Vibration Control of Relative Tool-Spindle Displacement for Computer Numerically Controlled Lathe With Pipe Frame Structure

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A new computer numerically controlled (CNC) lathe with a pipe frame bed has been developed. This structure is expected to have enough space between the truss bars to solve the space problem and have enough rigidity for machine tools. Therefore, a CNC lathe whose frame consists of pipes, joints, and diagonal braces has been developed with enough rigidity and space utility for chip evacuation. From the viewpoint of machine tool usage, real-time vibration control theory is applied to control the relative displacement between the tool post and the spindle to suppress specific relative vibration modes. [DOI: 10.1115/1.4027594]

Introduction

Conventional beds or frames for machine tools generally consist of cast iron or welded steel plates. These structures are suitable from the viewpoint of the machine tool's dynamic characteristics. Although the demand for downsizing machine tools (such as in desktop machines) is increasing, there is a strict volumetric limitation to obtain enough space for sufficient chip evacuation. In this case, there are problems of heat conduction and heat transfer. To solve these problems, we propose a traditional but new frame structure for machine tools that consists of pipes and connecting parts. We looked at the flexibility of the frame design using a pipe frame structure. The frame structure can be arranged with standardized pipe (diameter and length) and a connecting block. In the case of a lathe, this structure allows for a change in size based on the spindle unit and X-axis table, which are designed as standardized units.

For developing a CNC lathe using this structure [1], the following problems are assumed:

(1) structural vibration generated by the spindle motor unbalance and the cutting force disturbance,
(2) thermal deformation due to the low heat capacity,
(3) insufficient rigidity between the spindle and the X- and Z axis tables.

In this report, we focus on the vibration problem. There have been several prior studies [2–7] on the applications of vibration control for machine tools. In these research papers, the control methods have been applied to the tool post of the lathe machine. These results demonstrated the superiority of vibration control of the tool post displacement but do not consider the structural vibration modes that appear in the low-frequency range compared with the local vibration modes, such as the boring bar and tool post.

Developed CNC Lathe

A unique CNC lathe has been developed based on the idea of a pipe frame structure. The specifications are shown in Table 1, and a schematic view is shown in Fig. 1.

A maximum spindle speed of 10,000 min^{-1} can be obtained. The smallest resolution for the X and Z axes is set to 0.1 μm. The lathe weighs only 525 kg (including the covers) and fits within a space of 440 $W \times 1450$ $L \times 1050$ H mm^3, which is sufficiently compact. The CNC controller (Mitsubishi Electric Co., Ltd., M700V) is set beside the machine body.

Vibration Control Method

The most important vibratory motion for the CNC lathe is the relative displacement between the spindle and the tool post. The machined workpiece is often influenced by the antiphase motion between the spindle and the tool; this motion is generated by the resonant frequency. In this case, the antiphase motion is the most important behavior for the workpiece's geometric error. This motion can be evaluated by measuring the relative vibration between the spindle and the tool. Although this displacement should be measured directly by a displacement sensor during cutting, it is very difficult to position the sensor because of the space constraints, coolant, and chips. Figure 2 shows the typical compliance transfer function between the spindle head and the representative frame-connected block. There are several peaks. When we try to apply vibration control using this data, the control point is hardly determined. Then, two small acceleration sensors are used to detect the relative vibration mode. One is set on the spindle unit case and is measured as the signal $\ddot{x}_s$; the other is set on the tool post of the X-axis table and is measured as the signal $\ddot{x}_t$.

The relative vibration $\ddot{x}$ is expressed as follows:

$$\ddot{x} = \ddot{x}_s - \ddot{x}_t \tag{1}$$

Then, the difference between the two acceleration sensor measurements in Eq. (1) is calculated and used to extract the antiphase motion between them. The result of the calculated transfer function is shown in Fig. 3. A frequency peak at 45 Hz can be observed, and it can be considered a simple one-degree-of-freedom system. We can use this differential signal as the output of the vibration response.

Then, modal analysis can be used to obtain the vibration modes, to decide the target vibration mode, and to determine where to set the actuator for the vibration control. The synthesized mode shape vector at 45 Hz is shown in Fig. 4. The antiphase mode between the spindle and the tool post is actually seen from the vector

Contributed by the Manufacturing Engineering Division of ASME for publication in the Journal of Manufacturing Science and Engineering. Manuscript received February 20, 2013; final manuscript received April 11, 2014; published online May 21, 2014. Assoc. Editor: Tony Schmitz.

Table 1 Specifications of CNC lathe

Items		Specifications
Head stock	Chuck	Collet chuck
	Maximum speed	$10000\,\text{min}^{-1}$
Tool post type		Horizontal linear
Minimum resolution		0.0001 mm
Size		$440\,W \times 1450\,L \times 1050\,H$ mm
Machine weight		525 kg
CNC controller		Mitsubishi Electric Co., Ltd. M700V

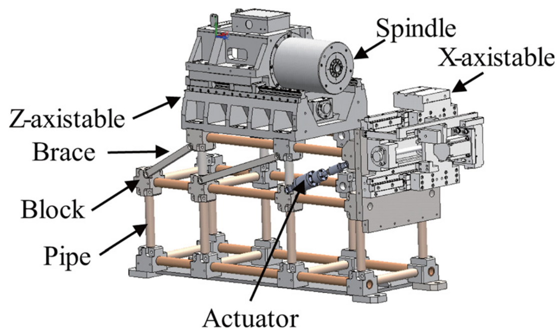

Fig. 1 Schematic view of developed CNC lathe

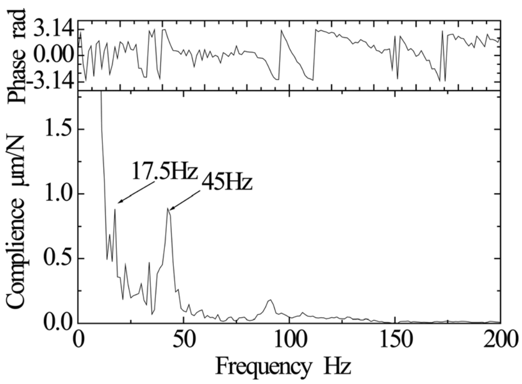

Fig. 2 Typical measurement result of compliance transfer function by conventional impulse excitation method on a representative point

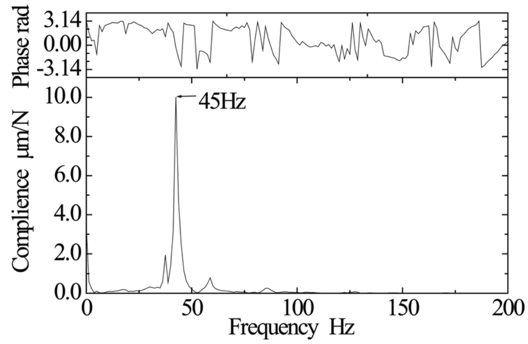

Fig. 3 Transfer function between input force and relative displacement (tool post and spindle)

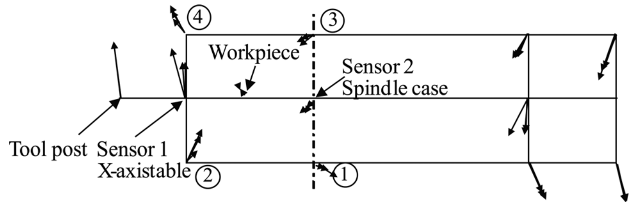

Fig. 4 Mode shape vector (top view)

Table 2 Identified modal parameter

Items	Value	Unit
Mass, m	214	kg
Damping factor, c	8.01	N/(m/s)
Spring constant, k	$1.62 \times 10^{+07}$	N/m
K_v	3.66×10^{03}	N/m/s
K_d	5.73×10^{02}	N/(m/s)

figure. The mode vectors at the head stock for the spindle (① and ③) are so small that the ends of the two actuators can be set at these points. Based on this result, the other ends of the actuators are set on connecting blocks below the X-axis table to control the antiphase vibration mode, as shown in Fig. 1. These two actuators generate the control forces to suppress the vibratory motion by the active vibration control method.

The relative displacement, velocity, and acceleration are defined as x, $\dot{x}$, and $\ddot{x}$, respectively. Assuming that x_d and $\dot{x}_d$ denote the desired relative displacement and the relative velocity, the relative displacement and the relative velocity are defined as

$$(x_d - x), (\dot{x}_d - \dot{x})$$

The controlled force can be expressed as follows:

$$K_v(\dot{x}_d - \dot{x}) + K_d(x_d - x) \tag{2}$$

where the weight coefficients K_d and K_v are used. The equation of motion is described as follows when a harmonic force $f_a e^{j\omega t}$ acts on this system:

$$m\ddot{x} + c\dot{x} + kx = f_a e^{j\omega t} + K_v(\dot{x}_d - \dot{x}) + K_d(x_d - x) \tag{3}$$

Because the desired relative displacement x_d and relative velocity $\dot{x}_d$ are to be zero, Eq. (3) is described as follows:

$$m\ddot{x} + (c + K_v)\dot{x} + (k + K_d)x = f_a e^{j\omega t} \tag{4}$$

In Eq. (2), substituting $f = f_a e^{j\omega t}$, the following equation can be derived:

$$\begin{Bmatrix} \ddot{x} \\ \dot{x} \end{Bmatrix} = \begin{bmatrix} -c/m & -k/m \\ 1 & 0 \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \ddot{x} \end{Bmatrix} + \begin{bmatrix} 1/m \\ 0 \end{bmatrix} f \tag{5}$$

This equation of motion is rewritten as the following state equation:

$$\dot{\mathbf{X}} = \mathbf{AX} + \mathbf{BF} \tag{6}$$

In the case of applying a control force that is proportional to the state vector X, the control force is described as $F = -KX$. Then, substituting this into Eq. (6), the following state equation is obtained:

$$\mathbf{X} = (\mathbf{A} - \mathbf{BK})\mathbf{X} \tag{7}$$

Once this equation can be obtained, the only thing to do is calculate the feedback gain K using MATLAB (MathWorks—MATLAB and Simulink). To calculate this coefficient K, the modal parameters m, c, and k must first be determined. We calculated these values using the simple "mass response method" because the vibratory motion can be assumed to be one-degree-of-freedom from the result in Fig. 3. Once the known mass is set on the tool

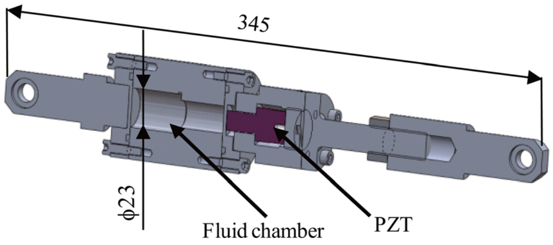

Fig. 5 Schematic diagram of PZT actuator for vibration control

Table 3 Experimental setup for vibration control

Items	Value	Unit
Spindle speed	2700	min^{-1}
Target frequency	45	Hz
Sampling frequency	3000	Hz
Depth of cut	0.1	Mm
Feed	0.02	mm/rev
Workpiece material	C3604BD	

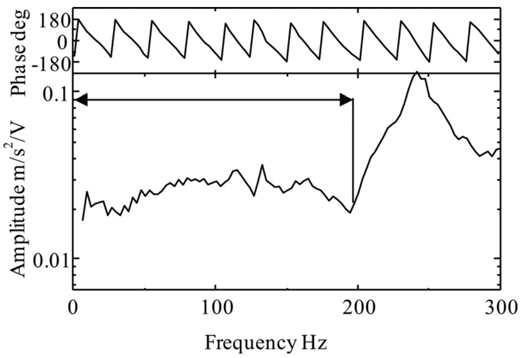

Fig. 6 Transfer function between input voltage of actuator and table displacement by impulse response

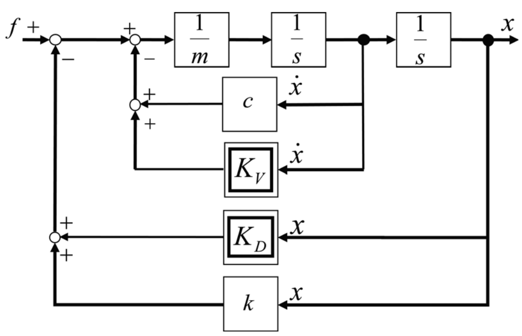

Fig. 7 Block diagram for vibration control

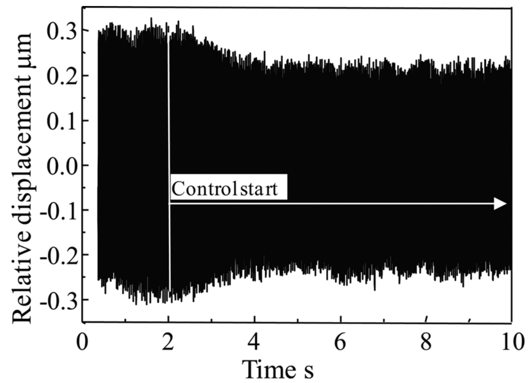

Fig. 8 Comparison of relative motion during cutting

post, the change in the resonant frequency is measured, and the modal parameters are obtained, as shown in Table 2. These are substituted into Eq. (7), and the feedback gain K is calculated according to linear-quadratic (LQ) control theory.

Vibration Control System

Lead zirconate titanate (PZT) devices are used in the vibration control system as the actuator source. Structural vibration control and motion control for machine tools use this device because of its large generated force and high-frequency response [8–11].

We developed an actuator that consists of a PZT, a fluid chamber with two different cross-sectional areas, and connecting rods, as shown in Fig. 5. The fluid chamber is used to transform the generated force from the PZT. Although the longitudinal displacement of the PZT is short enough, the displacement is reduced by the chamber area. This effect makes the generated displacement less than a submicron order. Two actuators are set between the connecting blocks as brace bars. The dynamic characteristics are evaluated using the impulse response method, and the result is shown in Fig. 6. This actuator can be used below 200 Hz. The effects of the vibration control are evaluated in Effects of Active Vibration Control section. Although the thermal elongation of the brace bars has to be considered when the long run operation, we do not consider the thermal deformation of the frame because of the short-time machining.

Effects of Active Vibration Control

The relative vibration between the spindle and the tool post is controlled by two actuators. In this case, according to LQ control theory, we have developed the control system to realize the control method represented by the block diagram in Fig. 7 and in Table 3. This system runs with a sampling speed of 3000 Hz. The relative vibration is generated by the unbalance of the spindle unit. Since the maximum displacement is observed at around 2725 min^{-1} (45.4 Hz), the target frequency is set to 45 Hz.

The experimental procedure has been executed during air cutting and during cutting. From the experimental result and frequency response of the actuator, the target-controlled frequency range is set from 10 to 100 Hz. A band-pass real-time filter is adapted to obtain the vibration signal and to suppress the disturbance noise and high-frequency signals.

To evaluate the effect of the active vibration control, two acceleration sensors due to the limitation of the installation space are used for the control. The relative vibration is numerically integrated to obtain the relative velocity and the relative displacement. The relative displacement itself during motion is measured by a laser displacement sensor that is set on the tool post and detects the relative displacement of the spindle unit.

The control result during finish cutting is shown in Fig. 8. The control parameters have been determined previously during air cutting. This condition is not always applicable to the condition during cutting because the workpiece and the tool contact each other, and the cutting force acts on the contact point. Although the modal parameters are not always constant, our control system is applied using the same conditions as air cutting, and the effect of the vibration control is evaluated during cutting.

When the control is started, the relative displacement decreases its amplitude. FFT analysis is used to determine the difference in the frequency element between with control and without control, as shown in Fig. 9. The solid line shows the result with control, and the dotted line shows the result without control. Almost 50% of the vibration amplitude around the target frequency of 45 Hz is suppressed by our developed system. The effect of an increase in rigidity by the actuator installed as a brace can be seen in the lower frequency range. On the other hand, the effect above 100 Hz cannot be seen because of the filter effect of the control

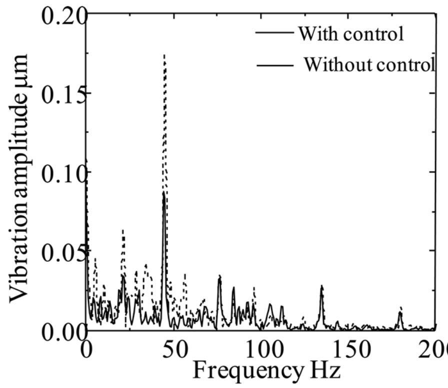

Fig. 9 Comparison of frequency analysis of relative vibration between tool and spindle during cutting

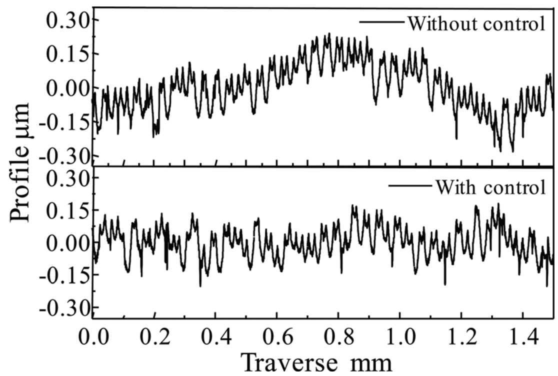

Fig. 10 Comparison of axial profile of machined workpiece measured by surface roughness tester without low-pass filter

system. These results demonstrate that this vibration control system used during finish cutting is suitable for our pipe frame CNC lathe. To improve the effect of the vibration control, the layout (including the direction of the actuator) should be considered, and further improvement of this actuator system is needed to suppress the vibration amplitude more efficiently.

Evaluation of Workpiece by Vibration Control

Since it has been confirmed that the vibration control is effective even during cutting, we evaluated the accuracy of the machined workpiece. Because the target control frequency was set to 45 Hz, the effect of the control result appears as surface waviness. Then, the surface profile was measured by the surface roughness tester without a high-pass filter. Figure 10 shows the measured results. The upper figure shows the profile without vibration control, and the lower figure shows the result with vibration control. The surface waviness is remarkably improved. The improvement of the waviness at this frequency range is related to the relative vibration mode that appears in this frequency range.

Figure 11 shows the harmonic analysis of the measured roundness result. Since the target frequency is set to the spindle rotational frequency, the effect of the vibration control generally appears on the roundness with two undulations per round. The measured result shows this tendency well. The amplitude of the undulations per round decreased by 50% from the amplitude without control.

These results show the effectiveness of our control system at suppressing the relative vibration mode between the spindle and the tool post and the improvement in the accuracy of the machined workpiece.

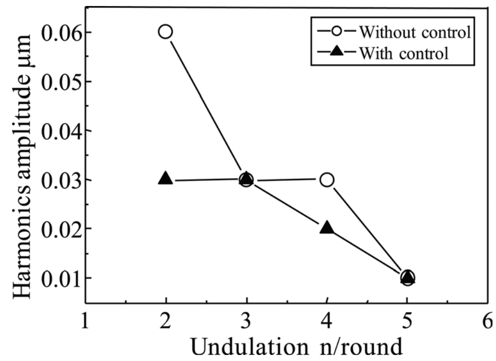

Fig. 11 Comparison of harmonic analysis of measured roundness

Conclusions

In this paper, the effectiveness of the real-time vibration control for the developed CNC lathe by a single degrees of freedom model has been evaluated using the differential signal between the spindle and the tool post. The main results obtained as follows:

(1) Almost 50% of the vibration amplitude around the target frequency of 45 Hz is suppressed by our developed system.
(2) The roundness amplitude of the undulations per round decreased by 50% from the amplitude without control.
(3) Results obtained show the effectiveness of the applied control system at suppressing the relative vibration mode between the spindle and the tool post.

Further investigations are required to improve the vibration control system, including the layout (the direction of the actuator) and actuator performance to suppress the vibration amplitude more efficiently.

Acknowledgment

The authors express their gratitude for the assistance of Kazuhiro Shimizu, who has supported our experiments. This study was partially supported by grant funding from the New Energy and Industrial Technology Development Organization.

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