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Paper:

Analysis of the Coupled Vibration Between Feed Drive Systems and Machine Tool Structure

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In this study, we have constructed a mathematical model that can analyze the coupled vibration of machine tool structure and feed drive systems. The model is proposed on the basis of the modal analysis of the actual machine tool structure. It consists of three translational and three rotational displacements of the bed, relative angular deformations between the bed and column, relative translational and angular deformations between the bed and saddle, and relative translational and angular deformations between the column and spindle head. In addition, each feed drive system is modeled using a vibration model, which has two degrees of freedom. The servo controllers of each axis are also modeled. To confirm the validity of the proposed model, frequency responses, motion trajectories of the feedback positions, linear scale positions, and the relative displacement between the table and head are measured and simulated. The effect of coupled vibrations on the tracking errors is examined with the help of both experiments and simulations. To investigate the effect of the servo systems on the vibration, both experiments and simulations are carried out by using feed drive systems in the following three conditions: mechanically clamped, servo-on, and servo-off. The results of experiments and the simulations show that the proposed model can express the mode of vibration and the influence of the condition of feed drive systems on the mode of vibration.

Keywords: machine tool structure, feed drive system, coupled vibration, frequency response, tracking error

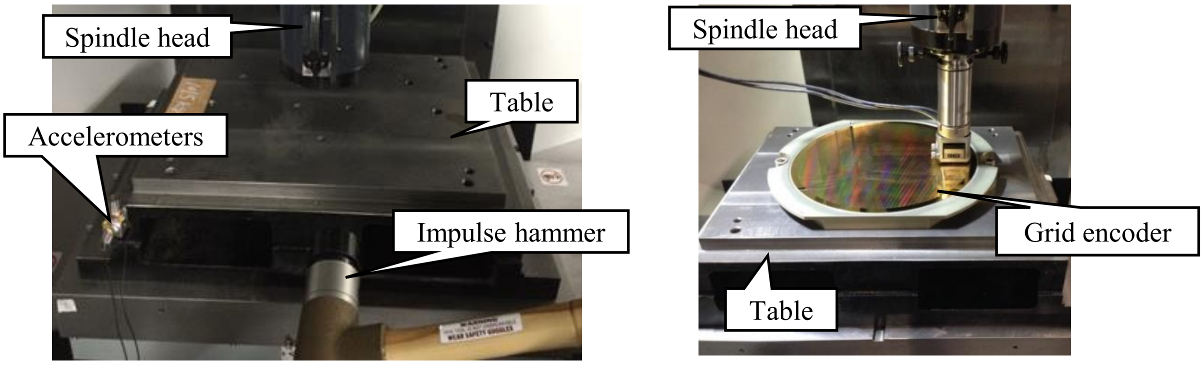
1. Introduction

Because the relative motion trajectories of the tool and workpiece are directly copied onto the workpiece geometry, higher motion accuracy is always required to achieve higher quality and productivity of the products. The most important factor affecting the accuracy of machine tools is the geometric accuracy of the machines, such as the straightness of the axis, positioning accuracy of the axis, squareness of the axes, etc. [1, 2]. These accuracies mainly depend on the accuracy of the mechanical components of the machine itself and its assembling accuracies. The geometric accuracy determines the accuracy of the relative motions between the tools and workpieces, regardless of whether they are undergoing low-speed or high-speed motion. Hence, there are many works of research regarding the evaluation of geometrical accuracy and compensation technologies [3].

Another important factor affecting the motion accuracy of NC machine tools is the feed drive system. This is because the axial motion of the moving axes in the machine is dominated by the feed drive system. From this point of view, many works of research concerning the modeling and control of feed drive systems have been presented [4]. One of the authors also proposed a feed drive simulator that could predict the motion accuracy of the systems [5] and controller tuning techniques that could improve the motion accuracies [6, 7]. The authors [8–10] also assessed how the motion error of feed drive systems influenced machined surfaces. The motion accuracy of the feed drive systems depends on the feed rate of the axes because the dynamics of the system determine the accuracy while undergoing contouring motion.

Currently, contouring motions of a considerably higher speed are required to the NC machine tools in the EMS (Electronics Manufacturing Service) companies that produce small parts for IT products such as smart phones, laptop PCs, and tablets. They require to reduce the production cost of the parts by reducing the processing time while maintaining the accuracy of machined parts. Even though higher-speed tracking motion is achieved, higher-precision tracking motion accuracy is always required to NC machine tools.

However, the motion accuracy is deteriorated because of the machine tool dynamics in high-speed tracking motion. The accuracy of the relative motion between the tool and workpiece depends not only on the motion accuracy of feed drive systems, but also on the structural vibration of the machine tools, such as bed, column, and saddle, which oscillate when acted upon by inertial forces. The inertial forces are induced when all the feed drivers located in the table, spindle head, and saddle are accelerated or decelerated by the servo systems.



(a) Frequency response (b) Relative position between table and head

Fig. 1. Measurement methods.

Rehsteiner et al. [11] analyzed the machine tool vibration under high-speed motion, and Matsubara et al. [12] investigated the influence of base vibrations on the motion accuracy. Bringmann and Maglie [13] proposed a method for evaluating the dynamic 3D motion path by using displacement sensors, and Nagaoka et al. [14] proposed a method for analyzing motion accuracy by using the synchronized measurement of tool-tip and feedback positions. A compensation method for analyzing the structural vibration on the basis of the measured data had also been proposed [15].

Simulation is known to be an effective tool for solving problems related to motion accuracy [16]. Zirn [17] explained the modeling strategies for machine tool feed drive systems, and Kono et al. compared the two modeling approaches– Finite Element Analysis (FEA) and Multi Body Simulation (MBS) [18]. They also found that the machine support influences the whole machine tool dynamics considerably [19] and tried to obtain the optimum machine support settings [20]. Apart from the research works mentioned above, many researchers have studied the structural vibration problems of the machine tools by using the Finite Element or Multi Body Analysis [21–25]. These research works mainly focused on "cross-talk" vibrations, which means the interference between the axes; however, the influence of the condition the feed drive systems which drive the axes and machine tool structure has not been studied adequately.

This study focused on the influence of the characteristics of feed drive systems on mechanical vibrations. This is an important factor in designing the feed drive system controllers. To achieve the purpose of designing the feed drive system controllers efficiently, an actual vertical type machining center is modeled by using the multibody dynamics approach, based on the modal analysis, with the number of DOF as low as possible. Feed drive mechanism models are installed in the model to analyze their own influence. The correctness of the model is verified by comparing the measured and simulated frequency responses and motion trajectories. The influence of the condition of the feed drive systems is discussed with the help of both experiments and simulations.

2. Experimental Set-up

2.1. Frequency Response

The motion characteristics of a small vertical type machining center are analyzed in this study. The machining center has three translational axes (X-, Y- and Z-axis). The work table is driven by the feed drive systems of the X- and Y-axes, and the X-axis feed drive system is attached to the Y-axis feed drive system. The spindle head is driven by the Z-axis feed drive system. Frequency responses are measured to evaluate the dynamic characteristics of the machine tool. The frequency responses are measured by using an impulse hammer and accelerometers, as shown in **Fig. 1(a)**.

The table is oscillated along both the X- and Y- axes by an impulse hammer, and the acceleration along the X-, Y- and Z-axis is measured at 48 points. Transfer functions between the table and the points can be obtained from the measured oscillation force and the accelerations. Modal analysis of the machine tool structure has also been carried out based on the measured the transfer functions between the table and each measurement point.

2.2. Motion Trajectory

Motion trajectories during corner tracking are measured by a grid-encoder, as shown in **Fig. 1(b)**. The grid encoder can measure the two-dimensional relative displacement between the table and the spindle head in the X–Y plane with a resolution of 1 nm.

The feed drive systems of the machining center considered in this study have a semi-closed control loop that is based on the obtained rotational angle of the motor. The obtained rotational angle of the motor can be converted to a positional dimension. These rotational angles that are converted to positional dimensions are called "feedback (FB) positions" in this study. Although the axes are controlled by semi-closed loop control systems, the relative displacement between the table and the saddle is measured by a linear encoder attached to the table, and the displacement between the saddle and the bed is measured by a linear encoder attached to the saddle. These relative

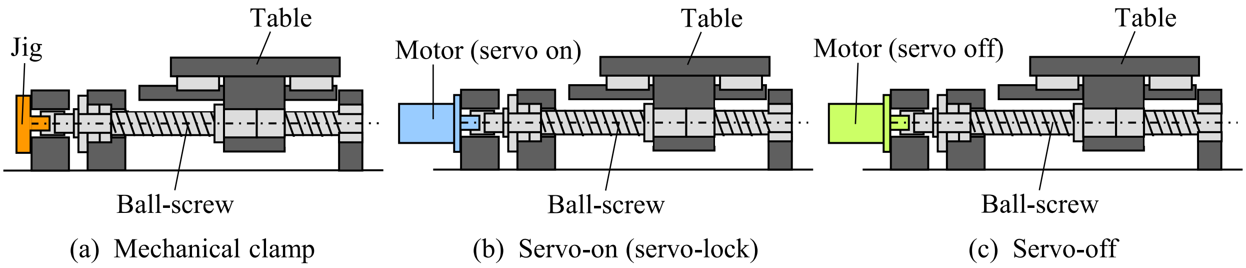


Fig. 2. Condition of servo derive systems.

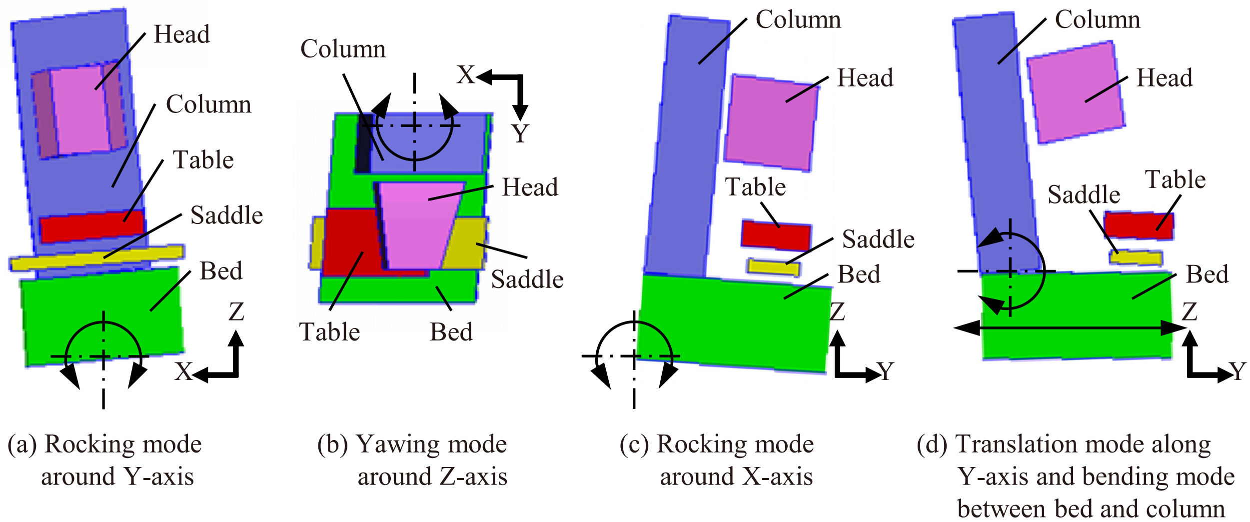


Fig. 3. Vibration mode of the machine tool structure.

displacements are called "linear scale positions" in this study. Both the feedback and linear scale positions are acquired by using a monitoring function provided in the controller.

2.3. Feed Drive System Condition

The feed drive system consists of a feed drive mechanism and a controller. The feed drive mechanism consists of an AC servomotor and a ball-screw in the machining center. The axial motion of each axis is guided by a set of linear ball guides.

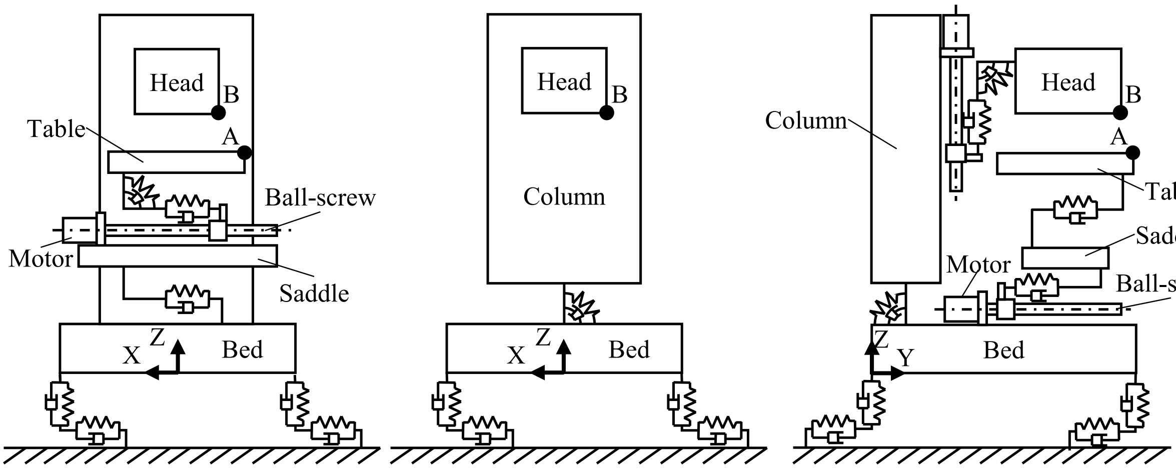
To investigate the influence of the feed drive systems on the vibration mode, both experiments and simulations are carried out by using feed drive systems in these three conditions: mechanically clamped, servo-on, and servo-off. **Fig. 2(a)** describes the mechanically clamped condition with a jig instead of the motor. In this case, the rotational motion of the ball-screw is mechanically clamped. **Figs. 2(b)** and **(c)** describe the feed drive system in the servo-on and off conditions, respectively. In these conditions, the motor is attached to the mechanisms. In the servo-on condition, the servo system is turned on. The rotational angle of the ball-screw is clamped by the feedback control system. Whereas, in the servo-off condition, the rotational angle of the ball-screw is not clamped.

3. Mathematical Model

3.1. Machine Tool Structure

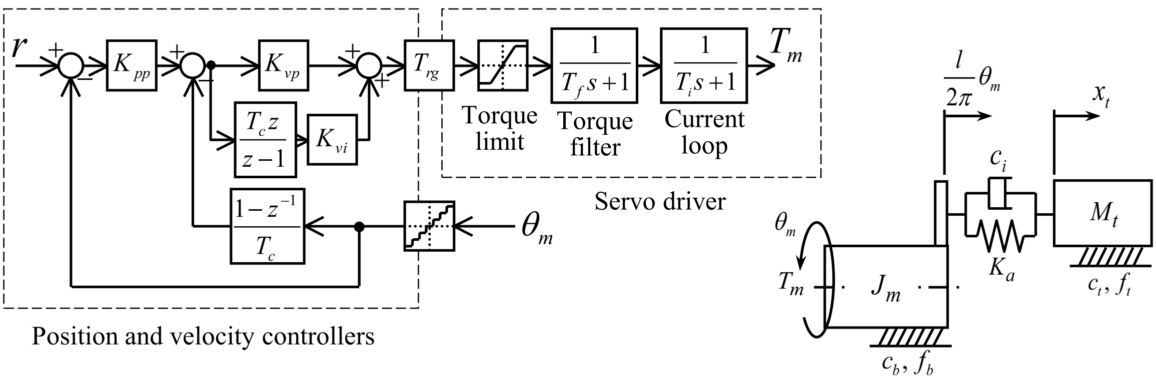
Figure 3 shows the analyzed vibration mode of the machine tool structure in the case of mechanically clamped feed drive mechanisms. According to the result of the modal analysis, the machine has a rocking mode along the Y- and X-axes at 30 Hz, yawing mode along the Z-axis at 70 Hz, and translation mode along the Y-axis of the bed and bending mode between the bed and column at 60 Hz. Based on the results, a mathematical model is proposed, as shown in **Fig. 4**. The model consists of a bed, saddle, table, column, and head. The stiffness of linear guides is also introduced between the bed and saddle, and the saddle and table; the stiffness along each axis between base and ground are also introduced. In addition, the torsional rigidities between bed and saddle, bed and column, and column and head are considered.

The mass and inertia of all the mechanical components can be obtained from the designed CAD models of each component. Translational and rotational stiffness between the components are estimated based on previous research works [26, 27]. However, since it is difficult to estimate the stiffness accurately, the stiffness is identified based on the measured frequency responses although the estimated values are applied to the model as the initial values. The damping coefficients are also determined based on the measured frequency responses.



(a) Z-X plane (b) Y-Z plane

Fig. 4. Vibration model of the machine tool structure.



(a) Controller and servo driver (b) Feed drive mechanism

Fig. 5. Mathematical model of feed drive systems [5].

3.2. Feed Drive Systems

Feed drive system models, which have two degrees of freedom, [5] are installed between the bed and the saddle (Y-axis feed drive system), the saddle and the table (X-axis feed drive system), and the column and the head (Z-axis feed drive system). The controllers and servo drivers are also introduced into the model. **Fig. 5** shows the block diagram of the controller and the vibration model of the feed drive mechanism.

In the figures, M_t is the moving mass driven by the systems, and J_m is the moment of inertia of the motor and the ball-screw. Values of these parameters can be determined based on the designed values. Axial stiffness and damping of the mechanism, K_a and c_i are identified from the measured frequency responses, and the coulombs and viscous friction parameters f_b, f_t and c_b, c_t are identified based on the measured motor torques during the feed motions. Friction forces and torques act between two components, such as saddle and table, in the X-axis feed drive mechanism. The controller models and their parameters are determined based on the model provided by the manufacturer and the setting values in the actual controller.

3.3. Block Diagram of the Whole System

Figure 6 shows a block diagram of the whole machine tool. The mechanical system has total of 19 degrees of freedom. The equation of motion can be obtained from the vibration models mentioned above, and the state equation expression is adopted in this study to conduct the simulations. The position and orientation of each mechanical component can be simulated by the model.

Feedback controllers are implemented in the block diagram on the basis of the simulated rotational angles of the motors, and the simulated motor torque by the controller and servo driver models can be input to the state equation model. Actual positional commands of each axis during the motion are acquired by using the monitoring function, and the commands are applied in the simulations.

4. Comparison of Measured and Simulated Results

4.1. Frequency Response

Figures 7 and **8** show the measured and simulated frequency responses at the measurement points A (on the table) and B (on the head) with the mechanical clamp of the

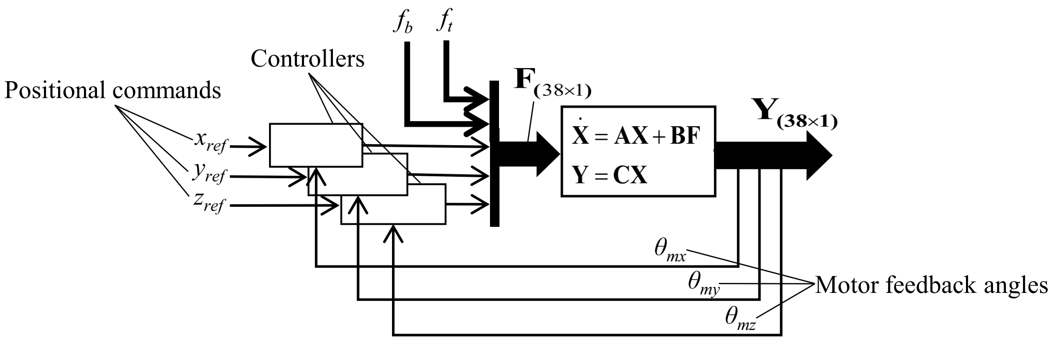


Fig. 6. Block diagram of the machine tool structure and feed drive systems.

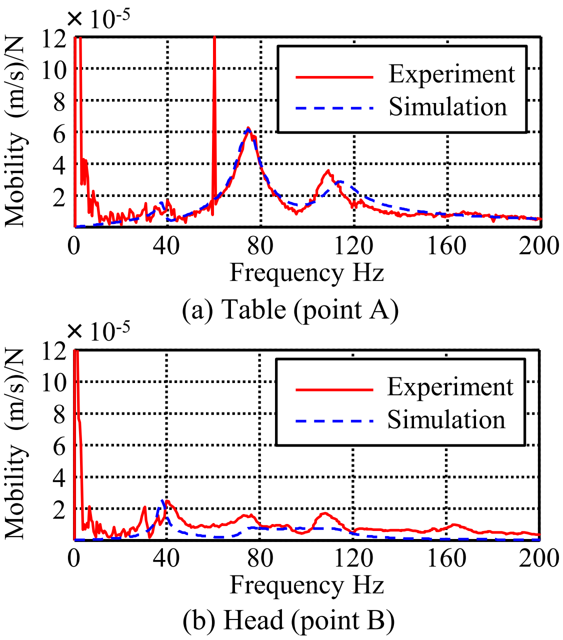


Fig. 7. Frequency responses along X-axis (mechanical clamp, excited along X-axis).

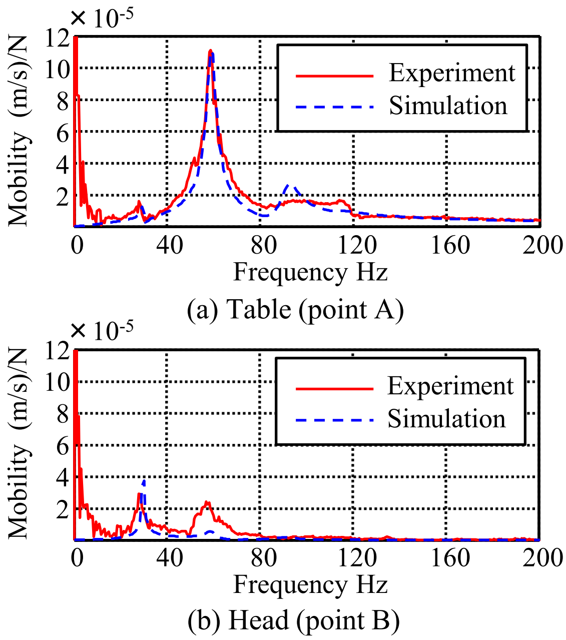


Fig. 8. Frequency responses along Y-axis (mechanical clamp, excited along Y-axis).

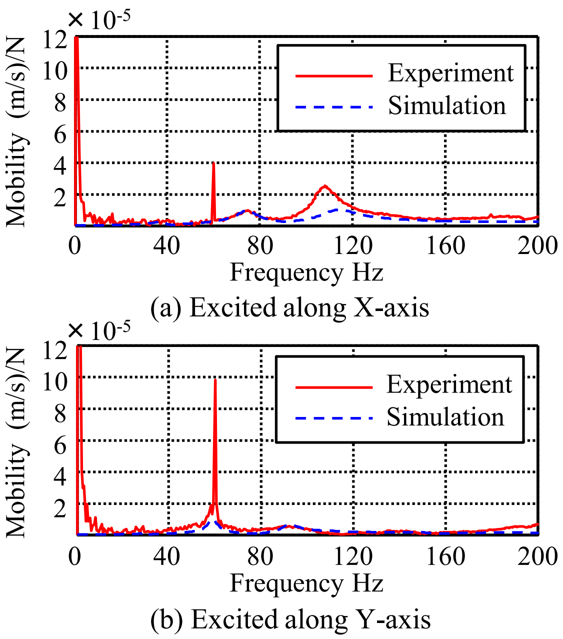


Fig. 9. Frequency responses of table (point A) along Z-axis (mechanical clamp).

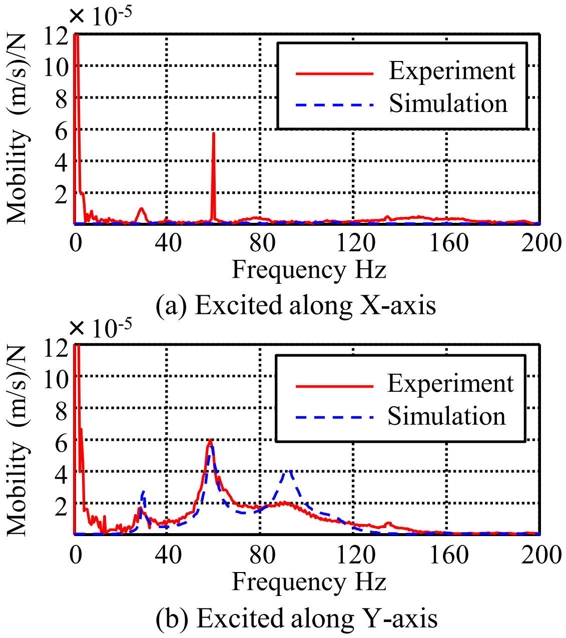


Fig. 10. Frequency responses of head (point B) along Z-axis (mechanical clamp).

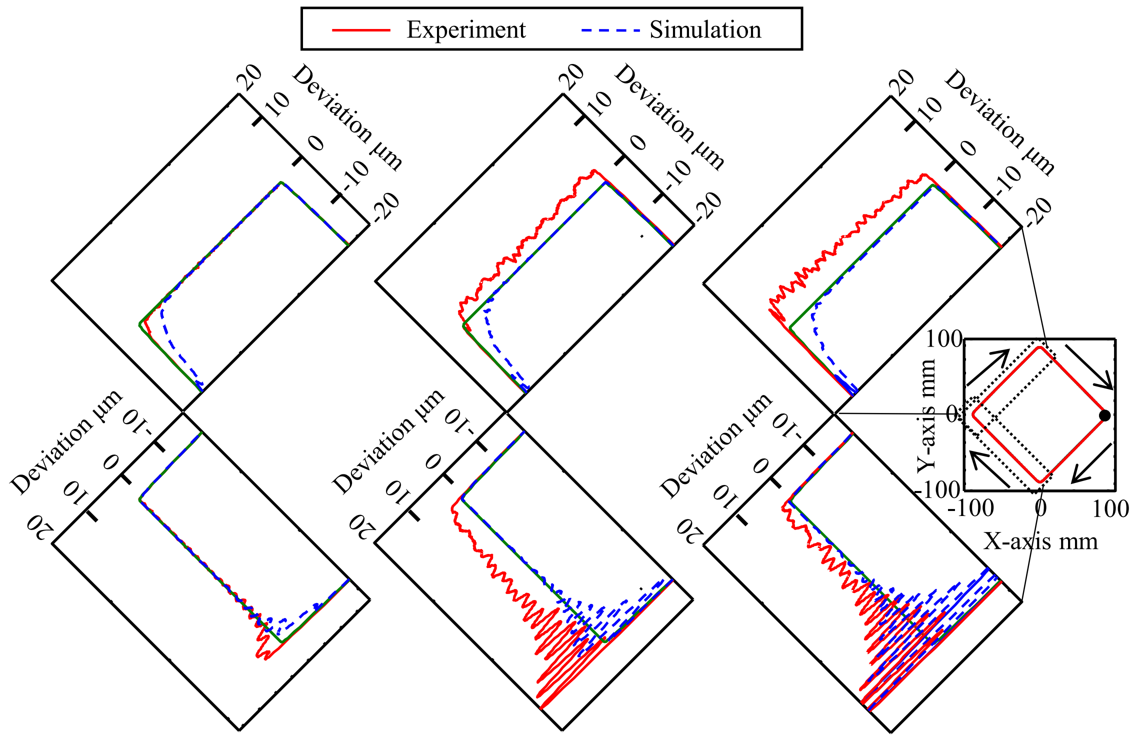


Fig. 11. Comparison of measured and simulated motion trajectories (Feed rate: 20000 mm/min).

case table hammered along the X- and Y-axes. From the measured and simulated results at point A, vibration peaks around 30 Hz, 70 Hz, and 110 Hz can be observed along the X-axis, and around 30 Hz, 60 Hz, and 90 Hz along Y-axis. The vibrations around 30 Hz along both of X- and Y-axes are the rocking mode of the whole machine, and the vibration around 70 Hz along the X-axis is the yawing mode around the Z-axis. The vibration around 60 Hz along the Y-axis is the bending mode between bed and column around the X-axis. In addition, vibrations around 110 Hz and 90 Hz are due to the axial stiffness of feed drive mechanisms.

It can also be observed from the figures that vibrations along the Y-axis around 60 Hz is the largest when the table is oscillated in the X–Y plane, and the vibrations of the head are relatively smaller than the vibrations of the table. Please note that the sharp peak observed at 60 Hz was caused by noise from the power line.

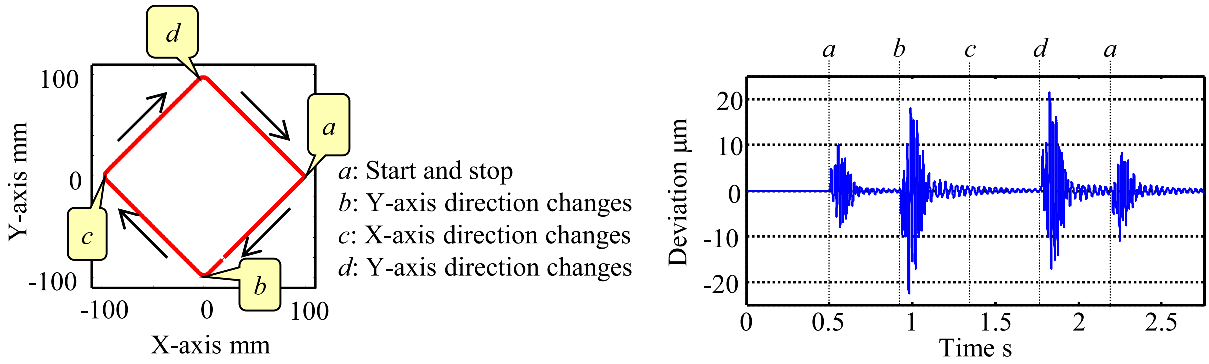
Although the motions are only in the X–Y plane, vibrations along the Z-axis occur and the vibrations influence the motion accuracy. **Fig. 9** shows the measured and simulated frequency response at measurement point A, and **Fig. 10** shows the frequency response at measurement point B, in cases where the table is hammered along the X- and Y-axes. It can be seen that the table does not oscillate much along the Z-axis although the head oscillates in the case where the table is hammered along the Y-axis. On the other hand, both the table and head do not vibrate much when the table is hammered along the X-axis.

The results show cross-talk vibrations exist between Y- and Z-axis in the machine evaluated in this study. It is also confirmed that the proposed model can adequately simulate the frequency response of the machine tool structure.

4.2. Corner Tracking Motion

Figure 11 shows the measured and simulated motion trajectories of the corner tracking motion. **Fig. 11(a)** shows the trajectory of feedback positions; **Fig. 11(b)** shows the trajectory of linear scale positions; and **Fig. 11(c)** shows the trajectory of relative displacements between the table and head measured by the grid encoder. Although the paths offset by several micrometers can be observed in **Figs. 11(a)** and **(b)**, these errors have not been discussed in this paper because the errors were caused from the pitch error of the ball-screws.

Focusing on the dynamic behavior, it is clear from the figures that larger vibrations occur at the relative displacement around the corner as shown in **Fig. 11(c)**. This result indicates that the vibration originated from the structural vibration of the machine. The vibration along the trajectory increases when the direction of motion of the Y-axis changes. It is expected that the vibration along the trajectory due to the vibration frequency occurs around 60 Hz. The vibration along the trajectory is mainly due to the bending mode between the bed and column around the X-axis (**Fig. 3(d)**). This vibration mode has the largest amplitude as shown in **Figs. 7–10**. Abbe offset between the rotational center of the vibration and the tool center



(a) Motion path (b) Simulated vibration along Z-axis

Fig. 12. Influence of Y-axis motion direction changes to Z-axis vibration.

point results in vibrations of larger amplitude.

On the other hand, clear vibration around the corner cannot be observed when the direction of motion of the X-axis changes. Because of this, the main vibration mode when the table is excited along the X-axis is the rocking mode of the entire machine structure around the Y-axis (**Fig. 3(a)**). This implies that the rocking mode of the entire machine structure does not cause the relative vibrations between the tool and workpiece.

Three-dimensional motion errors occur during the tracking motions because of cross-talk vibrations in the machine. **Fig. 12(a)** shows the motion path and **Fig. 12(b)** shows the simulated vibrations along the Z-axis during the corner tracking motion. The result is shown in the time domain. The results show that vibrations along the Z-axis occur at points a, b, and d where the direction of motion of the Y-axis changes. On the other hand, vibrations along the Z-axis do not occur at point c where the direction of motion of the X-axis changes. This implies that the magnitude of frequency response between the X-axis force and Z-axis vibrations is smaller than the cross-talk between Y- and Z-axes vibrations as shown in **Figs. 9** and **10**.

It can be concluded from the results that the coupled vibration between the axes causes three-dimensional vibrations between the tool and workpiece, and the proposed mathematical model can adequately simulate the behaviors.

5. Influence of Feed Drive System Condition

Figures 13 and **14** show the measured and simulated results of the frequency response along the X- and Y-axes on the table (point A). It can be seen that the frequency response depends on the condition of the feed drive systems. Resonance peaks around 70 Hz and 60 Hz increase in size when the ball-screw is clamped mechanically. Whereas, resonances around 60, 70, 100 and 120 Hz cannot be observed for the servo-off condition in both measured and simulated results. Because of this, the excited force on the table cannot be transmitted to the machine tool structure in the servo-off condition.

It is confirmed from the results that the condition of the feed drive systems influences the whole vibration mode of the machine tool structure, even though the vibration mode is not vibration came from the controlled displacements.

This fact indicates that the mechanical vibration of the machine tool structure can be controlled even though the vibration mode is not vibration came from the controlled displacements.

6. Conclusions

This study describes a mathematical model, which can simulate the coupled vibration of the machine tool structure and feed drive systems. The influence of the frequency characteristics on the motion trajectory is also investigated by the model. The conclusions can be summarized as follows:

1) The proposed model can adequately simulate the frequency response of the machine tool structure and motion trajectories.

2) The coupled vibration between the axes yields three dimensional vibrations between the tool and workpiece.

3) The condition of the feed drive systems influences the whole vibration mode of the machine tool structure, even though the vibration mode is not vibration came from the controlled displacements.

The proposed model can analyze the coupled vibrations between the feed drive systems and mechanical structure. It is confirmed in this study that the total optimization of the mechanical and controller design is required to achieve the optimum design because the control system and mechanical structure influence each other.

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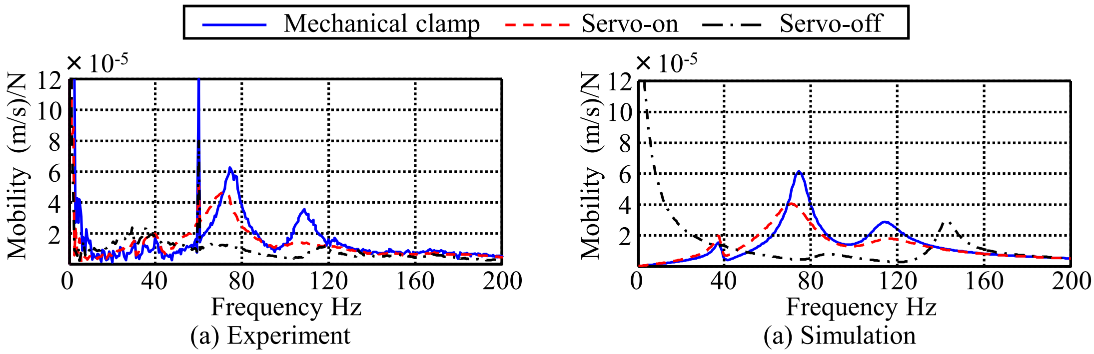

Fig. 13. Influence of feed drive system condition on frequency response of table along X-axis (Excited along X-axis).

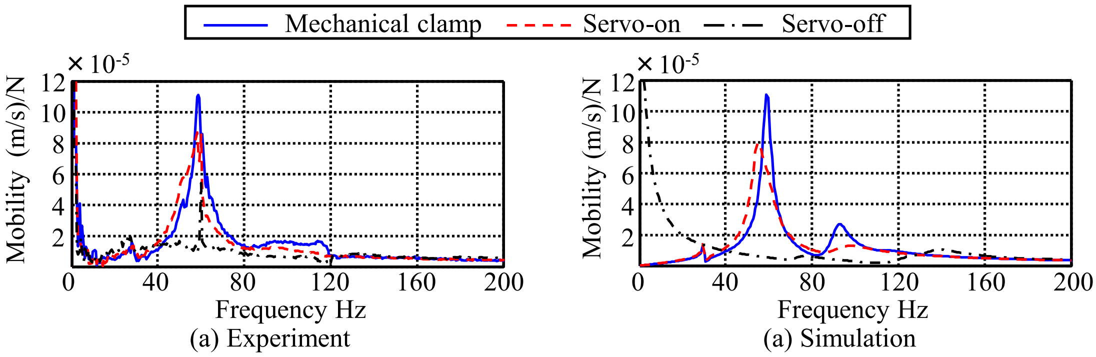

Fig. 14. Influence of feed drive system condition on frequency response of table along Y-axis (Excited along Y-axis).

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• "Generation Mechanism of Quadrant Glitches and Its Compensation of Feed Drive Systems for NC Machine Tools," Int. Journal of Automation Technology, Vol.6, No.2, pp. 154-162, 2012.

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• "Mathematical Model of Machine Tool Dynamics for Analyzing Coupled Vibration of Feed Drive Systems and Structural Configuration," Proc. of the 5th Int. Conf. of Asian Society for Precision Engineering and Nanotechnology (ASPEN2013), No.1270, 2013.
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Main Works:
• T. Shimada, K. Nakamoto, and K. Shirase, "Machining Strategy to Adapt Cutting Conditions under Digital Copy Milling Concept," Journal of Advanced Mechanical Design, Systems, and Manufacturing, Special Issue on Advanced Manufacturing Technology, Vol.4, No.5, pp. 924-935, 2010.
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• Society of Manufacturing Engineers (SME)
• Japan Society of Mechanical Engineers (JSME)
• Japan Society for Precision Engineering (JSPE)