Influence of linear ball guide preloads and retainers on the microscopic motions of a feed-drive system

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Abstract
This paper describes the effects of preloading and ball retainer conditions of linear ball guides on a feed-drive system operating on a microscopic scale, as this is important for applications that employ linear guides in precision machines. To begin, relationships between nonlinear spring behaviors of the guide and the behavior of quadrant glitches were analyzed based on the proposed simple friction model. The behavior of the quadrant glitches, nanometer step responses, and steady vibrations were also measured for three guide conditions that differed with respect to the ball retainers and preloading. These experiments were carried out by using a special feed-drive system that comprises eight-grooved linear ball guides, an AC linear servo motor, and a linear encoder with a high position resolution of 31.25 pm. This system was set on a vibration isolation table and driven by a linear current amplifier. The time constants of each of the step responses were also analyzed based on the friction and control system model. From the analysis and experiments, it is demonstrated that the behavior of quadrant glitches and step responses are strongly influenced by the friction characteristics of the guides, and that this behavior can be adequately estimated via analysis. Additionally, it is shown that steady vibrations are also influenced by the friction characteristics, and that the amplitude of the vibration is proportional to the compliance of the nonlinear spring behavior.

Keywords: Quadrant glitches, Linear ball guide, Nonlinear spring behavior, Microscopic motion, One-nanometer step response, Steady-state vibration

1. Introduction

Higher precision motion control is a consistent goal for manufacturers of machine tools and semiconductor manufacturing devices. A common method used to ensure higher accuracy is to implement a linear guide mechanism, which eliminates friction via the use of hydrostatic guides such as air bearings. Despite this advantage, according to a survey by the special committee on ultra-precision positioning, rolling ball/roller guides have become more popular in recent years (Oiwa et al., 2011, Oiwa and Katsuki, 2015).

Implementing rolling ball/roller guides in precision machines results in the friction characteristics of the guides inducing various motion errors such as quadrant glitches. Although many studies have been carried out on these glitches (Jamaludin et al., 2009; Mei et al., 2004; 2008; Sato et al, 2008, Tung et al, 1993), the studies focused on circular motions with a radius of larger than 10 mm. Sato (Sato, 2012) reported that the quadrant glitches cause the circular trajectory to become square for a 0.1 mm radius motion. It is also known that linear rolling guides induce microscopic-level nonlinear spring behavior (NSB) (Futami and Furutani, 1990, Kaneko et al., 2004, Otsuka and Masuda, 1998, Tanaka et al., 2009). Although the motion behavior of the feed-drive system is affected by NSBs, the relationships between the NSB and behavior is not sufficiently clarified.
This paper describes an investigation into the relationships between the NSB and microscopic motion behavior of a feed-drive system by using eight-grooved linear ball guides developed for ultra-precision machines. In order to investigate the influence of the NSBs, sinusoidal motion tests and step response tests were carried out for three guide conditions that differed with respect to the ball retainers and preloading. Moreover, the sinusoidal motion tests were carried out for an amplitude range of 10 to 1000 µm, and a nanometer-scale step height was utilized in the step response tests. The steady vibrations were also measured and investigated in this study. Lastly, to clarify the cause of the particular behaviors occurring on a microscopic level, theoretical analysis was performed as based on the proposed friction model for NSBs.

2. Eight-grooved linear ball guide

An eight-grooved linear ball guide (SPS25 series, THK Co.) for ultra-precision machines has been developed (THK, 2010). A schematic view of the SPS25 guide is shown in Figure 1. The guide has eight grooves, which is twice the amount of conventional linear ball guides, and the length of the carriage is also longer than its conventional counterpart to maintain load capability. The diameter of each ball is small as less than 3 mm, and the guide has retainers that purpose to maintain the relative position between the balls. The number of balls in contact with the rail in each guide is approximately 500. Additionally, the guide is capable of ultra-low waving at less than 20 nm in the horizontal plane, and less than 10 nm in the vertical plane. These values can be comparable to the accuracy of hydrostatic guides. The guides also have high stiffness of over 6.7 kN/m that surpasses even roller guides. The specifications of SPS25 are listed in Table 1 and illustrated in Figure 2, which shows a cross-sectional view and outline dimensional drawing of the guide.

![Fig. 1 Schematic view of the experimental SPS25 eight-grooved linear ball guide for use in ultra-precision machines (THK, 2010). The number of contacting balls is nearly five hundred.]

![Fig. 2 Cross-sectional view and outline dimensional drawing of SPS25 guide (THK, 2010).]

<table>
<thead>
<tr>
<th>Note</th>
<th>Length mm</th>
<th>Note</th>
<th>Length mm</th>
</tr>
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<tbody>
<tr>
<td>M</td>
<td>31</td>
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</tr>
<tr>
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<td>W₂</td>
<td>18.5</td>
</tr>
<tr>
<td>H₂</td>
<td>4.5</td>
<td>L</td>
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<tr>
<td>W</td>
<td>72</td>
<td>L₁</td>
<td>158.1</td>
</tr>
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</table>

Table 1 Specifications of the SPS25 eight-grooved linear ball guide (THK, 2010).
3. Experimental System

Figure 3 shows the experimental system. The experimental system consists of linear ball guides (SPS25, THK Co.), a linear AC servo motor (KOVERY Motor Inc.), a linear encoder (LIP281, DR. JOHANNES HEIDENHAIN GmbH), a three-phase linear amplifier, and a controller, as is shown in Figure 4. Although the motor has cores, it generates very small cogging and magnetic attraction forces because of the special structure of the cores. The maximum driving force, which is initiated by the amplifier, is 350 N, and the current reference of the motor is provided by the controller via an 18-bit DA convertor. The resolution of the linear encoder is 31.25 pm, and the mass of the moving part is 22.5 kg. The stage is mounted onto a vibration isolation table, which has a 1.3 Hz resonant frequency. Position and velocity controllers are composed as P control and PI control, respectively. A velocity feedforward controller is also implemented. The control parameters are set as $K_p = 1200 \ 1/s$, $K_v = 270 \ 1/s$, $T_i = 6.25 \ ms$, and $\alpha = 1 (= 100\% \ feedforward)$.

![Figure 3 Experimental system configuration. The system comprises a one-axis stage mechanism that is guided by the eight-grooved linear ball bearing and driven by a linear motor, an amplifier, and a controller. The position of the table is detected by using a high-resolution linear scale.](image)

![Figure 4 Block diagram of control system. In this control setup, the position is proportionally controlled, and the velocity is proportional and integral; the system also implements feedforward velocity control.](image)

4. Microscopic Sinusoidal Motion

The sinusoidal motion tests were carried out on a microscopic scale. Figure 5 shows the measured results for a frequency of 0.1 Hz. The motion amplitudes were set as 10, 100, and 1000 μm. The solid and dashed lines in the figure illustrate the positional deviation and displacement, respectively. It can be seen from the figure that the reference amplitude significantly influenced the shape of the positional deviation. It can also be seen that the waveform of the positional deviation did not differ between the first and second cycles.

Figure 6 shows how varying the frequency of motion (0.01, 0.1, and 1 Hz) affected the measured positional deviation for an amplitude of 100 μm. The horizontal axis represents the angular phase, $\theta = \omega t$, and the vertical axis represents positional deviation divided by the frequency. According to the figure, the waveform of the positional deviation does not depend on the frequency of motion, and the amplitude is proportional to frequency.

Although the experimental system had a single axis, it was possible to evaluate the circular trajectory by constructing a polar plot by using the measured deviation. Figure 7 shows the polar representation of the positional deviation under the conditions of a 1000 μm radius and 0.1 Hz frequency.

![Diagram](image)
Fig. 5 Displacement and deviation responses under the condition of 0.1 Hz sinusoidal motion for amplitudes of (a) 10 μm, (b) 100 μm, and (c) 1000 μm. The black dotted line and red line show the displacement and deviation, respectively.

Fig. 6 Comparison of positional deviations under the condition of 100 μm amplitude sinusoidal motion for frequencies of 0.01, 0.1, and 1 Hz. The deviations for 0.01 and 0.1 Hz are multiplied by 100 and 10, respectively. With the scale normalization, the responses of the three are observed to be nearly identical.

Fig. 7 Polar representation of positional deviation for 1000 μm amplitude and 0.1 Hz frequency. These deviations are referred to as quadrant glitches.
5. Relationship between Nonlinear Spring Behavior and Quadrant Glitches

Linear ball/roller guides behave as nonlinear springs when the motion is reversed; as previously mentioned, this behavior is referred to as NSB. Figure 8(a) shows the measured NSB of the experimental system. In this study, the NSB was approximated as the first-order system shown in Figure 8(b). In the figure, $f_m$ is the steady-state friction force, $L$ is the representative length that decides the steepness of the model, and $x$ is the displacement from the reversal position. The NSB model shown in Figure 8(b) can be expressed as Equation (1).

$$f_d = \begin{cases} 2f_m \left(\frac{1}{2} - \exp \left( -\frac{x}{L} \right) \right) & \text{if } v(t) \geq 0 \\ -2f_m \left(\frac{1}{2} - \exp \left( \frac{x}{L} \right) \right) & \text{if } v(t) < 0 \end{cases}$$  \hspace{1cm} (1)$$

![Figure 8 Nonlinear spring behavior, which is defined as the force-to-displacement relationship for sinusoidal position movement. The measured NSB was approximated via two first-order lag functions in order to simplify the analysis.](image)

Alternatively, the force disturbance $f_d$ is expressed via the block diagram shown in Figure 9(a). A simple equivalent transformation yields the block diagram illustrated in Figure 9(b), where the disturbance has the dimension of displacement. The displacement disturbance is expressed as Equation (2).

$$d = \frac{T_i}{K_p K_i M (1 + T_i)} f_d$$  \hspace{1cm} (2)$$

Because the integral time $T_i$ is 6.25 ms, and $T_i \omega$ may be negligible when the frequency of motion is less than 1 Hz, Equation (2) can be approximated as follows:

$$d \approx \frac{T_i}{K_p K_i M} f_d$$  \hspace{1cm} (3)$$

Equation (4) is obtained by substituting $f_d(x)$ into Equation (3), where $v(t)$ is velocity.

$$d(t) = \frac{T_i}{K_p K_i M} \cdot \frac{d}{dt} f_d$$

$$= \frac{T_i}{K_p K_i M} \cdot \frac{d}{dx} f_d \cdot \frac{dx}{dt}$$

$$= \frac{T_i}{K_p K_i M} \cdot v(t) \cdot \frac{d}{dx} f_d$$  \hspace{1cm} (4)$$
By substituting Equation (1) into Equation (4), the deformed disturbance \(d(t)\) is expressed as Equation (5) below.

\[
d(t) = \frac{T_i}{K_pK_rM} f_{d} \cdot \frac{v(t)}{L} \cdot \exp \left( -\frac{x}{L} \right)
\]  

(5)

Considering the third quadrant of clockwise motion, the displacement \(x\) and velocity \(v\) are written as Equation (6).

\[
x(t) = R \left( 1 - \cos \omega t \right) \\
v(t) = \omega R \cdot \sin \omega t
\]  

(6)

Then, by incorporating Equation (6) into Equation (5), \(d(t)\) becomes Equation (7).

\[
d(t) = \frac{T_i}{K_pK_rM} f_{d} \frac{R}{L} \cdot \omega \sin \omega t \cdot \exp \left( -\frac{R}{L} \left( 1 - \cos \omega t \right) \right)
\]  

(7)

Equation (7) can be divided into two parts: 1) \(d_{\text{amp}}\), which determines the magnitude of the positional deviation; and 2) \(d_{\text{shape}}\), which determines the shape of the deviation. This is mathematically described via Equations (8) and (9), respectively.

\[
d_{\text{amp}}(\omega) = \frac{T_i}{K_pK_rM} f_{d} \omega
\]  

(8)

\[
d_{\text{shape}} \left( \frac{R}{L} \theta \right) = \frac{R}{L} \sin \theta \cdot \exp \left( -\frac{R}{L} \left( 1 - \cos \theta \right) \right)
\]  

(9)

In Equation (9), the phase is expressed as \(\theta=\omega t\), and the following conclusions can be gleaned from Equation (8): The magnitude of a quadrant glitch is 1) proportional to the frequency, 2) proportional to the steady friction force, 3) inversely proportional to the moving-part mass, 4) proportional to \(T_i\), and 5) inversely proportional to \(K_p\) and \(K_r\).

Figure 10 illustrates the results of \(d_{\text{shape}}\) as calculated via Equation (9), with \(R/L = 2, 10,\) and 250; the horizontal axis is the phase angle. This phase angle corresponds to the displacement from the point after the motion direction changes. It can be seen from the results that the span of the phase angle of a quadrant glitch is wider for smaller values of \(R/L\), and narrower for larger values of \(R/L\).

Because \(L\) is a characteristic NSB parameter, the shape of a quadrant glitch is determined by the radius of the circular motion under the condition of constant frequency (i.e., constant angular velocity of the circular motion). It should be noted that the theoretical results mentioned above were found to be in agreement with the experimental results. Additionally, although the actual NSB cannot be expressed as a simple model, the trend of the relationship between the NSB and a quadrant glitch can be analyzed.

To experimentally confirm the analyzed trend, experimental tests were performed for three different guide conditions (Table 2). Figure 11(a) shows the measured quadrant glitches (positional deviation error) for the three preloading and retainer conditions; the green, blue, and red lines represent Condition 1 (CD1), Condition 2 (CD2), and Condition 3 (CD3), respectively.
It can be seen from the figure that, while the difference between CD1 and CD2 is not very significant, the results for CD3 are considerably different from those of CD1 and CD2. Figure 11(b) provides a close-up view of Figure 11(a), showing the time range of 7 to 9 s. This close-up view highlights the differences between the three conditions; for instance, at 7.5 s, the quadrant changed from four to three.

Figure 12 shows the measured NSB for each condition. The measured results for CD3 show the largest friction with the highest NSB stiffness. This results in the largest and sharpest quadrant glitches. Conversely, CD2 results, which correspond to the smallest friction force, yielded the smallest quadrant glitch. Consequently, these findings are in agreement with the analytically estimated trend.

Fig. 10 Shapes of quadrant glitches calculated via the theoretical equation expressed in Equation (9). $R/L$ is the ratio of the radius of the reference circle to the representative length of the first lag function. A larger $R/L$ corresponds to a sharper quadrant glitch. It is important to emphasize that the quadrant glitch exists in the wide-phase angle range for small values of $R/L$.

![Fig. 10 Shapes of quadrant glitches calculated via the theoretical equation expressed in Equation (9).](image)

![Fig. 11 Comparison of the positional deviations resulting from the three different experimental conditions presented in Table 2; the corresponding three different NSBs are shown in Figure 12.](image)

(a) Overall view (two cycles).  
(b) Close-up view of the time between 7 s and 9 s

Fig 11  

![Fig 12 Measured NSB for each of the three conditions described in Table 2.](image)

Table 2  

<table>
<thead>
<tr>
<th>Variable experimental conditions.</th>
<th>CD</th>
<th>Ball retainer with / without</th>
<th>Ball size large / small</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>With</td>
<td>Large</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>With</td>
<td>Small</td>
</tr>
<tr>
<td>3</td>
<td>3</td>
<td>Without</td>
<td>Large</td>
</tr>
</tbody>
</table>
6. Step Response

Figure 13 illustrates the measured step responses for the three different conditions of linear ball guides. As is shown in the figure, there is no significant difference between the CD1 and CD2 responses, whereas the CD3 response substantially differs from the other responses. Note that the time constants of the step responses were identified as 12, 9, and 100 ms, respectively.

This phenomenon can be analytically explained as follows: the plant dynamics of the system shown in Figure 4 should be changed from \( P(s) = 1/Ms^2 \) to \( P(s) = 1/(Ms^2+Ks) \) to measure the NSB of the linear ball guides on a microscopic scale, where \( M \) is the mass of the moving part and \( K \) is the spring constant of the NSB. In the case of \( P(s) = 1/(Ms^2+Ks) \), the characteristic equation of the closed-loop control system can be described via Equation (10).

\[
s^3 + K, s^2 + (K, K_p + \frac{K_p}{T_i} + \Omega^2)s + \frac{K, K_p}{T_i} = 0
\]

where \( \Omega^2 = K_s/M \).

This equation suggests that the NSB yields effects that are similar to those of a damper. The roots of the characteristic equation, which are composed of a real root and a vibrational root pair, are calculated as:

\[
\begin{align*}
CD1 & : -53.7, -108 \pm i 976 \\
CD2 & : -68.7, -100 \pm i 862 \\
CD3 & : -6.86, -131 \pm i 3546.
\end{align*}
\]

The calculated real roots were found to be in agreement with the measured time constants. The NSB stiffnesses were measured as 13.7, 9.0, and 270 N/μm, and calculated real roots are -53.7, -68.7 and -6.86, for CD1, CD2 and CD3, respectively. A higher stiffness of the NSB results in slower responses. It is a damper effect to make a response to be slow. These results clearly show that the NSB of the guide acts as a kind of damper and stabilizes the control system. This also explains why high-control gains could be set in the experimental system used in this study.

7. Steady Vibration

Steady vibrations, which refer to the vibrations at zero position, were evaluated for the three linear ball guide conditions. The amplitudes of the vibration were found to decrease in the order of CD2, CD1, and CD3, as is shown in Figure 13. In Figure 14, which illustrates the relationship between the compliance of NSB at the zero position and the standard deviation of the vibration, it can be seen that the standard deviation is proportional to the compliance. This result indicates that the vibrational force disturbance due to electrical noise may act on the moving part of the stage, and that the steady vibration induced by this force disturbance is suppressed by the NSB of the linear guides, which have relatively high stiffness.
The moving part of the stage near the zero position can be modeled as Figure 15, where $x_s$ is the displacement and $f_{\text{noise}}$ is the disturbance force. The evaluated stiffness for each zero-position NSB was 13.7, 9.0, and 270 N/μm for CD1, CD2 and CD3, respectively. A schematic representation of the frequency response of the mass-spring system is shown in Figure 16. Because the angular resonant frequency $\omega_n$ is defined as $(K_s/M)^{0.5}$, changing the spring constant changes the amplitude and resonant frequency. Under the assumption that the force disturbance $f_{\text{noise}}$ is represented as white noise, the amplitude of the steady vibration should be influenced by the NSB of the linear guides.

To investigate the influence of the NSB on the step response and steady vibration, force disturbance simulations were carried out. Figure 17 shows the block diagram of the feed-drive system, which is a simplified model of NSB at the zero position, with a force disturbance $f_{\text{noise}}$ and stiffness $K_s$. In the simulation, the disturbance force $f_{\text{noise}}$ is a white-noise signal with amplitude of 0.015 N; this amplitude was selected such that the amplitude of the steady vibration was equivalent to that of the measured vibrations.

Figure 18 shows the nanometer step responses achieved by implementing the force disturbance simulations with the stiffnesses of the linear ball guides. It can be seen from the figure that, as was observed in the experimental results shown in Figure 13, the time constant of the response and amplitude of steady vibration are influenced by the stiffness of NSB. Figure 19 illustrates the simulated relationship between the NSB compliance at the zero position and the standard deviation of the vibration. As is shown in Figure 19, the standard deviation is proportional to the compliance; this finding was also observed in the experimental results shown in Figure 14.

Thus, it can be concluded from the results mentioned above that the NSB of the linear ball guides increases system damping on a microscopic level, and that a lower NSB compliance can suppress the steady-state vibrations of the system.
8. Conclusions

In this study, in order to investigate the relationships between the NSB and microscopic motion behavior of a feed-drive system by using the eight-grooved linear ball guides developed for ultra-precision machines, sinusoidal motion tests and step response tests were carried out for three guide conditions that differed with respect to the ball retainers and preloading. Steady vibrations were also measured and investigated. The conclusions can be summarized as follows:

1) The relationship between the NSB and a quadrant glitch was theoretically investigated under the assumption that NSB can be approximated via a simple first-lag function. The analytic results were supported by the experimental results.
2) The shape and height of quadrant glitches are influenced by the NSB; additionally, the largest quadrant glitch was found to occur under the condition that the guide does not have a ball retainer.
3) On a microscopic scale, NSB acts as a damper, and a higher stiffness of NSB results in slower step responses.
4) NSB can suppress steady vibrations caused by force disturbances, and a higher stiffness of the NSB results in smaller steady vibrations.

This study has served to verify that the NSB of linear ball guides plays a significant role in ultra-precision feed-drive systems driven by a linear motor via experimentation and simulations. Although it yields larger quadrant glitches, a higher stiffness of the NSB corresponds to increased system stability. Thus, users should choose linear ball guides that will yield the most suitable NSB for their purpose. In the future, the authors will try to clarify the strategy required to achieve the desired system characteristics via optimized guide selection and controller design.

References