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Paper:

Machine Bed Support with Sliding Surface for Improving the Motion Accuracy

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The purpose of this study is to develop a new machine bed support mechanism for reducing the vibration generated during the high-speed tracking motion of numerical control machine tools. In order to achieve this, the frequency response and motion trajectory of a machine tool with the proposed machine bed, which has a sliding surface, are measured and compared with that of the conventional support. Based on the modal analysis of the machine tool structure, a mathematical model representing the influence of the machine bed characteristics on the vibration is also developed. The model consists of a bed, saddle, table, column, and spindle head. Every component has three degrees of freedom for each of the translational and rotational axes. In order to evaluate the characteristics of the machine bed, the mathematical model determines the stiffness and damping along the X-, Y-, and Z-axis between the bed and the ground. The frequency response curves simulated by using the mathematical model are compared with that of the measured ones. From the results of the experiments and simulations, it is confirmed that the vibration generated during high-speed tracking motions can be reduced by using the proposed machine bed with a sliding surface.

Keywords: NC machine tools, machine bed, motion accuracy, tracking motion, vibration, mathematical model

1. Introduction

In the manufacturing industry, a numerical control (NC) machine tool is one of the important facilities for performing machining operations. In recent years, companies manufacturing components for the IT industry (electronic manufacturing services) replaced metal stamping with cutting because of its high flexibility and accuracy in shape generation. In order to improve the productivity and quality of machining processes, machine tools with high tracking speed and accuracy are required. However, the maximum speed of a machining tool is limited because, at higher speeds, the inertial force induces mechanical vibrations decreasing the accuracy of motion of the tool. In particular, the inertial force excites a rocking vibrational mode with translational and rotational vibrations of the entire machine structure at a low frequency [1, 2]. Because the structural vibration deteriorates the accuracy of motion of machine tools, it is necessary to suppress the vibrations at low frequencies.

Several research works have been reported on the design of machine tools and the mechanical vibration of the spindle head and table [3-5]. It has been clarified that the machine bed of the machine tools has a significant influence on the mechanical vibration [6-11]. The estimation and tuning strategy of the stiffness of machine beds have been investigated by Kono et al. [6-8]. The placement of machine beds for machine tools has been studied using a stiffness model [8, 11]. In addition, a structural analysis of machine tools using finite element models has been carried out [12-18].

Although Mori et al. [19] investigated the effect of machine beds with additional dampers on the frequency response of machine tools, no research works focused on the reduction of vibration during high-speed tracking motions of machine tools by changing the machine beds.

This study proposes a new machine bed support mechanism with a sliding surface that can reduce the vibration generated during high-speed tracking motions. A mathematical model representing the influence of the characteristics of the support mechanism on the mechanical vibration is also developed. In order to evaluate the effectiveness of the proposed model, the frequency response curves, displacement of the bed, and motion trajectories are simulated and compared with that of the measured values.

2. Experimental Method

2.1. Machine Bed

A vertical machining center is used in this study. The total mass of the machine is approximately 2000 kg. The machine is supported by four machine beds. This study tests two types of machine beds shown in **Fig. 1**.

Figure 1(a) illustrates the conventional machine bed comprising a leveling bolt and a support. The proposed

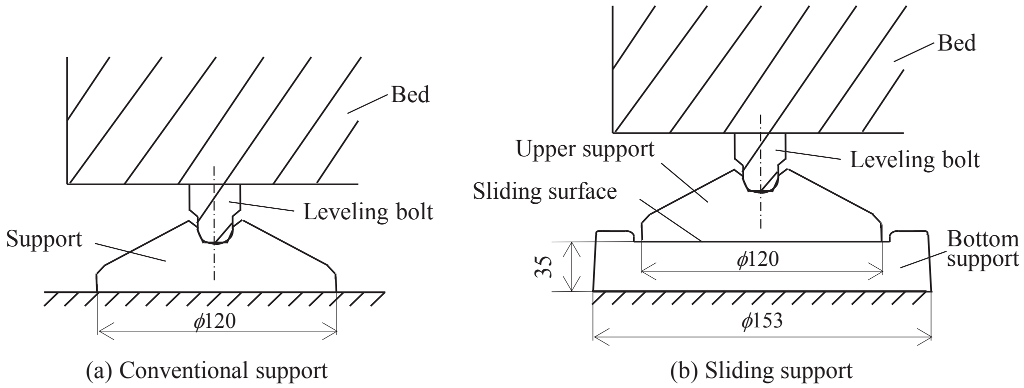


Fig. 1. Machine bed support.

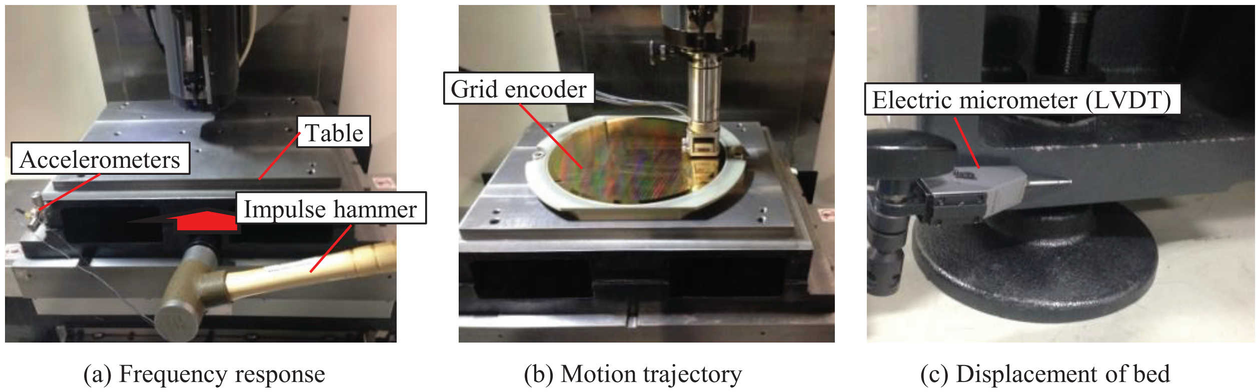


Fig. 2. Measurement method.

machine bed with a sliding surface is shown in **Fig. 1(b)**.

The proposed machine bed consists of upper and bottom supports; the upper support is made of carbon steel (JIS S45C), and the bottom support is made of cast iron (JIS FC250). The sliding surface between the two supports is lubricated using a typical lubricant oil, and the surface roughness of the sliding surface is maintained at approximately 1 μm. Thus, when the machine tool moves at a high speed, the sliding surface can slide along the X–Y plane. The diameter of the upper support is 120 mm, and the clearance between the upper and bottom supports is 5 mm. Thus, the sliding stroke is 10 mm in the X and Y directions.

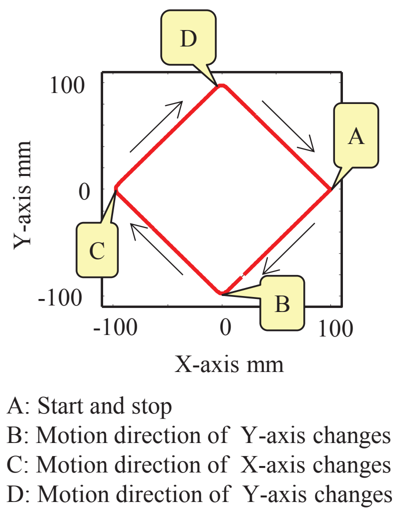


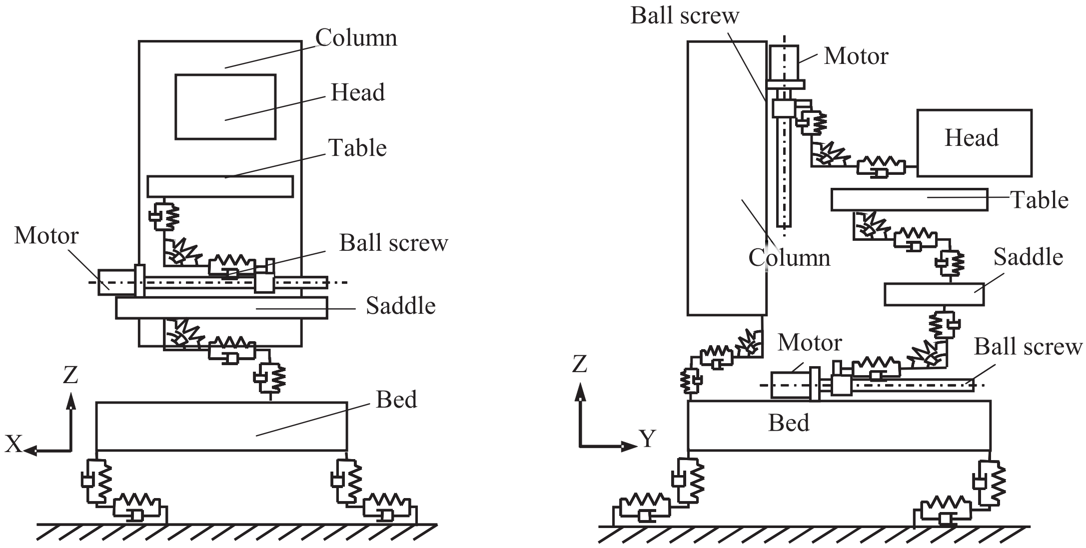
Fig. 3. Motion path for the tests.

2.2. Hammering Test

In order to analyze the vibration mode and frequency response of the machine tool structure, the table is oscillated along the X- and Y-axis by using an impulse hammer as shown in **Fig. 2(a)**. Accelerations at 48 points along the X-, Y-, and Z-axis are measured by using accelerometers. The frequency response of the exciting force and the acceleration is calculated by using a FFT analyzer. The frequency response is indicated in terms of the compliance in order to observe the vibration characteristics around 100 Hz. The measurement frequency range is set as 0–200 Hz. The impulse tests are repeated five times at each point, and the averaged data is used for the analysis.

2.3. Motion Trajectory and Displacement of Bed

In order to evaluate the effectiveness of the proposed support mechanism, the motion trajectory and the displacement of the bed during the tracking motion are measured as shown in **Figs. 2(b)** and **(c)**. The motion trajectory of the table and spindle head is measured by using a grid encoder (KGM 182, Heidenhain GmbH). The measurement system can measure the two-dimensional rela-



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

Fig. 4. Dynamics model of machine tool structure and feed drive mechanism.

tive positions between the table and the spindle with a resolution of 1 nm. In the case of the proposed machine bed with a sliding surface, the axial displacements of the bed along the X- and Y-axis are measured during high-speed tracking motions by using an LVDT as shown in **Fig. 2(c)** in order to confirm that the bed slides along the X–Y plane.

Figure 3 illustrates the motion path for measuring the accuracy of motion in tracking the corners. This motion path is a diamond-shaped path with a diagonal length of 200 mm. A simultaneous motion along the X- and Y-axis is required for this motion path. The motion starts and ends at point A. The direction of motion of the Y-axis changes at points B and D, and the direction of motion of the X-axis changes at point C. Therefore, the direction of motion changes only at corners B, C, and D. The maximum acceleration at these corners is approximately 15 m/s^2 and the feed speed is 20000 mm/min.

3. Simulation Method

3.1. Mathematical Modeling

In this study, in order to evaluate the influence of the machine bed on the motion characteristics of the machine tool, a simulation is performed based on a mathematical model. Based on the modal analysis of the machine tool structure, a mathematical model representing the influence of the machine support on the motion characteristics of the machine tool is developed as shown in **Fig. 4**.

This model consists of a bed, saddle, table, column, and spindle head and considers the torsional vibration of the bed and the motion characteristics of the feed drive systems [20, 21]. Every component has three degrees of freedom for each of the translational and rotational axes. A feed drive mechanism with friction is installed between the saddle and the table, the saddle and the bed, and the column and the spindle head. The equation of motion can be obtained from the dynamics model shown in **Fig. 4**, and the state equation can be obtained from the equation of motion.

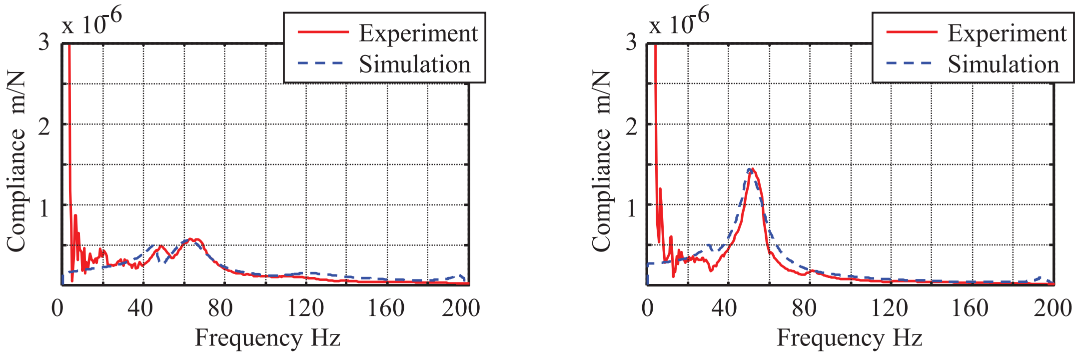
In order to consider the characteristics of the machine bed, this model considers the stiffness and damping along the X-, Y-, and Z-axis between the bed and the ground. Thus, it is assumed that the difference in the characteristics of the machine bed can be expressed by changing the stiffness and damping.

3.2. Parameter Identification

In the mathematical model, the mass and inertia of the mechanical components of the machine tool can be determined by using the specifications and a CAD model of the machine tool, and the stiffness and damping are obtained from calculations [20, 21]. The torsional stiffness of the bed and column is estimated by referring to the polar moment of inertia calculated using the CAD model. In addition, the stiffness and damping of the machine bed are determined based on calculations by considering the center of gravity of the machine obtained from the CAD model.

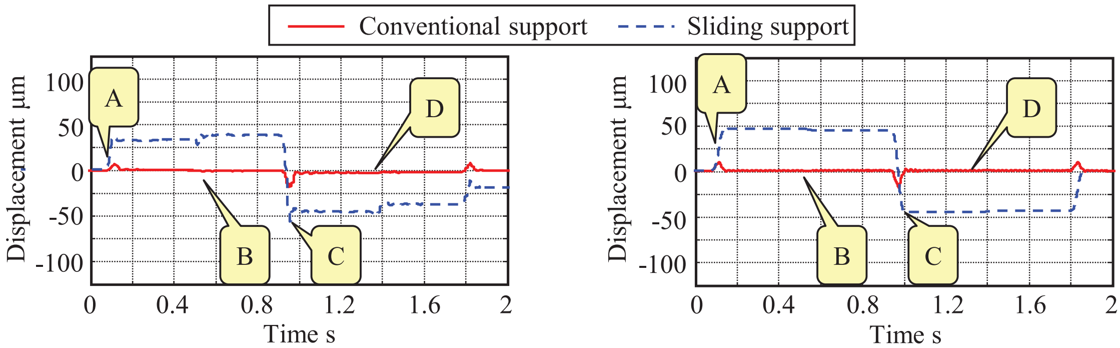
In order to compare the simulated and measured results, the stiffness and damping are identified based on the measured frequency response. The calculated values are used as the initial values of identification. The stiffness and damping are identified using a trial-and-error method by fitting the frequency response curves.

Figure 5 shows the comparison of the measured and simulated frequency response of the table in the case of the conventional support. It can be observed that the model with the identified values can express the frequency response along the X-axis around 65 Hz and along the Y-axis around 50 Hz. In the direction of the X-axis, the vibration around 65 Hz is the vibration mode with the entire machine rotating around the Y-axis (rocking mode).



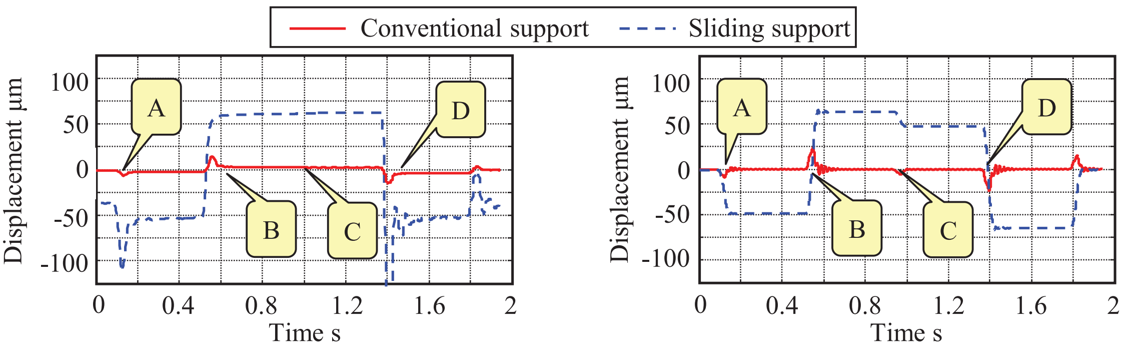
(a) X-axis direction. (b) Y-axis direction.

Fig. 5. Comparison of frequency responses of table in the case of the conventional support installed.



(a) Experiment (b) Simulation

Fig. 6. Comparison of displacement of bed along X-axis.



(a) Experiment (b) Simulation

Fig. 7. Comparison of displacement of bed along Y-axis.

In the direction of the Y-axis, the vibration around 50 Hz is the vibration mode with the column rotating around the X-axis. It is confirmed that those vibration modes affect the accuracy of motion of the machine tool, and the proposed mathematical model can simulate the vibration modes adequately. It is also confirmed that the vibration modes are not influenced by the proposed sliding support.

4. Effect of the Proposed Machine Bed Supports

4.1. Displacement of Bed

Figures 6 and **7** illustrate the measured and simulated displacements of the machine bed along the X- and Y-axis during the high-speed tracking motion. The results shown in **Figs. 6(a)** and **7(a)** indicate that, in the case of the proposed machine bed with a sliding surface, the machine bed moves approximately 80 μm along the X-axis at point C; here, the direction of motion of the X-axis changes, and the bed moves approximately 110 μm along the Y-axis at points B and D at which the direction of motion of the Y-axis changes.

On the other hand, in the case of the conventional support, the machine bed moves approximately 10 to 20 μm at the points where the direction of motion changes and moves back to the original position after 0.1 s approximately. From the results, it can be concluded that, in the case of the proposed sliding support, the machine tool is

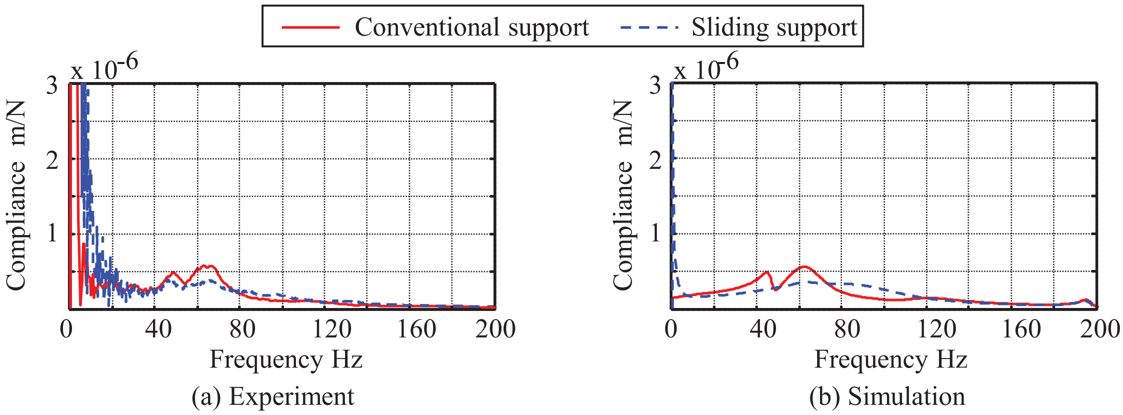


Fig. 8. Comparison offrequency responseof table along X-axis.

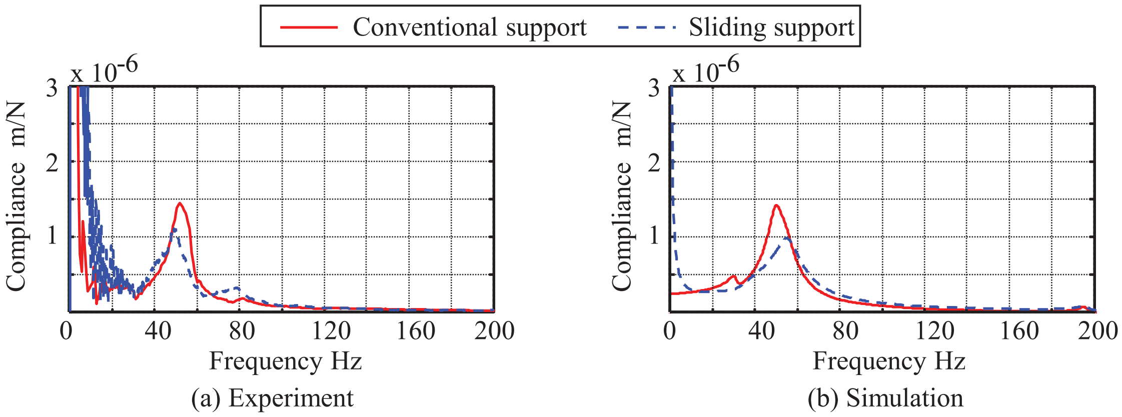


Fig. 9. Comparison offrequency responseof table along Y-axis.

displaced because of the inertial force when the direction of motion changes during the high-speed tracking motion.

In the mathematical model, the displacement of the machine bed with a sliding surface can be expressed by changing the stiffness and the damping between the bed and the ground. In consequence, the stiffness along the X- and Y-axis is set to zero, and the damping along the X- and Y-axis is adjusted by matching the displacement of the bed during motion.

The proposed model with the adjusted values of stiffness and damping can simulate the characteristics of the machine bed. It can be observed from **Figs. 6(b)** and **7(b)** that the simulated displacement of the bed is approximately 80 μm at point C and approximately 110 μm at points B and D.

It can be observed from the figures that the proposed model with the adjusted values of stiffness and damping between the bed and the ground represents the characteristics of the machine bed, and the model can express the displacement of the bed during high-speed tracking motions.

From the simulated results shown in **Fig. 7(b)**, a small displacement can be observed along the Y-axis around point C at which the direction of motion of the X-axis changes. It is expected that the rotational motion about the Z-axis is generated by the inertial force along the X-axis, and the motion influences the position of the measurement point along the Y-axis. In addition, from the measured results shown in **Fig. 7(a)**, the displacement has overshoots and vibrations when the direction of motion along the Y-axis changes at points A and D. Unfortunately, the reason behind these phenomena could not be clarified in this study. It is observed from **Fig. 7(a)** that an initial displacement of approximately 30 μm exists because the motion history influences the initial position of the supports.

4.2. Frequency Response

Figures 8 and **9** show the measured and simulated frequency response curves of the table. **Fig. 8** shows the response along the X-axis, and **Fig. 9** shows the response along the Y-axis. It can be observed from the measured and simulated results that, in the case of the proposed machine bed, the amplitude of vibration decreases around 65 Hz along the X-axis and around 50 Hz along the Y-axis. It can be concluded from the results that the proposed bed can reduce the vibration of the table. Although the amplitude of vibration increases around 80 Hz along the Y-axis in the case of the sliding support, the reason behind this phenomenon has not been clarified.

On comparing the measured and simulated results, it can be observed that the natural frequency in the X- and Y-axis is not completely expressed by the proposed model. For example, the peak vibration around 80 Hz can be observed in the measured results as shown in **Fig. 9(a)**. The

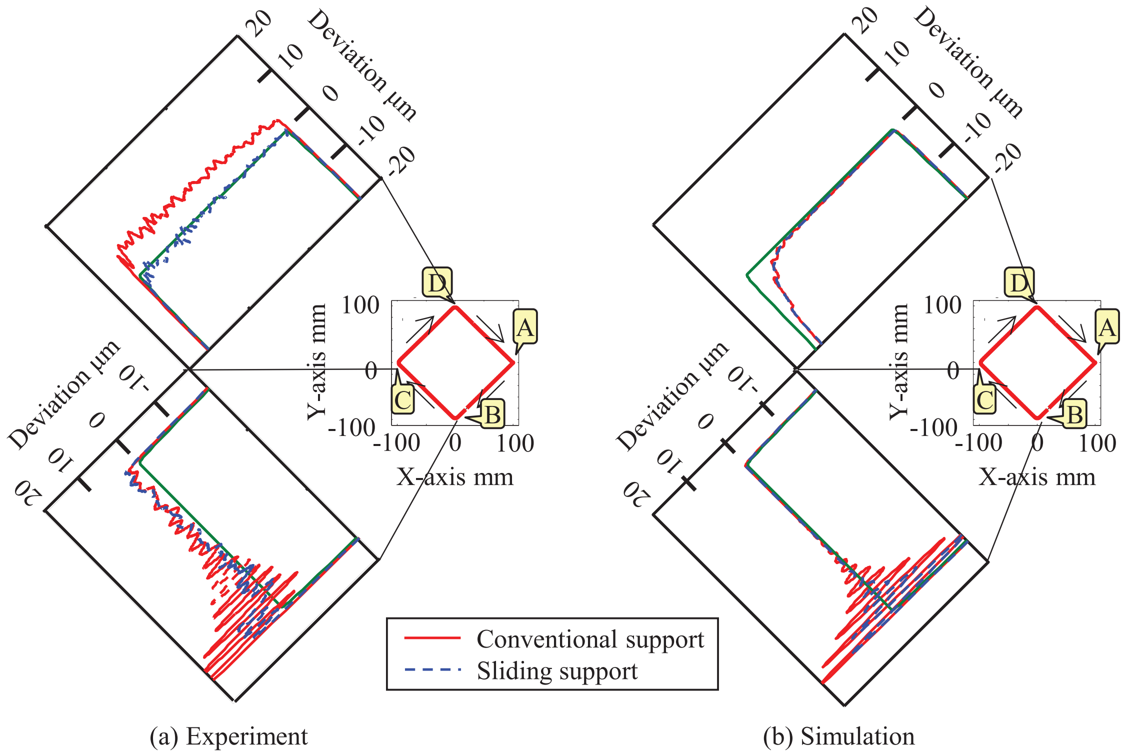


Fig. 10. Comparison of motion trajectories.

vibration mode is also influenced by the machine bed, and the amplitude of vibration is increased when the proposed support is installed. In the simulated results shown in **Fig. 9(b)**, however, the peak vibration around 80 Hz cannot be observed. The authors will continue to analyze the problem and improve the accuracy of simulation in the future works.

4.3. Motion Trajectory

Figure 10 illustrates the measured and simulated motion trajectories, which represent the relative position between the spindle head and the table on the X–Y plane.

In the measured results shown in **Fig. 10(a)**, vibrations can be observed around the corners (point B) at which the direction of motion of the Y-axis changes. It can also be observed that the vibration is smaller in the case of the proposed sliding support when compared to the conventional support because the oscillated energy is consumed by the displacement of the machine when the sliding support is installed. It is confirmed from the results that the vibration generated during high-speed tracking motions can be reduced by the proposed machine bed.

Vibrations can also be observed around the corners (point C) at which the direction of motion of the X-axis changes. It is similar for both the cases of conventional and sliding supports. It is expected that the moving mass along the X-axis is smaller than the mass along the Y-axis, and the effect of inertia on the high-speed tracking motion decreases. In addition, the trajectory between points C and D in the case of the conventional support has an offset from the desired path. Unfortunately, the reason behind this phenomenon has not been clarified.

In the simulated trajectories shown in **Fig. 10(b)**, it can be observed that the vibration around the corner is reduced when the proposed machine bed is installed although the effect is small when compared to the experimental results.

It is expected that the non-linear friction characteristics of the sliding surfaces affect the behaviors of the vibration because the non-linear friction characteristics [22] are not introduced into the simulation model. The authors will try to develop a non-linear frictional model for the proposed sliding support as a non-linear spring [23].

5. Conclusion

This study proposes a new machine bed support mechanism with a sliding surface for reducing the vibration generated during high-speed tracking motions and a mathematical model expressing the influence of the machine support. The conclusions can be summarized as follows:

(1) The proposed model with the adjusted values of the stiffness and damping between the bed and the ground represents the characteristics of the machine bed.

(2) The model can express the displacement of the bed during high-speed tracking motions.

(3) The proposed machine bed with a sliding surface can reduce the vibration generated during high-speed tracking motions.

The proposed machine bed support mechanism is an effective method for reducing the vibration during high-speed tracking motions. The authors will continue to improve the simulation accuracy and investigate the effect of surface roughness and lubrication state between two sliding surfaces on the vibration. The authors will also try to apply the proposed support to larger and different types of machines.

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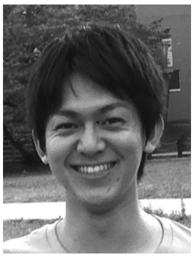

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