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Title: VIBRATION SUPPRESSION IN CUTTING TOOLS USING COLLOCATED PIEZOELECTRIC SENSORS/ACTUATORS WITH AN ADAPTIVE CONTROL ALGORITHM

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ABSTRACT

The machining process is very important in many engineering applications. In high precision machining, surface finish is strongly correlated with vibrations and the dynamic interactions between the part and the cutting tool. Parameters affecting these vibrations and dynamic interactions, such as spindle speed, cut depth, feed rate, and the part's material properties can vary in real-time, resulting in unexpected or undesirable effects on the surface finish of the machining product. The focus of this research is the development of an improved machining process through the use of active vibration damping. The tool holder employs a high bandwidth piezoelectric actuator with an adaptive positive position feedback control algorithm for vibration and chatter suppression. In addition, instead of using external sensors, the proposed approach investigates the use of a collocated piezoelectric sensor for measuring the dynamic responses from machining processes. The performance of this method is evaluated by comparing the surface finishes obtained with active vibration control versus baseline uncontrolled cuts. Considerable improvement in surface finish (up to 50%) was observed for applications in modern day machining.

INTRODUCTION

Unwanted vibration in precision manufacturing machinery is a major source of poor surface finish, increased tool wear or failure, and poor dimensional accuracy. There are several sources of vibrations in machining processes, such as self-regenerated vibration within a process (referred to as chatter), rigid body motion of the machine caused by the internal processes including spindle rotations or the dynamic cutting process, and externally induced vibration from the foundation or acoustic coupling. These vibrations are usually smaller in magnitude and higher in frequency than those typically found in other mechanical systems. Numerous vibration reduction techniques have been therefore proposed in the past relying on vibration-awareness design or changing machining parameters during operation.

An alternative, modern technique to address these vibration issues is the use of active or passive damping treatments. The most popular approach is to suppress the unwanted vibration with a control algorithm and an actuator which typically employs active materials, such as piezoelectric elements. Piezoelectric materials experience an elastic strain when exposed to an electric field and are excellent candidates for vibration control devices. These materials have been increasingly used to reduce the vibrations in both a passive and an active manner. In machining processes, piezoelectric materials are usually used in the form of a stack configuration acting as an actuator for the implementation of a fast tool servo (FTS) for improving surface quality in manufacturing machinery. FTS refers to auxiliary servos that are specially adopted to control a fast-acting tool holder with high resolution and fast dynamic response for ultra-precision turning or facing operations. Piezoelectric materials are very suitable for FTSs because of their high frequency bandwidth (up to kHz ranges), high dynamic stiffness (greater than 100 N/μm), compact size, no backlash, and the capability of producing relatively high force (greater than 10 kN ranges). These devices have been successfully used for active vibration reduction, dynamic machine error compensation, and axis-asymmetric machining, resulting in improved surface integrity and increased geometrical accuracy. The application of piezoelectric materials for the FTS design is well summarized by Park et al. [1]. A review of some of the more noteworthy efforts follows.

Research at North Carolina State University set the foundation for using the piezo-stack actuator for diamond turning of micro-surface applications [2,3,4]. Their FTS design utilized a piezoelectric ring-type stack actuator and a pair of high-bandwidth capacitance sensors for displacement measurements. With the implementation of a PI control algorithm, the results of the FTS on a parallel axis ultraprecison lathe indicated that it could actively correct thermally induced spindle error motions, maintain the error of the tool position with respect to the reference signal to within 2% of the input amplitude, and reduce the surface roughness. Follow-on work from this group produced several prototypes of the FTS and used by other researchers as a base design.

Kim and Kim [5] and Kim et al [6] designed a piezoelectric stack actuator-based FTS, which improves tracking performance up to 0.15 μm peak-to-peak error level. Crudele and Kurfess [7] published a design that integrates a piezo-based FTS servo with repetitive control for facing applications. In repetitive control, a controller is designed to identify certain periodic patterns in a process. Based on these patterns, the controller's output is a function of the expected pattern coupled with any errors from the previous cycles. This technique is particularly applicable to machining where the material and tool conditions change relatively slowly and the process is relatively repeatable. The benefit of the repetitive controller was the tracking ability of surface waviness. A substantial reduction in waviness, up to 62%, was observed.

The piezoelectric-based FTSs have seen increasing use in micro-machining applications and are considered a fairly mature technology with some commercial realization. While the traditional application of piezo-based FTS has focused on diamond turning applications which requires relatively small chip loads and small cutting force disturbances, Zhu et al [8] and Woronko et al [9] addressed the use of a piezo-based fast tool servo for precision shaft machining in conventional CNC turning machines. They employed an adaptive sliding-mode controller to compensate for uncertainties due to cutting disturbances and hysteresis in the stack actuator. The authors' motivation was to provide rough, semi-finish, and ultra-precision cutting using a single conventional CNC machine. The rough and semi-finish operations were performed on a tool with a conventional CNC machine and the ultra-precision cutting was accomplished by the same machine with a piezo-based FTS on the CNC turret. A significant improvement on the surface quality was obtained that cannot be attainable with traditional CNC machine.

The use of a piezoelectric actuator for conventional machine tool vibration reduction has also been substantially investigated in the past via active and passive damping treatments. Fung and Yang [10] used a forecasting control technique with a two-dimensional piezo-actuated tool motion system to compensate for spindle error motion. Pan and Su [11] used a piezoelectric actuator mounted on to a tool holder for chatter suppression during turning. The piezoelectric actuator regulates the tool displacement with a robust adaptive controller which accounts for the hysteretic nonlinearity and results in a significant reduction in chatter. Lee et al [12] used a piezoelectric inertia actuator acting as a passive tuned vibration absorber to suppress chatter in turning operations. They demonstrated that the cutting stability is increased six times using this approach.

In this paper a new piezoelectric actuator-embedded cutting device is coupled with an adaptive control algorithm to achieve improved surface finish in the presence of static and dynamic deformations of the tool assembly, thermal displacements, and unknown process variations. When using piezoelectric devices the implementation of efficient control strategies becomes a significant factor in obtaining desirable surface finishes. In the machining process, the parameters affecting control performance can vary in real-time and could adversely affect the quality of finished products. Large temperature changes, uncertainties in the dynamic interaction between part and cutting tool, and unknown process variations have the potential to alter the machining dynamics, and therefore detune an active controller. To account for this variability, the control strategy must be robust enough to account for unexpected changes while providing sufficient control performance, or it must be designed specifically to adapt to, and account for, system changes. In this study we used a modified version of the controller proposed by Creasy et al. [**Error! Bookmark not defined.**,**Error! Bookmark not defined.**], developing an adaptive positive position feedback (PPF) filter in series with a low-pass Butterworth filter to actively absorb unwanted vibration in the machine tool. The adaptation scheme used in this study relies on a self-tuning regulator (STR) to track system changes through the feedback signal and update the control parameters in real-time accordingly.

In many control applications, the precise measurement of displacement or velocity is critical. When piezoelectric actuators are incorporated in the machining process, the most common feedback device is a capacitive gap sensor with nanometer resolution. The use of laser interferometers [9,13,14], high-resolution strain gauges [15], eddy current probes [16] are also reported. However, these devices are rather expensive, bulky, and the performance of such sensitive devices is not guaranteed under the harsh manufacturing environmental and operational conditions. In order to overcome such limitations, we proposed to use low-cost active material-based sensing systems that perform comparable to those of traditional sensors typically used in manufacturing industry. We utilize the sensing capabilities of the piezoelectric stacks, which is collocated at the actuator, therefore forming a self-sensing actuator. The stack can directly and reliably measure the dynamic response of the tool assembly as a sensing element, allowing for the compact and self-contained design of the proposed active tool holder. This proposed piezoelectric-driven adaptive vibration suppression system has shown to substantially improve the surface finish in turning operations by actively suppressing unwanted vibration and chatter. The theory behind these methods, experimental setup and results are described in the following sections.

ACTIVE TOOL HOLDER DESIGN

A schematic of the active tool holder is shown in Figure 1. The tool holder consists of the cutting tool piece, tool holder assembly, diaphragm, ring-type piezoelectric stack actuator and supporting shell structure. A high voltage piezoelectric ring-type stack actuator is housed under preload between diaphragm and a rear endcap by a steel preload rod that slips through the center of the stack actuator. This piezoelectric ring actuator was manufactured by Peizomechanik, Inc., and has 25 mm outside-, 15 mm inside-diameter, 27 mm length, 400 N/μm stiffness, and maximum stroke of 25 μm. Displacement of the stack is directly transmitted to the tool assembly at higher than several kHz upon the command signal provided through a high voltage amplifier. The overall size of this active tool holder is 75 mm diameter with 90 mm long, with the tool extending from the center of the holder.

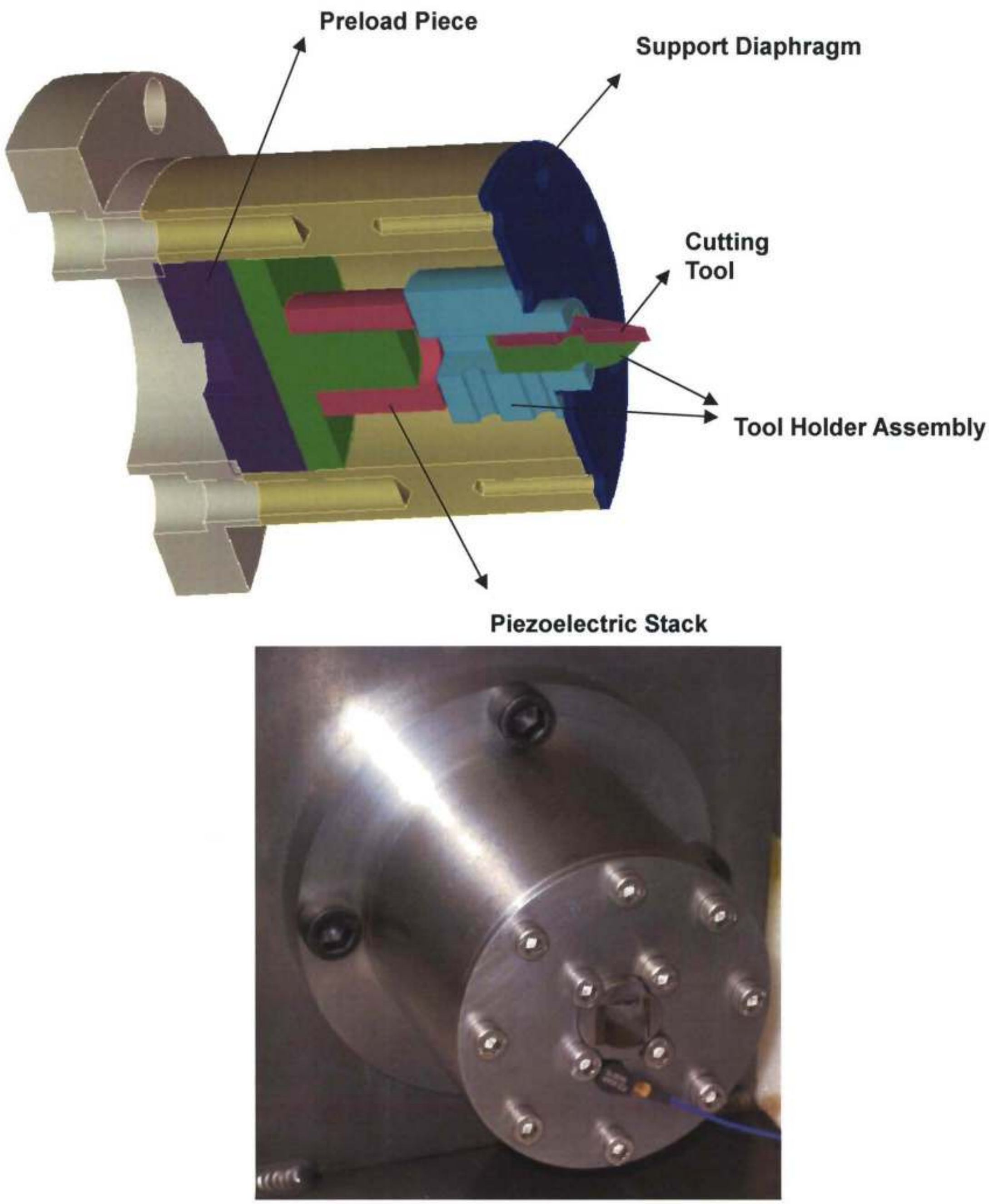

Figure 1. The active tool holder

As can be seen in the Figure 2, the ring stack actuator is equipped with a sensing element that directly measures the dynamic strain that the tool holder experiences. The actuator has four wire leads. The red(+) and back(-) leads are connected to the actuator and the light-blue terminals are the sensor output connections. This configuration eliminates the need of external gap sensors for control, making the entire design compact and self-contained for ease of integration with conventional lathe assemblies. Furthermore, the collocation of sensing and actuation allows the control signal to be applied at the point of measured response, thereby eliminating the capacitive coupling between the sensor and actuator elements [17]. Goh and Coughley [18] presented results which demonstrate that, in the absence of finite actuator dynamics, structures controlled with collocated velocity feedback are unconditionally stable at all frequencies.

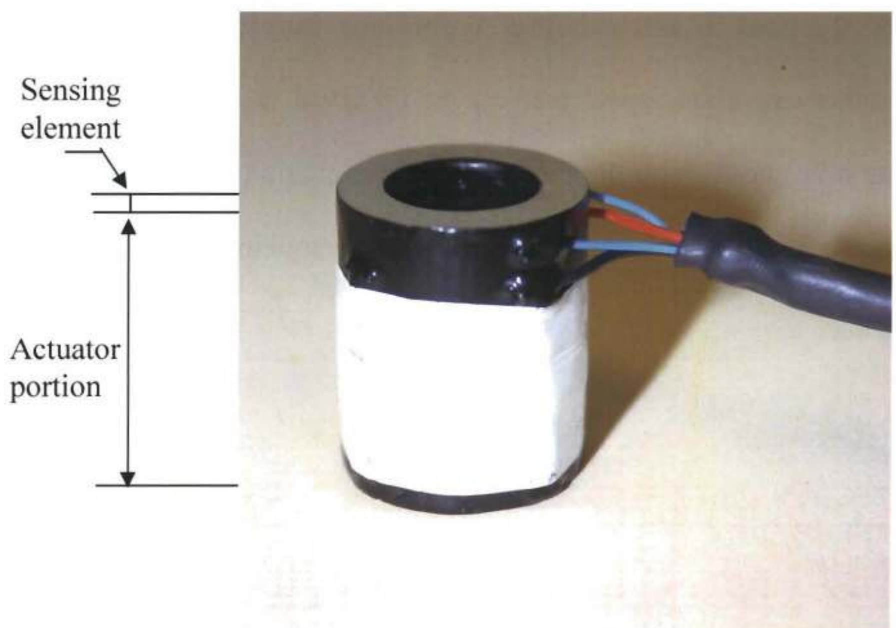

Figure 2. The ring-type piezoelectric Stack actuator for used in the active tool holder

The sensitivity of the collocated piezoelectric sensor was investigated by comparing it with that of an external sensor. The sensitivity was tested by applying a 500 Hz sinusoidal voltage in increasing increments to the fully assembled active tool holder via an amplifier. A laser vibrometer sensor was used to measure the displacement of the tool tip. The sensing voltage from the collocated piezoelectric sensor was also measured using a DACTRON data acquisition system. The voltage was increased in increments of 100 V until reaching 500 V. The experimental results are shown in Figure 3. As can be seen in the figure, the slope of the piezoelectric and vibrometer responses are essentially the same. Around 500 V excitation, a slight change in the slope from the piezoelectric sensor was observed. Therefore, under the proposed design scheme, it can be concluded that the collocated piezoelectric sensors could produce a reliable sensing voltage for actuation signals up to 400 V (up to 10 μm output displacement). As stated, the use of a piezoelectric transducer as a collocated sensing element will eliminate the need of externals gap sensors that were traditionally used in the previous studies. Another important aspect that can be observed in this figure is that the active tool holder actuation is linear. With the increase in the excitation voltage, there is a corresponding and linear increase in the measurement amplitude by looking at the response from the laser vibrometer.

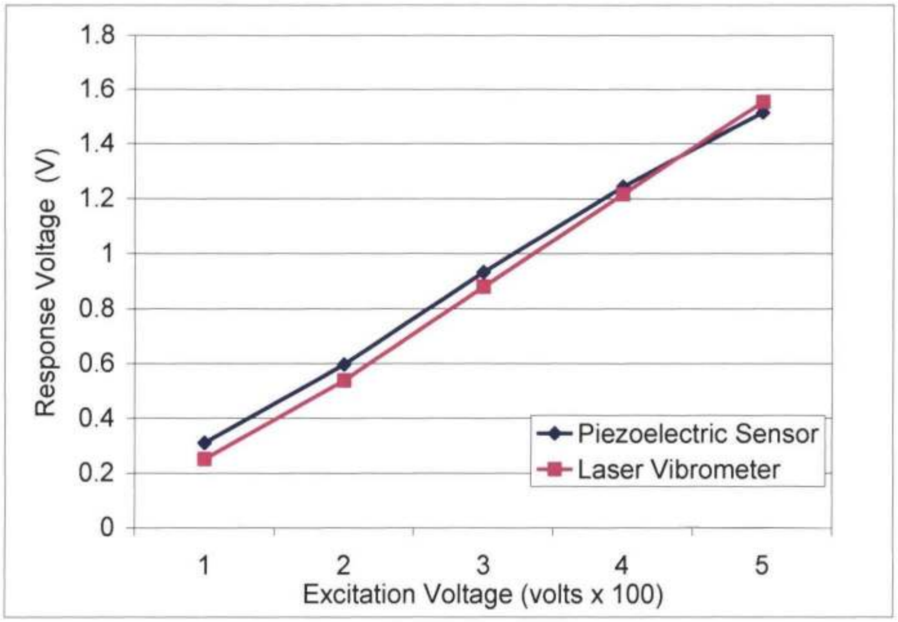

Figure 3. Piezoelectric sensor vs. laser vibrometer response at 500 Hz excitation.

In experimental vibration testing, the first mode of this tool holder was identified 3.7 kHz. A finite element model of this tool holder was also constructed to better understand the dynamics of this active tool holder, and the analytically predicted modal frequencies were all within 10% of the experimentally identified modal frequencies [19].

ADAPTIVE POSITIVE POSITION FEEDBACK CONTROL

When using an active control technique to obtain high-quality surface finishes, there must be a robust control strategy in place to compensate for unknown disturbances and process variations. An adaptive positive position based feedback controller was therefore used in this study for active vibration control. This form of control was chose because it has been used successfully in vibration and acoustic reduction applications, and can be used with frequency tracking algorithms to dynamically adapt to changes in the system such as varying operational and environmental conditions that often occurred during a machining operation.

The concept of positive position feedback was proposed by Fanson and Caughey [20] who began by first defining a second-order expression for the modal coordinate of the system,

$$\ddot{q} + 2\zeta\omega\,\dot{q} + \omega^2 q = g\omega^2\eta \quad , \tag{1}$$

along with the second order equation for the filter coordinate,

$$\ddot{\eta} + 2\zeta_f\omega_f\dot{\eta} \; + \omega_f^2\eta_i = \omega_f^2 q_i(t) \,. \tag{2}$$

Thus, a transfer function of this PPF filter takes the form

$$K_{ppf}(s) = \frac{g\omega_f^2}{s^2 + 2\zeta_f\omega_f s + \omega_f^2} \tag{3}$$

where g is the gain of the transfer function, ω_f is the filter frequency, and ξ_f is the filter damping ratio. Equation (3) shows that the filter adds two poles to the system for each PPF filter used.

PPF control works to suppress unwanted vibrations by driving the actuator with a highly damped signal that is in-phase with the disturbance signal. This allows the actuator to couple with the disturbance, increasing the effective damping within the system. This serves to significantly reduce the relative motion between the tool and workpiece, equalizing the interaction force between each and thus reducing the amplitude of vibration. This effect can be easily conceived for a basic turning operation on bar-stock with a long length to diameter ratio (length/diameter > 3), where the workpiece usually has more torsional rigidity than bending rigidity. Therefore, if cutting inconsistencies occur, the workpiece could begin to chatter, entering into a self-excited vibration that significantly reduces the structural damping between the workpiece and the cutting tool. By using a PPF based controller, the active tool holder would increase the effective damping of the system, preventing chatter and improving surface finish quality.

Figure 4 shows a schematic of the adaptive, self-tuning regulator PPF controller used in this research, which was previously developed by Creasy et al [**Error! Bookmark not defined.,Error! Bookmark not defined.**]. For the sake of completeness, the adaptive PPF algorithm is briefly described here.

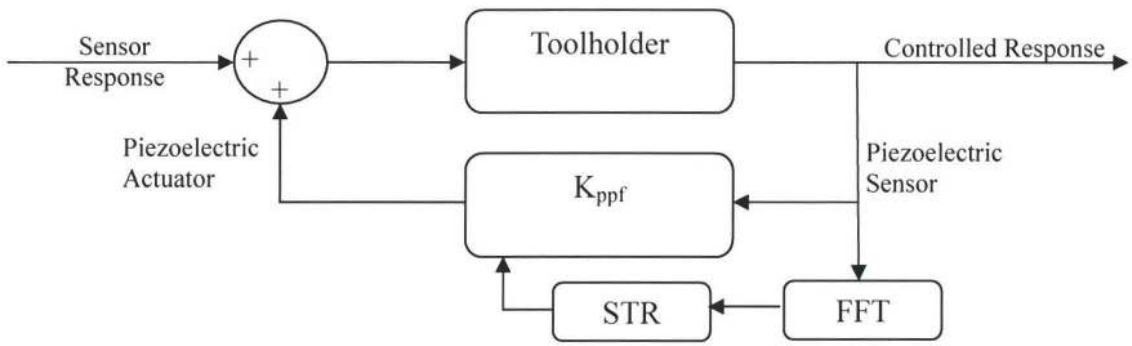

Figure 4. Adaptive Positive Position Feedback Control

The adaptive portion of the controller tracks the resonant frequency of the structure and dynamically tunes its transfer function to this frequency. The adaptive PPF filter is described as having two loops, (1) the conventional controller loop with variable parameters and (2) a self-tuning algorithm loop that identifies the system and changes the variable PPF parameters of the controller in the first loop. The conventional controller loop contains the K_{ppf} block with a parameter varying transfer function. The parameter variations can be the gain, damping, or resonance frequency of the positive position feedback filter and can be updated in real time. The second loop determines the varying parameters for the filters. The time domain data of the collocated sensor signal is first transformed to frequency data. A small buffer is used to accumulate the collocated measurements to perform the fast Fourier transform (FFT) of the data in real time. The magnitude of this FFT data is sent to an algorithm that searches for the minimum magnitude and their associated frequency values which coincide with the zeros of the system. The frequency values of the zeros are used to update the varying parameters of the transfer functions, as explained in [**Error! Bookmark not defined.**,**Error! Bookmark not defined.**].

Before implementing this adaptive PPF filter to the active holder, a simple beam experiment was performed in order to show the effectiveness of this control algorithm in the face of changing operational conditions as would be expected during machining. The test setup is comprised of a polycarbonate beam (43 x 300 x 2 mm) with two piezoelectric patches attached to the beam, as shown in Figure 5. This beam allowed for inspection of controller performance and easy tuning of the PPF filter. The beam was controlled with a MATLAB Simulink model, which was ported to an xPC Target real-time system. A National Instruments data acquisition system and a commercially-available inverting amplifier were used to control the actuator and measure the sensor response. The piezoelectric patch (30 x 30 x 0.2 mm) on the left in Figure 5 was used as sensor while the larger patch (30 x 72 x 0.2 mm) on the right was used as an actuator.

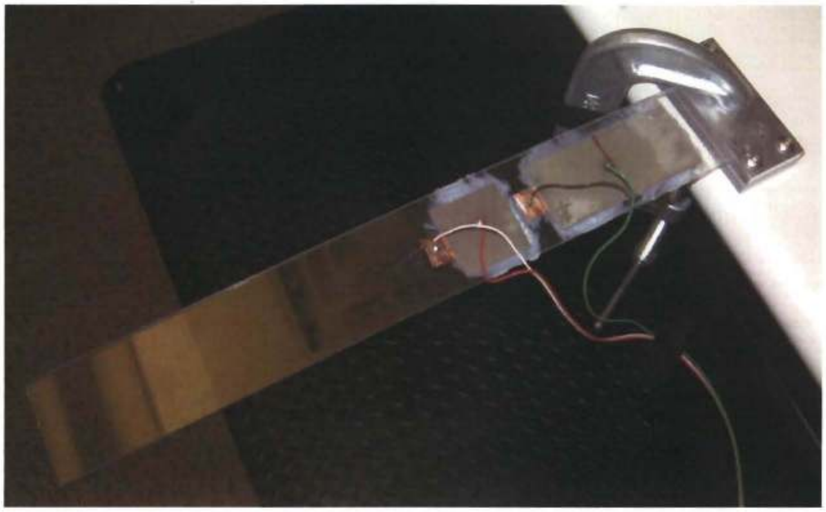

Figure 5. A snap shot of test setup with a cantilever beam

The cantilevered beam itself was subjected to an initial condition excitation. The free tip of the beam was displaced by 5 mm and released, allowing the cantilever to vibrate freely. The system was first evaluated without control, then with control applied. The controller was tuned by adjusting the controller gain, damping ratio, and filter frequency to obtain optimum performance. The resonant frequency of the first bending mode of the beam was measured using a DACTRON data acquisition board and the RT Pro modal analysis software. The measured first resonant frequency of the beam was 6 Hz. By following the procedure outlined in [**Error! Bookmark not defined.**] and with slight adjustments, the optimal values were identified as g=1.5, ξ_f=0.4 and $\omega_f = 0.9*\omega_n$ for the controller.

Implementing the self-tuning regulator algorithm in Simulink required modeling the control transfer function discretely. The controller was digitized by performing a z-transform on the continuous transfer function $K(s)$. Tustin's approximation, shown in Equation (4), was used for finding the discrete-domain form of the Laplace-domain PPF controller function.

$$s = \frac{2}{T}\frac{(z-1)}{(z+1)} \tag{4}$$

where T is the sampling period of the control system. By substituting equation (4) into equation (3), the following z-domain transfer function is obtained,

$$H(z) = \frac{\left(\frac{\omega_f^2}{d}\right)z^2 + \left(\frac{2\omega_f^2}{d}\right)z + \left(\frac{\omega_f^2}{d}\right)}{z^2 + \left(\frac{\frac{-8}{T^2} + 2\omega_f^2}{d}\right)z + \left(\frac{\frac{4}{T^2} - \frac{4\xi_f\omega_f}{T} + \omega_f^2}{d}\right)} \tag{5}$$

where $$d = \frac{4}{T^2} + \frac{4\xi_f\omega_f}{T} + \omega_f^2$$

The adaptive portion of the controller in Simulink was implemented by taking the FFT of the input signal, writing a custom script to detect the first resonant mode in frequency domain, and then dynamically updating the transfer function with the new frequency value. To ensure controller stability, the maximum filter frequency change between subsequent FFT runs was set at 10% of the previous frequency. The performance of the adaptive controller was tested by changing the mass of the beam by adding a large washer to the free end. The non-adaptive controller was tuned to operate at 6 Hz (the natural frequency of the polycarbonate beam without additional mass) as previously described. With the added mass, the beam's natural frequency decreased to 3 Hz, a 50% change.

The experimental results from both the conventional and the adaptive PPF filter is illustrated in

Figure 6. The first response shows the free response of the beam without implementing the control. The second figure shows the response using the conventional PPF filter. There is certain reduction in settling time, but the performance is deteriorated as the mass loading detuned this PPF filter. The third plot shows the performance improvement of the adaptive version of the controller over the non-adaptive controller. The adaptive vibration controller showed an improvement over the non-adaptive case by demonstrating its ability to dynamically and adaptively tune itself to the changes in the system.

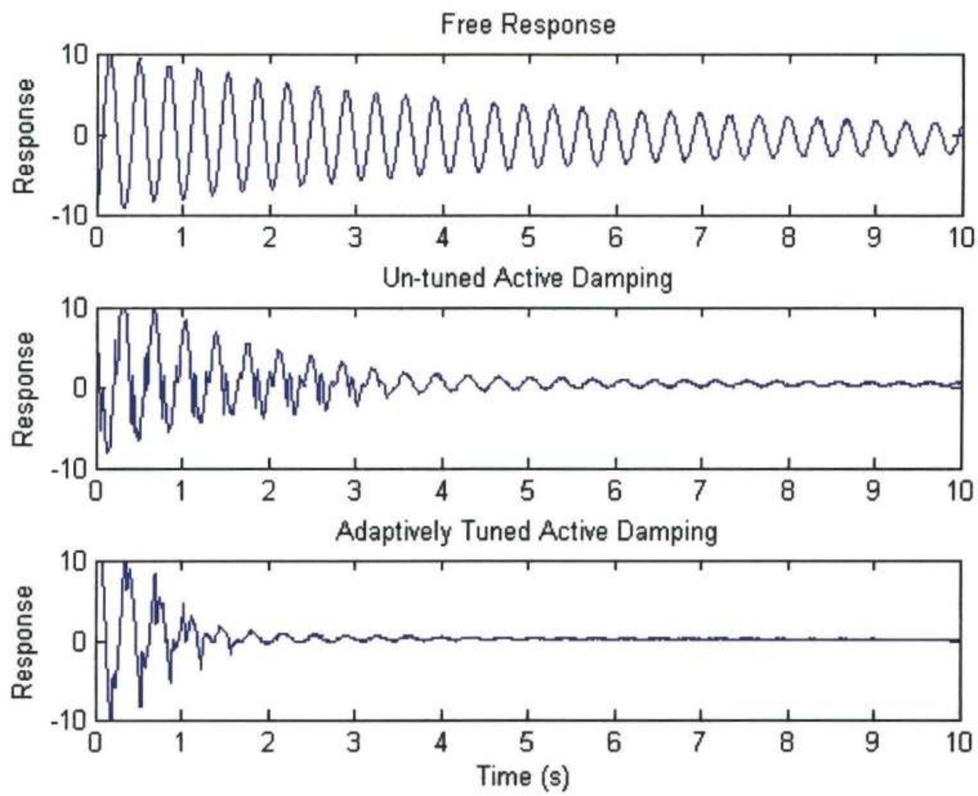

Figure 6. System response for Adaptive PPF control with added mass. (Top) Free vibration case. (Center) Static PPF control tuned to 6 Hz. (Bottom) Adaptive PPF control.

Damping ratios, estimated using log-decrement calculations on the time responses from Figure 6, are presented in Table 1. These damping ratios provide a quantitative comparison of the adaptive versus non-adaptive controller for the test case. Comparing these ratios, it can be seen that the adaptive system provides a substantial improvement (more than 100%) over non-adaptive control when the system mass or stiffness is perturbed.

Table 1. Estimated damping ratios for low-power control experiments.

Test	1st Bending Mode	Damping Ratio
Free vibration		1.2%
Static PPF control tuned at 6 Hz	3 Hz	3.2%
Adaptive PPF control		7.4%

EXPERIMENTAL IMPLEMENTATION WITH THE ACTIVE TOOL HOLDER

The active tool holder was tested with the same hardware and software setup; the only alteration was that the polycarbonate beam was replaced by the active tool holder. Testing the vibration reduction tool holder was first done by exciting the tip of the cutting blade with an impact hammer and inspecting the time response of the holder as measured from the collocated piezoelectric sensor, while the control force was applied by the piezoelectric actuator.

Tuning the controller for the tool holder was similar to tuning the controller for the beam. The target frequency of FFT was 3.7 kHz, which is the first resonant frequency of the holder. Figure 7 provides the time response before and after the adaptive PPF algorithm was applied. As can be seen in the figure, the implementation of the control algorithm provides a substantial reduction in settling time, up to 85%.

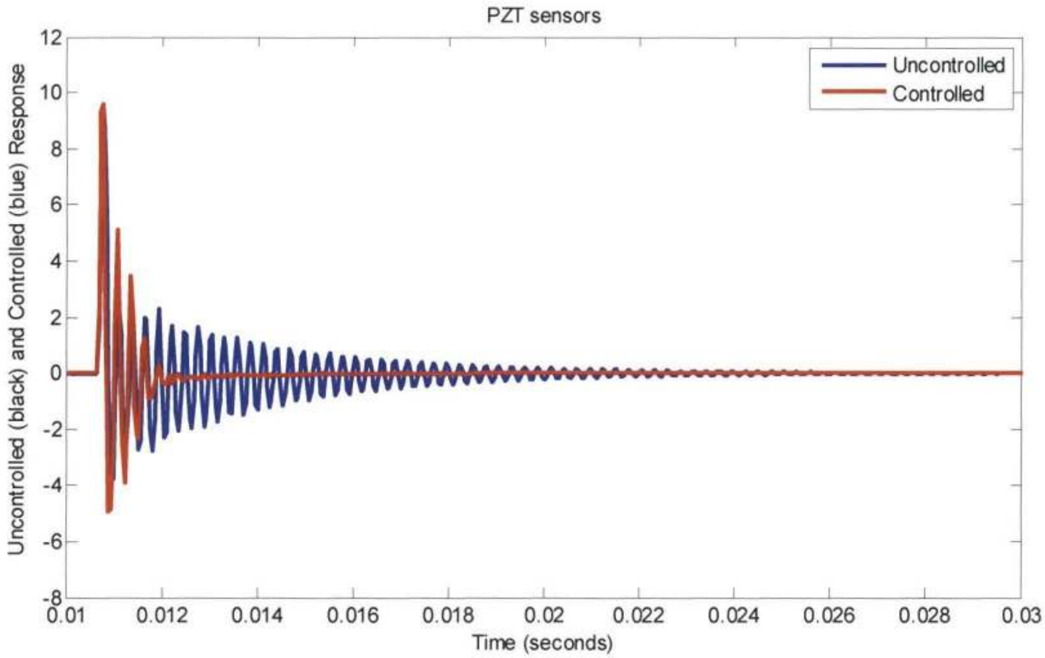

Figure 7. Vibration reduction in the active tool holder under impact testing.

Further testing of the tool holder was done by applying random noise to the actuator. The free and controlled responses from this test were analyzed in the frequency domain through a frequency response function (FRF) which relates the sensor measurement to the excitation signal. The first bending mode of the vibration response was reduced effectively with the same transfer function constants that were applied during the impact analysis. However, the amplitude of the higher frequency regions was increased due of spillover into the higher modes. Therefore, a low-pass Butterworth filter was added to the PPF controller, reducing almost all of the high frequency spillover without affecting the performance of the PPF controller. With this design, approximately 10 dB of amplitude reduction was observed across the first resonant mode of the tool holder, as shown in Figure 8. These tests showed a significant increase in damping in the first resonant frequency of the tool holder, without any loss in stability. This single adaptive PPF coupled with a low pass filter was used as the controller for the actual machine testing on a conventional lathe.

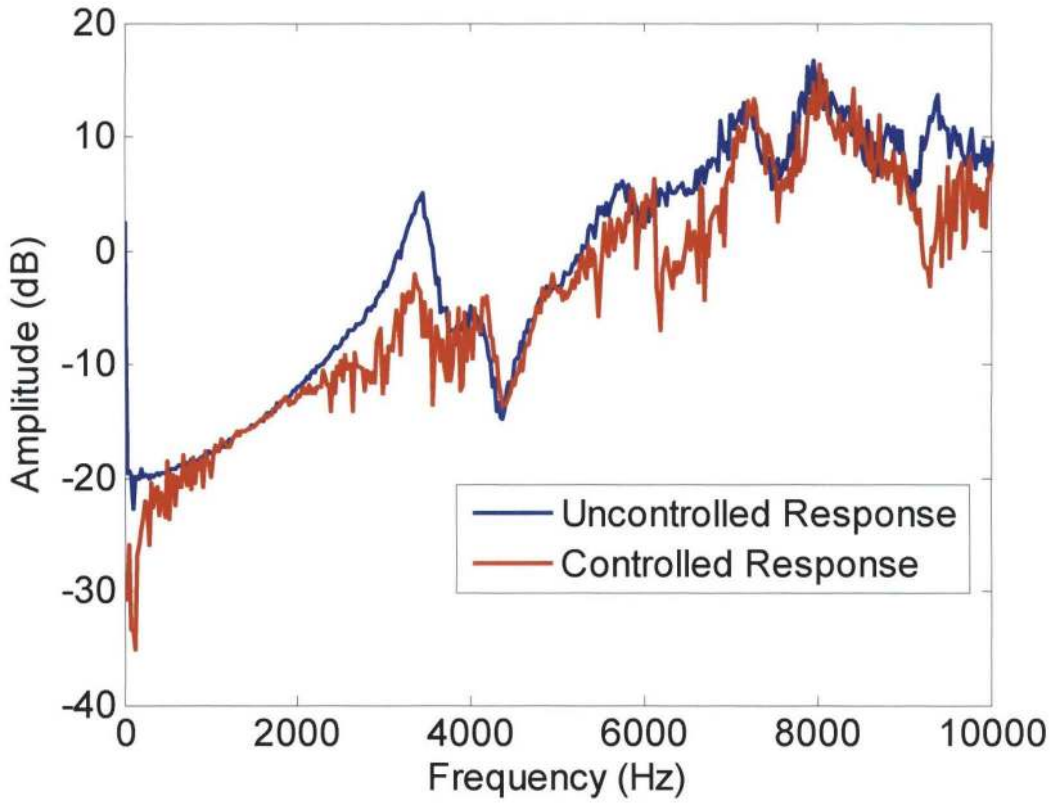

Figure 8. Tool vibration reduction of the 1st mode under random excitation.

The proposed tool holder with an adaptive PPF filter was implemented on a traditional manual lathe. First, the specimens were cut without implementing control. Then, the control electronics were turned on and further tests were performed. On-line performance inspection was done by running an FFT of the measured sensor time signal and comparing the magnitude of the response against the uncontrolled baseline response. Surface finish roughness for each cut was also measured using surface profilometry gages. Tool wear was assumed to be negligible during the testing because of the low hardness of the workpiece material and the relatively slow cutting speeds.

The turning setup consisted of using Aluminum 6061 T6 bar stock (50 mm diameter) mounted in a three jaw chuck on a shop-floor Kent USA manual lathe. Turning tests on the manual lathe were done next with a bar stock that exhibited chatter when turned at 1200 rpm. Turning cut was done on the free end of bar stock where the length to diameter ratio was greater than three. Several cuts were done with different cutting control parameters for the adaptive controller. Contact surface profilometry was recorded after each turning to compare surface finishes.

As can be seen in Figure 9, the first peak frequency during turning occurs at 515 Hz. This lightly damped resonant peak is attributed to machine chatter, a self-excited state that develops between the tool and the workpiece a certain cut speeds and feed rates. This condition leads to a decrease in structural damping that further exacerbates the unstable behavior, producing unwanted, visible surface finish defects, commonly known as chatter marks. If the tool holder or workpiece are not stiff enough to support the machining forces, then chatter will occur. Chatter often limits the depth for boring operations and the depth of cut and feed rates for turning and facing operations.

During the initial testing, it was discovered that response amplitude was an order of magnitude less than what was measured during random noise and impact tests in the laboratory. To address this issue a small amplitude random noise signal was overlaid on the control signal, producing a reference signal for the sensor measurement. A low pass filter was included with a cut-off frequency of 6000 Hz. It was found that, by providing the random signal to the actuator, the signal strength measured by the sensor was increased significantly, without amplifying measurement noise.

Figure 9 shows the frequency response of the sensor signal while taking a 0.1016 mm deep cut with 1200 rpm spindle speed at 127 μm/rev feed rate on the aluminum bar stock. Tuning the controller targeting the first mode (3.7 kHz) frequency caused a decrease of 5 dB in the peak magnitude and completely eliminated the 515 Hz chatter frequency. Surface finish improvement was dramatic. The uncontrolled cut yielded a roughness average of 1.818 μm, while the controlled cut yielded an average of 1.02 μm.

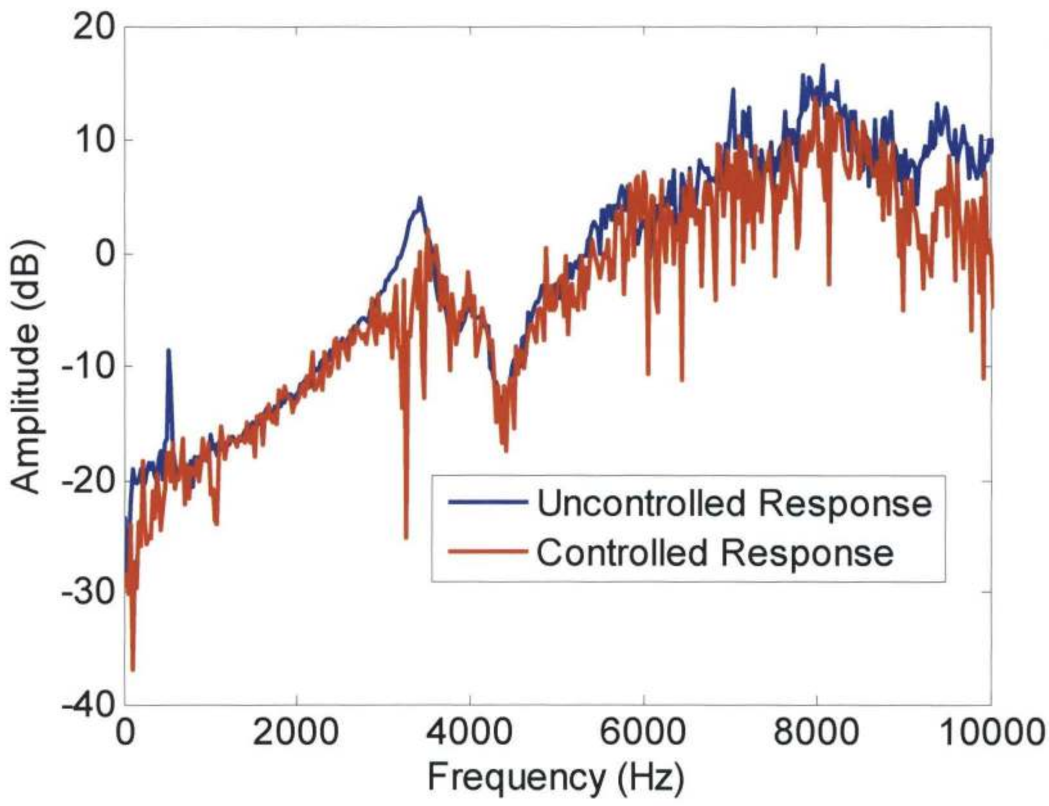

Figure 9. Frequency response measured at the lathe

It was found that the small random signal provided to the actuator significantly increased the signal strength and frequency bandwidth enough for the controller to engage and perform in a stable manner. Tests taken at several different combinations of cut depth and spindle speed showed an average roughness increase of 0.05 μm by the small random signal. This negative effect is perfectly nullified when the controller is performing because of the complete elimination of the 515 Hz chatter frequency.

Several different turning tests were performed with varying cut depths in order to characterize the performance range of the active tool holder with adaptive PPF filter. Two different types of amplifiers were also used for the piezoelectric actuator, a 200 V (PCB 790 series) and a 1000 V (PI E480) max amplifier.
Figure 10 shows a plot of surface roughness of cuts taken at different depths with the 200 V amplifier and 1000 V amplifier against baseline uncontrolled tests. The results show significant improvement for radial cut depths of 50 μm to 381 μm. Performance was limited to 254 μm for the 200 V amplifier because deeper cuts caused its output to become saturated as it could not supply the power required to get the necessary tool displacement. Slight performance degradation at 25.4 μm cut-depth is most likely attributed to the effects of the chatter response signal. Performance benefits are decreased after 381 μm because the displacement of the tool is limited to around 2 μm, which is equal to the displacement caused by the chatter for an uncontrolled cut at 381 μm cut depth. When the chatter displacement exceeds the maximum displacement of the tool holder, the controller was found to be completely ineffective.

Figure 11illustrates the different surface finishes obtained with and without the applied control for visual comparison. It should be noted that all the results shown in
Figure 10 were performed after the adaptive PPF controller had been tuned for 100 μm in cut depth and the control parameters were not changed for each cut, which proves the robustness and adaptability of the controller. Baseline tests determined that the spindle speed would need to be decreased by about 50% to approximately 600 rpm in order to achieve similar surface finish quality as was achieved with the active tool holder, which shows that the actively controlled tool

holder could allow a machinist to double spindle speed on chatter limited cuts, which could provide significant cost savings.

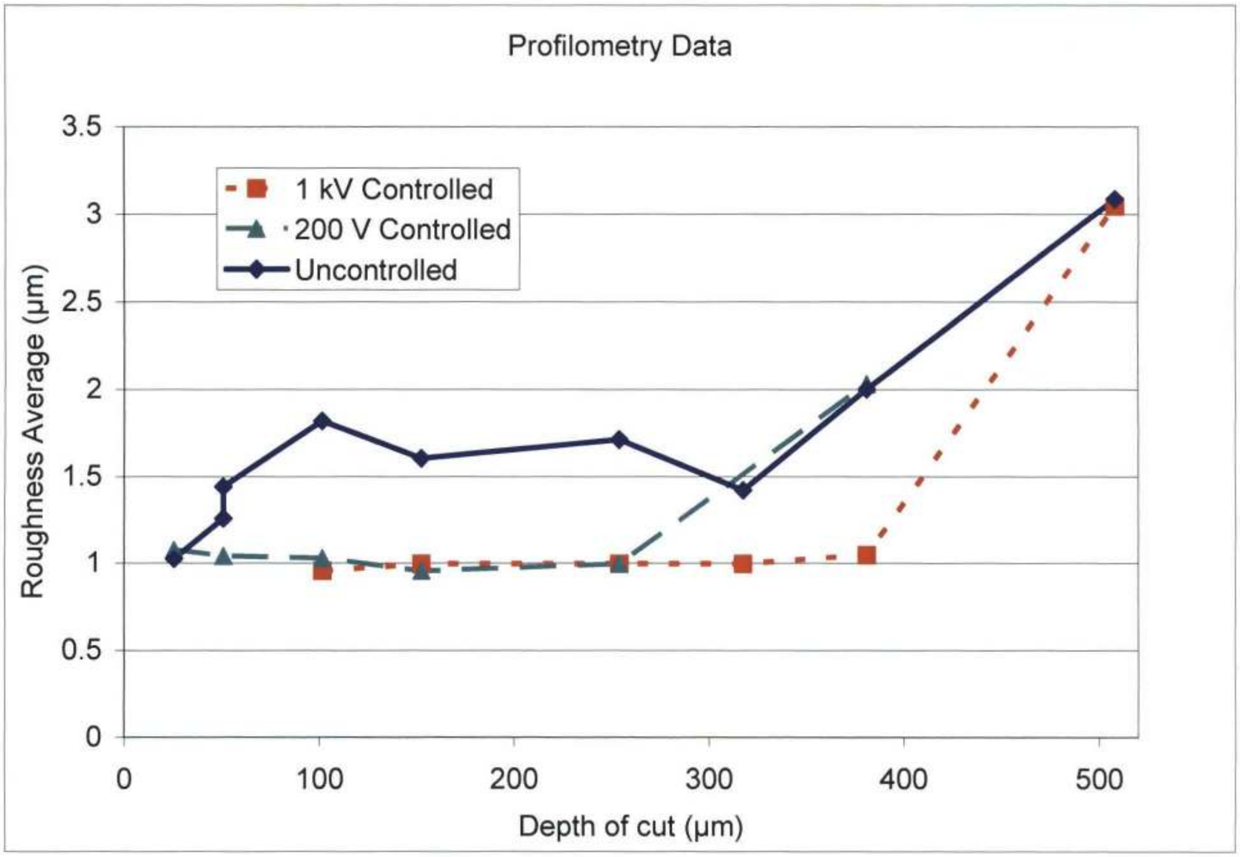

Figure 10. Surface Roughness Performance (1200rpm, 127 μm/rev feed)

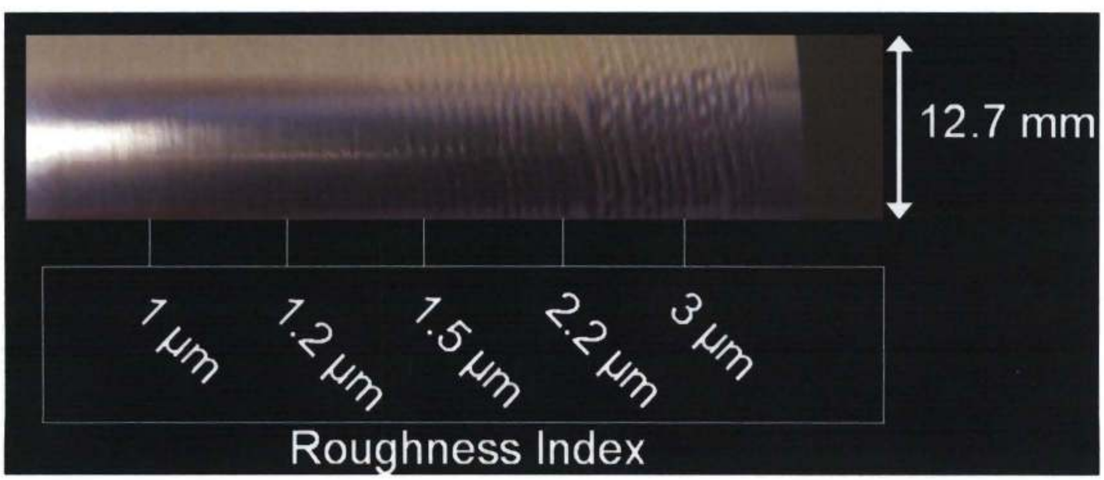

Figure 11. Comparison of different surface quality

CONCLUSION

An active vibration control tool holder is designed and tested in this study. The tool holder employs a collocated piezoelectric stack actuator and sensor with an adaptive positive position feedback control algorithm. This configuration eliminates the need of external gap sensors and makes the entire design compact and self-contained for ease of integration with conventional machining assemblies. The experimental results show that the proposed device significantly increases surface finish quality and reduces the occurrence of chatter. Demonstration of the tool holder proved its cost effective application to standard turning operations. Future work of this research includes the implementation of more advanced control algorithms, the amplification of the active tool displacement, and the use position control to improve part tolerances as well as surface quality.

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