Friction compensation in contouring control for biaxial feed drive systems and experimental verification

Dinh Ba BUI, Naoki UCHIYAMA¹ and Shigenori SANO

¹ Department of Mechanical Engineering, Toyohashi University of Technology, Japan, uchiyama@tut.jp

Abstract:

Friction is a primary disturbance in mechanical motion control of high precision machines. Because the friction causes undesired effects such as steady state and tracking motion errors of industrial machines, control systems should be designed to compensate for frictional effects. In this paper, we propose a simple yet really efficient method for estimating the friction property in biaxial feed drive systems. Our method employs the conventional circular contouring control results in estimating the friction property for contouring controller design. The effectiveness of the proposed method is demonstrated by experimental results for different contour curves in different motion speeds.

Keywords: Machine tool, Friction compensation, Biaxial feed drive dynamics, Contouring control, High-precision machining.

1. Introduction

In machine tool control, position error with respect to each feed drive axis is generally used as feedback signals. However, regarding to machining, error components orthogonal to the desired contour curves are more important than errors regarding to the feed drive axes. Error components orthogonal to the contour curves are called contour errors. There are many controllers proposed for reducing contour error and its efficiency has been proved by experiment.

Among the many proposals for reducing contour error [1], Koren proposed a cross-coupling controller that calculated contour error from drive axis tracking error and reduced contour error [2]. Advanced optimal and nonlinear controllers applied to contouring controller design have had their effectiveness verified [3][4], although they have disadvantage that both contour and tracking errors in a feed drive axis are used to calculate control input, and this compromise contour tracking performance. Considering contour and tracking errors simultaneously, creates difficulties in adjusting controller parameters.

Lo and Chung proposed contouring control for biaxial feed drives based on a coordinate transformation [5], in which feed drive axis tracking error is converted to error components orthogonal and tangential to the desired contour curve. This obtains two decoupled single input/output systems for the orthogonal and tangential directions, and enables controllers for each direction to be designed independently, and hence simplifies controller parameter adjustment. Chiu and Tomizuka proposed a similar approach, whereby a task coordinate frame is defined at the desired position of the feed drive system, and the control system dynamics are reformulated with respect to this coordinate frame correspond to the orthogonal and tangential directions of the desired contour curve, and the orthogonal error component is regarded as an approximation of the actual contour error

[6]. Although this approach provides control system stability for any contour curve, the difference between the actual contour error and the determined orthogonal error component may result in an increase in the contour error. A similar approach for feed drive system with fourth-order dynamics has been presented in [7].

In this paper, we improve the contouring controller design for biaxial feed drive systems developed by one of the authors [8] by including the friction compensation. Many friction models have been proposed thus far, and can be divided into two categories: kinetic and dynamic friction models. The kinetic friction models take into account major friction effects such as the viscous friction, the Coulomb friction and the Stribeck effect [9]. The dynamic friction models such as the LuGre model consider the natural mechanism of friction generation [10]. Although there have been many sophisticated friction models, we verify the contouring control performance by using a simple friction model consisting of the viscous and the Coulomb ones to clarify its effectiveness and limitations. A contouring control result based on a typical circular motion is used for the estimation of friction parameters. The control input is divided into two terms: a nominal dynamics term and a control term. Considering only the control term, we can estimate the friction parameters in a simple manner. Based on the estimated friction parameters, we include a simple friction compensator in a contouring controller. By applying this compensator, the control input profile coincides with the nominal dynamics term in repetitive circular motion experiments, and more than 70% reduction of the contour error is achieved on average. Applying the designed friction compensator with the same friction parameters to several non-circular motion contours in the experiment, we confirm that the effectiveness is maintained and around 70% reduction of the contour error is achieved as well.

2. Feed drive dynamics

2.1 Definitions of tracking errors and contouring control

Fig. 1: Definition of contour error.

Fig. 1 schematically explains the relation between the tracking errors in each feed drive axis and the contour error. The coordinate frame Σ_w , whose axes X_1 and X_2 correspond to the feed drive axes, is a fixed frame. The curve c in the figure is the desired contour curve of the point of a machined part driven by the feed drive system. The symbol $r = [r_1, r_2]^T$ is the desired position of the point of the machined part at time t , and defined in Σ_w . The actual position of the feed drive system is assumed to be $x = [x_1, x_2]^T$, which is also defined in Σ_w . A local coordinate frame Σ_l is also defined, its origin is at r and axes are Σ_{li} ($i = 1, 2$), as shown in the figure. The axis Σ_{l1} is in the tangential direction of c at r , and the direction of l_2 is perpendicular to l_1 . The tracking error vector e_w , which consists of the tracking errors of both feed drive axes, is defined as follows:

$$E_w = [e_{w1}, e_{w2}]^T = x - r \quad (1)$$

This error vector can be expressed with respect to Σ_l as follows:

$$E_l = [e_{l1}, e_{l2}]^T = R^T E_w \quad (2)$$

$$R^T = \begin{bmatrix} \cos\theta & -\sin\theta \\ \sin\theta & \cos\theta \end{bmatrix}$$

where the inclination θ of Σ_l to Σ_w is as shown in Fig. 1. Regard to machining, error component orthogonal to the desired contour curve are more significant than the tracking errors of the feed drive axes. Thus, a controller design is proposed that allows the control performance to be adjusted with respect to each axis of Σ_l independently for feed drive systems with fourth-order dynamics [8].

Because the error component tangential to the contour curve is less significant than the orthogonal one, the controller gain should be set for reducing the error along l_1 to lower than the gain for the error along l_2 , and the proposed approach allows such gain assignment.

2.2 Feed drive dynamics

The controller design presented here for biaxial feed drives driven by servomotors, commonly used in industrial applications, involves feed drive dynamics represented by the following decoupled second-order system:

$$M\ddot{x} + C\dot{x} = v$$

$$M = \text{diag}\{m_i\}, C = \text{diag}\{c_i\}, i = 1, 2, \quad (3)$$

$$v = [v_1, v_2]^T,$$

where $m_i (> 0)$, $c_i (\geq 0)$ and v_i are the table mass, viscous friction coefficient and driving force on the drive axis i , respectively. The symbol $\text{diag}\{a_i\}$ denotes a diagonal matrix with elements a_i at the i th diagonal positions.

3. Friction compensation

This section presents a coordinate transformation-based contouring controller design with friction compensation. The following items are assumed for the controller design:

1. The desired trajectory r_i and its derivative $\dot{r}_i$ and $\ddot{r}_i$ are available.
2. The inclination of the local coordinate frame Σ_l in Fig. 1 is available.
3. The position of the feed drive system x and its derivative $\dot{x}$ are measurable.
4. Feed drive dynamics parameter values for m_i and c_i are available.

Under these assumptions and from reference [8], we have the following control:

$$v = M\{\ddot{r} - R(K_{vl}\dot{e}_l + K_{pl}e_l) - \ddot{\theta}I_e e_w + \dot{\theta}^2 I_e \dot{e}_w - 2\dot{\theta}I_e \dot{e}_w\} + C\dot{x} \quad (4)$$

$$I_e = \begin{bmatrix} 0 & 1 \\ -1 & 0 \end{bmatrix}$$

where I and $\ddot{r}$ are 2×2 identity matrix and reference acceleration of desired contour, respectively. The symbols K_{vl} and K_{pl} are the velocity and position feedback gain matrices. They are assumed to be diagonal matrices with positive elements. This control considers only viscous friction compensation $C\dot{x}$. From Eqs. (3) and (4), we propose the following contouring control with the Coulomb friction term:

$$v = M\{\ddot{r} - R(K_{vl}\dot{e}_l + K_{pl}e_l) - \ddot{\theta}I_e e_w + \dot{\theta}^2 I_e \dot{e}_w - 2\dot{\theta}I_e \dot{e}_w\} + \eta_1 \text{sgn}(\dot{x}) + \eta_2 \dot{x} \quad (5)$$

$$\eta_1 = \text{diag}\{\eta_{i1}\},$$

$$\eta_2 = \text{diag}\{\eta_{i2}\}, \quad i = 1, 2$$

where $\eta_{i1} (\geq 0)$ and $\eta_{i2} (\geq 0)$ are the Coulomb friction force and viscous friction coefficient on the axis i , respectively.

4. Experiment

4.1 Circular contouring control for estimating friction coefficients

Fig. 2: Experimental setup.

The contouring controller in Eq. (4) was first applied to an X1-X2 table driven by DC servomotors and ball screw drives as shown in Fig. 2. The X1-X2 table position is measured by rotary encoders of servomotors whose resolution for position measurement is $0.025\mu\text{m}$. The velocity of each drive axis is calculated by the backward difference operation of position measurement. Sampling period is 5ms. We have the following parameter values of the experiment system:

$$\begin{aligned} m_1 &= 0.045\text{Vs}^2/\text{mm}; m_2 = 0.061\text{Vs}^2/\text{mm} \\ c_1 &= 0.21\text{Vs}/\text{mm}; c_2 = 0.24\text{Vs}/\text{mm} \end{aligned} \quad (6)$$

and controller gains are set as follows:

$$\begin{aligned} K_{v1} &= 2\omega, K_{p1} = \omega^2, \omega = \text{diag}\{\omega_{1i}\}, i = 1, 2, \\ \omega_{11} &= 40\text{rad/s}, \omega_{12} = 60\text{rad/s} \end{aligned} \quad (7)$$

The following reference trajectories with speed $\pi\text{mm/s}$ are used for the experiments:

$$\begin{aligned} r_1 &= 5\cos\frac{2\pi}{10}t \text{ mm.} \\ r_2 &= 5\sin\frac{2\pi}{10}t \text{ mm.} \end{aligned} \quad (8)$$

Equation (4) is rewritten as follows for the friction analysis:

$$\begin{aligned} v &= \alpha + \delta \\ \alpha &= [\alpha_1, \alpha_2]^T = M\ddot{r} + C\dot{r}, \\ \delta &= [\delta_1, \delta_2]^T = M\{-R(K_{v1}\dot{e}_1 + K_{p1}e_1) - \ddot{\theta}_1 e_w + \dot{\theta}^2 I_e e_w \\ &\quad - 2\dot{\theta} I_e \dot{e}_w\} \end{aligned} \quad (9)$$

Hereafter, we call α and δ are a nominal dynamics term and a control term, respectively. With $\omega_{11}=40\text{rad/s}$, $\omega_{12}=60\text{rad/s}$. We have the following results of the experiment:

Fig. 3: Contour profile in conventional control with circular contour error enlarged by 400 times, $R=5\text{mm}$, Speed $\pi\text{mm/s}$.

The profile of the contour error in the X1-X2 plane is shown in Fig. 3, in which the magnitude of the contour errors is enlarged by 400 times and the significant contour error is confirmed.

(a) X1 axis

(b) X2 axis

Fig. 4: Control input profiles in conventional control.

Control input profiles in conventional control are shown in Fig. 4. Note that $\delta_i \approx 0$ when tracking error is close to zero in Eq. (3), and then $\alpha_i \approx v_i$ is obtained. In this case, however, α_i does not match v_i because the nominal dynamics is very different from the actual one due to friction.

Fig. 5: Relation between control input and velocity on X1 axis.

(a) Estimation of friction property

(b) Comparison between control input and nominal dynamics

Fig. 6: Friction coefficient estimation for X1 axis.

Next we consider friction parameter estimation. Figure 5 shows the relation between the velocity and the control input. In Fig. 6a, the term $v_1 - m_1 \ddot{r}_1$ is regarded as the disturbance compensation control. This property can be approximated by linear functions with respect to positive and negative velocities. Therefore, the friction $\eta_{11} \text{sgn}(\dot{x}) + \eta_{12} \dot{x}$ is obtained approximately as shown in Fig. 6a. The values of η_{11} and η_{12} correspond to an intersection with the vertical axis and inclination of the graph. Figure 6b shows the comparison between control input v_1 and nominal dynamics term α_1 , in which α_1 is changed as $\alpha_1 = m_1 \ddot{r}_1 + \eta_{11} \text{sgn}(\dot{x}_1) + \eta_{12} \dot{x}_1$. Applying the same method to X2 axis, we have the following values for friction that depend on the sign of $\dot{x}_1$ and $\dot{x}_2$:

$$\begin{aligned} \eta_{11} &= 0.90\text{V}, \eta_{12} = 0.19\text{Vs/mm}, \text{ when } \dot{x}_1 \geq 0 \\ \eta_{11} &= 0.76\text{V}, \eta_{12} = 0.23\text{Vs/mm}, \text{ when } \dot{x}_1 < 0 \\ \eta_{21} &= 0.67\text{V}, \eta_{22} = 0.31\text{Vs/mm}, \text{ when } \dot{x}_2 \geq 0 \\ \eta_{21} &= 0.83\text{V}, \eta_{22} = 0.19\text{Vs/mm}, \text{ when } \dot{x}_2 < 0 \end{aligned}$$

Fig. 7: Contour error profile in proposed control ($R = 5\text{mm}$, Speed 3.14 mm/s).

(a) X1 axis

(b) X2 axis

Fig. 8: Control input profiles in proposed control.

After applying Eq. (5) with these parameters, we obtain the result in Fig. 7, in which α is changed as $\alpha = M \ddot{r} + \eta_1 \text{sgn}(\dot{x}) + \eta_2 \dot{x}$. The contouring error is significantly reduced. The control input profile and its α_i and δ_i components are shown in Fig. 8. Because the tracking error is very small, we have $\delta_i \approx 0$ for both drive axes. This concludes that applying simple friction compensation considering the Coulomb and viscous one achieves the ideal motion state and improves control performance significantly.

4.2 Experimental results for several contour curves

We compared the proposed controller with friction compensation with conventional one by applying them to different contour curves in different speeds.

Fig. 9: Circular contour ($R = 15\text{mm}$, Speed 6.28mm/s).

Figure 9a shows the comparison results of conventional controller and proposed controller. Figure 9b and Figure 9c show the tangential and normal errors to the contour, in which the proposed method reduced the normal error significantly. Figure 9d is the contour error magnitude for 10 times trial, and repeatability is confirmed.

Fig. 10: Parallelogram Contour (Speed 3.14mm/s).

Fig. 11: Triangle Contour (Speed 3.14mm/s).

Figures 10 and 11 show comparison results for noncircular contour curves by using the same control parameters with those in Fig. 9. The meaning of each figure (a)-(d) is the same as those in Fig. 9. This result confirms the effectiveness of proposed method, which gives a simple way for improving the contouring performance of feed drive systems.

5. Conclusion

Using the circular contouring control for friction analysis of biaxial feed drive dynamics and enhancement of contouring performance by friction compensation have been considered in this study. Experiments of applying to different curves in different speeds confirm the effectiveness of the proposed controller.

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