Simultaneous Identification of Linear Parameters and Nonlinear Rolling Friction for Ball Screw Driven Stage

Tadashi Takemura and Hiroshi Fujimoto
The University of Tokyo
5-1-5, Kashiwanoha, Kashiwa, Chiba, 227-8561 Japan
Phone: +81-4-7136-3881
Fax: +81-4-7136-4132
Email: take@hfllab.k.u-tokyo.ac.jp, fujimoto@k.u-tokyo.ac.jp

***Abstract*—Ball screw driven stages have nonlinear friction and its influences appear in frequency characteristic and positioning. Generally, linear parameters are estimated in conditions without nonlinear friction effect, and nonlinear friction characteristic is obtained in conditions without linear parameters effect. In this paper, the method which can identify linear parameters and nonlinear rolling friction characteristic simultaneously is proposed. Experimental results show the accuracy of proposed linear parameters identification and effectiveness of rolling friction compensation which derives from the characteristic extracted by proposed method.**

I. Introduction

Ball screw driven stages are used for industrial equipments such as machine tools and semiconductor equipments. High speed and precision positioning is necessary to enhance productivity and micro fabrication technology of the system.

Ball screw driven stage includes ball screw and rolling guide. These components cause nonlinear friction because of their rolling elements. In particular, nonlinear friction which is called rolling friction deteriorates positioning accuracy while motion reversal and settling. There are two approaches to compensate rolling friction, one is model-based control and the other is iterative-based control. In model-based control approach, it is necessary to model rolling friction and precise modeling is effective to improve tracking performance. Heretofore, GMS model [1], Rheology model [2], data-based model [3], and so on have been proposed as the precise rolling friction model. On the other hand, in iterative-based control approach, it is possible to decrease tracking error by iteration [4]. Therefore, rolling friction modeling is unnecessary. In author's research group, model-based approach and iterative-based approach have been proposed. Moreover, combination control method of these two approaches has been proposed [5].

Also nonlinear friction causes gain lowering and phase lead phenomena in low frequency of the frequency response[6]. Generally speaking, in designing feedback or feedforward controller, designers measure frequency characteristic of the system at first. Then designers determine linear model of the system, and adjust parameters of the model to satisfy measurement characteristic. However, in ball screw driven system, modeling error arises by using this technique because frequency response includes effects of nonlinear friction. Thus, in order to remove effects of nonlinear friction, designers need to input large amplitude signal in measuring frequency response. However, the input signal is limited by the motor ratings etc., and it is difficult to remove the influence of the nonlinear friction strictly. Model identification by step response is not suitable for the same reason. Constant speed examination can identify the viscous friction but number of trails gets large to identify accurately.

Likewise in designing nonlinear friction compensator, designers fit the model to obtained characteristic. Nonlinear frictions which exist in ball screw driven stage are Stribeck effect and rolling friction. In contrast, rolling friction appears when reversal or settling motion happens. In most case, tracking accuracy is deteriorated by rolling friction. Therefore, in order to improve tracking performance, modeling of rolling friction is essential. Although designers have to obtain rolling friction characteristic for compensation, this characteristic can be obtained only at ultra low speed. Thus rolling friction identification is a time-consuming task.

As stated above, the problem is that linear parameters and nonlinear rolling friction cannot be identified in the same trial. In this paper, simultaneous identification method of linear parameters and rolling friction based on equation of motion and rolling friction characteristic is proposed as this solution. In addition, rolling friction compensator which is constructed from the characteristic obtained by proposed method is applied to position control system. Experimental results show the effectiveness of this compensator as well as conventional one.

II. Experimental stage

The experimental X-Y ball screw driven stage is shown in Fig. 1. The ball screw is directly connected with the shaft of the servo motor through the coupling. The servo motor is equipped 2^{20} pulse/rev (resolution 11.4 nm) absolute encoder. In addition, high speed current feedback control

Fig. 1. Experimental stage.

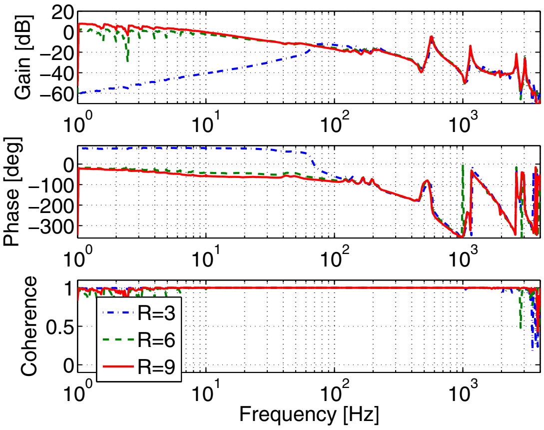
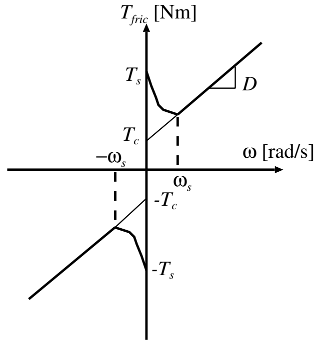
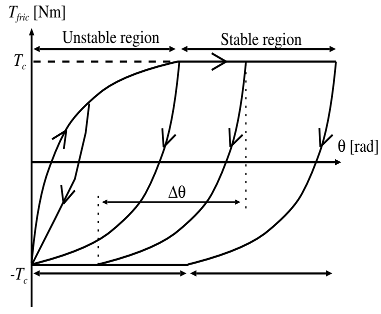


Fig. 2. Frequency response.



(a) ω–T_{fric}



(b) Rolling friction characteristic.

Fig. 3. Friction characteristics.

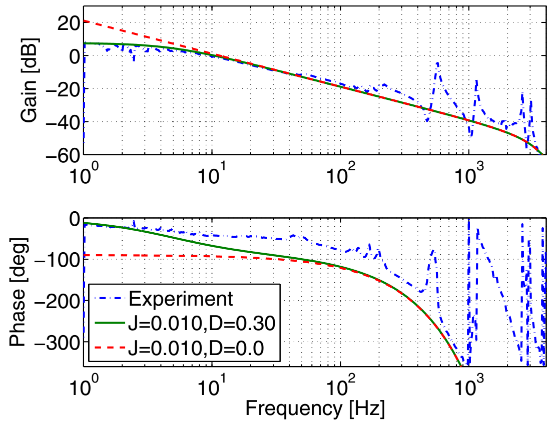


Fig. 4. Curve fitting.

is implemented to the servo motors. In this paper, position control is examined only X stage using the mounted encoder.

Fig. 2 shows the frequency response when the current reference $i_{ref} = R\sin(2\pi ft)$ is given to the servo motor by using FFT analyzer. Gain lowering and phase lead phenomena caused by nonlinear friction appear when R is small.

III. Nonlinear Friction

The nonlinear frictional force T_{fric} in the ball screw driven stage is caused by the rolling elements which exist in a ball screw and a linear guide. T_{fric} is attributed to elastic deformation of the balls and lubricant condition. T_{fric} can be divided into two kinds of frictional forces which depend on velocity and displacement.

The frictional force which depends on velocity shows the characteristic drawn in Fig. 3(a). At this time, frictional force is represented as sum of Stribeck effect, Coulomb friction and viscous friction.

$$T_{fric} = \begin{cases} u & (\omega = 0) \\ \{D|\omega| + T_c + (T_s - T_c)e^{-\alpha|\omega|}\}\mathrm{sgn}(\omega) & (\omega \neq 0) \end{cases}$$

Here, D is viscous friction, ω is motor angular velocity, u is motor torque, T_c is Coulomb friction, T_s is static friction, α is a coefficient which determine the damping of the Stribeck effect and sgn$(\cdot)$ is signum function, respectively. In static condition, T_{fric} is equal to u. Once u overcome T_s, motor starts moving. After initiation, T_{fric} decreases from T_s exponentially. This is called Stribeck effect. Stribeck effect only appears in $\omega < \omega_s$, here ω_s is ultra low speed. When ω is fast ($\omega > \omega_s$), T_{fric} is represented as sum of viscous friction and Coulomb friction.

On the other hand, frictional force which depends on displacement from motion reversal is called rolling friction. Rolling friction has hysteresis property shown as Fig. 3(b). Here, T_{roll} is defined as rolling friction. After motion reversal, friction force varies, just like nonlinear spring, depending on displacement from the position which motion reversal happened (unstable region). When displacement gets larger than $\Delta\theta$, friction force shows Coulomb friction (stable region). Here, it is assumed that Stribeck effect just influences at ultra low speed. Hence, only rolling friction deteriorates the positioning performance at practical use speed.

IV. Identification of Linear Parameters and Nonlinear Rolling Friction

A. Conventional identification

1) Linear parameters identification: In general, designers have to identify the plant before designing controllers. In order to identify linear parameters of the plant, curve fitting, a technique that designers get frequency response of the plant and

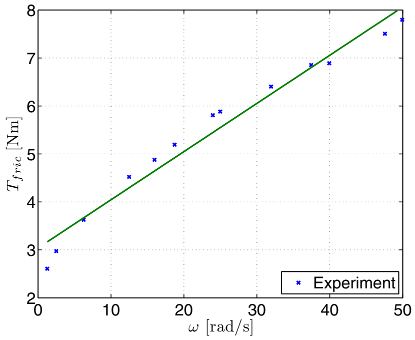


Fig. 5. Results of constant velocity examination.

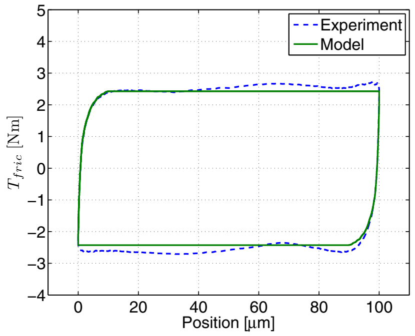


Fig. 6. Rolling friction characteristic and its model (conventional).

fit the model which designers determine to the plant frequency response, is often adopted. However, gain lowering and phase lead phenomena as seen in Fig. 2 happen in ball screw driven stage. If designers use curve fitting to the less gain frequency response, modeling error occurs at viscous friction D. Thus, curve fitting is not suitable technique to identify the linear parameters for ball screw drive stage. Identification using step response is also influenced by nonlinear friction. For these reasons, viscous friction is often identified by constant speed examination.

In this paper, transfer function from current to velocity is treated as a first order system.

$$P(s) = \frac{\omega}{i} = \frac{K_T}{Js + D} \tag{1}$$

Here, K_T is torque constant, J is inertia, D is viscous friction, respectively. Fig. 4 show the results of curve fitting using $R = 9$ of Fig. 2 and parameters are $K_T = 0.72$, $J = 0.010$ and $D = 0.10$.

Meanwhile, results of constant speed examination and it's first-order approximation are shown in Fig. 5. Slope of this line, viscous friction D, is $D = 0.14$.

2) Nonlinear friction indentification: High positioning accuracy is disturbed by rolling friction. Thus, rolling friction compensation is required for improvement of tracking performance. In order to design rolling friction compensator, rolling friction characteristic has to be extracted by ultra low speed experiments.

Fig. 6 presents the rolling friction characteristic and its model of the experimental stage. At this time, sinusoidal position reference whose amplitude and frequency are 50 μm and 0.05 Hz was given. Here, horizontal axis is translatory position which is calculated from motor position and transfer coefficient $\frac{12}{2\pi}$ mm/rad. From this figure, nonlinear spring behavior is confirmed within 10 μm after motion reversal.

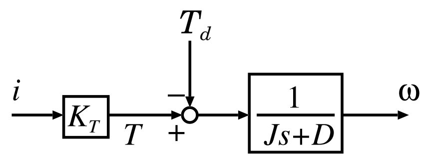


Fig. 7. Block diagram of plant.

B. Simultaneous identification of linear parameters and nonlinear rolling friction (Proposed method)

Problems of conventional identification of linear parameters and rolling friction characteristic are summarized as follows.

- **Identification linear parameters by curve fitting**
  Modeling error happens if fitting frequency response is influenced by rolling friction.
- **Identification of viscous friction D by constant speed examination**
  In order to identify D precisely, number of experiments has to be large.
- **Identification of nonlinear rolling friction by ultra low speed examination**
  It costs time because this operation is ultra low speed.

For these reasons, simultaneous identification of linear parameters and nonlinear rolling friction which is based on motor equation of motion and rolling friction characteristic is proposed.

Fig. 7 shows the block diagram from current to velocity. From the motor equation of motion, disturbance torque T_d is represented as

$$\begin{aligned} T_d &= K_T i - J\dot{\omega} - D\omega & (2) \\ &= K_T i - (J_n + \Delta_J)\dot{\omega} - (D_n + \Delta_D)\omega, & (3) \end{aligned}$$

where J_n and D_n are nominal inertia and viscous friction, Δ_J and Δ_D are moeling error of J and D, respectively. Under ideal situation, $\Delta_J = \Delta_D = 0$, disturbance torque is assumed as only rolling friction T_{roll}.

$$\begin{aligned} T_d &= K_T i - J_n\dot{\omega} - D_n\omega \\ &= T_{roll} & (4) \end{aligned}$$

From (3) and (4), T_d is derived as

$$T_d = T_{roll} - \Delta_J\dot{\omega} - \Delta_D\omega. \tag{5}$$

Conventional identification of rolling friction is operated with ultra low speed because modeling errors, Δ_J and Δ_D, can be ignored under $\dot{\omega} = \omega = 0$. In other words, rolling friction cannot be extracted in practical use speed. In contrast, in proposed method, rolling friction characteristic is reconstructed by parameter tuning. The goal of parameter tuning is to satisfy

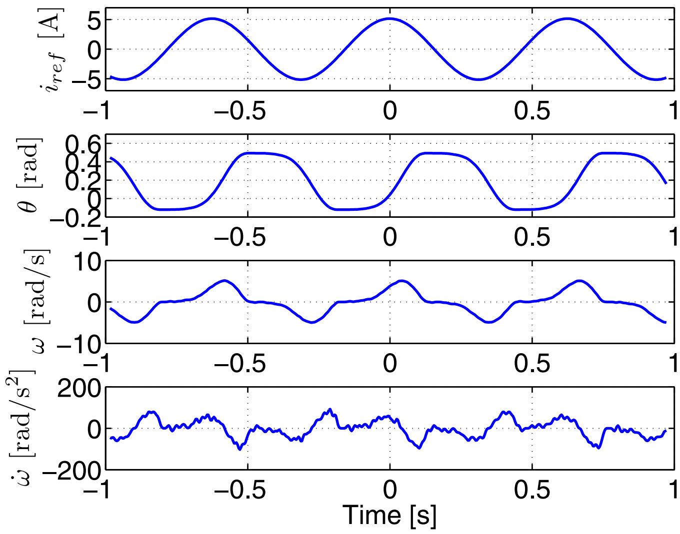
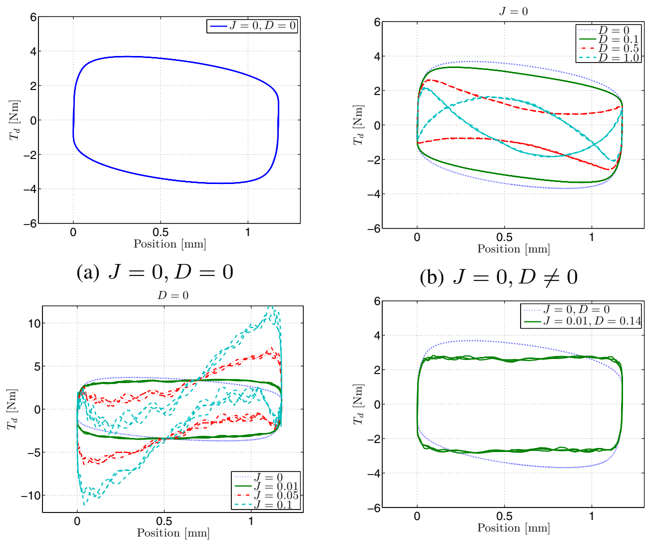


Fig. 8. Time responses on current control.



(a) $J = 0, D = 0$
(b) $J = 0, D \neq 0$
(c) $J \neq 0, D = 0$
(d) $J = 0.010, D = 0.14$

Fig. 9. Shape variation of T_d by J and D (proposed).

(4). Here, free parameters are J and D. When rolling friction characteristic T_{roll} is reconstructed, $J = J_n$ and $D = D_n$ is accomplished. Therefore, proposed method can identify linear parameters and rolling friction simultaneously. Proposed method has an advantage that identification can be executed not only ultra low speed drive condition. However, there are two restrictions. First is that drive condition has to include motion reversal in order to appear T_{roll}. Second is that it has to include the time span whose velocity and acceleration are same sign in order to prevent $\Delta_J \dot{\omega} = -\Delta_D \omega$. Sinusoidal current or position drive are examples drive condition which satisfied these restrictions.

One of identification results using proposed method is described as follows. Fig. 8 shows the time responses of i, θ, ω and $\dot{\omega}$ when current reference $i_{ref} = 5\sin(2\pi 1.6t)$ is given. Note that ω and $\dot{\omega}$ were calculated by central difference of θ. Additionally, high frequency content in all wave profiles is removed by zero phase low pass filter. From these profiles and (2), J_n and D_n are identified according to following steps.

1) T_d is calculated by substituting i, $\dot{\omega}$ and ω into (2). At this time, the values of J and D are determined as $J = D = 0$. Then T_d is plotted like Fig. 9(a), where x-axis is θ and y-axis is T_d.
2) Let us consider the case J is fixed and D is changed. Then T_d is deformed by variation of D like Fig. 9(b). In this step, in order to become parallel upper and lower side of wave shape, D was determined ($D = 0.01$).
3) Likewise previous step, let us consider the case J is changed and D is fixed. Then T_d is deformed by variation of D like Fig. 9(c). At this time, in order to become parallel T_d and x-axis, J was determined ($J = 0.1$).
4) In order to satisfy (4), determined parameters are fine tuned. When fine tuning is accomplished, T_{roll} is reconstructed from T_d and linear parameters are identified.

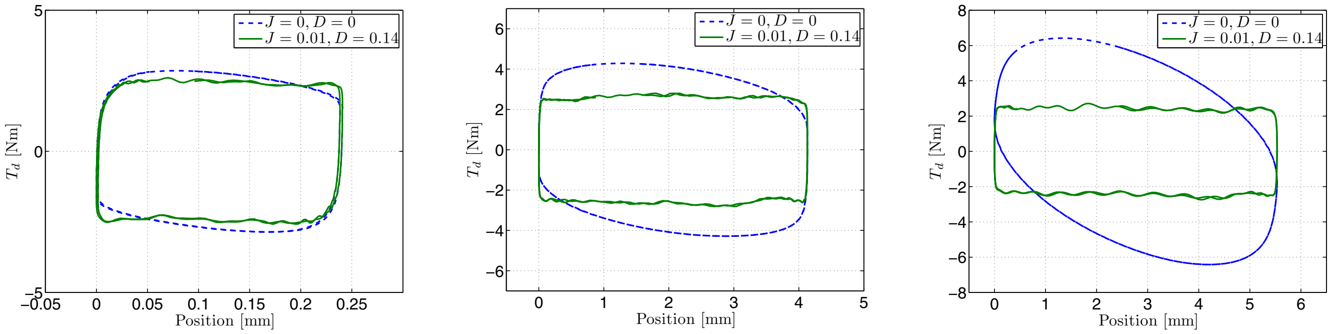
According to these operations, the value of J and D was identified as $J_n = 0.010$ and $D_n = 0.14$. These values were same as the results which were identified using curve fitting and constant speed examinations. Fig. 10 represents the identification results in other drive conditions. In all conditions, T_{roll} is reconstructed when $J_n = 0.010$ and $D_n = 0.14$. For these results, proposed method is effective to linear parameters identification. Furthermore, from Fig. 10(a), displacement within 20 μm from motion reversal of reconstructed characteristic shows nonlinear spring behavior. Thus it can be said that proposed method can identify rolling friction simultaneously.

V. EXAMINATIONS

In this section, it is verified that rolling friction compensator which is derived from proposed method is effective for tracking performance. Here, conventional and proposed model are defined as the models which are constructed from ultra low speed examination and proposed model method, respectively. Both were modeled with data-based friction model[5] which is a table of displacement from motion reversal and friction force. Data-based friction model can compensate rolling friction at arbitrary trajectory if displacement from motion reversal and current position were given.

Block diagram of control system is illustrated in Fig. 11. This system is consisted of feedback controller C_{fb}, multirate feedforward controller C_{ff} and data-based friction model. Under ideal condition, perfect tracking is achieved [7]. C_{fb}, PID feedback controller, was designed whose position band frequency becomes 40 Hz using pole placement method. C_{fb} and C_{ff} are respectively designed using $J_n = 0.010$ and $D_n = 0.14$.

Fig. 12, Fig. 13 and Fig. 14 shows the experimental results of position reference, tracking error and control input, respectively.



(a) $i_{ref} = 4\sin(2\pi 2t)$ (b) $i_{ref} = 6\sin(2\pi t)$ (c) $i_{ref} = 9\sin(2\pi 2t)$

Fig. 10. Estimation results of different situations.

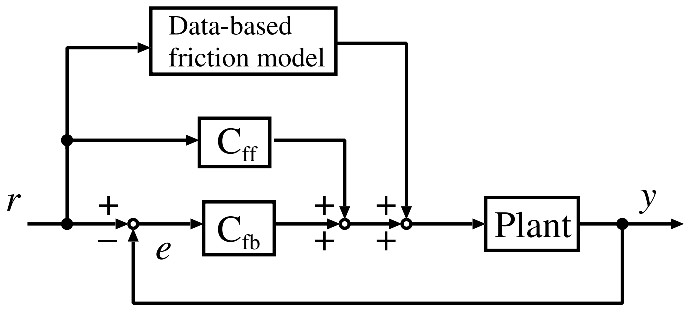


Fig. 11. Block diagram of control system.

Sinusoidal waves represented in Fig. 12 were given as a position reference. Fig. 13 and Fig. 14 show the tracking errors and control input, respectively. Therefore, it is verified that proposed method can identify rolling friction characteristic even if under practical use speed.

VI. Conclusion

In this paper, simultaneous identification method of linear parameters and nonlinear rolling friction is proposed. Although linear parameters and nonlinear rolling friction are conventionally identified in different examinations, proposed method makes it possible to identify these in the same examination. Identified parameters obtained by proposed method were same value as conventional method. Furthermore, rolling friction characteristic obtained at the same instant is effective for rolling friction compensator and it performs as well as conventional one obtained from ultra low speed experiment.

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References

[1] Z. Jamaludin, H. Van Brussel, J. Swevers, “Quadrant Glitch Compensation using Friction Model-Based Feedforward and an Inverse-Model-Based Disturbance Observer”, 10th IEEE InternationalWorkshop Advanced Motion Control, 26-28, pp.212–217, 2008.

[2] Y. Maeda, M. Iwasaki, “Improvement of Settling Performance by Initial Value Compensation Considering Rolling Friction Characteristic”, 36th Annual Conference of the IEEE Industrial Electronics Society, pp.1896-1901, 2010.

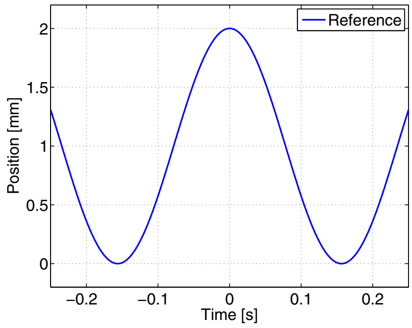
[3] T. Takemura, T. Shiraishi, H. Fujimoto, “Examination of Friction Compensation for NC Machine Tools Driven by Ball Screw Stage-Comparison of Mathmatical-based Model with Data-based Friction Model-”, IIC-10-161, pp.7-12, 2010

[4] H. Asaumi, H. Fujimoto, “Proposal on Presice Positioning Control of Ball Screw Stage Based on ILC with PTC”, IIC-07-128, pp.71-76, 2007

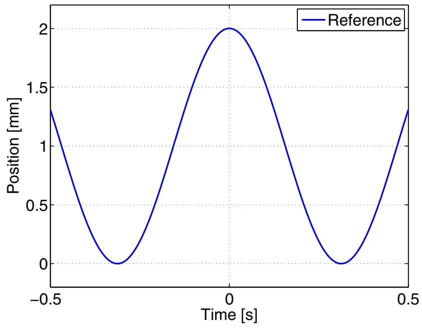
[5] T. Takemura, H. Fujimoto, “Proposal of Novel Rolling Friction Compensation with Data-based Friction Model for Ball Screw Driven Stage”, 36th Annual Conference of the IEEE Industrial Electronics Society, pp. 1926-1931, 2010.

[6] Y. Maeda, M. Iwasaki, M. Kawafuku, H. Hirai, “Nonlinear Modeling and Evaluation of Rolling Friction”, IEEE International Conference on Mechatronics, pp.1-6, 2009.

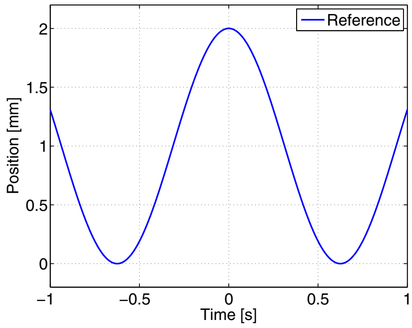
[7] Hiroshi Fujimoto, Yoichi Hori, Atsuo Kawamura, “Perfect Tracking Control based on Multirate Feedforward Control with Generalized Sampling Periods”, IEEE Trans. Industrial Electronics , vol.48, No.3, pp.636–644, 2001



(a) $f = 3.2$ Hz

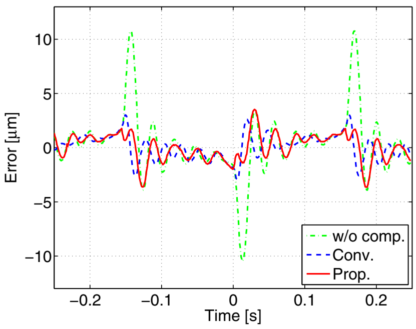


(b) $f = 1.6$ Hz

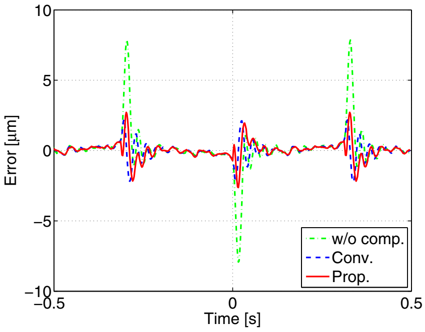


(c) $f = 0.8$ Hz

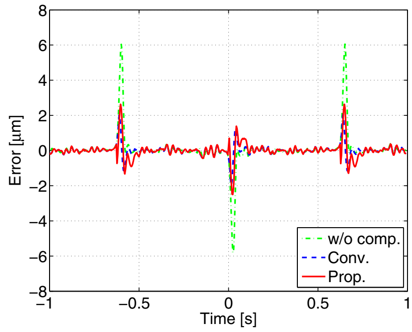
Fig. 12. Target trajectory.



(a) $f = 3.2$ Hz

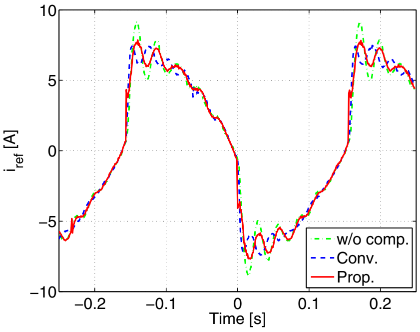


(b) $f = 1.6$ Hz

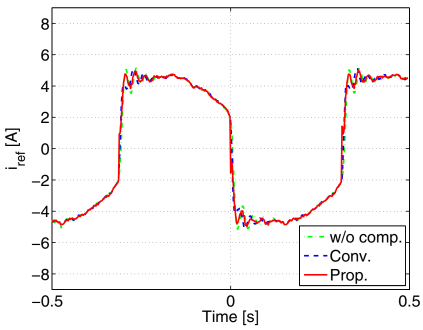


(c) $f = 0.8$ Hz

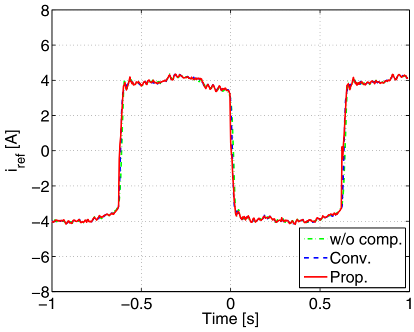
Fig. 13. Tracking error.



(a) $f = 3.2$ Hz



(b) $f = 1.6$ Hz



(c) $f = 0.8$ Hz

Fig. 14. Control input.