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TESTING OF AN ACTIVELY DAMPED BORING BAR FEATURING STRUCTURALLY INTEGRATED PZT STACK ACTUATORS*

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This paper summarizes the results of cutting tests performed using an actively damped boring bar to minimize chatter in metal cutting. A commercially available 2 inch diameter boring bar was modified to incorporate PZT stack actuators for controlling tool bending vibrations encountered during metal removal. The extensional motion of the actuators induce bending moments in the host structure through a two-point preloaded mounting scheme. Cutting tests performed at various speeds and depths of cuts on a hardened steel workpiece illustrate the bar's effectiveness toward eliminating chatter vibrations and improving workpiece surface finish.

1. INTRODUCTION

In this paper, results of metal cutting tests using an actively damped boring bar to suppress chatter vibrations are summarized. Chatter is a regenerative vibration that is driven by surface undulations that result from previous tool vibrations recorded on the workpiece surface.[1] For relatively stiff workpieces, chatter immunity is dictated by the by the bar's modal amplification factors,[2] which are a function of the static stiffness and the damping properties of the tool. Consequently, from both a precision and a stability viewpoint, it is desirable to use the a tool with a low over hang ratio (Length/Diameter). Many boring applications, however, necessitate the use of tools with high L/D for which it is difficult to achieve stability in the cutting process due to the inherent flexibility of the tool.

Passive and active vibration absorbers tuned to a specific operating condition can provide relief in some boring operations.[3-5] Previous, cutting tests have indicated that

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the performance of these devices is heavily dependent on the cutting conditions. Some conditions favor a damper orientation normal to the workpiece surface while others favor a tangential orientation. Thus, changing materials or cutting conditions can require removal and disassembly of the cutting head to properly orient the damper. To remedy this situation, an active tool holder capable of damping tool vibrations in two directions simultaneously was subsequently developed, and good performance was demonstrated for a range of cutting conditions on different materials.[5] However, this approach has the added complication of interfacing the active tool holder to the machine, and may require custom interfacing hardware for each machine considered.

As an alternative to the previously mentioned vibration control schemes, the approach adopted in this project is to integrate lead zirconate titanate (PZT) stacks into the bar structure to enhance the damping through feedback control. Since no specialized mounting hardware is required, this technique provides two-axis control without the complications of additional machine interfacing. An added benefit of this approach is realized in actuator placement which is most effective near the tool root, far removed from the cutting zone. The following section summarizes the actuation methodology. A description of the test set-up is given in section 3, followed by a description of some select results in section 4. Finally, some concluding remarks are provided in section 5.

2. INTERFACE MODEL

Controlling bending vibrations in thick bars and plates requires greater control authority than is available from conventional surface mounted actuators such as PZT patches. Instead, we propose to actively damp the boring bar through the use of PZT stack actuators. Mounted in material cutouts near the bar root and offset from the neutral axis, the actuators induce bending moments in the bar through their extensional deformation. To facilitate understanding of the coupling of the actuators into the modal dynamics of the bar, a simple model for the actuator/bar interaction is now developed.[6]

A conceptual representation of a bar with an actuator mounted in a cutout on the surface is shown in Figure 1. The actuator is offset from the bar center line by a distance b, making contact only at the end points denoted z_1 and z_2 as measured from the bar root. The rounded tip pre-loaded mounting scheme assures that the bending stresses in the bar are not transmitted to the actuator stacks. Rather, bending deformation of the bar results in axial deformation of the actuators as a consequence of the offset distance.

Ignoring shear and rotary inertia effects for simplicity, the dynamic displacement of a simple bar in bending is given by

$$u(z, t) = \sum_{r=1}^{\infty} \phi_r(z) q_r(t) \qquad \text{(EQ 1)}$$

in which $\phi_r(z)$ are the mass normalized mode shapes and $q_r(t)$ are the modal displacements for a cantilevered bar. The mode shapes reflect only the dynamics of the structure including the material cutouts intended to accommodate actuators. The modal control forces are the projection of the external forces onto the modal sub-space given by the inner product

$$Q_r(t) = \int_0^L \phi_r(z) f(z, t) dz \qquad \text{(EQ 2)}$$

where $f(z, t)$ is the external force per unit length.

The total actuator force comprised of both active and passive components is given by

$$F_a(t) = K_a(pd_{33}V(t) - \Delta L(t)) \qquad \text{(EQ 3)}$$

in which K_a is the actuator stiffness, p is the number of wafers in the stack, d_{33} is the piezoelectric constant, $V(t)$ is the input voltage, and $\Delta L(t)$ is the axial deformation of the actuator resulting from bar bending deformations of the beam. F_a is positive when a compressive load is present in the actuator. Direct coupling between the actuator and the bar is assumed by neglecting the compliance of the mounting hardware.

Actuator deformations are approximated geometrically by

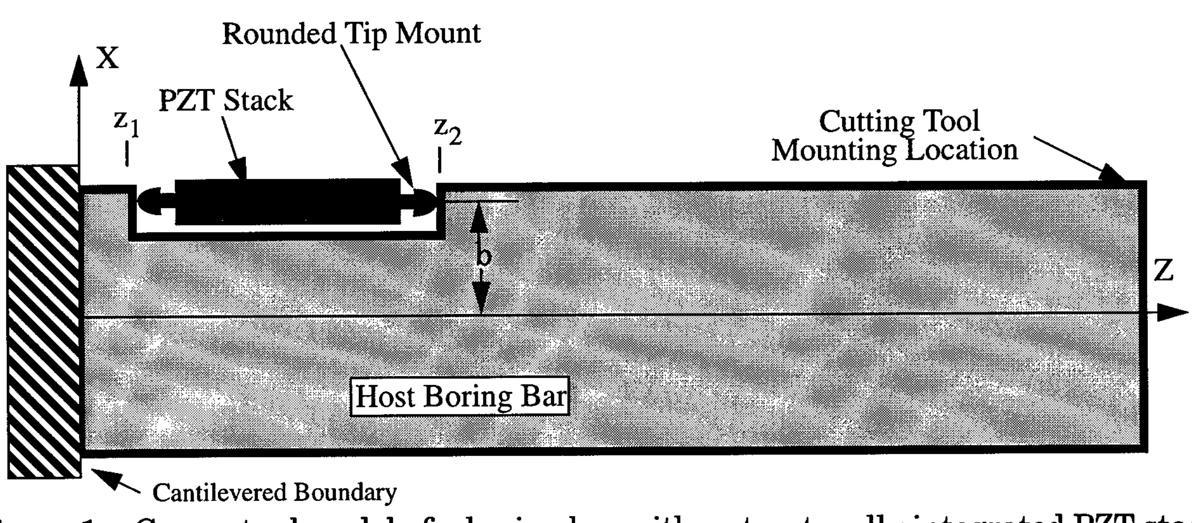

Figure 1. - Conceptual model of a boring bar with a structurally integrated PZT stack actuator located at the bar root.

$$\Delta L(t) = b[\theta_y(z_1, t) - \theta_y(z_2, t)] \tag{EQ 4}$$

in which the beam slope $\theta_y(z, t)$ is obtained from the first spatial derivative of the displacement given in Equation 1. This leads to the alternate expression for the actuator deformations

$$\Delta L(t) = b\left[\sum_{s=1}^{\infty} (\phi_s'(z_1) - \phi_s'(z_2))q_s(t)\right]. \tag{EQ 5}$$

The effect of the actuator force can be approximated as moments concentrated at the actuator-bar interfaces. Accordingly, the moment per unit length is given by

$$m(z, t) = bF_{a+}(t)(\delta(z - z_1) - \delta(z - z_2)) . \tag{EQ 6}$$

in which δ is the Dirac delta function. An alternate expression for the moment distribution in terms of the actuator parameters and the modal deformations is given by substituting Equations 3 and 5 into Equation 6 to yield

$$m(z, t) = bK_a\left[pd_{33}V(t) - b\sum_{s=1}^{\infty} (\phi_s'(z_1) - \phi_s'(z_2))q_s(t)\right](\delta(z - z_1) - \delta(z - z_2)). \tag{EQ 7}$$

Finally, exercising the relation $f(z, t) = -\partial m(z, t)/\partial z$, substituting Equation 7 into Equation 2 and integrating by parts gives the modal control forces as

$$Q_r(t) = bK_a[\phi_r'(z_1) - \phi_r'(z_2)]\left[pd_{33}V(t) - b\sum_{s=1}^{\infty} (\phi_s'(z_1) - \phi_s'(z_2))q_s(t)\right] \tag{EQ 8}$$

which includes both the active and passive effects of the actuator. The control input $V(t)$ is governed by an appropriate feedback control algorithm. Note that the degree of coupling of the control to a particular mode depends on the first spatial derivative of the mode shape at the interface locations. Therefore, the modal control forces can be maximized by placing the actuators in locations of peak modal strain energy.

3 EXPERIMENTAL SET-UP

Cutting tests were performed on a Binns and Barry horizontal lathe located at Sandia National Laboratories. The prototype boring tool, shown during a cut in Figure 2, consisted of a two inch diameter steel shank Valenite bar (model S32-MCLNL6) fitted with a titanium nitride coated Valenite cutting insert (model CNMP643 SV4). All cutting tests were performed with a bar L/D of 6 on a rigid 4340 steel workpiece that had been heat treated to a hardness of 42 Rockwell 'C' scale.

The bar was modified to incorporate Physik Instrumente PZT stack actuators (model 840.1) near the bar root as shown in Figure 3. The actuators were mounted using mechanical preload, with the rounded actuator tips inserted into the concave pocket ends. Two actuators were mounted on the lower and aft sections of the bar to suppress bending vibrations tangential and normal to the workpiece surface, respectively. Tip mounted Endevco accelerometers (model 22) provided the input to the two independent rate feedback control algorithms running on the dSpace controller. Performance was monitored using a Hewlett Packard 3565s system operating at 4096 samples per second.

A comparison of the normal and tangential frequency response functions is given in Figure 4 and 5, respectively. A double pole is evident in the vicinity of 330 Hz, indicating the presence of the first bending mode in both the normal and tangential directions. A 10-15 db reduction in the resonant peaks result from the application of the Active Vibration Control (AVC). The estimated first mode damping ratio in the tangential

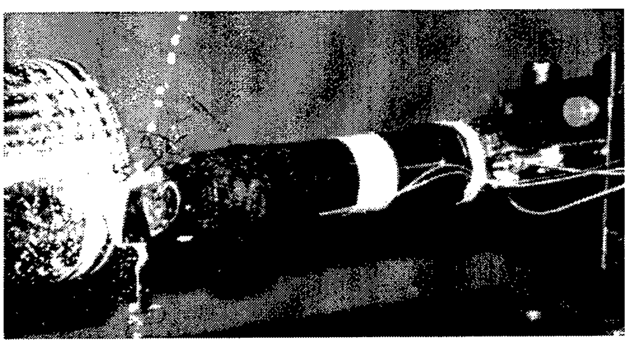

Figure 2 - Prototype Valenite Boring Bar with L/D of 6.

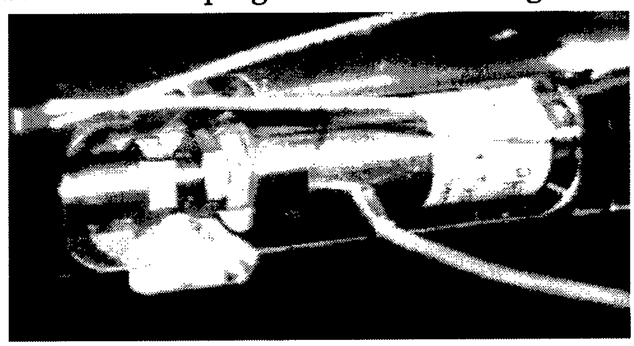

Figure 3 - Physik Instrumente Actuator Mounted in Cutout Near the Bar Root.

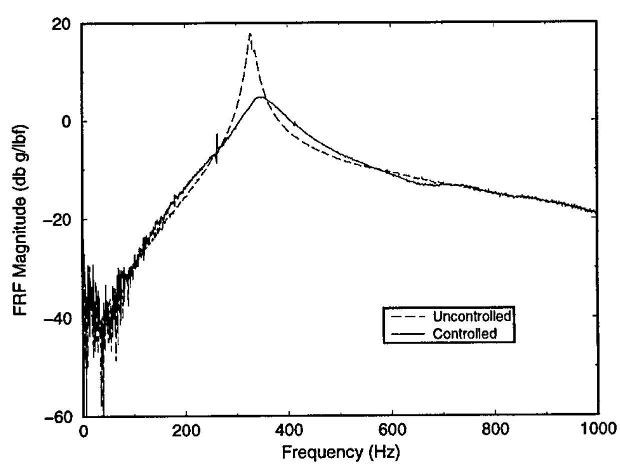

Figure 4 - Tangential Tip Driving Point FRF Magnitude for L/D=6.

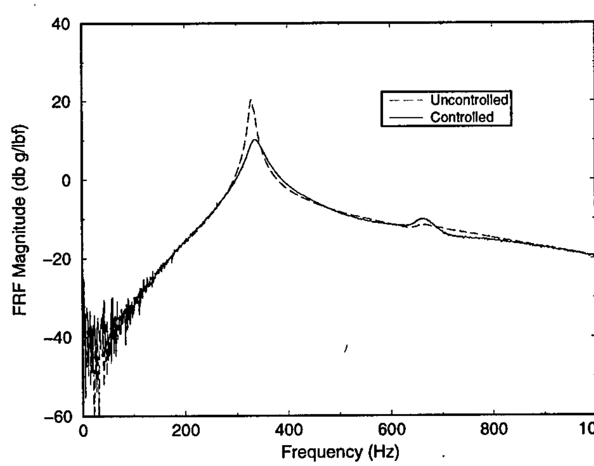

Figure 5 - Normal Tip Driving Point FRF Magnitude for L/D=6.

direction moved from 2.1% to 9.0% critical while an increase from 1.5% to 4.0% was realized in the normal direction.

4. CUTTING TESTS

In this section, two different cutting conditions are examined to illustrate the difference in bar vibration between the control-on and control-off cases. In the absence of AVC, the data clearly indicates the existence of chatter for both cases considered. Chatter is eliminated when the loop is closed, with the residual vibration amplitude dependent on the cutting conditions. Note that although these examples provide insight into how the AVC influences behavior, they do not adequately reflect the potential of the AVC. For some cutting conditions, AVC may be unable to eliminate chatter once its initiated because the actuator stroke capability can be exceeded. However, under the same cutting conditions, the presence of AVC from the start of the cut may prevent chatter from developing at all. Therefore, the successful chatter prevention envelope under AVC is actually much larger than the chatter elimination envelope. As a result, a more comprehensive assessment of system performance is being conducted, and results will be described in subsequent communications.

4.1. Depth of Cut 0.010 inches

With the cut depth set at 0.010 inches and the AVC off, the cutting speed and feed rate were adjusted until a heavy chatter was sustained at 127 rpm and 0.0045 inches/rev. Acceleration measurements were recorded for 10 seconds prior to activating the AVC, after which 10 more seconds of data were recorded. Power spectra for these conditions are shown in Figures 6 and 7. While the existence of the strong resonant peak and several harmonics for the uncontrolled case give strong evidence of chatter, the absence of peaks in the presence of AVC indicate that the chatter was completely eliminated. The vibrational energy at the resonant peak for this case was reduced by nearly 60 db. Consequently, the estimated displacement portrait given in Figure 8 shows that the tool tip displacements were dramatically reduced. The surface finish improved from 140 μinches for the uncontrolled case to 54 μinches in the presence of AVC.

4.2 Depth of Cut 0.020 inches

As shown in Figures 9 and 10, chatter was established for a 0.02 inch cut depth with a workpiece speed of 79 rpm and feed rate of 0.003 inches/rev. The chatter was again eliminated upon activation of the AVC, although the resonant peak dominates the

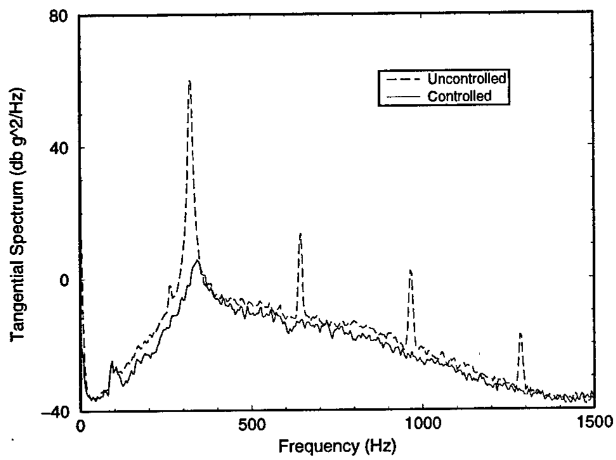

Figure 6. - Tangential Acceleration Spectrum for 10 mil Depth of Cut.

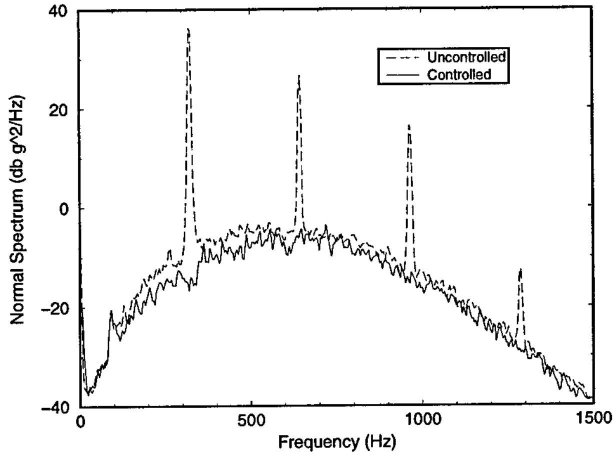

Figure 7. - Normal Acceleration Spectrum for 10 mil Depth of Cut.

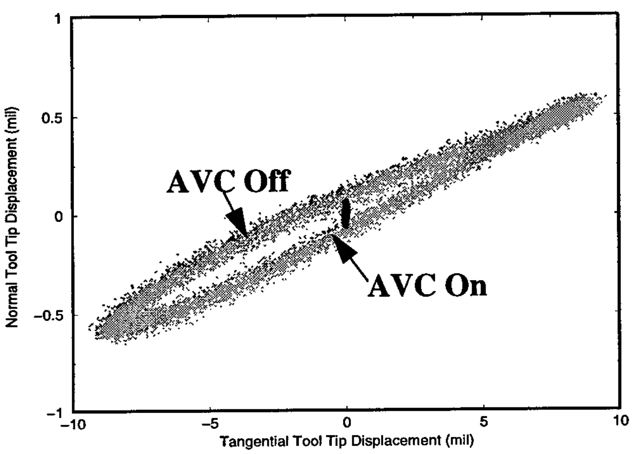

Figure 8. - Tool Tip Displacement Portrait for 10 mil Depth of Cut.

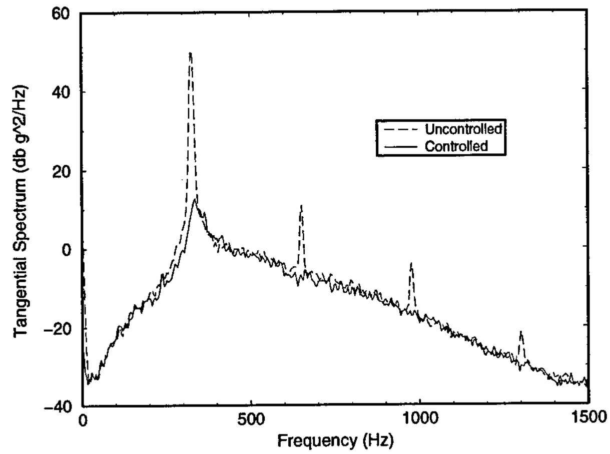

Figure 9. - Tangential Acceleration Spectrum for 20 mil Depth of Cut.

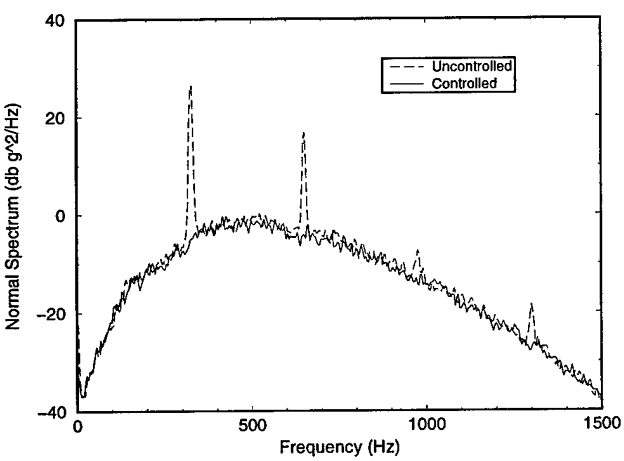

Figure 10. - Normal Acceleration Spectrum for 20 mil Depth of Cut.

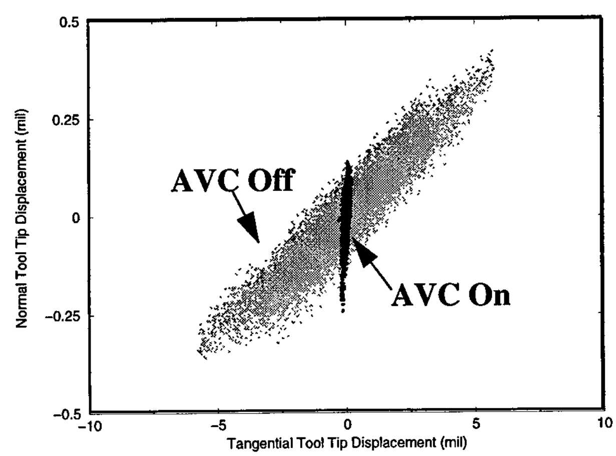

Figure 11. - Tool Tip Displacement Portrait for 20 mil Depth of Cut.

residual vibration. The surface roughness for this example was reduced from 167 μinches to 85 μinches as a consequence of the reduction in amplitude shown in Figure 11.

5. CONCLUDING REMARKS

A technique for actively suppressing regenerative chatter vibration in boring has been described. Piezoelectric stack actuators were integrated into the body of a commercially available boring bar to provide vibration control both normal and tangential to the workpiece surface. Unlike previous approaches featuring active tool holders, the structurally integrated actuators enable biaxial control while utilizing conventional tool mounting hardware. Axial strain induced in the actuators through application of closed-loop control produces bending moments in the boring bar to counter cutting induced vibrations. The efficacy of this control approach was demonstrated in a series of cutting tests performed on a hardened steel workpiece. For the case of a bar with L/D of 6, rate feedback control yielded dramatic reductions in tool vibrations for moderate metal removal rates. Workpiece surface roughness was reduced by a factor of three in many cases considered. With relatively high metal removal rates, however, the stroke capacity of the tangential actuator was exceeded due to large deformations of the tool. Thus the ability of the system to eliminate sustained chatter was compromised by high removal rates.

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