel bar. Axially aligned and offset from the bar's neutral axis, the actuators generate control moments in the bar through differential loading. Adequate control of first bending mode vibrations while maintaining compressive loading in the actuators is demonstrated. Dynamic tests reveal that significant levels of damping can be achieved in the first bending mode within the stroke limitation of the actuators.

The paper is divided in the following manner. In the next section, a potential benefit of incorporating active vibration control strategies in machine tools is described. This is followed by a description of the experimental testbed designed to demonstrate the proposed method of damping bending vibrations in bars and plates using integrated PZT stack actuators. For clarity, the coupling of the actuators to bending vibrations in bars is summarized in the fourth section using a pin-force type model. To augment the damping in the first vibrational mode of the bar, a Positive Position Feedback (PPF) control law is then described, followed by a summary of the experimental results. Finally, some concluding remarks and a description of work in progress is provided.

ACTIVE DAMPING IMPLICATIONS

In metal removal processes, the cutting forces generated at the tool-workpiece interface excite bending modes of the tool and supporting structures. Such vibrations adversely affect workpiece surface finish and can lead to regenerative chatter vibrations. Although the high frequencies and small amplitude that are typical of tool vibrations make them difficult to target with active vibration control strategies, significant enhancements to machine tools could result from damping a single mode. For example, the limiting depth of cut in turning, boring, and milling at which regenerative chatter can occur is approximated by (Smith and

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Tlusty, 1987)

$$b_{lim} = \frac{-1}{2K_s\Re(G)_{min}} \qquad (1)$$

in which K_s is the cutting stiffness of the workpiece material and $\Re(G)_{min}$ is the minimum of the real part of the tool's oriented transfer function relating tip force to tip displacement. Note that this asymptotic limit is defined by the mode with the least dynamic stiffness. With a single degree of freedom system, this quantity is given by

$$\Re(G)_{min} = \frac{1}{4K\xi(1+\xi)} \qquad (2)$$

where K is the static stiffness and ξ is the damping ratio.Therefore, assuming no loss of static stiffness, the ratio of the limiting depth of cut for a particular mode that results from an increase in tool damping is given by

$$\frac{b_{lim2}}{b_{lim1}} = \frac{\xi_2(1+\xi_2)}{\xi_1(1+\xi_1)} \qquad (3).$$

In practice, the benefits represented by equation 3 are tempered by the existence of other modes of vibration. As the dynamic stiffness of the limiting mode is increased, the limiting depths of cut defined by other modes are likely to become more prominent. However, in many cases significant performance improvements can be achieved through modest increases in the damping of a single vibratory mode. This is the focus of the experiment described in the following section.

EXPERIMENTAL SETUP

For this experiment, a circular steel bar 12 inches long and 2 inches in diameter was cantilevered from a steel mounting plate as shown in Figure 1. Two PZT stacks aligned with the bar axis were mounted in material cutouts near the bar root. Nominally 2 inches long, 0.75 inches wide, and 0.75 inches deep, the rectangular cutouts were located 0.5 inches from the mounting plate. EDO Corporation Model E200P-5 actuators were selected based on force and stroke capabilities as well as their low price and ready availability. These actuators operate in a -20 to 400V range, have a sensitivity of 0.048 μm/volt, and a stiffness of 100N/μm. A 20 volt DC bias applied to the actuators minimizes the possibility of their exposure to potentially harmful tensile loads.

A force transducer was located in series with the actuators to monitor control force inputs. The assembled bar exhibited good separation between the first and second bending modes in the x-y plane at approximately 214 and 885 Hertz which include the stiffening effects of the actuators. These frequencies are quite low compared to those obtained from a finite element model of a truly cantilevered bar (268 and 2213). The differences, however, have been attributed to the discrepancies in modeling the boundary conditions at the bar-mounting plate interface.

A three axis Endevco accelerometer was placed at the bar tip and a Micro-Measurements strain gage was placed near the bar root. Both signals were tested in feedback control schemes, but the accelerometer proved to be a more reliable measurement of the low amplitude vibrations due to its higher signal to noise ratio. Tip acceleration, root strain, and transducer force signals were monitored and each were tried as feedback signals. The feedback signal was fed into a Microstar DAP3200 board utilizing an on-board 486 based 100MHz processor. Voltage outputs were augmented by Kron Hite amplifiers before being applied to the actuators.

STRUCTURE AND CONTROL COUPLING MODEL

In order to better understand the function of the control system, a pin-force model of the coupling between the axially aligned actuators and the bending motion of the bar is provided. First we will consider the bar's dynamics without the effects of the actuator. Using a normal mode expansion, the displacement in the y direction resulting from lateral bending vibrations can be

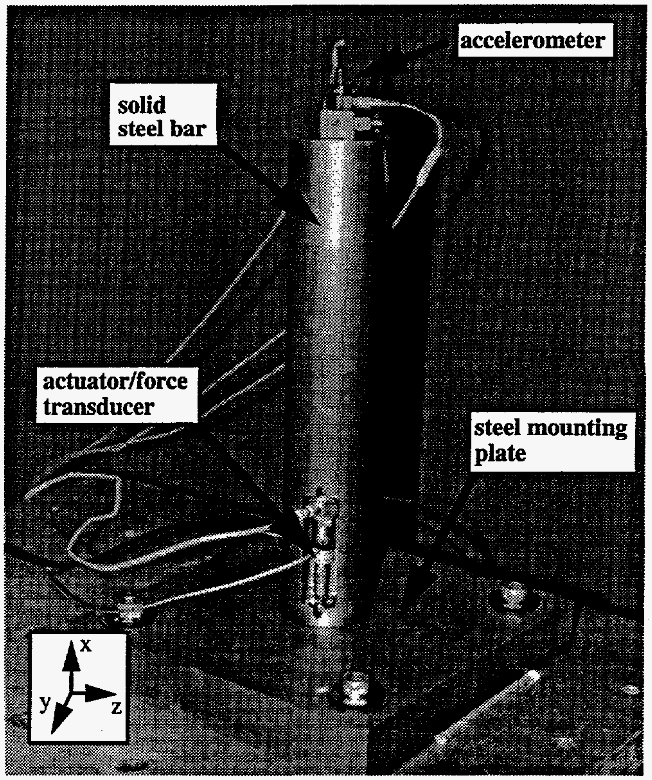

FIGURE 1. - TEST SETUP FOR BAR DAMPING EXPERIMENT.

expressed as

$$u(x,t) = \sum_{r=1}^{\infty} \phi_r(x)\, q_r(t) \qquad (4)$$

in which $\phi_r(x)$ are the mass normalized mode shapes and $q_r(t)$ are the modal displacements. In general, the time dependence of the modal coordinates can be described by

$$\ddot{q}_r(t) + 2\zeta_r\omega_r\dot{q}_r(t) + \omega_r^2 q_r(t) = Q_r(t) \qquad (5)$$

where ζ_r are the modal damping factors arising from the internal structural damping of the bar, ω_r are the undamped natural frequencies, and $Q_r(t)$ are the modal control forces. The modal control forces are the projection of the external forces onto the modal subspace given as

$$Q_r(t) = \int_0^L \phi_r(x) f(x,t)\, dx \qquad (6)$$

where $f(x,t)$ is the force per unit length applied in the y direction. For the system described in the previous section, the modal forces result from a combination of the elastic and the voltage induced deformations of the PZT stack actuator assemblies.

The effect of the actuator is to produce equal and opposite forces (moments) concentrated at the actuator-bar interfaces. Note from Figure 1 that the actuators lie in the x-y plane offset from the x axis. Through differential loading, the actuators affect bending about the z axis as illustrated in Figure 2 with both the actuator stiffness and the converse piezoelectric effect contributing.

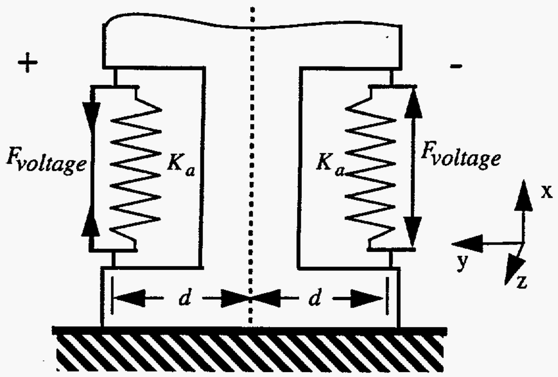

FIGURE 2. - SCHEMATIC DIAGRAM OF ACTUATOR/BAR INTERFACE.

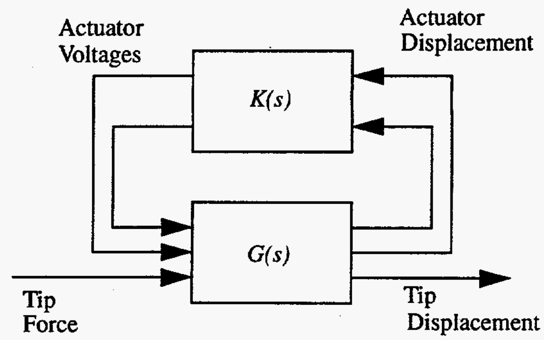

FIGURE 3. - BLOCK DIAGRAM OF CLOSED-LOOP BAR SYSTEM WITH PPF COMPENSATOR.

Neglecting actuator hysteresis and the voltage induced by actuator deformation, the total actuator force is given by

$$F(t) = K_a(\eta V(t) - \Delta L(t)) \quad (7).$$

In equation 7, K_a represents the actuator stiffness, η is the actuator sensitivity, $V(t)$ is the input voltage, and $\Delta L(t)$ is the axial elastic deformation of the actuator resulting from the beam deformation. The force generated by the actuator is not directly applied to the system but is reduced as a result of the compliance of the force transducer and associated mounting hardware. To simplify this discussion, direct coupling between the actuator and the bar is assumed by neglecting the compliance of these additional components.

Assuming that the actuator length is small relative to the modal node spacing in the actuator vicinity, the axial actuator deformations are approximated geometrically by

$$\Delta L_+(t) = d[\theta(x_1, t) - \theta(x_2, t)] \quad (8)$$

and

$$\Delta L_-(t) = d[\theta(x_2, t) - \theta(x_1, t)] \quad (9)$$

in which d represents the actuator offset distance relative to the beam neutral axis. The subscripts + and - refer to the sign of the y coordinate defining each actuator's location. The axial coordinates x_1 and x_2 denote the actuator-bar interface locations as measured from the bar root. The beam slope $\theta(x, t)$ is obtained from the first spatial derivative of the displacement given in equation 4 which leads to the alternate expression for the actuator deformations

$$\Delta L_+(t) = -\Delta L_-(t) = d\left[\sum_{s=1}^{\infty} (\phi_s'(x_1) - \phi_s'(x_2))\, q_s(t)\right] \quad (10).$$

The effect of the total force (elastic and electro-mechanical) supplied by the actuators to the bar can be approximated as moments concentrated at the actuator-bar interfaces (Choe and Baruh, 1992). Since the actuators are intended to supply equal and opposite forces to the system, therefore doubling the effect of a single actuator, we make the simplifying assumption that $F_-(t) = -F_+(t)$. Then, lumping the contributions from both + and - actuators, the moments at the interfaces are given by

$$M(x_1, t) = -M(x_2, t) = 2dF_+(t) \quad (11).$$

The spatial moment distribution is then described as

$$m(x, t) = 2dF_+(t)\,(\delta(x - x_1) - \delta(x - x_2)) \quad (12).$$

Finally, recalling that $f(x, t) = -\partial m(x, t)/\partial x$ (Baruh and Tadikonda, 1991), substitution of equations 10 and 7 into equation 12 and using the result in equation 6 yields the modal control forces as

$$Q_r(t) = 2dK_a[\phi_r'(x_1) - \phi_r'(x_2)] \left[\eta V(t) - d\left[\sum_{s=1}^{\infty} (\phi_s'(x_1) - \phi_s'(x_2))\, q_s(t)\right]\right] \quad (13)$$

which includes both the converse piezoelectric effects and the elastic effects of both actuators. While the piezoelectric effect of the actuators provide control authority through the applied voltage $V(t)$, the elastic component couples the bar's natural modes. An alternative derivation includes the elastic component of the actuator in a modal expansion of equation 4, using a set of system mode shapes and frequencies. This approach eliminates the coupling terms, leaving only the control term in equation 13. However, the approach presented verifies the expected stiffening effects of the actuators.

Note that the modal control forces depend on the actuator offset distance as well as the first spatial derivative of the mode shapes. Therefore, the modal control forces can be maximized by locating the actuators in locations of peak modal strain energy and maximizing the actuator offset distance. However, as seen in equations 8 and 9, the elastic deformations of the actuators are also directly proportional to the offset, and the potential for exceeding the stroke capabilities of the actuator must be carefully considered.

CONTROL DESIGN

A block diagram of the experimental system is shown in Figure 3. The plant transfer function $G(s)$ contains the dynamics of the passive system. For simplicity, the elastic component of the actuator forces are absorbed into the plant. The system is subjected to a disturbance force in the form of a force applied at the bar tip. Analogous to machine tools, this disturbance excites the vibrational modes of the bar which in turn cause oscillatory motion of the tip. As previously described, the objective of this experiment is to augment the damping in the first mode without causing instability in the higher frequency modes. The Positive Position Feedback control strategy provides a useful approach for targeting a specific vibrational mode through active control (Fanson and Caughey, 1990). Collocation of the sensors and actuators is a basic assumption of this approach and guarantees that all control spill-over is stabilizing in the absence of actuator dynamics. As shown in Figure 3, a displacement-like measurement collocated with the actuator is fed through a second order compensator $K(s)$ to produce actuator excitation voltages. Appropriate feedback signals include measurements such as strain

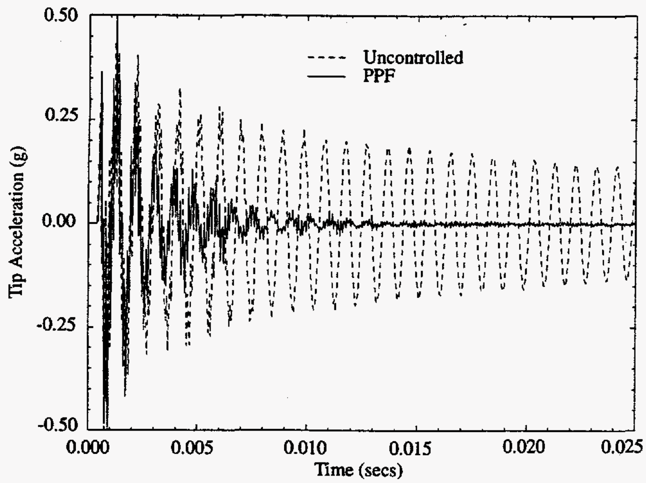

FIGURE 4. - COMPARISON OF TIP IMPULSE RESPONSE ACCELERATION HISTORIES.

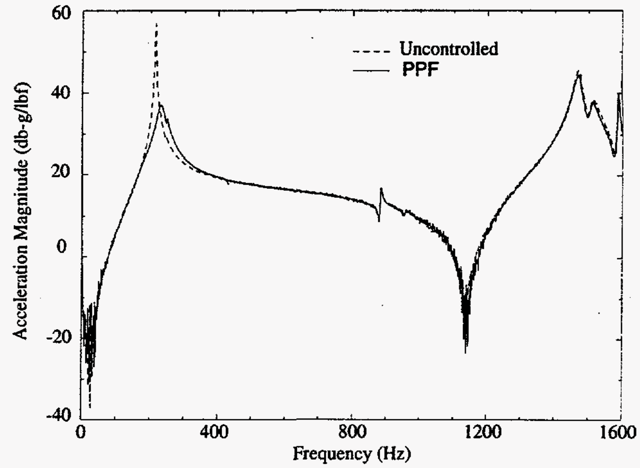

FIGURE 5. - MAGNITUDE OF TIP FORCE TO TIP ACCELERATION TRANSFER FUNCTION.

that are in phase with actuator displacement. Other measurements such as acceleration can be used if the appropriate phase lag is incorporated into the feedback signals. In addition, non-collocation can be handled for a specific mode by accounting for the phase difference between the actuator and sensor locations. However, robustness to control spillover is no longer guaranteed since a non-minimum phase zero is introduced.

Assuming second order dynamics, the general form of the compensator is given as

$$K(s) = \frac{g\omega_f^2}{s^2 + 2\zeta_f\omega_f s + \omega_f^2} \qquad (14).$$

The filter poles are placed in the s-plane to effect a desired migration of the closed loop system poles as the compensator gain is increased. Typically, each local feedback loop is used to target a single mode, but multiple modes can be targeted through judicious placement of the compensator poles.

After much experimentation with a variety of sensors, an accelerometer mounted at the tip of the bar was selected for the feedback signal to damp the first bending mode. Assuming that the mode shapes were characteristic of a cantilevered beam, the accelerometer was essentially collocated with the odd numbered bending modes, but the potential for instability resulting from non-collocation relative to the even numbered bending modes limited the achievable feedback gain. A collocated strain gage mounted adjacent to one of the actuators was tested, but the sensor signal to noise ratio was insufficient to adequately represent the low strain levels (~10µstrain) experienced during impact tests. To effect a migration of the first mode pole further into the left half plane, the undamped filter frequency coincided with the bar frequency at 214 Hertz and the filter damping ratio was set at 0.3. Since, the signal from the tip accelerometer signal was essentially in phase with one actuator displacement and 180 degrees out of phase with the other, one sensor was used to control both actuators. The phase shift was accommodated by flipping the polarity of the actuation drive signal relative to the 20 volt bias. Through trial and error, the filter gain was set at *** which provided reasonable damping of the first mode without introducing higher mode instabilities.

RESULTS

Tip impact tests on the open and closed-loop systems were conducted to judge the effectiveness of the control scheme in damping the first mode. Frequency response functions relating tip force to tip acceleration were determined by averaging 5 impact tests. Typical impulse response functions shown in Figure 4 indicate that both the open and closed-loop histories were dominated by first mode vibration. However, the closed-loop response indicates that a significant increase in first mode damping

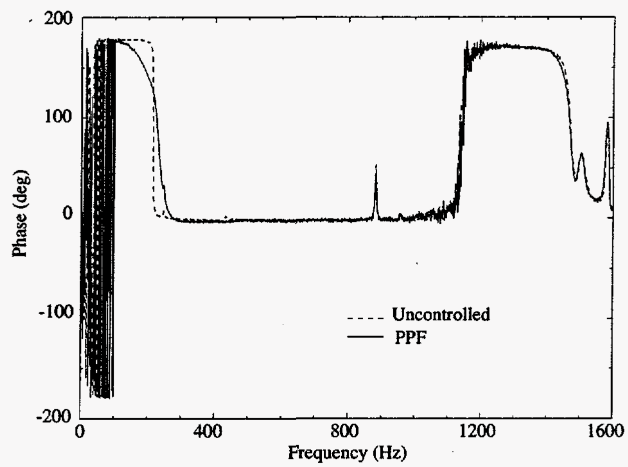

FIGURE 6. - PHASE ANGLE OF TIP FORCE TO TIP ACCELERATION TRANSFER FUNCTION.

was achieved using the PPF control previously described. As anticipated, the higher modes appeared relatively unaffected by the addition of the control scheme as a consequence of the careful selection of the compensator parameters.

Further evidence of the ability of the control system to damp the first mode without altering the higher modes is provided by the frequency response function. As shown in Figure 5, a 20 db reduction in first mode amplitude was achieved with minor differences between the open and closed-loop curves evident throughout the remainder of the frequency range shown. The small peak near 240 Hz corresponds to the first bending mode in the z direction and results from the cross sensitivity of the accelerometer. First mode damping increased from an estimated 2% to 17% of critical as a consequence of the control system. Substitution of these factors into equation 3 indicates that a comparable increase in tool mode damping in metal cutting increases the chatter depth of cut for that mode by a factor of 9.75. Of course, this degree of improvement would not be realized in practice since tool heating and limits defined by other modes would likely be encountered. However, this simple example demonstrates the potential of improving performance using the proposed method.

The phase angle of the frequency response function relating tip impact force to acceleration is shown in Figure 6. Slight differences in the phase were realized at higher frequency as a result of the control spill-over. As previously stated, non-collocation of the actuator and sensor in this experiment produced instability at a sufficiently high gain.

CONCLUDING REMARKS

An experimental demonstration of a vibration control methodology suitable for thick bar and plate-type structural elements exhibiting high frequency low amplitude vibrations has been described. Axial aligned PZT stack actuators mounted in material cutouts were used to augment damping in the first bending mode of a circular steel bar. Test results verify that significant increases in damping were achieved using the proposed method coupled with a Positive Position Feedback control law. This preliminary experiment suggests that the control strategy can be useful for damping augmentation in thick bars and plates which are not conducive to piezoelectric patch and film treatments. In particular, the proposed damping method is being further developed for damping vibrations in a photolithography platen.

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