Influence of Rolling Friction in Linear Ball Guideways on Positioning Accuracy

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Linear ball guideways have been used recently in precision or ultra-precision positioning devices. However, when the inner balls begin to roll or the moving direction reverses, these guideways are subject to rolling friction or nonlinear spring behavior. An ultra-precision device with a linear motor, referred to as a 'tunnel actuator' (TA), has been constructed to measure these phenomena. The application of a TA is beneficial for two reasons: it mostly cancels the attractive magnetic force between the stator and mover (armature), and its magnetic flux leakage is very low. The influence of the nonlinear spring behavior in ball guideways was investigated in this study using the pure driving force from a TA. The equilibrium between the driving force from the TA and the nonlinear spring force provided great accuracy for a positioning stage using a linear ball guideway.

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NOMENCLATURE

x = displacement of the positioning stage
F = applied force

1. Introduction

The required accuracy and resolution for precision and ultra-precision positioning has recently approached the order of nanometers or sub-nanometers. Conventional precision positioning devices use either an AC or a DC servomotor as the actuator, a ball or sliding screw as the transmission mechanism to change from a rotational to a straight movement, and linear guideways for rolling or sliding. Because these devices have frictional characteristics, the accuracy of their positioning can be disturbed. To address this issue, researchers introduced a linear motor with a pneumatic linear guideway. However, problems related to disturbances remained because the pneumatic guideway was not sufficiently stiff.

To achieve precision and ultra-precision positioning, researchers developed an apparatus composed of a linear motor and linear rolling guideways. Because this apparatus produced friction only from the guideways, techniques to compensate for that friction became very important to achieve precision or ultra-precision positioning. Futami *et al.*[1,2] found that the spring constants can be separated into three regions according to the relationship between the applied force and displacement. Moreover, they found that the nonlinear spring behavior expressing the relationship between the applied force and resulting displacement differed significantly in each area. However, they did not determine how the nonlinear spring behavior influenced the positioning accuracy.

The goal of this study is to construct an apparatus composed of a linear motor with two linear ball guideways and to measure how the nonlinear spring behavior in the linear rolling guideways affected the positioning accuracy of the device.

2. Explanation of Nonlinear Spring Behavior

Nonlinear spring behavior, which is the relationship between the applied force F and resulting displacement x, appears in precision positioning devices that utilize a rolling element such as a linear ball guideway.[3] As shown in Fig. 1, a preload is applied in a vertical direction to the upper race of the rolling element. In addition, a horizontal force $F = F_1 \sin(2\pi ft)$ is applied to the upper race with a low frequency f to prevent inertial forces. The relationship between the applied force and the relative displacement of the upper and lower races can be separated into two regions as follows.

(I) When the amplitude of force F_1 is equal to or less than the rolling friction F_0, the relationship between the applied force F and resulting displacement x appears as hysteresis loops like AA', BB', and CC' in Fig. 2. This area is referred to as the 'nonlinear spring area' and the behavior of hysteresis loops in this area is referred to as the 'nonlinear spring behavior curve' (NSB curve).

(II) When the amplitude of force F_1 is greater than the rolling friction F_0, the relationship between the applied force F and resulting displacement x appears as a curve like $D_1D_2D_3D_4$, which differs significantly from the NSB curve. The displacement x is substantially greater in parts of curves D_2D_3 and D_4D_1, where the rolling elements

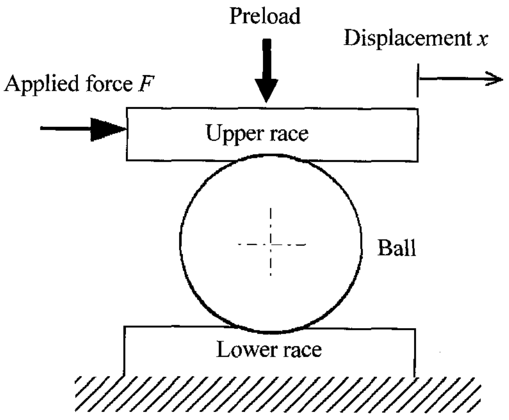


Fig. 1 Rolling element

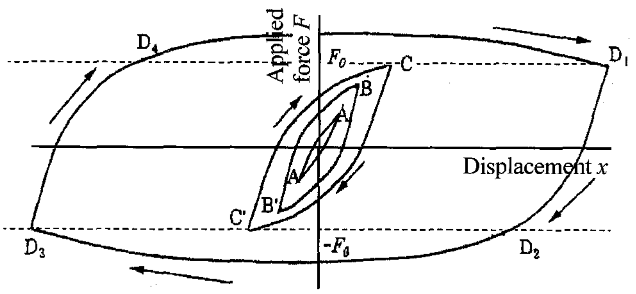


Fig. 2 Nonlinear spring behavior

are in motion. The areas of the rolling state are referred to as the 'rolling area'.

The frictional behavior and frequency characteristics in these two areas differ significantly from each other.

3. Experimental Setup

Figure 3 illustrates the experimental apparatus, which was composed of a linear motor and linear rolling guideways to obtain ultra-precision positioning. Figure 4 presents a cross-section of a linear guideway that uses balls as the rolling elements. The linear ball guideways were manufactured by THK Co., Ltd., and had a length of 510 mm and a width of 25 mm. They also had attached seals and a normal preload. Figure 5 shows details of the linear motor, which was a tunnel actuator (TA) developed by Dr. Houng-Joong Kim of Hitachi, Ltd.[4] A TA has many advantages over conventional linear motors: its assembly is much easier because it mostly cancels the attractive magnetic force between the stator and mover (armature), it is more efficient because its magnetic flux leakage is very low, it allows for a small core equipped motor, and it provides a smaller detent for the maximum force. This study only considered the influence of the ball guideway NSB due to the pure driving force of the TA. The displacement sensor used for the stage displacement feedback had a resolution of 20 nm and a scale length of 460 mm. The total mass of the stage, which consisted of an aluminum alloy, four guideway blocks, and the linear motor mover, was 11.5 kg.

Figure 6 illustrates the control system used for the experiments. Position x was measured using a linear encoder attached to the apparatus. Error Δx was calculated using position x and reference position x_{ref}. The controller board was a P-MAC product with closed loops for displacement and velocity. Proportional and integrated controls were used for the displacement loop, and proportional control was used for the velocity loop. In the figure, K_{pp} is the proportional gain of the displacement, K_{pi} is the integrated gain of the displacement, and K_{vp} is the proportional gain of the velocity; these gain parameters were determined by auto-tuning the controller. The TA was moved by an amplified current, and the linear ball guideways

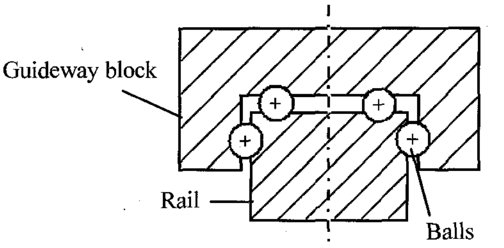
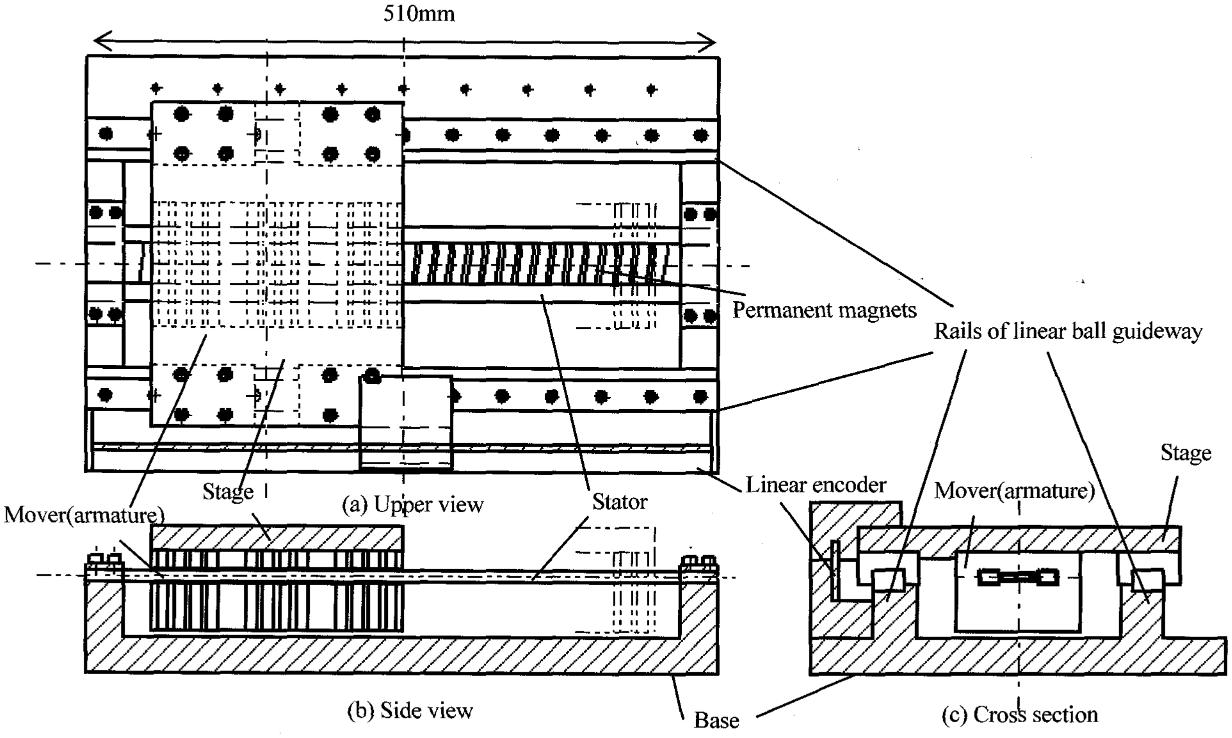


Fig. 4 Diagram of a linear ball guideway cross-section



(a) Upper view

(b) Side view

(c) Cross section

Fig. 3 Experimental apparatus

had rolling friction. The frictional behavior was simulated from the velocity using the Bristle model.[5] This model can express sliding frictional behavior, but in a previous study, it was also used to express rolling frictional behavior.[5] Because of the existence of frictional forces, frictional behavior was caused by the velocity based on the method shown in Fig. 6.

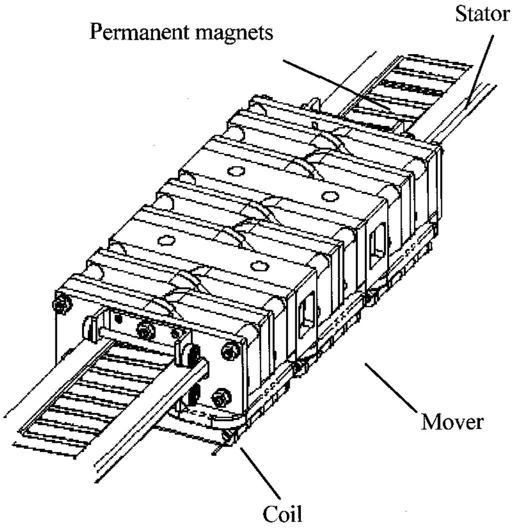


Fig. 5 Linear motor details

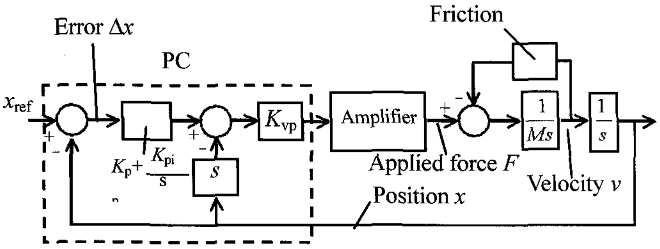


Fig. 6 Control system

4. Experimental Methodology and Results

4.1 Nonlinear spring behavior

As discussed in Section 2, the nonlinear spring behavior can be expressed by the relationship between the displacement x in μm and applied force F in N. When x is $x_1 \sin(2\pi f t)$, the frequency f is 0.1 Hz. The inertial force is almost negligible because the stage acceleration is very small.

Figure 7 illustrates the experimental results. The ordinate expresses the applied force F, and the abscissa is the displacement x. Measurements were taken over three periods; only the second is shown. The influences of starting and finishing each measurement can be excluded. Figure 7(a) illustrates the behavior in the nonlinear spring area and Fig. 7(b) illustrates the rolling behavior. In Fig. 7(a), the displacement reference of the nonlinear spring behavior has an amplitude x_1 of 1, 2, 3, 5, 10, and 15 μm. As displacement amplitude increased, the hysteresis loop slope became smaller. This finding is similar to the results of a previous study [5] and is mainly caused by the decreasing spring hardness due to the decreased adhesive area between the balls and contact surfaces such as rails.[6] In Fig. 7(b), the displacement reference amplitude ranges from 20 to 100 μm every 20 μm and 160 μm. When the applied force was less than 27 N (=F_0), the trajectory of the measurement results indicated a nonlinear spring behavior, but when the applied force was greater than 27 N, the trajectory of the large displacement indicated a rolling behavior. As a result, the nonlinear spring and rolling areas were bounded by an applied force F_1 of 27 N and displacement of 70 μm, respectively.

4.2 Step response

4.2.1 Displacement behavior

The step response is one expression of dynamic behavior. The reference position x_{ref} was set at 10, 100, and 200 μm. Figure 8 shows the measurement results. The left ordinate is the position x in μm, the right ordinate is the applied force F in N, and the abscissa is the time t in s. In all three cases, an applied force of 5 N was generated for the nonlinear spring behavior of the linear ball guideway at the start of the test and position x = 0 was maintained.

(a) 10-μm step response

Figure 8(a) presents results for a 10-μm step response. An applied force F was generated to drive the step, but a large force was not required since the reference position of 10 μm was very small. The maximum applied force was 30 N. After achieving a 10-μm step response, an applied force of 15 N was generated to maintain the position. Therefore, the spring force of the position in the nonlinear spring area was balanced by the applied force for balancing positioning, which was achieved without any overshoot.

(b) 100-μm step response

Figure 8(b) presents the results for a 100-μm step response. The stage moved by generating a momentary applied force of 300 N and positioning was completed at the reference position of 100 μm after an overshoot of 2 μm. Then, an applied force of 10 N was generated to balance the spring force of the nonlinear spring behavior and maintain the position.

(c) 200-μm step response

Figure 8(c) presents the results for a 200-μm step response. Once again, the stage moved by generating a momentary applied force of 300 N and positioning was completed at the reference position of 200 μm after an overshoot of 53 μm. However, almost no applied force was generated after the final positioning, indicating that the spring force of the guideway at the reference position was balanced.

4.2.2 Relationship between displacement and applied force

Figure 9 shows the results in the same manner as the nonlinear step behavior using the three different step responses. The ordinate is the applied force F in N and the abscissa is the displacement x in μm. These results give the dynamic behavior whereas the nonlinear spring results were static.

(a) 10-μm step response

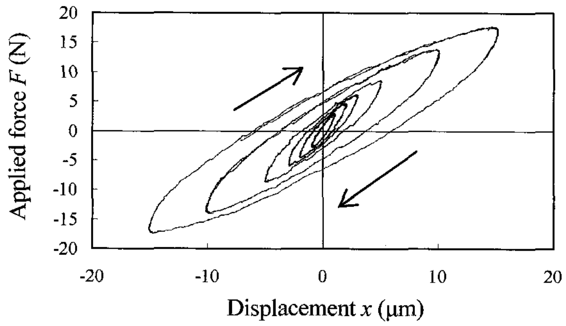
Figure 9(a) presents the results for a 10-μm step response. The positioning was accomplished with an applied force F of 14 N, which is illustrated by the nonlinear spring behavior at position 10 μm. The position was maintained by generating this applied force because of the influence of the nonlinear spring behavior. The positioning stage should return to 10 μm when the applied force becomes 0 N by turning off the servo because of the spring force, as shown in Fig. 7(a).

(b) 100-μm step response

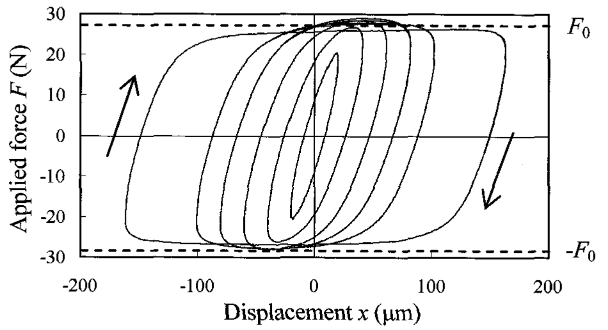
Figure 9(b) presents the results for a 100-μm step response. Here, the positioning was accomplished with an applied force of 10 N after an overshoot 2 μm. In this case, the position returned to 5 μm due to the influence of the nonlinear spring force since the generated applied force was 10 N.

(c) 200-μm step response

Figure 9(c) presents the results for a 200-μm step response. In this case, positioning was accomplished with an applied force of 0 N after an overshoot of 50 μm. The position should not return because the applied force is balanced with the force of the nonlinear spring behavior.

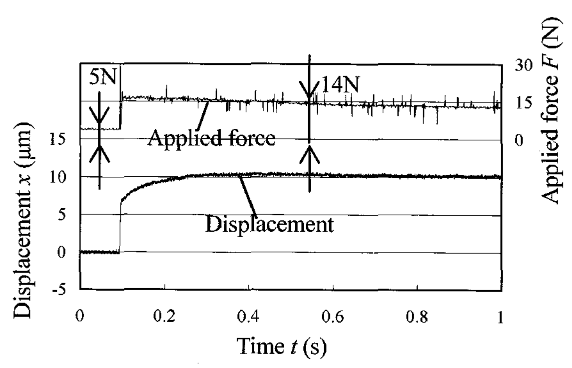


(a) Nonlinear spring area

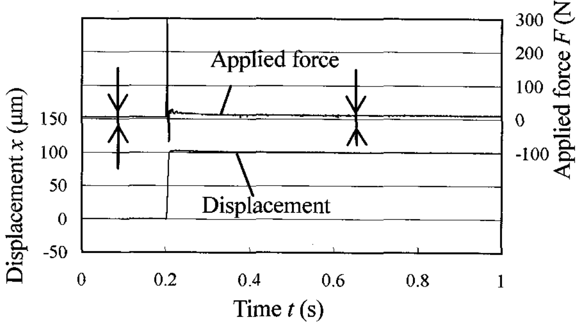


(b) Rolling area

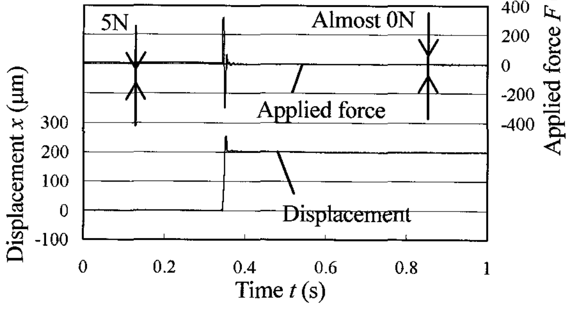
Fig. 7 Measured nonlinear spring behavior



(a) $x_{ref} = 10$ μm

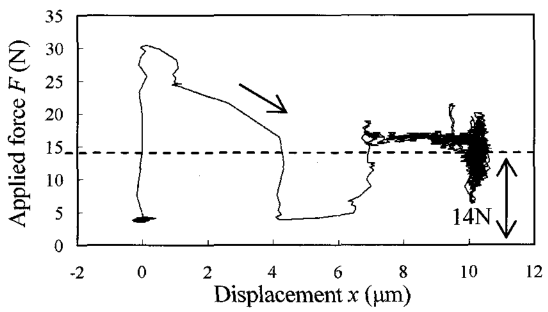


(b) $x_{ref} = 100$ μm

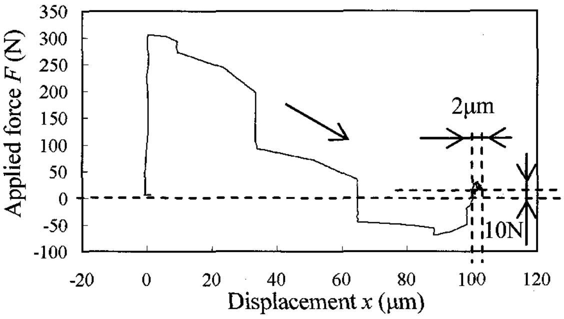


(c) $x_{ref} = 200$ μm

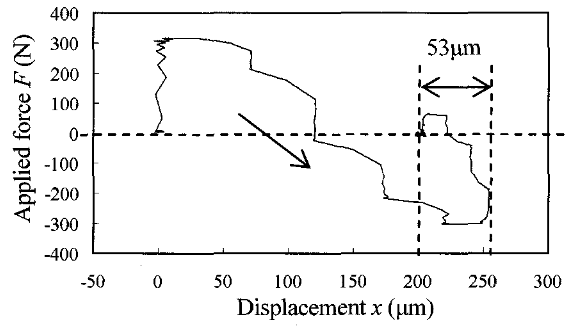
Fig. 8 Step response results



(a) $x_{ref} = 10$ μm



(b) $x_{ref} = 100$ μm



(c) $x_{ref} = 200$ μm

Fig. 9 Relationship between applied force and displacement of the step response

In all the results, the displacement route from start to finish was strongly influenced by the controller, the details of which could not be obtained from the controller manufacturer. This was likely caused by the controller dead-band.

These results indicate that the position was maintained when the spring force of the linear ball guideway in the nonlinear spring area was balanced with the applied force after the positioning of the step response was completed. However, if positioning is completed after a rolling behavior appears, the applied force is not generated by the spring force.

4.3 Trapezoid acceleration and deceleration

When Jog-sending was used for the positioning, the acceleration and deceleration exhibited a trapezoidal velocity waveform, as shown in Fig. 10. Here, the left ordinate is the velocity v in mm/s, the right ordinate is the applied force F in N, and the abscissa is the time t in s. The reference position was 100 mm, the maximum velocity was 100 mm/s, and the acceleration and deceleration times were 50 ms. When the velocity was 0 mm/s, the positioning stopped for 0.2 s, but an applied force of 10 N was generated. When a displacement of several micrometers occurred, an applied force was generated by the nonlinear spring behavior of the linear ball guideway. The position began moving in the positive direction at 0.2 s. An applied force of 150 N was generated to overcome static frictional force that existed in

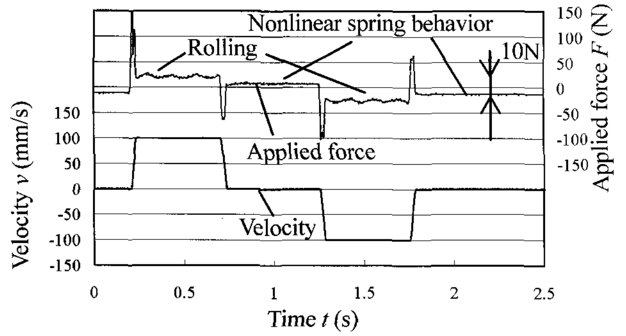


Fig. 10 Results of the trapezoid wave velocity command

the linear ball guideway for the purpose of acceleration. After the acceleration, the stage moved at a constant velocity of 100 mm/s. The purpose of deceleration was to give the motor a negative current and create friction in the guideways. The stage stopped at the position where the velocity was 0 mm/s, but an applied force of 10 N was generated by the spring force of the nonlinear spring behavior of the guideway. The results were similar for driving in the negative position.

The applied force was smaller during deceleration than during acceleration: the average maximum applied force was 167 N during acceleration and 86 N during deceleration, as shown in Fig. 10. This was because the frictional force of the linear ball guideway acted as an obstruction during acceleration and assisted during deceleration.

Finally, an applied force of 10 N was generated after 1.8 s. This resulted due to the stop while balancing the spring force of the guideway because the applied force was generated to maintain the position, similar to the other results.

5. Conclusions

An experimental apparatus was constructed using a tunnel actuator linear motor and two linear ball guideways. Measurements were recorded of the nonlinear spring behavior exhibited by the static behavior relationship between the applied force and resulting displacement, the step response expressed in terms of the dynamic behavior, and the behavior of trapezoid acceleration and deceleration velocity. The results yielded the following conclusions.

1) A positioning apparatus using two linear ball guideways can be implemented with 10 μm-positioning.
2) If a rolling element such as a linear ball guideway is used, positioning in the nonlinear spring area is accomplished when the spring force of the rolling element is balanced with the applied force of the linear motor.

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