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Sato, Ryuta
Hayashi, Hideaki
Shirase, Keiichi

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Active vibration suppression of NC machine tools for high-speed contouring motions

Ryuta SATO*, Hideaki HAYASHI* and Keiichi SHIRASE*

*Department of Mechanical Engineering, Kobe University

1-1 Rokko-dai, Nada-ku, Kobe 657-8501, Japan

E-mail: sato@mech.kobe-u.ac.jp

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Abstract

In this study, we describe a novel mechanical vibration suppression method for high-speed contouring motions. In this method, vibration compensation torque was applied to suppress mechanical vibrations during high-speed contouring motions. The parameters of the compensation signal were determined using commanded acceleration and time for acceleration and deceleration, which were set as numerical control (NC) parameters. We also propose an autonomous compensation-torque-generation algorithm based on the acceleration command from NC. The effectiveness of this method was assessed by measuring and simulating rectangular corner contouring motions in which the compensation torque is applied. The results confirmed that the proposed method can effectively suppress the vibration when the motion direction changes. This method can also be effectively performed at various feed speeds by automatically adapting the torque based on the proposed criteria.

Keywords: Numerically controlled (NC) machine tools, Feed drive system, High-speed contouring motion, Vibration suppression

1. Introduction

Numerically controlled (NC) machine tools is an important technology in the manufacturing field. To improve productivity and quality it is highly desirable that these tools perform at higher speeds and accuracy. However, mechanical vibration caused by the inertia of moving mass during high-speed contouring motion greatly influences the relative distance between the tool and workpiece, thus deteriorating machining accuracy.

Several studies have previously dealt with mechanical vibration suppression methods. The first approach is to design feedback control systems. Iwasaki et al. (1996) proposed a two-degree-of-freedom feedback control system to suppress vibrations. Sato and Tsutsumi (2005) suggested a controller tuning method based on the feed drive system model. Other approaches that have also been proposed to effectively suppress vibrations. These include acceleration feedback (Kai et al., 2018), designing feed rate and acceleration/deceleration profiles (Erkorkmaz and Altintas, 2001; Altintas and Erkorkmaz, 2003), and designing an input command filter that can avoid exciting the resonances (Chen and Lee, 1998). Attempts have been made to apply filtering approaches to the contouring motions (Altintas and Khoshdarregi, 2012; Sencer et al., 2015, and Tajima et al., 2018). A frequency separation approach and tool position estimation method using a Kalman filter have also been suggested (Denkena et al., 2015). However, the effectiveness of the aforementioned strategies is limited by the characteristics of the axis with maximum oscillation, this is because the frequency characteristics of the axes have to be matched with the contouring control accuracy, and any mismatch deteriorates the contouring controlled accuracy (Sato et al., 2007). Although input command design to suppress the residual vibration is possible for positioning purposes by a method called “posicast control” (Smith, 1957) or “input shaping” (Singer and Seering, 1990; Singhose et al., 1996), it is difficult to apply to contour control motions.

Vibration suppression has also been attempted using mechanical approaches. It was found that the placement of machine bed support influences the residual vibrations (Mori et al., 2015). Shirahama et al. (2016) proposed a sliding support that can reduce the residual vibration by consuming the vibration energy at the supports. Mori et al. (2018) proposed using viscoelastic damper supports under the bed. Although these approaches can reduce vibrations, they do not satisfy the requirements. Active vibration control approaches have also been applied to suppress chatter vibrations (Zaeh et al., 2017; Kleinwort et al., 2018a, 2018b; Konigsberg et al., 2018; Mancisidor et al., 2018). In these studies, additional sensors and actuators were added to the spindle or mechanical structures. The effectiveness of the active vibration control approach for contouring control motions has not been investigated. In addition, it is better to suppress the vibration without any additional actuators and sensors, because they increase the cost.

The purpose of this study was to develop an active vibration suppression method for high-speed contouring motions, without any additional actuators or sensors for control. To achieve this goal, we propose a mechanical vibration suppression method by applying compensation torque to cancel the mechanical vibration during high-speed contouring motion. In the proposed method parameters of the compensation signal are determined based on the commanded acceleration and time for acceleration and deceleration, which are set as NC parameters. We also suggest an autonomous compensation torque generation algorithm based on the acceleration command from NC. To evaluate the effectiveness of the proposed method rectangular corner contouring motions using the vibration compensation torque were measured and simulated.

2. Experimental method and mathematical model

A vertical type of machining center was used in the experiment in this study. Each axis of the machining center was driven by an AC servomotor and ball screw. An XY table was arranged on the table side, and the X-axis was placed on the Y-axis. To assess the effectiveness of the method the trajectory during the tracking motion was measured as shown in Fig. 1. The trajectory between the table and spindle head was measured using a grid encoder (KGM 182, HEIDENHAIN GmbH). The system could measure the two-dimensional relative positions between the table side and spindle side with a resolution of 1 nm. Simultaneously, an accelerometer was installed on the table to measure the acceleration of the table in the Y-axis direction. Additionally, the position command, position feedback, and motor torque of each axis was also measured. The path of the corner tracking motion test for this study was diamond-shaped with a 200 mm diagonal length for each. Simultaneous motions along the X-axis and Y-axis are required to realize this motion.

We also carried out simulation based on a mathematical model consisting of five components: bed, saddle, table, column, and spindle head (Fig. 3; Sato et al., 2015). All components had three degrees of freedom for each of the translational and rotational axes. Feed drive system models with two degrees of freedom were installed between the table and saddle (X-axis), saddle and bed (Y-axis), and column and spindle head (Z-axis). These models also had the controller models, which could represent the influence of servo gain settings etc.

In the model, the mass and inertia of the mechanical components of the machine could be determined by the specification and a CAD model of the machine tool, while stiffness and damping were estimated by adjusting trial and error by fitting the frequency responses. Figure 4 shows the comparison between measured and simulated frequency response along the X and Y axes. The model with the identified values can depict the frequency response of ~65 Hz along the X-axis and ~50 Hz along the Y-axis; these vibration modes determine the motion accuracy. Hence, our model can be applied for evaluation of the proposed vibration suppression method.

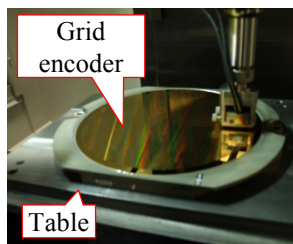


Fig. 1 Measurement of motion trajectory.

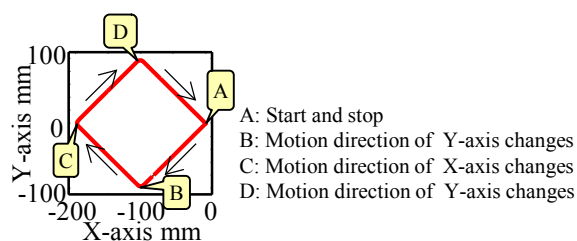


Fig. 2 Motion path for the test.

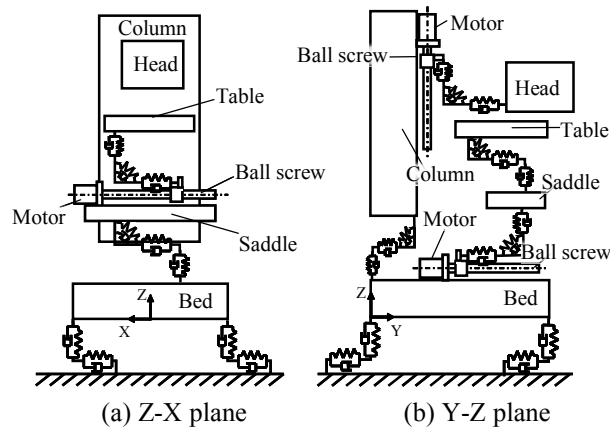


Fig. 3 Vibration model of machine tool structure.

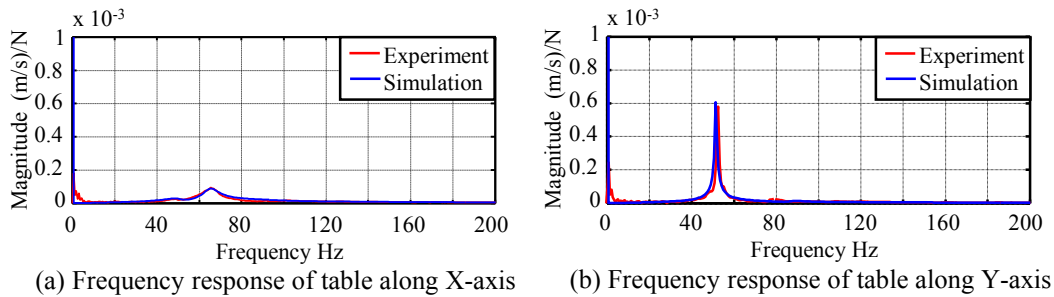


Fig. 4 Comparison of measured and simulated frequency responses.

3. Active vibration suppression method

Figure 5 shows a block diagram outlining the proposed mechanical vibration suppression method. This includes a position control loop and a velocity control loop in the feed drive system. The position controller is proportional (P) control and the velocity controller is proportional–integral (PI) control. In addition, the position controller is semi-closed control based on the rotation angle of the motor as a feedback signal.

In the proposed method, at the instance of acceleration or deceleration, a compensation signal is applied to the torque command output from the velocity controller. Based on the signal, vibration with the same amplitude and opposite phase is superposed with the vibration caused by acceleration and deceleration. As a result, the vibration can be canceled out. In Fig. 5, R is a coefficient for converting the rotation angle of the servomotor into the axial displacement.

The proposed compensation signal is outlined in Fig. 6. It is a rectangular signal expressed as a superposition of step inputs, and its components are compensation magnitude M_s and duration T_l . The time at which the acceleration command (second-order derivative of position command) attains the set threshold value A_t is set as the reference time T_s , and a compensation signal is generated after the compensation input time T_d seconds therefrom. A_t should be larger than the resolution of the acceleration command signal. In addition, if A_t is set too high, the vibration suppression performance will deteriorate. When a rectangular torque command containing a high-frequency component is applied to the torque command, undesired vibration and noise are generated. Therefore, a biquadratic infinite-impulse-response (IIR) low-pass filter is applied to the compensation signal. The cutoff frequency of the filter F_c is set to the natural frequency in the Y-axis of the machining center used in the experiment. Additionally, the Q-value of the filter is set as 0.5 to make the compensation torque have critical damping.

During practical application of the proposed method it is necessary to automatically determine the parameters of the compensation signal in response to various commands and to automatically generate the compensation signal. The proposed active vibration control method that can automatically generate the compensation signal is outlined in Figure 7. Here, G_1 is a transfer function from position command Y_{ref} to vibration X_1 due to acceleration and deceleration, and G_2 is a transfer function from compensation torque command T_{comp} to oscillated vibration X_2 . The positional command

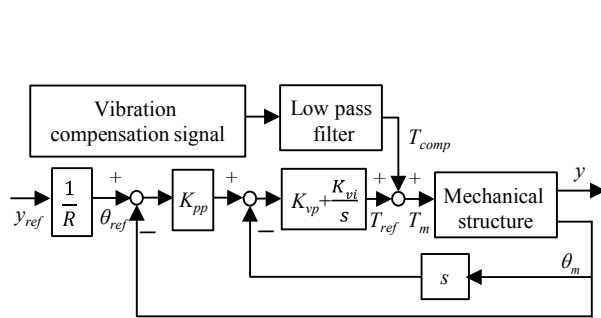


Fig. 5 Proposed vibration suppression method.

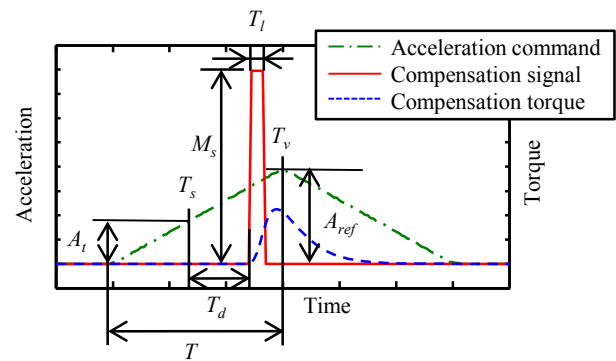


Fig. 6 Vibration compensation torque.

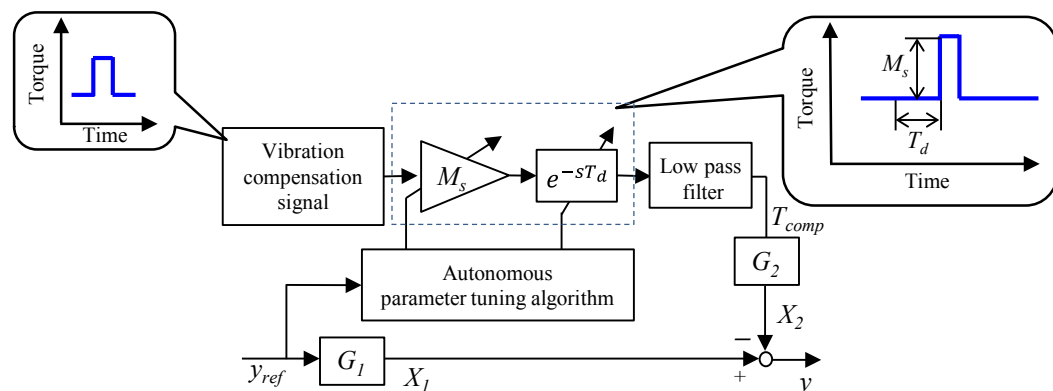


Fig. 7 Outline of the active vibration suppression method.

during feed motion is input to an autonomous compensation generation algorithm. The magnitude M_s and input timing T_d of the compensation signal are also calculated. Hence, the vibration y (i.e., the difference between X_1 and X_2) can be reduced by using the transfer functions G_1 and G_2 acquired in advance. Finally, the compensation signal is generated based on the calculation results.

4. Automatic generation method of compensation signal

4.1 Prediction of residual vibration

Prediction of the residual vibration phase and amplitude from positional command is necessary for calculating the compensation parameters and generating the compensation signal. Acceleration and deceleration times were set as the parameter of acceleration and deceleration in the control system of the machining center. The time for acceleration and deceleration was constant during feed motions. Based on the above assumption we analyzed the phase of the residual vibration when the acceleration command was a triangular wave (Fig. 8).

We define acceleration and deceleration start point T_b as the time when the acceleration command rises, the maximum acceleration point T_v as the time when the acceleration command reaches maximum, and the acceleration and deceleration end point T_e as the time when the acceleration command reduces to zero. We consider residual vibration as a superposition of vibrations occurring at T_b , T_v , and T_e , where the jerk command (derivative of the acceleration command) varies stepwise. Therefore, to calculate the residual vibration phase the vibration phase that occurred at each point is defined.

We define the positive direction of the phase delay time and the phase angle as the direction in which the phase is advanced. The phase Φ [rad] of the acceleration and deceleration time T [s] is calculated by Eq. (1) using the period T_n [s] of natural vibration that occurs by acceleration and deceleration.

$$\Phi = 2\pi(T/T_n) \quad (1)$$

In the case where the natural vibration that occurred is expressed as a sine wave without damping, the vibration Y_b , Y_v and Y_e excited respectively at the acceleration and deceleration start point T_b , the maximum acceleration point T_v , and

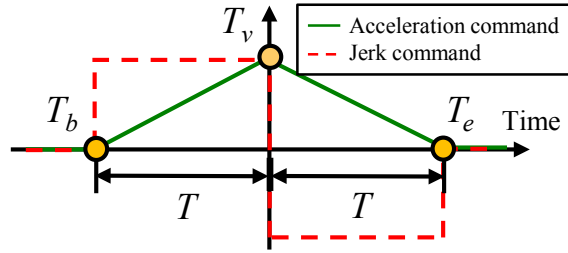


Fig. 8 Triangular acceleration and jerk commands.

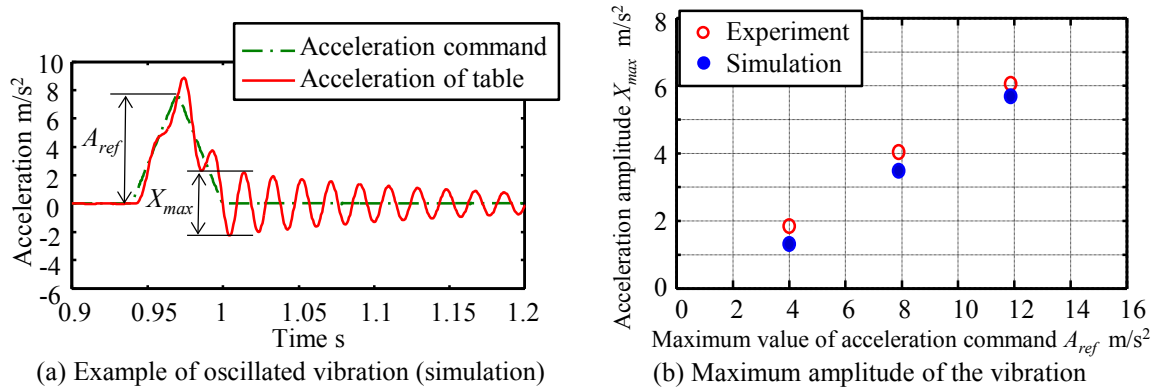


Fig. 9 Relationship between acceleration command and amplitude of vibration.

the deceleration end point T_e are expressed as:

$$Y_b = A \sin(\omega t + \Phi) \quad (2)$$

$$Y_v = 2A \sin \omega t \quad (3)$$

$$Y_e = A \sin(\omega t - \Phi) \quad (4)$$

where, A is the amplitude that occurred at the acceleration and deceleration start point, and ω is natural angular frequency.

Residual vibration X , caused by acceleration and deceleration, is expressed as the sum of Y_b , Y_v , and Y_e as:

$$X = 2A(1 - \cos \Phi) \sin(\omega t \pm \pi) \quad (5)$$

The phase of the residual vibration differs from the phase of the vibration that occurred at T_v by half a period. Therefore, a residual vibration phase can be estimated based on the acceleration maximum point T_v of the acceleration command. In order to characterize the oscillated vibration accurately it is required to analyze the vibration based on transfer functions. Practically, however, it is difficult to analyze the vibration characteristics precisely. Hence, in this study we assume that the vibration has no damping, and simple sine wave vibrations occur at each jerk changing point, with focus only on predicting the phase delay of the vibration.

We also investigate the method of predicting the amplitude of residual vibration. Where acceleration and deceleration time is constant during feed motions the resulting amplitude of vibration increases in proportion to a maximum acceleration command value A_{ref} (Fig. 5). A verification experiment confirmed that there is a proportional relationship between the maximum acceleration command value A_{ref} and the vibration amplitude caused by acceleration and deceleration.

Figure 9 (a) shows an example of the simulated vibration due to the acceleration command with X_{max} indicating maximum amplitude of the acceleration. Figure 9 (b) plots the measured and simulated values of the maximum vibration amplitude X_{max} for respective input values of reference acceleration command A_{ref} . From this relationship it is possible to predict the residual vibration amplitude based on the maximum value of the acceleration command.

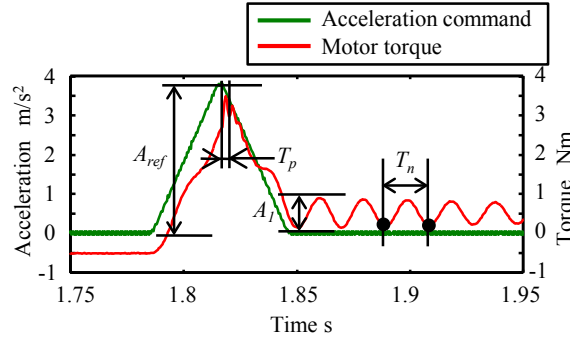


Fig. 10 Acceleration and torque commands during reference feed motion.

4.2 Signal input timing T_d

The residual vibration caused by acceleration and deceleration is in opposite phase to the vibration at the maximum acceleration point T_v (Fig. 8). Therefore, the time of maximum acceleration is based on the phase reference time T_v of the residual vibration. In this study we propose parameter determination for the compensation signal without using an external sensor, such as an acceleration sensor. The motor torque output T_m , is used for the parameter determination (Fig. 10). The parameters of the compensation signal are adjusted to cancel out the vibration that appeared on the torque output because of acceleration and deceleration. There is a phase difference T_p [s] between the acceleration command and the motor torque because of the frequency characteristics of the control system and the mechanical structure. Hence, the residual vibration X_1 that appeared on the torque output T_m can be expressed as follows, where A_m is the amplitude of torque vibration.

$$X_1 = A_m \sin \left(\omega \left(t - (T_v - T_p) \right) \pm \pi \right) \quad (6)$$

In the machine tool used in this study the vibration amplitude along the Y-axis was larger than that along the X-axis. Therefore, we identified the natural vibration along the Y-axis as the suppression target for evaluation. We calculated the phase difference T_p as follows. First, the phase characteristic between the position command y_{ref} and the motor torque output T_m was obtained by simulation. The frequency characteristics included the influence of positional and velocity control loops. In this case the Y-axis had high resonance at 50 Hz (Fig. 4 (b)). This was also reflected in the acceleration and torque waveforms (Figs. 9 (a) & 10). The phase angle at the resonance frequency φ_p was obtained from the simulated frequency characteristics. The acceleration command is the second-order differentiation of the position command, and the phase advances by 180° with respect to the position command. Therefore, we calculated the phase difference T_p between the acceleration command and the torque output of the natural vibration component 50 Hz using the natural vibration period T_n as:

$$T_p = T_n((-180 - \varphi_p)/360) \quad (7)$$

However, a phase difference T_k [s] also exists between the compensation signal and the oscillated vibration X_2 on the motor torque output T_m . X_2 for the compensation signal applied at input timing T_d with the phase difference T_k is expressed as:

$$X_2 = A_m \sin \left(\omega \left(t - (T_d - T_k) \right) \right) \quad (8)$$

T_k is the sum of the following three phase differences.

(1) T_r generated by the compensation signal due to superposition of a positive step input and a negative step input. It is calculated using the relationship between the natural vibration period set as a target for suppression and the compensation signal duration; (2) T_f produced by low-pass filter which is based on the frequency response of the filter; and (3) T_a due to frequency characteristics of control system and mechanical structure, calculated using the phase angle α [°] of the natural vibration component between the compensation torque T_{comp} and the Y-axis motor torque output T_m

obtained by the simulation as:

$$T_a = T_n(\alpha/360) \quad (9)$$

Based on the above, T_k is expressed as:

$$T_k = T_f + T_a + T_r \quad (10)$$

Compensation signal input timing T_d can be calculated by comparing Eq. (6) expressing the residual vibration X_1 caused by acceleration and deceleration and Eq. (8) representing vibration X_2 oscillated by the compensation signal, so that each vibration is in the opposite phase. Input timing T_d can be calculated as:

$$T_d = T_v - T_p + T_f + T_a + T_r \quad (11)$$

4.3 Signal magnitude M_s

It is possible to predict the amplitude of the residual vibration by reading the maximum value from the acceleration command during feed motion by recording the residual vibration amplitude when the reference acceleration command during the reference feed motion is input in advance. In addition, there is a proportional relationship between the compensation signal magnitude M_s and maximum amplitude of oscillated vibration X_{max} (Fig. 11). Hence, by recording the oscillated vibration amplitude from the reference compensation signal it is possible to determine M_s proportionally to the predicted vibration by using these relationships.

The reference feed motion was a simultaneous two-axis control reciprocating motion on the XY plane with feed rate 5000 mm/min. The reference acceleration command along the Y-axis, and the Y-axis motor torque at the point where motion direction changes during reference feed motion are shown in Fig. 10. We recorded the maximum value A_{ref} of the reference acceleration command. From the residual vibration on the motor torque we also recorded the maximum vibration amplitude A_1 and the natural vibration period T_n .

The magnitude of the reference compensation signal was M_{sr} , and the cutoff frequency of the filter was set to the same value as the natural frequency acquired during the reference feed motion. Figure 12 shows the Y-axis motor torque oscillation when the reference compensation signal was applied to the Y-axis motor torque command. From the vibration on the motor torque, the initial vibration amplitude A_2 was recorded. The vibration oscillated by the compensation signal was damped. Therefore, it is necessary to adjust the compensation magnitude T_c from the compensation input time T_d to the start of the residual vibration. The damping ratio ζ is calculated from the vibration as:

$$\zeta = 1/(2m\pi) \log(a_n/a_{n+m}) \quad (12)$$

where, a_n is the maximum half amplitude of the oscillated vibration, and a_{n+m} is a half amplitude after m periods (Fig. 12).

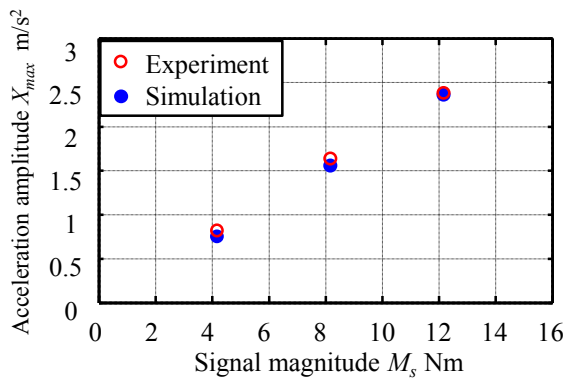


Fig. 11 Relationship between signal magnitude T_c and acceleration amplitude of vibration.

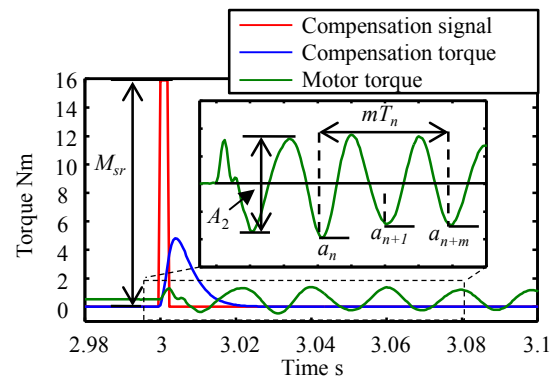


Fig. 12 Relationship between reference compensation signal and oscillated vibration.

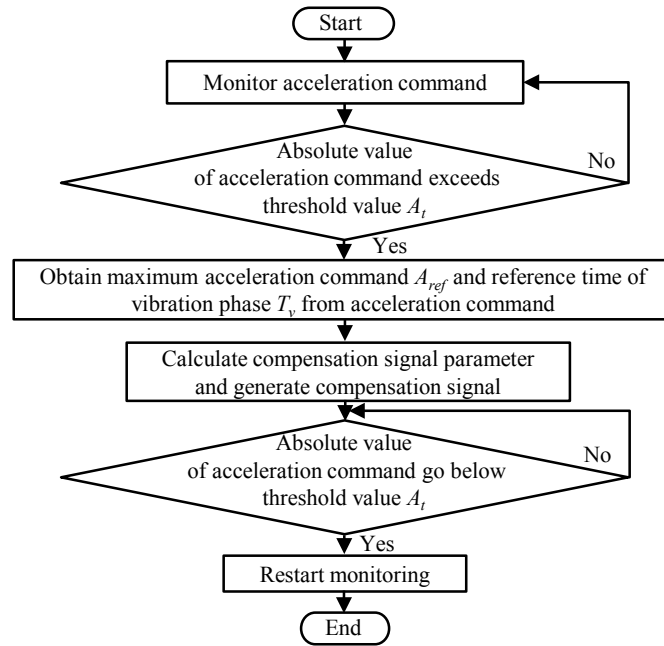


Fig. 13 Flowchart of proposed active vibration suppression system.

The vibration damping time t is defined as the time difference from the input timing of the compensation signal to the start time T_e of the residual vibration (i.e. when the acceleration command reduces to 0). Assuming the natural vibration oscillated by the compensation signal as damped free vibration the damping magnification for vibration damping time t seconds and using the natural vibration frequency F is expressed as $1/e^{-2\pi\zeta Ft}$. Thus, the magnitude M_s of the compensation signal can be calculated as:

$$M_s = A_{ref} \times (A_1/A_s) \times (T_{cs}/A_2) \times (1/e^{-2\pi\zeta Ft}) \quad (13)$$

4.4 Active vibration suppression system

In the proposed active vibration suppression system the acceleration command during feed motion is constantly monitored, and the compensation signal is generated when the acceleration command exceeds threshold value A_t . Subsequently, the maximum acceleration command value A_{ref} and the phase reference time T_v are obtained from the acceleration command. Thereafter, the magnitude M_s and the input timing T_d of the compensation signal are calculated using Eqs. (11) and (13). Finally, the compensation signal is generated automatically based on the calculated results. A flowchart of the proposed method is shown in Fig. 13.

5. Experiment and simulation

To investigate the vibration characteristics occurring at the corner we measured and simulated the acceleration command and Y-axis acceleration of the table around point B where the motion direction in the Y-axis changed. The feed speed during high-speed tracking motion was 10000 mm/min. Results indicated that 50 Hz vibration occurred on the table because of the acceleration and deceleration (see Fig. 14). This was a vibration mode in which the collapse of the column and the characteristics of the feed drive system were coupled (Sato et al., 2015). We also measured and simulated the vibration excited by applying the compensation torque during a uniform motion in the Y-axis (Fig. 15). We found that by applying the compensation torque to the torque command the same frequency vibration as that caused by acceleration and deceleration was generated. Hence, with the proposed vibration control system the vibration with desired amplitude and phase can be generated and the vibration caused by acceleration and deceleration can be canceled out by applying the compensation torque.

Low frequency changing of the acceleration can be observed in the measured values of acceleration (Figs. 14 (a) & 16 (a)). Although it is possible that this phenomenon occurs due to the frequency characteristics of the ball-screw and linear guide the actual reason could not be ascertained in this study.

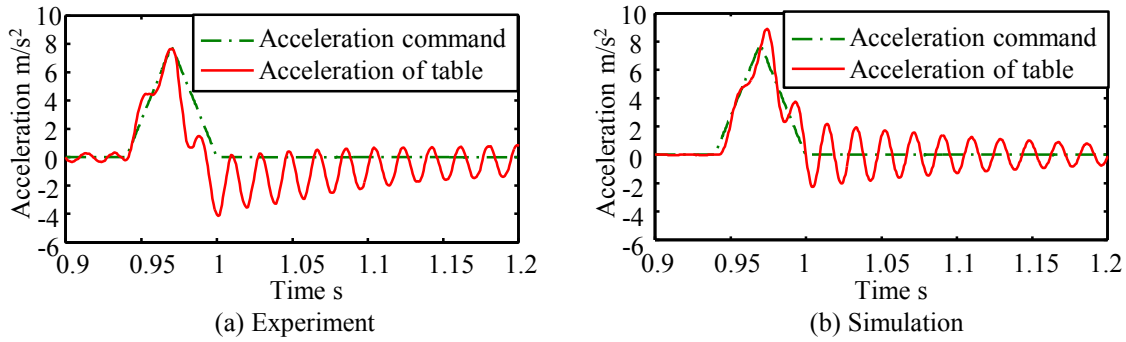


Fig. 14 Acceleration of table around corner B where motion direction of Y-axis changes.

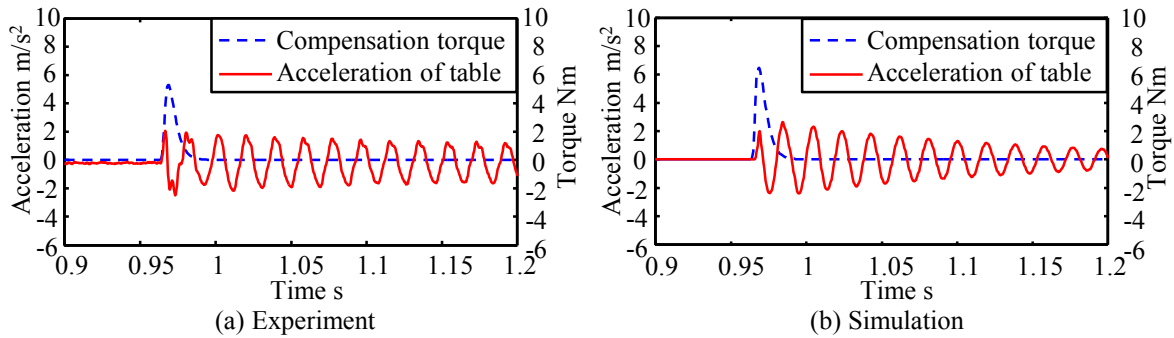


Fig. 15 Oscillated vibration by compensation torque.

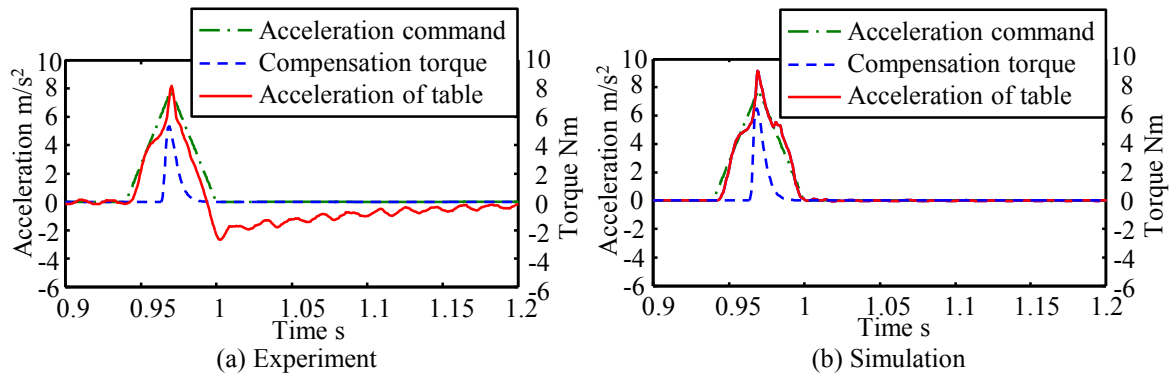
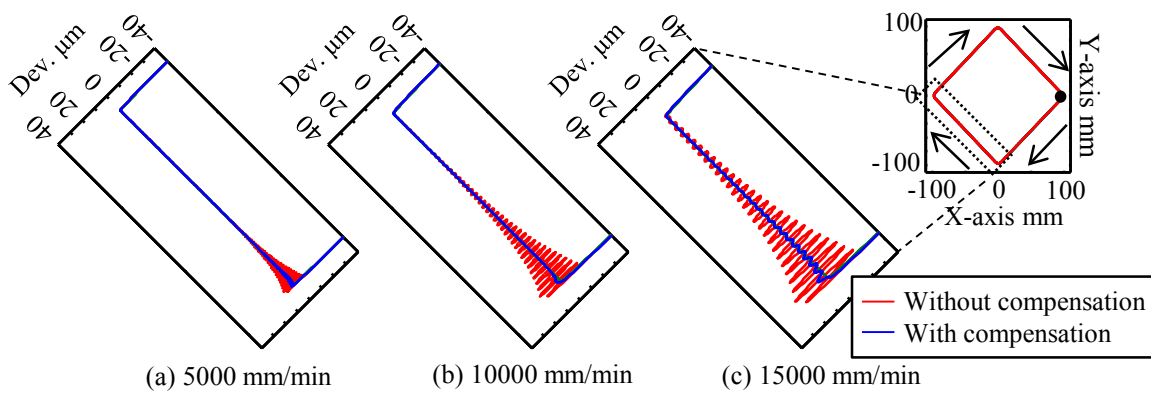
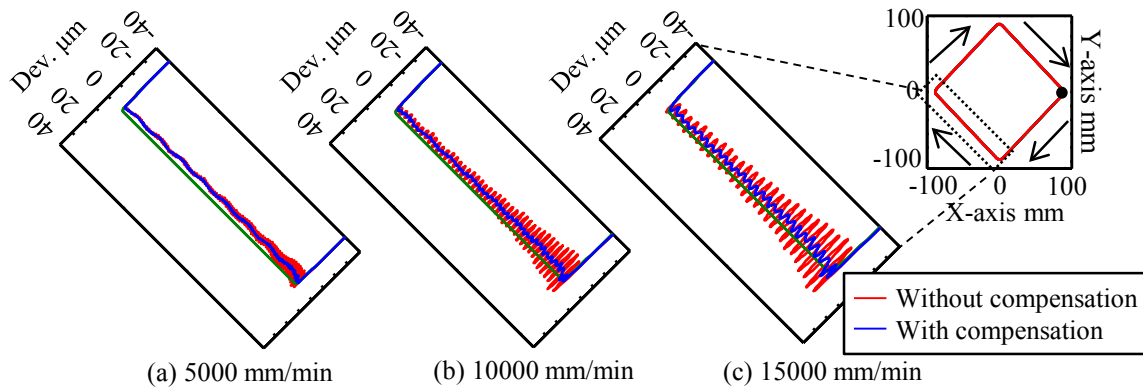


Fig. 16 Acceleration of table around corner B with proposed method.

Next, we measured and simulated the acceleration when compensation torque was applied corresponding to acceleration and deceleration (Fig. 16). Results show that vibrations caused by acceleration and deceleration, and vibrations oscillated by compensation torque cancel each other out. Thus, the vibration that occurred when the compensation torque was not applied (Fig. 14) was significantly reduced in both the experiment and simulation.

Finally, we also measured and simulated the motion trajectory by applying the proposed active vibration control method for the contouring motion (Figs. 17 & 18). To investigate the effectiveness of the proposed automatic parameter tuning method when the feed rate changed, feed rates were set to 5000, 10000, and 15000 mm/min in the experiment and simulation. As noted earlier, the Y-axis vibration oscillated at the corner, and its influence could be observed as the normal and tangential direction errors of the feed motion. Since the normal directional errors are copied onto the machined workpieces, Figures 17 and 18 show only the normal directional errors. Since the X-axis kept a constant velocity it was not oscillated at the point.

According to the results, in the case where the proposed system was not applied the vibration oscillated because of the motion direction change in the Y-axis with large mass. However, in the case where the proposed system was applied, the amplitude of the vibration was significantly reduced in the experiments and simulations. This confirmed that the proposed active vibration control method can suppress the vibration during high-speed contouring motions. It was also confirmed that the vibration can be suppressed effectively, even if the feed rate is changed.



6. Conclusions

In this study we proposed an active vibration suppression method for high-speed contouring motions without any additional actuators or sensors for the control. In this method, mechanical vibration was successfully suppressed by applying a compensation torque to cancel the mechanical vibration during high-speed contouring motion. In this method parameters of the compensation signal were determined using commanded acceleration and time for acceleration and deceleration, which were set as NC parameters. An autonomous compensation torque generation algorithm based on the acceleration command from NC was also proposed. To evaluate the effectiveness of the proposed method rectangular corner contouring motions when applying the vibration compensation torque were analyzed. Analysis confirmed that this method can suppress the vibration during high-speed contouring motions. It was also confirmed that vibration on the trajectory can be suppressed effectively, even if the feed rate is changed.

Since the proposed method does not involve any changes of positional commands or dynamic characteristics of the axis it is potentially an effective technique to suppress the vibration during high-speed contouring motions. However, the proposed method can only be applied in cases where the desired vibration can be oscillated by the impulse torque signal. Also, the acceleration command has to be the jerk limited command to apply the method. The authors plan to investigate the above two limitations by conducting further studies in future.

The key component of the proposed method is the input timing of the compensation signal, although the amplitude and frequency of the oscillated vibration by the compensation signal can automatically be matched with those of the vibration that requires to be suppressed. The authors also plan to develop a method that can be adjusted according to the natural vibration frequency of NC machine tools, because the changes in the natural frequency influence the optimum timing to input the compensation signal.

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