Sensor-less estimation of positioning reversal value for ball screw feed drives

Sato, Ryuta

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Highlights:

- A sensor-less estimation method of positioning reversal value for ball screw feed drive systems based on servo signals is proposed.

- The reversal value of the positioning accuracy can be adequately estimated from the measured frequency responses of the feedback position and the motor torque against the positional command.

- Since the proposed estimation method can easily be implemented into commercial NCs, it is expected that the proposed method can be an effective tool for managing the condition of feed drive systems, as it is capable of evaluating system condition without any additional sensors.

Research paper

Sensor-less estimation of positioning reversal value for ball screw feed drives

Ryuta SATO
Department of Mechanical Engineering, Kobe University,
1-1 Rokko-dai, Nada, Kobe 657-8501, JAPAN
TEL/FAX: +81-78-803-6326
E-mail: sato@mech.kobe-u.ac.jp

Abstract
Motion accuracy of the feed drive systems deteriorate due to wear of the bearing and ball screw during the long term use. It is known that the wear makes larger positioning reversal value. In this study, a sensor-less estimation method for the positioning reversal value of ball screw driven feed drive systems based on the servo signals is proposed. The proposed method analyses the relationship between amplitude of feedback position and motor torque during swept sine wave motion. It is experimentally confirmed that the proposed method can estimate the positioning reversal values adequately.

Keywords: feed drive systems, ball screw, support bearing, wear, positioning reversal value, sensor-less estimation

1. Introduction

Feed drive systems consisting of AC servomotors and ball screws are generally used in NC machine tools, wire bonders, tip mounters, and the like [1] [2]. Since feed drive systems in these applications are among the most important components of these types of equipment, higher motion and positioning accuracy and speed are consistently required of the feed drive systems. However, over long-term operation, the accuracy of feed drive systems are generally deteriorated due to the pre-load decreasing that caused wear of ball screws and support bearings [3]. Since inaccurate motion of the feed drive system can result in issues during the production processes, the deterioration of accuracy should be detected before the trouble occurs. From this point of view, renewal and maintenance planning becomes a key issue in the manufacturing field [4].

In machining operations, various types of monitoring methods for cutting processes using sensors or servo signals have already been proposed [5]. For fault-diagnosis of the feed drive systems driven by ball screws, several monitoring technologies have also been developed [6-10]. Möhring and Bertram [6] proposed a pre-load monitoring system for the ball screw nut by a sensor installed into the nut. Several researchers developed condition monitoring method based on the vibration analysis [7-9]. Industrial company also tries to monitor the condition [10]. The vibration based condition monitoring is a popular approach for the ball bearings [11]. It can successfully be applied to the condition monitoring of spindle bearings [12]. Those methods can avoid the critical breakdown of the machines.

A feed drive system consists of a feed drive mechanism and a controller. Feed drive systems typically have velocity and position control loops [13]. The velocity controller is a PI (Proportional and Integral) controller based on the detected angular velocity from the rotary encoder attached to the motor. The position control loop is a P (Proportional) controller. There are two kinds of position control loops: semi-closed loop and full-closed loop. In a semi-closed loop control system, the position controller uses the detected rotational angle of the motor by the rotary encoder to position the drive. In this case, the wear of the bearing and ball-screw directly influences positioning accuracy. On the other hand, in the full-closed loop control system, wear of the bearing and ball screw does not directly influence positioning accuracy because the position controller detects table position via an independent linear encoder attached to the table. It is also possible to monitor the condition based on the detected table position. Although the full-closed loop control system has many advantages, the semi-closed loop control system is still popular especially in the smaller machine tools mainly used in EMS industries because of the cost efficiency.

It is necessary to keep the positioning accuracy of the semi-closed loop controlled feed drive systems because the accuracy deterioration makes inferior products. However, the proposed monitoring methods mentioned above cannot evaluate and improve the positioning accuracy even though the methods can avoid the critical breakdown of the machine. The positioning and motion accuracy of the feed drive systems can be evaluated by the measurement system such as laser displacement sensors [14] [15]. However, since several hundreds or thousands machines are working continuously, it is difficult to check the accuracies in the real field.

The purpose of this study is to develop a sensor-less estimation method of positioning reversal value for ball screw feed drive systems based on servo signals, without requiring any additional measurement systems. The decrease in the pre-load caused by the wear of ball screws and support bearings mainly influences the positioning reversal value which is defined as the difference in the positioning values in the back and forth motions [15].

The positioning reversal value can be compensated by using the backlash compensation functions implemented in the NCs [16] [17]. The proposed sensor-less estimation method can also be implemented into the NCs. Figure 1 describes the expected effect of the proposed estimation method.

Although the positioning accuracy deteriorates during the long-term operation, the accuracy can be maintained by changing the compensation parameters based on the estimated reversal values as shown in Figure 1.

2. Feed drive mechanism

Figure 2 (a) shows the schematic of the feed drive mechanism with AC servomotor and ball screw. In this mechanism, the rotational motion of the motor is converted to the translational motion of the table through the ball screw. Note the axial position of the ball screw is fixed by the support bearing. As a result, the positioning accuracy of the table is extremely sensitive to the wear of the ball screw and support bearing. It has already been shown that this feed drive mechanism can be simply modelled as a 2-DOF vibration model as shown in Figure 2 (b) [18]. In this figure, J_m is the moment of inertia of motor rotor, coupling, and ball screw shaft. M_t is the mass of table, nut, and nut bracket. It has also been confirmed that this model expresses the first order mode of vibration. This means that the feed drive mechanism can be divided into motor side and table side. Stiffness between the motor side and table side K_a can be estimated as a serial connection of the stiffness of the components, the torsional stiffness of the coupling and ball screw shaft, and the axial stiffness of the support bearing, ball screw shaft, and nut. Additionally, c_b and f_b are the viscous and Coulmb's friction around the ball screw shaft, c_t and f_t are the viscous and Coulmb's friction along the table motion, and c_i is the internal damping coefficient of the mechanism.

3. Estimation method of positioning reversal value

Figure 3 describes the relationship between the vibration amplitude of the motor, table, and motor torque in the sinusoidal motion. Motor torque is influenced by both of the inertial force and frictions. However, in the vibrated motions, the inertial force becomes dominant compared with the influence of the frictions. Hence the frictions and damping shown in Figure 2 (b) are ignored in Figure 3. This fact suggests that the influence of the non-linear effect of the frictions can be reduced by applying the vibrated motions.

Wear of the bearing and ball screw can be simply modelled as a dead band between the motor and table side components. If vibration amplitude of the motor being larger than the dead band, as shown in Figure 3 (a), the motion is transmitted to the table side, and the motor torque includes the inertial forces of both the motor and the table side. On the other hand, in case of motor vibration amplitude being smaller than the dead band, the motion is not transmitted to the table side. As a result, the table does not move and the motor torque does not include the inertial force of the table side.

Vibration amplitude of the motor depends on the input frequency based on the closed loop frequency response of the feed drive system. The frequency response in the lower frequency range mainly be dominated by the position loop gain G_p [13]. Figure 4 shows a simplified block diagram of the system. The vibration amplitude A_{xt} which depends on the frequency ω can be obtained from the block diagram as Equation (1). $G_x(j\omega)$ is the frequency transfer function from the positional command r to the displacement.

$$A_{xt} = \left|G_x(j\omega)\right| = \frac{1}{\sqrt{1+\left(\omega/G_p\right)^2}} \tag{1}$$

The motor torque depends on the acceleration of the motion. Hence the amplitude of motor torque A_{Tm} can be expressed as Equations (2) and (3) based on the vibration amplitude A_{xt}.

In case of vibration amplitude is larger than dead band:

$$A_{Tm} = \frac{\omega^2}{\sqrt{1+\left(\omega/G_p\right)^2}}\left(J_m + \frac{l}{2\pi}M_t\right) \tag{2}$$

In case of vibration amplitude is smaller than dead band:

$$A_{Tm} = \frac{\omega^2}{\sqrt{1+\left(\omega/G_p\right)^2}}J_m \tag{3}$$

Figure 5 (a) shows the calculated frequency dependent amplitude of the motor side vibration when sinusoidal motion with amplitude of 150 μm is commanded. Amplitude of the positional command should be larger than the expected reversal value. The typical range of the reversal value is less than 100 μm in the feed drive systems for machine tools. Figure 5 (b) also shows the calculated amplitude of the motor torque during the same sinusoidal motion. In Figure 5 (b), Line A describes the motor torque amplitude when the motor torque includes the inertial forces of both the motor and table sides, while Line B describes the motor torque amplitude when the motor torque does not include the inertial force of the table side. The figure clearly shows that the motor torque becomes smaller in the case that does not include the inertial force of the table.

According to the simplified schematic model shown in Figure 3, the motor torque includes the inertial forces of both the motor and table sides when the vibration amplitude is larger than the dead band. Hence, it can be inferred that the amplitude of the motor torque begins on Line A shown in Figure 3 (a) and Equation (2). The amplitude of the motor vibration becomes smaller when the vibration frequency becomes higher; thus, it can also be inferred that the amplitude of the motor torque shifts toward Line B shown in Figure 3 (b) and Equation (3) when the vibration amplitude becomes smaller than the dead band. As a result, the amplitude of the motor torque will assume the minimum value at the frequency that matches the vibration amplitude of the motor with the dead band.

As mentioned above, the wear of the bearing and ball screw, modelled as the dead band, can be detected by measuring and comparing the two frequency responses: the response between the positional command and the feedback position, and the response between the positional command and the motor torque. The size of the dead band can be estimated as the vibration amplitude at the frequency that has the minimum motor torque amplitude.

4. Experiment

To confirm the correctness of the proposed method, experimental tests of worn support bearing behavior are carried out. As it requires quite a long time to reproduce the worn ball screw and support bearing in the laboratory, in this study pre-load on the support bearing is intentionally reduced to simulate a worn bearing condition for the experimental tests.

Figure 6 (a) illustrates the detailed construction of the support bearing used in this study. The support bearing consists of angular contact bearings with back-to-back duplex. The inner diameter of the inner ring of the bearings are 20 mm. In addition, lead and diameter of the ball screw of the mechanism used in this study are 16 mm and 28 mm. The ball screw nut is an over-sized ball preload type. Rated power, mass of the moving part, and resolution of rotary encoder are 1 kW, 62 kg, and 17bit, respectively.

The axial preload is supplied by lock nuts that push on the inner ring of the bearing. Therefore, it is possible to control the pre-load by changing the comparative widths of the outer and inner rings. In this study, shim plates with the thickness of several 10 μm are placed between the two inner rings to reduce the pre-load. Measurement tests of the positioning accuracy compliant with ISO 230-2 standard [15] are then carried out. Figure 6 (b) shows the measurement set-up for the positioning accuracy. A grid encoder (KGM181, Heidenhain) is used for the measurements.

Figure 7 shows the positioning accuracies measured in the experiments. The results are measured with a loading mass of 26 kg on the table. It is also confirmed that the static positioning accuracy is not influenced by the loading mass. Figure 7 (a) shows the measured results without the shim plates. As shown in the figure, the reversal value at the stroke center is 2.2 μm. It can also be seen that the repeatability of the positioning is quite high; less than 1 μm. Figures 7 (b) and 7 (c) show the measured results with shim plates of 30 μm and 40 μm, respectively. As the figures indicate, the reversal values at the stroke center are 20 μm and 32 μm, respectively. Repeatability of the positioning is also quite high in both results. The positioning reversal values and the thickness of the shim plates are not identical because of the support bearings are preloaded to make higher stiffness. It can also be expected that the support bearings do not have complete dead-band because the steel balls always contact with the inner and outer rings. Required estimation accuracy for the reversal value depends on the type and the purpose of the machine tools. In this study, however, estimation of around several 10 μm orders are aimed. It is a typical positioning error level should be compensated.

Figure 8 shows the measured frequency responses of the feed drive system in the three different conditions. For the condition without shim plates, the frequency that matches the minimum motor torque amplitude is 195 Hz as shown in Figure 8 (b). The amount of dead band can then be estimated from Figure 8 (a) as the measured amplitude at 195 Hz. The estimated value is 2.8 μm, quite an accurate value compared with the measured result shown in Figure 7 (a). For the conditions with shim plates, the motor torque amplitude is at a minimum at 80 Hz and 70 Hz, respectively. The vibration amplitude of the motor at those frequencies is 23 μm and 36 μm, respectively. Although these values are slightly larger than the measured reversal values shown in Figures 7 (b) and (c), the differences are less than 5 μm.

It can be confirmed from these results that the proposed method adequately estimates the reversal value of the positioning accuracy from the measured frequency responses between the positional command, feedback position, and motor torque. The estimation accuracy is quite enough for the typical machining centers.

According to Figure 8 (b), amplitude of the torque becomes maximum around 50 Hz in cases with shim plates. On the other hand, however, amplitude of the torque becomes maximum around 100 Hz in case without the shim plates. In addition, the torque increases rapidly around 250 Hz in case without the shim plates. Although it can be expected that the resonance of the feed drive mechanism has some influences on to the behaviors, the reason of the differences is not clarified now.

Figure 9 shows measured results of motor torque amplitude with several loading mass attached on the table. The total thickness of shim plates is 30 μm. It can be seen from the results that the motor torque amplitude is at a minimum at 80 Hz as same as the results shown in Figure 8 for all loading mass. It is clear from the figure that the motor torque amplitude does not depend on the loading mass on the table in case of the frequency is higher than 80 Hz, because the table does not move. This fact can be a good confirmation of the simplified schematic model shown in Figure 3. It can also be confirmed that the motor torque amplitude depends on the loading mass in the lower frequency region, because of the displacement amplitude of motor is larger than the dead band, as expressed as Equation (2).

5. Conclusion

In this study, a sensor-less estimation method of positioning reversal value for ball screw feed drive systems based on servo signals is proposed. It is experimentally confirmed that the reversal value of the positioning accuracy can be adequately estimated from the measured frequency responses of the feedback position and the motor torque against the positional command. Since the proposed estimation method can easily be implemented into commercial NCs, it is expected that the proposed method can be an effective tool for managing the condition of feed drive systems, as it is capable of evaluating system condition without any additional sensors.

On the other hand, it is expected that the estimation accuracy of the proposed method depends on the resolution of rotary encoder and torque. Frequency response of the feed drive system may also influence the estimation accuracy. The author will try to clarify the influence of the factors and the applicable range for the proposed method.

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Figures:

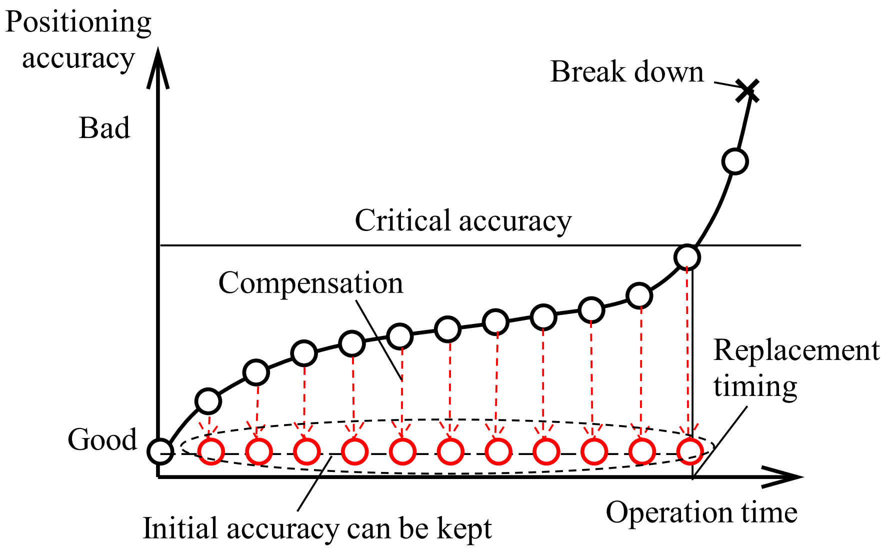
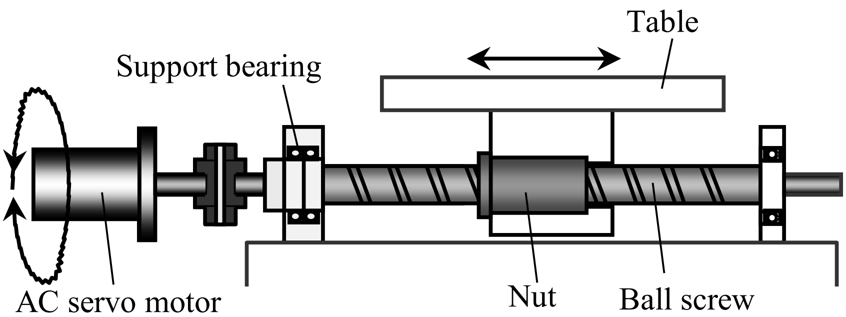
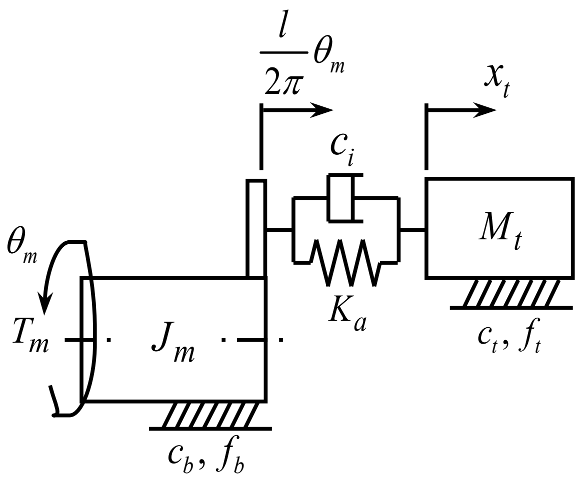


Figure 1: Expected effect of the proposed estimation method.

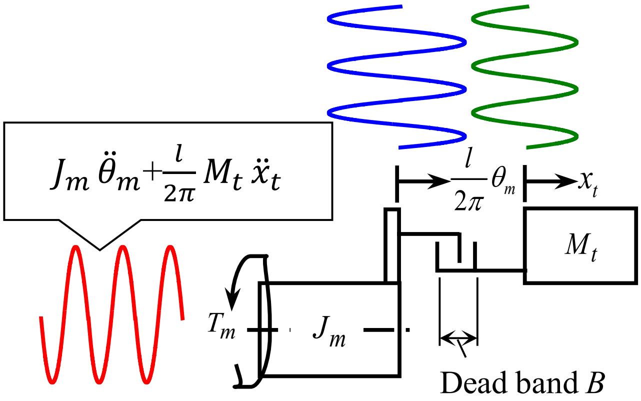


(a) Schematics of feed drive mechanism.

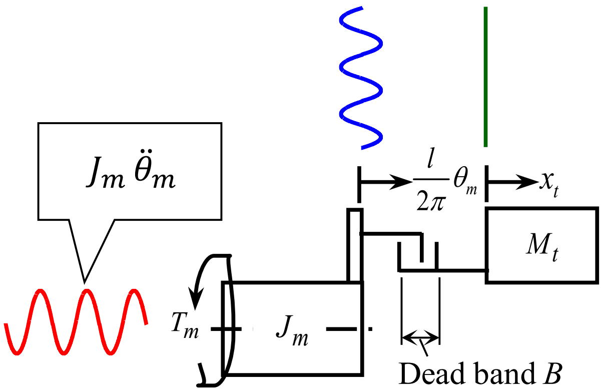


(b) 2DOF model of feed drive mechanism.

Figure 2: Mechanical structure of ball-screw feed drive mechanism and its vibration model.



(a) In case of vibration amplitude is larger than dead band



(b) In case of vibration amplitude is smaller than dead band

Figure 3: Schematic explanation for the relationships between vibration amplitude of motor angle and motor torque influenced by a dead band.

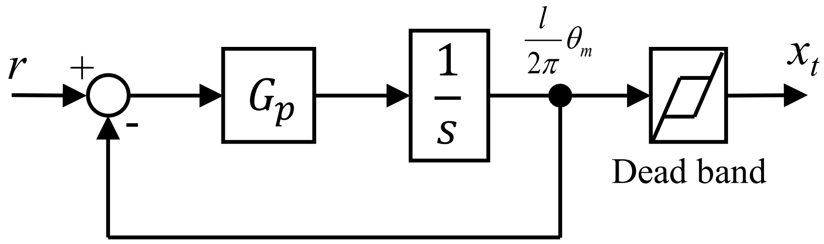
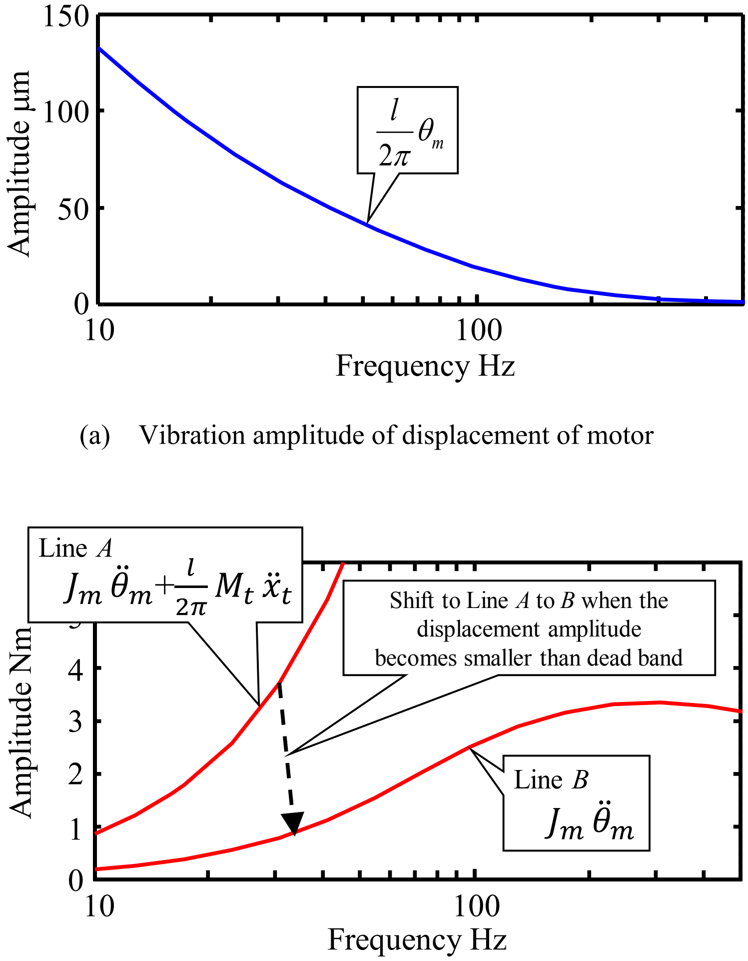


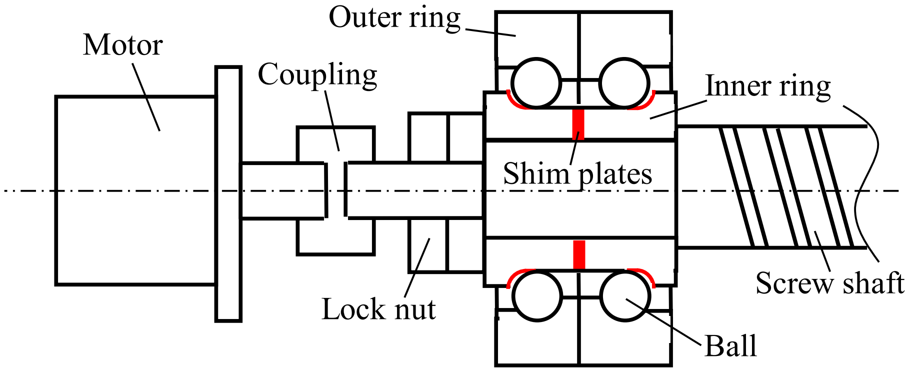
Figure 4: Simplified block diagram of a feed drive system with a dead band.



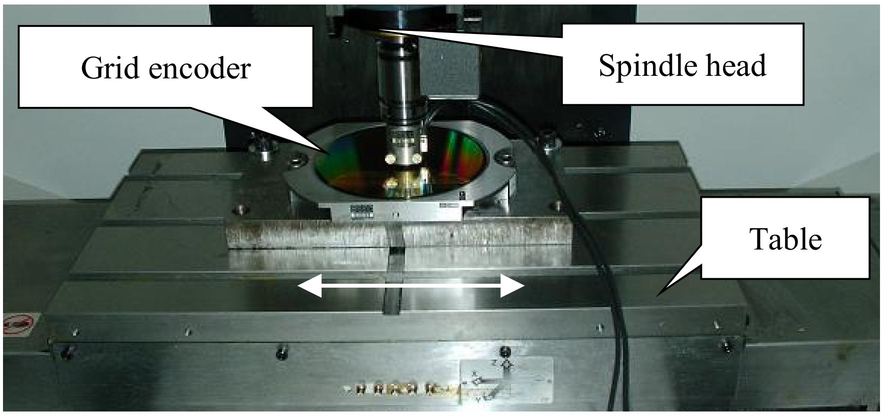
(a) Vibration amplitude of displacement of motor

(b) Vibration amplitude of motor torque

Figure 5: Calculated frequency responses of a feed drive system.

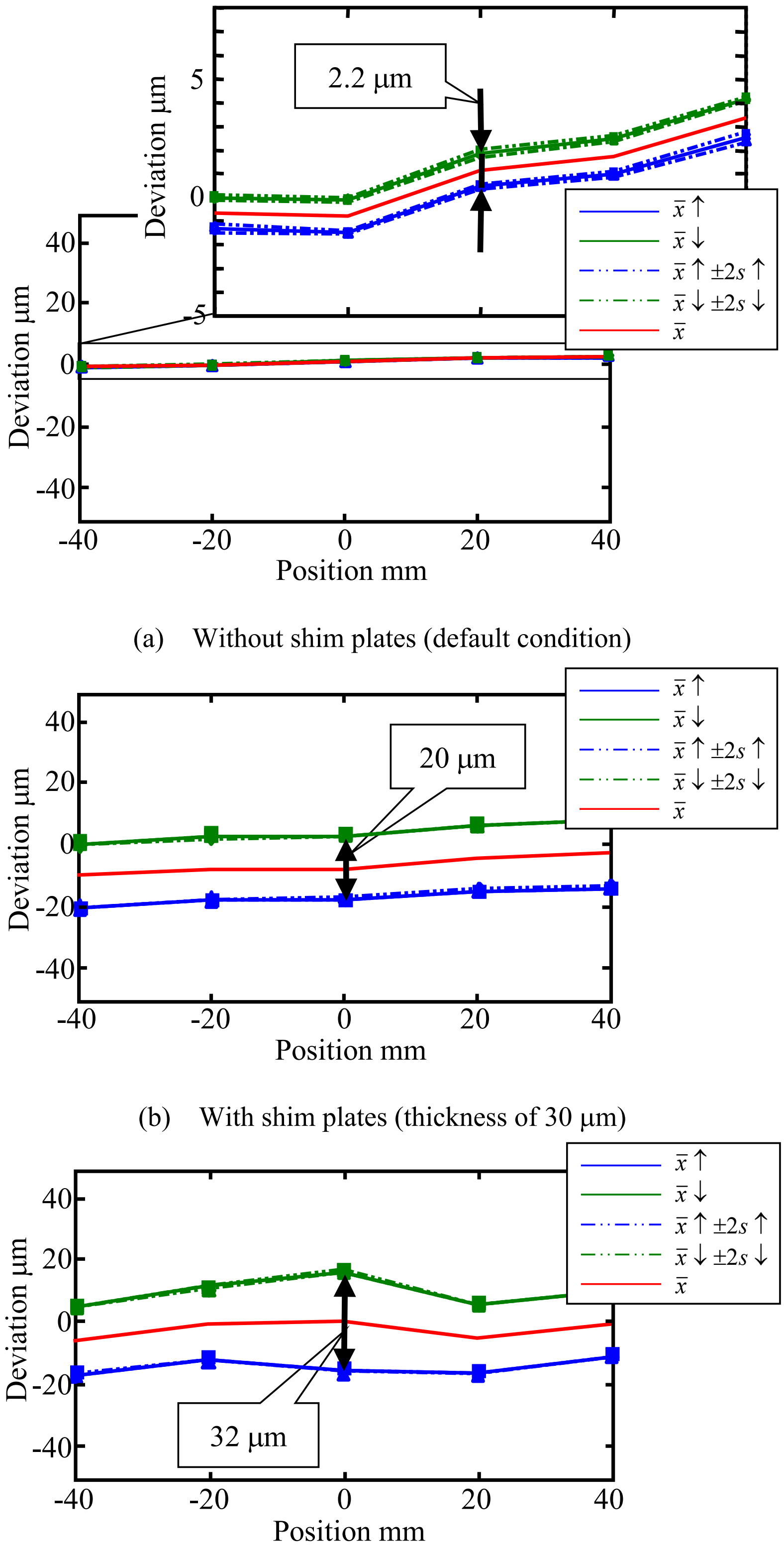


(a) Schematic view of support bearing with shim plates



(b) Measurement method for displacement of table

Figure 6: Experimental method to confirm the correctness of proposed method.

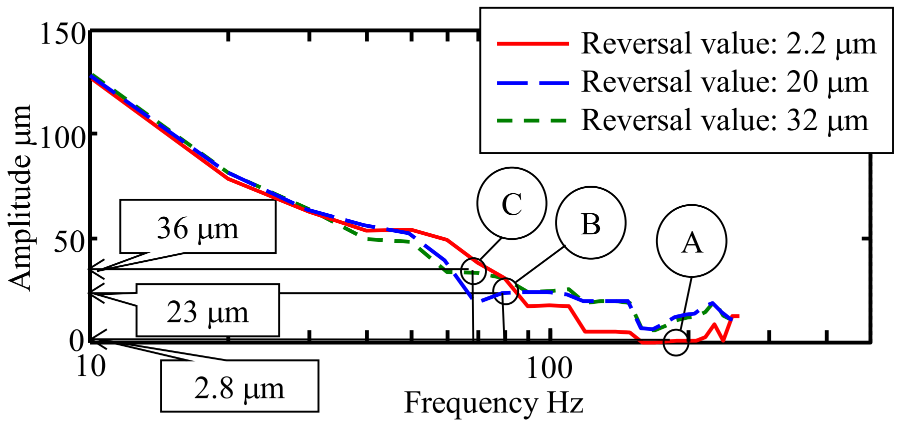


(a) Without shim plates (default condition)

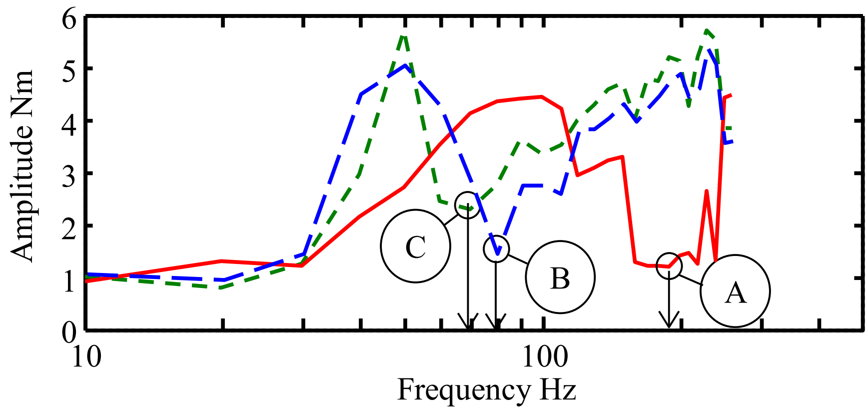
(b) With shim plates (thickness of 30 μm)

(c) With shim plates (thickness of 40 μm)

Figure 7: Measured positioning accuracies with different condition of support bearing.
(with loading mass of 26 kg)



(a) Vibration amplitude of displacement of motor



(b) Vibration amplitude of motor torque

Figure 8: Measured frequency responses with different condition of support bearing. (with a loading mass of 26 kg)

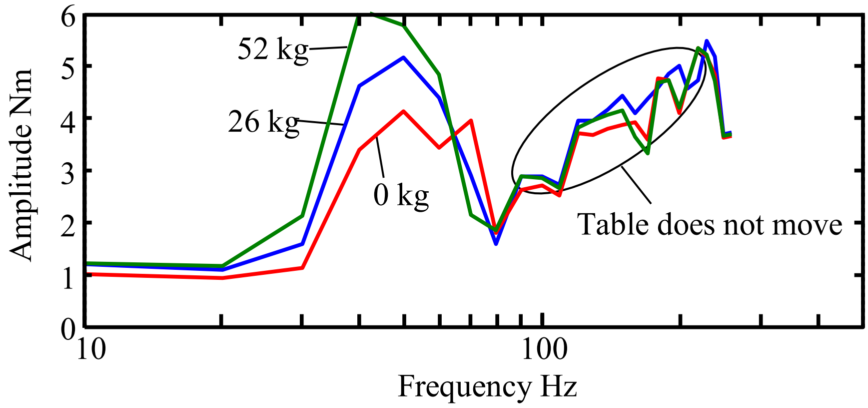


Figure 9: Influence of loading mass on the table onto the measured frequency responses. (total thickness of shim plates is 30 μm)