Piezoelectric Active Damper for Surface Roughness Improvement in Hard Turning Processes

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Abstract

Surface roughness and profile accuracy on rolling or sliding surfaces are critical for the wear and fatigue of a component. A high roughness or poor profile accuracy results in higher friction and higher wear rate of the surface. One of the major factors affecting the surface roughness is chatter vibrations. This paper presents a novel design and development of an active damper for chatter suppression of hard turning processes using a piezoelectric actuator and strain signal for chatter detection. The active damper consists of a piezoelectric actuator with an embedded strain gauge for measuring the vibration displacement. In this work, the radial strain signal as a result of radial chatter vibrations from the strain gauge is used as a feedback signal to the actuator using a feedback controller. The experimental results showed a significant suppression in chatter vibrations and improvement of surface roughness using the proposed active damper. The details of the tool design, control design, hardware implementation and system validation are given hereinafter.

Introduction

Hard Turning is a single point cutting process of a workpiece having a hardness in a range between 45 up to 70 HRC. In certain applications the Hard-Turning Process can be an alternative to Grinding process. Therefore, the surface roughness achieved is of crucial importance in this kind of process. Chatter suppression and surface roughness improvement in machining has been the focus of many researchers in the past two decades. Atsushi Matsubara et al. [1] actively suppressed the vibrations of a boring bar using a piezoelectric device shunted with a serial electrical inductor-resister circuit (LR). The acceleration signal from the boring bar was fed to the command voltage for the piezoelectric actuator. Fan Chen et al. [2] designed and developed an electromagnetic actuator to increase the damping and stiffness of a boring bar. The magnetic actuator comprised of four identical magnetic actuating units which can be moved in x, y and θ (torsion) directions by driving the current directions for each unit. Deqing Mei et al. [3] developed an active chatter suppression damper for boring bars using magnetorheological (MR) fluid. By surrounding the boring bar with the MR fluid, the stiffness and natural frequency of the boring bar was continuously changed by changing the intensity of the magnetic field of the MR damper proportional to the excitation frequency. Jérémie Monnin et al. [4] developed an active actuation system for controlling the position of the front spindle in milling operations. The actuating system is composed of two pairs of piezo actuators working in push–pull configuration and orthogonally located in the radial plane over the front bearing. An accelerometer was used to detect the vibration signals for the feedback controller and the actuation system. Mikael Lundblad [5] reduced bending vibrations in cutting tools by introducing flat-shaped piezo-elements on the sides of the cutting tool shank. The bending vibrations were dampened actively via a source of electricity to the piezo-elements and a logic control circuit. Timothy J. et al. [6] developed an automatically tuneable anti-vibration system for boring operations. The vibration of the boring bar is controlled by changing the stiffness of a resilient spring connected to a sliding slug via a linear actuator. The signal to the actuator is calculated using a vibration detector and a controller.

In the work of Lockwood S. et al [7] and Haase F. et al. [8], theories and digital implementations for active vibration control have been covered in details including modelling, digital control algorithms, and digital filters. Active vibration control with various developed sensor-actuator systems has been studied for CNC machines, such as in [9-11] for turning processes and in [12-14] for milling processes. In [9-10], piezoelectric actuators were used and the feedback signal was the acceleration signal on the tool. Especially in the work of Abis C. et al. [9] a multilayer piezoelectric actuator was used for a high actuation power to suppress chatter. Andren L. et al. [11] used a piezoceramic actuator and accelerometer embedded toolholder to implement the active vibration control. Moreover, Ford D. et al. [14] studied PZT actuators, force or acceleration signals and adaptive control algorithms for active vibration control.

For milling processes, F. Aggogeri et al. [12] proposed a piezo-based smart platform to improve the surface finishing of the workpiece by exploiting a broadband active vibration control strategy. F. Kochtbene et al. [13] studied several control algorithms for the active control of a vibrating beam in a milling process. K. Shao, et al. [15] developed a sliding mode controller to eliminate chattering of a linear motor (LM).

Majority of the active vibration suppression systems in the literature mainly focus on damping the bending vibrations of the tool holders. This paper concentrates on active vibration control of the tool holder in the radial direction which is perpendicular to the surface of the workpiece. In addition to this, there is hardly any existing work on active vibration control using strain signal for chatter detection and control system's input. In the proposed system in this paper, a strain signal was used as the feedback signal to facilitate the active vibration control with a PZT actuator.

The final surface roughness obtained on a workpiece can be affected by different parameters such as tool condition, workpiece material, stiffness of the cutting tool, etc. The parameters mentioned influence the process stability and can result in vibrations that can deteriorate the surface roughness of the produced parts. Vibration of the cutting tool in the radial direction is one of the major causes of poor surface roughness in hard turning processes. The present section concerns in-process improvement of surface roughness in hard turning, using active damping technology.

Active Vibration Damper Design

A customized cutting tool with a tool fixture was designed and manufactured to incorporate the actuation system in the cutting tool. A piezoelectric actuator was mounted to the tool fixture, in order to damp the vibration of the tool in the radial direction. Special stiffeners were used in the design of the tool to provide flexibility of a few microns in the radial direction of the cutting, and sub-micron deflections in the remaining directions (feed and cutting force directions). Preventing deflections in the remaining directions was needed in order to avoid damage to the piezoelectric stack and to avoid any additional effects on the surface roughness from deflections in these directions. A clamp was used to align the piezoelectric actuator to the radial direction. The tool fixture design is illustrated in Fig. 1.

The cutting tool is easily replaceable in order to get the correct tool geometry for each machining application. The tool holder block is clamped to the fixture through two screws. The importance of the tool holder block is that it isolates the piezoelectric actuator from the cutting tool, which allows replacement of the cutting tool without disturbing the preloaded piezoelectric actuator.

The piezoelectric actuator includes a strain gauge which is used to measure the excitation that the cutting tool will experience during the cutting process. Using such a co-located strain sensor is better than using a separate vibration sensor, because it reduces the phase shift between the real strain and the measured one. Fig 2 illustrates the active damper setup mounted on the turret of a CNC lathe machine.

An FEA (Finite Element Analysis) was conducted in order to find out the flexibility levels of the whole fixture design. The simulation was performed with applying forces in three directions that were comparable to the forces that the fixture experiences during a real cutting process. Fig. 3 illustrates the FEA model of the deformed fixture subjected to cutting forces at the tip of the insert in three directions; feed, cutting and radial.

The results of the dynamic simulation using FEA are given in Table 1. Two objectives needed to be met here; making sure that the flexibility of the tool is of around five microns in the radial direction and in sub-microns for the other directions as mentioned, above. Moreover, make sure that the piezoelectric actuator will not move more than few sub-microns in the feed and cut (tangential) directions (if not, this may lead to bending the actuator and damaging it). It can be noticed from results presented in Table 1 that the both criterions were met. A circular clamp was used to clamp the piezoelectric actuator from both sides at its ends. Slots were made at the back of the fixture block to allow for the adjustment of the PZT clamp (4 slots for clamping the clamping block and one threaded hole in the middle, where a bolt was used to push/preload the piezoelectric from behind) as shown in Fig. 2.

Table 1. Flexibility levels of the whole fixture design from FEA simulation

	Radial Displacement with a radial force of 200N	Feed displacement with a feed force of 100N	Tangential displacement with a cutting force of 100N
Tool max. displacement		0.07 - 0.2µm	0.4µm
Piezoelectric actuator max. displacement	4.2 - 4.7µm	0.1 - 0.2µm	0.2 – 0.4µm

2.1 Active damping control design with H₂ optimal control method

The active damping control structure is illustrated in Fig. 4, where the plant $P(s)$ is composed of a rigid part and several flexible modes and its model is generally expressed as

$$P(s) = P_0(s) + \sum_{i=1}^p \frac{b_{1i}s + b_{0i}}{s^2 + 2\xi_i\omega_i s + \omega_i^2}$$

where $P_0(s)$ stands for the rigid part and others represent p flexible modes at the frequencies $\omega_i = 2\pi f_i$ ($i = 1, 2, 3, \dots, p$). Fig. 5 shows the frequency responses of $P(s)$, with the dominant mode around 1 kHz, which is easily excited by vibrations.

From Fig. 4, with the controller $C(z)$, the sensor signal y is written as

$$y = \frac{1}{1 + P(z)C(z)} \cdot d$$

The controller $C(z)$ is to be designed so that the sensor signal y is minimized for any input u in the presence of the vibration d . Bearing this in mind, the design of $C(z)$ is formulated as an H_2 control problem. That is to design a controller $C(z)$ such that the H_2 norm of the transfer function from u to the sensor output y , denoted by $\|T_{yu}\|_2$, is minimized. An LMI (linear matrix inequality) approach is used to solve the H_2 control problem. The proposed H_2 control method is a systematic and effective method for the active damping controller design.

In the design of $C(z)$, the loop time-delay and the PZT driver need to be considered in the model $P(s)$, in addition to the flexible resonance mode to be controlled. For our specific application to deal with the vibration focusing around 1 kHz, the designed controller $C(z)$ is a 4th order controller, and Fig. 6 shows the frequency responses of the designed controller. Fig. 7 illustrates that when the controller is on, the magnitude of the sensor signal around 1 kHz is reduced significantly.

2.2 Piezo Hardware Setup

The detailed setup for the hardware required to drive the piezo, includes Compact Rio 9035 (a data acquisition device from National Instruments) FPGA, power amplifier and piezo (the actuator includes also a strain sensor). The main functions of each device are as such: (1) Compact Rio FPGA detects strain from strain gage which was embedded inside the piezo and calculates the required compensation in voltage and outputs it to the amplifier. (2) Amplifier takes in this signal and amplifies it by 10 times so that the voltage is in the working range of the piezo. (3) Piezo actuator would then expand or contract

based on the input signal and alters the tool displacement with respect to work piece. The maximum voltage of 0 to 100V corresponds to a maximum stroke of 10 μ m in both directions. Detailed connection of each component is illustrated below in Fig. 8. A computer was also needed for programming and setting parameters for FPGA and also records the data from FPGA.

Compact Rio FPGA is the processing unit for data acquisition and calculations. CRio 9035 was used together with three data acquisition units (NI 9218, NI9234, NI9232) and one Analog output unit (NI9263). Fig. 9 illustrates the compact Rio setup. 9263 was connected to amplifier which outputs the control signal. 9232 can be used to monitor amplifier output (limited to 30V) and also record signals coming from 9263. 9234 was used to record the impulse hammer signals used for system identification. Lastly, 9218 acquires strain input from the piezo strain gage.

E-617.00F Amplifier was used due to its higher power output. Wire connections were made to the amplifier via the removable solderable socket. NI PS-15 was used as power supply for the amplifier. Piezo output voltage from amplifier was connected to a BNC adaptor and connected directly to the piezo via the BNC adaptor.

Piezo actuator was secured between tool holder and back mounting wall. A preload was applied by tightening the back screw. The amount of preload was such that the strain becomes almost zero when a 50V voltage was supplied to the piezo. This allows the piezo to work at the full range in both directions. Supporting back screws were then tightened to secure the structure. Wire connection for piezo strain gage is as illustrated in Fig. 10. Strain gage output signal was connected to the LEMO connectors via a screw terminal. The white wire from the strain gage splits to five wires. Red and black goes to 1 and 3a which provides the excitation. Green white gives the resistor potential difference and goes to 6 and 7. The shielding wire is connected to the ground (soldered to a thick green wire).

Results & Discussion

All tests were conducted systematically with simulation followed by validation. This includes the modal simulation from the final tool design and the natural frequency were validated by chirp excitation and impulse hammer test. Upon tool fabrication, multiple iterations of controller design simulation and validation were carried out and the results are highlighted in the following section.

3.1 Chirp signal results

Chirp signal was generated using a sine sweep from 100Hz to 10kHz in a period of one second. The results are plotted and analysed as shown in Fig. 11. Chirp signal generation was done on the cDAQ (another data acquisition device from National Instruments). Hardware setup for cDAQ was exactly the same as cRio. Once the signal was recorded, the data was processed to give the bode plot which shows the gain and phase response of a given system for different frequencies.

The top image shows the magnitude plot of the output signal over input signal in frequency domain while the lower image is the actual time domain signal. Right image shows the phase shift in degrees and the drop is quite significant. A resonance of 1.2kHz was observed and it was confirmed later by an accelerometer that the direction is parallel to the piezo actuation direction. Frequency response signal from the chirp signal test was used to generate the model used for control simulation.

3.2 Hammer test results

Dytran Impulse hammer (5800B4) was used to excite the tool holder in the X cutting direction. The hammer test is different from the chirp signal test as hammer test acts as an external disturbance while chirp test excites the plant to give the transfer function of the amplifier, piezo and strain gage. External disturbance from the hammer was necessary to test the performance of the closed loop system with control. The impact signal act as input signal while measured strain acts as output. Response between input and output was calculated similar in the chirp signal analysis just that the time period is shorter at 0.05s as transient behaviour of the system was very short. Upon generation of control parameters from the simulation, hammer test was carried out to verify that control was working before proceeding to actual cut. This was done by comparing strain signal from impacts with and without control. The results are shown in Fig. 12.

Active control has decreased the transient response of the tool natural vibration dramatically. Peak resonant frequency at 1.2kHz was reduced by 6-7dB with active damping. Initially, the hammer test was done while the fixture was outside the machine. After securing the fixture to the machine, the resonant frequency actually dropped to 1.1kHz as shown later in the actual cutting tests. The model was re-estimated and new control was implemented in the actual cutting tests.

3.3 Cutting test results

With the positive results from the impact hammer test, the actual test was carried out and cutting parameters were adjusted such that to create chatter. The final cutting parameters used were V_c 80m/min, doc 0.1mm, feed 0.025mm/rev. The strain measurements corresponding to the two cases with and without active damping were recorded and displayed in Fig. 13.

The parameters used induced a tool chatter from which high pitch chatter noise could be heard. Once the active damping was activated, the chatter was completely removed. This is seen in Fig. 13 and Fig. 14 where the controller was activated at 200s and strain variation was greatly reduced (Fig. 13, a). The power spectra density of the strain was computed and the amplitude drop was significant at 1.1kHz which was the chatter frequency.

Surface finish of the finished workpiece was inspected and compared. The test was repeated several times and a new cutting insert was used for every test. In area without active damping, an average

roughness (Ra) of 0.721 μm was measured and with active damping, average Ra was found to be much lower at 0.123 μm . The roughness difference was observable with visual inspection and a magnified image using a microscope of the surface is shown in Fig. 15. The results clearly show the improvement in the roughness with active damping control.

Conclusion

The active damper with the piezo actuator feedback vibration control system was capable of significantly reducing the surface roughness (Ra) related to chattering by almost 83%. The surface roughness (Ra) was reduced from 0.721 μm (without control) to 0.123 μm (with control) during the cutting process. This research also presented a novel fixture design for the active damper which only suppresses the vibrations in the radial direction perpendicular to the surface of the workpiece. In addition to this, using the strain signal from the embedded strain gauge inside the actuator as the feedback signal to the controller has shown to be an effective and efficient method in chatter vibration suppression. It should be noted that the active damper system is only a prototype model at this stage. More research needs to be conducted on the integration of the active chatter control systems into the machine tools suitable for industry applications.

Declarations

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Figures

Figure 1

Tool Fixture Design, a) CAD fixture, b) Manufactured and assembled fixture, c) Fixture mounted on CNC lathe turret

Figure 2

Piezoelectric actuator clamping

Figure 3

FEA model of the fixture deformation under cutting loads

Figure 4

Block diagram of active damping for vibration control with strain signal.

Figure 5

The frequency responses of $P(s)$

Figure 6

Frequency responses of the designed $C(z)$

Figure 7

System response with and without active damping control

Figure 8

Illustration of hardware and software setup

Figure 9

CRio FPGA with (from left to right) Strain input 9218, IEPE input 9234, Analog input 9232 and Analog output 9263

Figure 10

Piezo setup on tool holder and fixture and wire connections

Figure 11

(top left) Chirp signal frequency response, (bottom left) Strain signal in time domain, (right) phase plot

Figure 12

Hammer test results, (top left) magnitude plot of strain response with and without control. (top right) time domain strain signal without control. (bottom right) time domain strain signal with active control

Figure 13

(a) Strain signal during actual cutting, (b) Control signal, Controller was activated at 200s

Figure 14

Effect of the control system on strain signals, (Grey) strain signal without control, (Black) strain signal with control

Figure 15

(left) Surface finish with active damping, (right) without active damping showing chatter marks