Robust Motion Controller Design for High-Accuracy Positioning Systems

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Abstract—**This paper presents a controller structure for robust high-speed/high-accuracy motion control systems. The overall control system consists of four elements: a friction compensator, a disturbance observer for the velocity loop, a position loop feedback controller, and a feedforward controller acting on the desired output. A parameter estimation technique coupled with friction compensation is used as the first step in the design process. The friction compensator is based on the experimental friction model and it compensates for unmodeled nonlinear friction. Stability of the closed-loop is provided by the feedback controller. The robust feedback controller based on the disturbance observer compensates for external disturbances and plant uncertainties. Precise tracking is achieved by the zero phase error tracking controller. Experimental results are presented to demonstrate performance improvement obtained by each element in the proposed robust control structure.**

I. Introduction

Modern mechanical systems such as machine tools, microelectronics manufacturing equipment, mechanical manipulators, and automatic inspection machines must be supported by motion controllers, which ensure robust, high-speed, and high-accuracy positioning/tracking performance. High speed operation is often required to yield high productivity. Precision/accuracy requirement becomes more and more stringent because of the reduced size of components in modern mechanical devices or microelectronics products. The achievable accuracy is set by measurement. If an encoder is used, the goal is to make positioning errors near the encoder resolution including during transients. Robustness is implied not only for stability robustness but for performance robustness. This is an important requirement for avoiding fine tuning of controller parameters when dynamic characteristics vary from one unit to another and/or when the characteristics of a unit vary during operation.

In motion control, two major sources of uncertainties are *friction*, which includes static friction (stiction), Coulomb friction and viscous friction, and *inertia*. In particular, the stiction and Coulomb friction forces are bounded nonlinear functions of the velocity, and the linear control theory is not suited for handling them most effectively. Therefore, the first element introduced in the proposed control scheme is *friction compensation*. Friction compensation is based on a reasonably accurate model for nonlinear friction which can be obtained by off-line experiments.

An alternative approach which regards the bounded nonlinear friction terms as disturbances is the PI (proportional-plus-integral) velocity servo in traditional servo systems. But an I-action in the velocity servo may cause a limit cycle around a target position in point-to-point control, and may magnify the tracking error when motion direction reverses. The latter results in so-called "lost motion" or "quadrant glitches" in machining applications [8], [12], [13]. Further discussion on friction compensation can be found in [11].

A better approach for handling disturbances in motion control is the *disturbance observer* introduced by Ohnishi [7] (also see [6]) and refined by Umeno and Hori [14]. The disturbance observer is not limited to dc-disturbances, and the bandwidth for disturbance rejection can be adjusted. The disturbance observer and friction compensator are not mutually exclusive and may coexist in the overall system.

If major uncertainties are removed by the disturbance observer, the linear feedback control theory can be utilized for constructing an asymptotically stable position feedback loop. In this paper, this element is a PD (proportional-plus-derivative) controller. Although the module can be made of a higher order compensator, PD-control appears to be sufficient for a large class of mechanical systems.

The closed loop dynamics sets the limit of tracking performance if the desired output is directly used as the reference input. To recover the dynamic delay, the desired output needs to be processed by a feedforward controller. The closed loop system, which include the disturbance observer and the position loop controller, remains robust in the presence of disturbances in the specified bandwidth as well as under the variation of inertia and viscous friction parameters. Therefore, the nominal closed loop dynamics is a reliable model for the design of feedforward controllers for precise trajectory tracking. In many motion control applications, the desired output is given in advance. Based on this fact, the feedforward controller utilizes available future values of the desired output and synthesizes the reference input for the closed loop system. The zero phase error tracking controller (ZPETC) proposed by Tomizuka [9] is used in this paper. In [10], Tomizuka also summarized the past and present research on the design of digital tracking controller.

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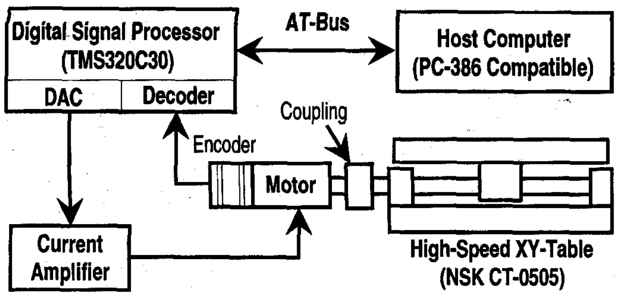


Fig. 1. Experimental setup for high-speed motion control.

In this paper, details of each control element will be described after introducing a precision motion control system and describing identification of system parameters. Digital implementation is assumed, and elements will be either introduced in the discrete time domain or transformed to a digital form if introduced in the continuous time domain. Experimental results will be presented to examine performance of control systems with different combination of elements.

II. Experimental Motion Control System

Fig. 1 depicts an experimental precision motion control system. Each system component is designed for high-speed operations such as semiconductor chip wire-bonding. Under linear assumption, a fourth order transfer function model can be obtained to relate the input voltage to the motor, $u(t)$, to the velocity of the table, $v(t)$. Ignoring the amplifier time constant, which is less than 1 msec, and the mechanical coupling constant which is assumed to be large, the fourth order model can be simplified to a first order model [4]

$$P_n^v(s) = \frac{v(s)}{u(s)} = \frac{1}{Js + B} \tag{1}$$

where J is an equivalent moment of inertia and B is an equivalent viscous friction coefficient of the system. (1) implies the following equation of motion

$$J\frac{dv(t)}{dt} = u(t) - Bv(t). \tag{2}$$

Similarly, the transfer function from the input voltage to the table position is

$$P_n^y(s) = \frac{y(s)}{u(s)} = \frac{1}{Js^2 + Bs}. \tag{3}$$

The reduced order model in (1) ignores nonlinear friction forces. While there have been many friction models proposed [1], a practical approach is to regard the friction force as a *static nonlinear function* of the velocity, i.e., $F_f(v(t))$. Then the reduced order model should be

$$J\frac{dv(t)}{dt} = u(t) - F_f(v(t)) \tag{4}$$

$$F_f(v(t)) = Bv(t) + F_{\text{fn}}(v(t)) \tag{5}$$

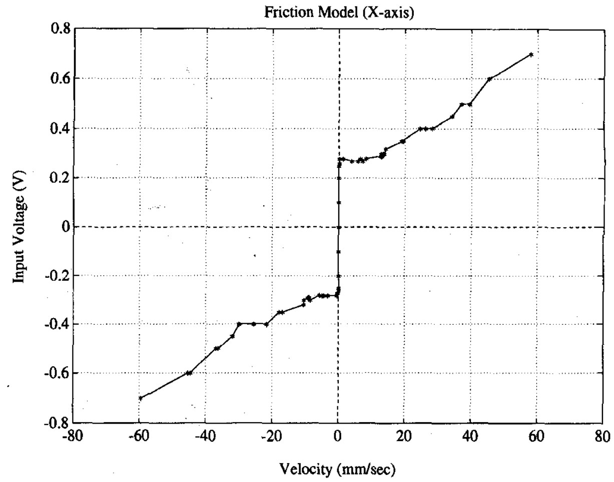


Fig. 2. Friction model: velocity versus friction force (X-axis).

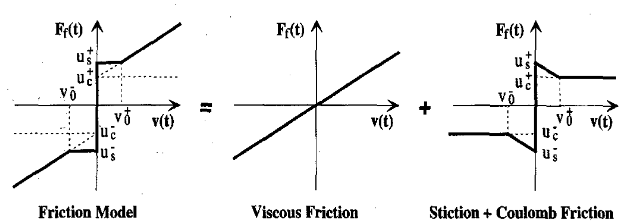


Fig. 3. Equivalent friction model.

where $Bv(t)$ is the linear (viscous) friction term and $F_{\text{fn}}(v(t))$ is the nonlinear friction term.

Notice that if the input, $u(t)$, is held constant u_s, the velocity reaches the steady state v_s. When this happens, $u_s = F_f(v_s)$ must be satisfied. By repeating such experiments for various u_s's, $F_f(v)$ can be experimentally obtained. Fig. 2 shows such a relation obtained for the X-axis of the experimental system. A friction model for Y-axis has been obtained similarly.

III. Friction Compensation and System Parameter Identification

Friction force, especially its nonlinear component, may degrade the performance of any motion control system. A simple but effective method for overcoming problems due to friction is to introduce a cancellation term for the friction force. Such approaches are called friction compensation approaches. If we are to attempt friction compensation by computer control, a look-up table may be utilized to represent $F_f(v)$. However, $F_f(v)$ in Fig. 2 can be parameterized as shown in Fig. 3. Such parameterizations make a table unnecessary. An additional advantage of parameterization is that it separates the linear component (viscous friction), $F_{fl}(v(t)) = Bv(t)$, and the nonlinear component (Coulomb friction and stiction), $F_{fn}(v(t))$. Since the nonlinear component degrades the system performance under linear control, we apply the friction compensation to cancel the nonlinear component. In other words, the control input, $u(t)$, becomes

$$u(t) = u^*(t) + \hat{F}_{fn}(v(t)) \tag{6}$$

where $u^*(t)$ is the output of a linear controller yet to be designed, and ^ has been placed to emphasize that the estimate of nonlinear component of friction is utilized. If the *estimate* agrees with the actual nonlinear component, the transfer function from u^* to v becomes as given by (1), and further design based on the linear control theory becomes effective.

Note that the slope of the friction model corresponds to the viscous friction coefficient term B, if the Coulomb friction and stiction are cancelled based on the friction model. In this paper, the following values were used for friction compensation.

- X-axis:

$$u_s^+ = 0.28 \text{ (V)}, \quad u_c^+ = 0.20 \text{ (V)} \quad \text{and} \quad v_0^+ = 10 \text{ (mm/sec)}$$
$$u_s^- = -0.28 \text{ (V)}, u_c^- = -0.20 \text{ (V) and } v_0^- = -20 \text{ (mm/sec)}$$

- Y-axis:

$$u_s^+ = 0.20 \text{ (V)}, \quad u_c^+ = 0.10 \text{ (V)} \quad \text{and} \quad v_0^+ = 20 \text{ (mm/sec)}$$
$$u_s^- = -0.24 \text{ (V)}, \ u_c^- = -0.16 \text{ (V) and } v_0^- = -10 \text{ (mm/sec)}.$$

Note that the friction characteristics changes when the direction of motion reverses.

If $F_f(v)$ has been estimated as outlined above, use the input as given by (6) and apply any standard parameter identification algorithm for linear systems utilizing $u^*(t)$ and $v(t)$ as input and output data, respectively. This procedure will identify two system parameters, i.e., J and B (for both X- and Y-axis). Alternatively, if the full version of the friction model is utilized in friction compensation, identification algorithms can be run for J only, and B is found as the slope of the friction model as shown in Fig. 3.

The first approach described above is used. Standard parameter identification techniques utilize the sampled input and output sequences, i.e., $\{u(k)\}$ and $\{v(k)\}$ [5]. Spectral analysis on input-output pair yields the frequency response data in a nonparameterized form. By fitting the frequency response data by a first order transfer function model, we obtain J and B. In the fitting process, emphasis should be at frequencies where the first order model is justified. High frequency data which exhibits the ignored higher order dynamics are not considered. The parameters are estimated as

$$J_x = 3.285 \times 10^{-4} \text{ V/(mm/sec}^2\text{)}$$

and

$$B_x = 8.837 \times 10^{-3} \text{ V/(mm/sec)},$$
$$J_y = 3.469 \times 10^{-4} \text{ V/(mm/sec}^2\text{)}$$

and

$$B_y = 9.366 \times 10^{-3} \text{ V/(mm/sec)}$$

for the X- and Y-axis, respectively. More detailed identification procedures can be found in [4].

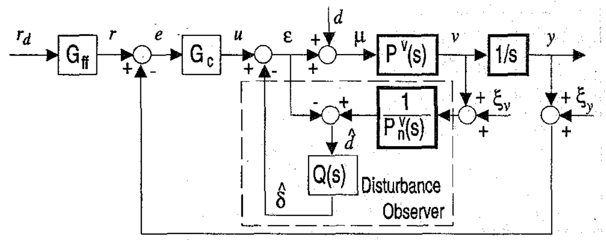


Fig. 4. Robust digital tracking control based on a disturbance observer.

IV. Design of Tracking Controllers

A. Controller Structure

Fig. 4 shows that the tracking controller consists of three components: a feedback controller (G_c) which is a proportional-plus-derivative (PD) controller, a zero phase error tracking controller (ZPETC) as a feedforward controller (G_{ff}), and a disturbance observer. Note that the disturbance observer is applied in the velocity loop. Endo *et al.* [3] proposed the similar control structure by applying the disturbance observer in the position loop instead.

B. Disturbance Observer

In practice, the actual system will not remain the same as a model no matter how the model is obtained. The disturbance observer regards the difference between the actual output and the output of the nominal model as an equivalent disturbance applied to the nominal model. It estimates the equivalent disturbance and the estimate is utilized as a cancellation signal. The structure is shown in Fig. 4 inside the dashed box. This structure follows the one suggested by Umeno and Hori [14].

To understand how the disturbance observer works, first let $Q(s) = 1$ in Fig. 4. Then it can be easily verified that

$$\hat{\delta} = \hat{d} = \left(1 - \frac{P_n^v}{P^v}\right)u + \frac{1}{P^v}\xi_v + d \tag{7}$$

where P^v is the actual plant transfer function for the velocity output, P_n^v is the nominal transfer function, the disturbance term (d) includes the nonlinear friction force if it is not compensated or the estimation error for the nonlinear friction force if it is compensated, $\hat{d}$ represents the estimate of d, u (or u^* if the nonlinear friction force is compensated) is the external input to the velocity loop and ξ_v is the measurement noise. The velocity output is then expressed as

$$v = P^v(u - \hat{d} + d) = P_n^v u - \xi_v. \tag{8}$$

Notice that the input-output relation between u and v is characterized by the nominal model.

The disturbance observer cannot be implemented if $Q(s) = 1$. Notice that $1/P_n^v(s)$ is not realizable by itself but that $Q(s)/P_n^v(s)$ can be made realizable by letting the relative degree of $Q(s)$ be equal to or greater than that of $P_n^v(s)$. Furthermore, $Q(s)$ may attenuate the effect of measurement

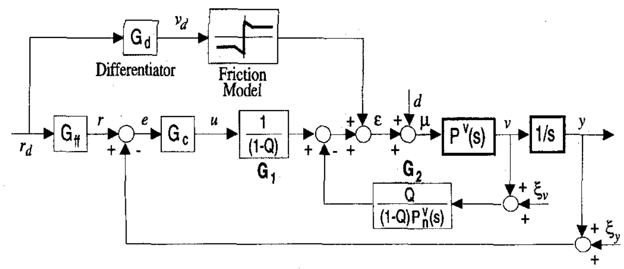


Fig. 5. Disturbance observer structured for digital implementation and feedforward friction compensator.

noise appearing in (7) and (8). From the block diagram in Fig. 4, v is expressed as

$$v = G_{uv}(s)u + G_{dv}(s)d + G_{\xi_v v}(s)\xi_v \tag{9}$$

where $G_{uv} = \frac{P^v P_n^v}{P_n^v + (P^v - P_n^v)Q}$, $G_{dv} = \frac{P^v P_n^v (1-Q)}{P_n^v + (P^v - P_n^v)Q}$, and $G_{\xi_v v} = -\frac{P^v Q}{P_n^v + (P^v - P_n^v)Q}$. If $Q(s) \approx 1$, the three transfer functions in (9) are $G_{uv} \approx P_n^v, G_{dv} \approx 0$, and $G_{\xi_v v} \approx -1$, and the relation in (8) approximately holds. This implies that the disturbance observer makes the actual plant behave like a nominal plant, and this provides robustness to the control system. On the other hand, if $Q(s) \approx 0$, the three transfer functions are $G_{uv} \approx P^v, G_{dv} \approx P^v$, and $G_{\xi_v v} \approx 0$, and the open loop dynamics is observed. Therefore, a sensible choice is to let the low frequency dynamics of $Q(s)$ close to 1 for disturbance rejection and model uncertainties. The high frequency dynamics must be close to 0 anyway because the relative degree of $Q(s)$ must be equal or greater than that of $P_n^v(s)$. Sensor noise rejection may be another consideration in selection of $Q(s)$. Note that $(1 - Q(s))$ and $Q(s)$ can be interpreted as a sensitivity function and complementary sensitivity function for the velocity feedback loop, respectively. A third order binomial filter which satisfies above stated properties has been chosen for this research

$$Q(s) = \frac{3(\tau s) + 1}{(\tau s)^3 + 3(\tau s)^2 + 3(\tau s) + 1}. \tag{10}$$

The general form and selection of binomial filters for $Q(s)$ has been suggested by Umeno and Hori [14].

The disturbance observer can be implemented digitally in several ways. Experimental results in Section V were obtained by transforming it into the structure shown in Fig. 5 and by applying bilinear transformation to $G_1(s)$ and $G_2(s)$ to convert them into digital filters. The feedforward friction compensator in the figure will be explained later. Another way of implementation is demonstrated in [2].

C. Position Loop Feedback Controller

The disturbance observer makes the dynamics between the external control input, u, to the velocity loop and the velocity output, v, robust in the presence of external disturbances and parameter variations affecting the low frequency dynamics. This implies that the transfer function from u to y remains close to the one in (3) with nominal J and B. The design of the position loop controller can be based on the zero order hold equivalent of (3).

The primary focus in the design is the stability of the closed loop system. A simple control law to satisfy this objective is

$$G_c(z^{-1}) = K_c\left[1 + \frac{T_d}{T_s}(1 - z^{-1})\right] \tag{11}$$

where T_s is the sampling time, K_c and T_d are constants. This controller approximates the continuous time PD (proportional-plus-derivative) controller which is given as

$$G_c(s) = K_c(1 + T_d s). \tag{12}$$

D. Feedforward Controller

The feedforward controller is based on the inversion of the closed loop transfer function so that the product of the feedforward controller and the closed loop transfer function comes close to unity for arbitrary desired output. From Fig. 4, with $Q(s) = 1$, the closed loop transfer function from the reference input, $r(k)$, to the output, $y(k)$, is

$$\begin{aligned} G_{ry}(z^{-1}) &= \frac{G_c(z^{-1})P_n^y(z^{-1})}{1 + G_c(z^{-1})P_n^y(z^{-1})} \\ &= \frac{z^{-d}B^-(z^{-1})B^+(z^{-1})}{A(z^{-1})} \end{aligned} \tag{13}$$

where z^{-d} represents a d-step pure delay, $B^-(z^{-1})$ contains uncancelable zeros, which include unstable zeros and some zeros close to $z = -1$, and $B^+(z^{-1})$ contains cancelable zeros. The zero phase error tracking controller (ZPETC) is represented as [9]

$$G_{\rm ff}(z^{-1}) = \frac{r(k)}{r_d(k)} = \frac{z^d A(z^{-1})B^-(z)}{[B^-(1)]^2 B^+(z^{-1})} \tag{14}$$

where $B^-(z)$ is obtained by replacing every z^{-1} in $B^-(z^{-1})$ by z and $[B^-(1)]^2$ is a scaling factor normalizing the low frequency gain of the overall transfer function from $r_d(k)$ to $y(k)$ to unity. The ZPETC cancels the poles and cancelable zeros of the closed loop system and compensates for phase shifts induced by uncancellable zeros. This implies that the overall transfer function has zero phase shift characteristics at all frequencies. For further details of ZPETC, see [10].

E. Friction Compensation Based on Desired Velocity

In tracking control, if the closed loop transfer function is preceded by the feedforward controller and the actual velocity may be close to the desired velocity in the absence of nonlinear friction force, the input to the friction compensator may be the desired velocity instead of the actual velocity. In this case, the friction compensator is a feedforward controller instead of a feedback controller. This approach may be advantageous because 1) the feedforward control is more robust than feedback control, and 2) any delay caused by velocity estimation in the absence of a direct velocity transducer is not an issue.

Furthermore, in the presence of the disturbance observer, an additional performance improvement attained by feedback friction compensation is expected to be minimal. The disturbance observer is a feedback compensation scheme, and its inherent performance limitation in digital implementation is due to one step sampling delay, which makes it logical to implement friction compensation based on the desired velocity. Fig. 5 shows a block diagram of this approach. Experimental results will be shown in the next section to demonstrate the effectiveness of this scheme.

V. Experimental Results

Three different controllers were implemented to compare their tracking performance: the first controller was with ZPETC plus PD-controller in the position loop (ZP), the second was with ZPETC plus PD plus Friction Compensation based on the actual velocity (ZPFC) and the third with ZPETC plus PD plus Disturbance Observer (DOB). The sampling time for the two-axis experiments is 0.3 msec and 2500 samplings (750 msec) are taken to complete the trajectory. The resolution of encoder is 4000 counts per revolution, i.e., 1.25 μm per pulse. Fig. 6 shows tracking performance of ZP and DOB for drawing a tiny circle with the radius of 100 μm (i.e., a diameter of 0.2 mm). The dashed line corresponds to the tracking performance under ZP and the solid line represents that under DOB. The radial tracking error is defined as

$$e_{\mathrm{rt}} = \sqrt{(x_d - x_c)^2 + (y_d - y_c)^2} - \sqrt{(x - x_c)^2 + (y - y_c)^2} \tag{15}$$

where $(x_d, y_d), (x, y)$ and (x_c, y_c) are the coordinates of the desired trajectory, the actual trajectory, and the center of the circle, respectively. The adverse effect of stiction and Coulomb friction on the tracking performance of ZP is evident. ZP cannot compensate the errors due to Coulomb friction and the "stick" around the near-zero velocity. The maximum radial tracking error is around 4 μm for this case. Note that there exist rather big peaks at every quarter period (i.e., "quadrant glitches"). The tracking performance of DOB shows that the quadrant glitches and periodicity of the tracking errors are removed. The maximum tracking error is less than 2 μm which implies less than two encoder counts. We see that DOB compensates effectively for stiction and Coulomb friction in this rather slow and small-movement tracking experiment.

The experimental results in Fig. 7 shows the ellipse drawing capability of the three controllers: ZP, ZPFC, and DOB. The ellipse has a 2-mm major axis and a 0.5-mm minor axis. ZP shows the effect of stiction and Coulomb friction. Although ZPFC demonstrates slightly better performance than ZP by moving the error trend to the near zero-error line, it cannot completely remove the friction effect. This is due to especially slow movement in the y-direction and the limitation in the resolution of the velocity measurement. In other words, the velocity information is easily lost in the low speed range with a very high sampling rate. Since the minimum speed which can

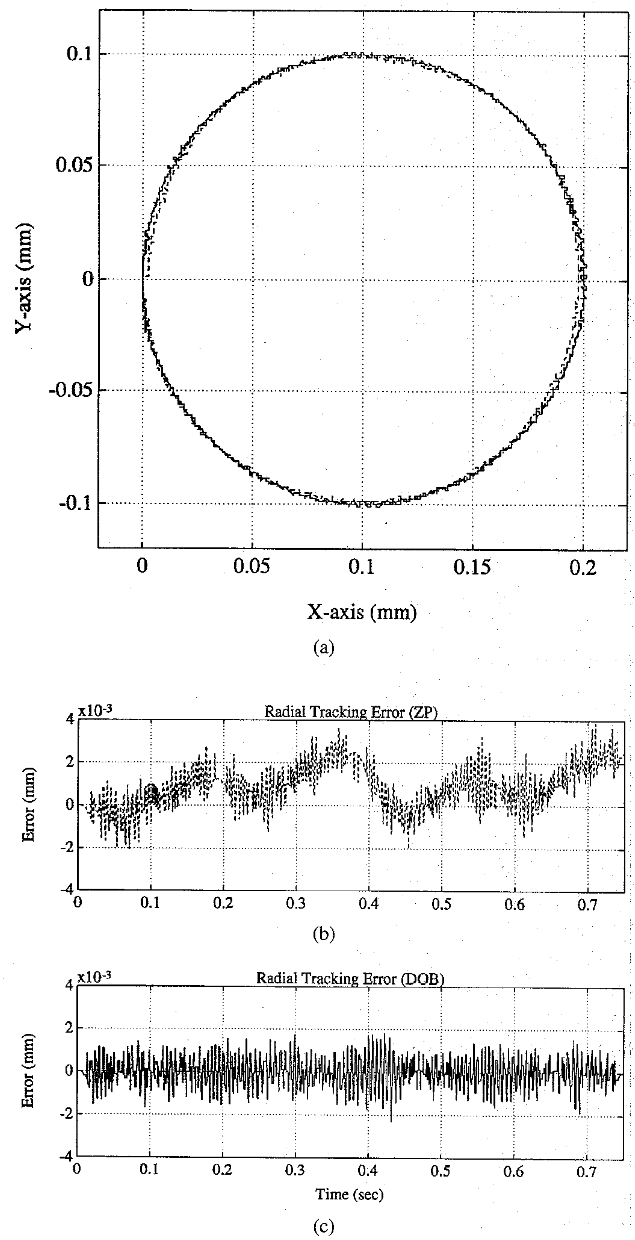


Fig. 6. Small circle tracking experiment: Tracking performance and radial tracking errors (solid line: DOB; dashed line: ZP).

be estimated by the optical encoder is 4.167 mm/sec (= 1.25 μm/0.0003 sec), the friction compensator can not generate an appropriate compensation signal below this level. By adjusting the magnitude of the friction compensation around the zero velocity, the radial tracking error can be reduced further. However, it has been observed that this created a serious chattering problem, when we tried to achieve a better tracking below the error level shown in Fig. 7. As we see in the figure, DOB exhibits the best tracking performance by compensating the stiction and Coulomb friction. Overall tracking error is less than 2 μm which is less than two encoder counts.

The next set of experimental results show how well the above-mentioned controllers compensate the step disturbance.

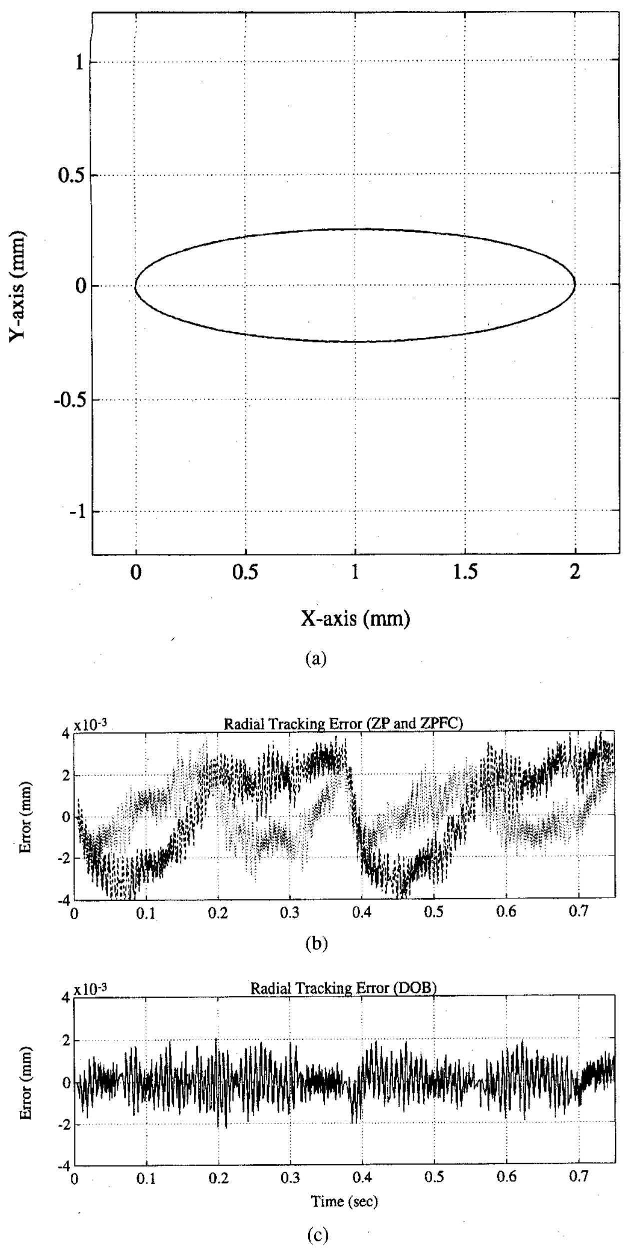


Fig. 7. Ellipse tracking experiment: Tracking performance and radial tracking errors (solid line: DOB; dashed line: ZP; dotted line: ZPFC).

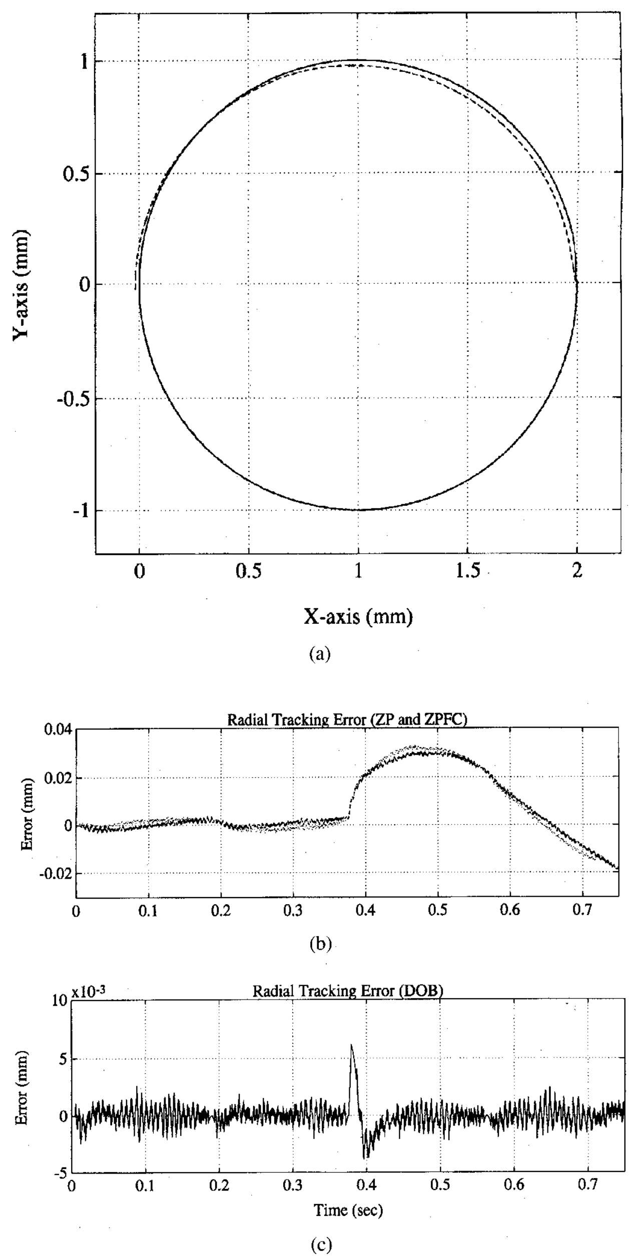


Fig. 8. Circle tracking experiment: Tracking performance and radial tracking errors under step disturbance (solid Line: DOB; dotted line: ZP; dashed line: ZPFC).

A step disturbance of 1.0 V magnitude was injected into the control signal at 0.375 sec after the start time. Since the voltage output range of the digital-to-analog converter (DAC) of the controller is ± 3.0 V, this is a rather big disturbance. Fig. 8 shows the tracking performance and tracking errors of the three controllers: the solid line is for DOB, the dotted line for ZP, and the dashed line for ZPFC. As we see in the figure, only DOB correctly compensates the step disturbance. Although ZPFC performs better during the transient and steady-state period compared with ZP, it shows slightly bigger oscillations after the step disturbance is given. It has been shown again that the robust digital controller based on the disturbance observer (DOB) has satisfactorily performed for both unmodeled dynamics (i.e., nonlinear friction terms) and external disturbance cases.

The final experiment shows the effectiveness of the controller in Fig. 5, i.e., DOB plus Feedforward Friction Compensation, which is designated as DOBFC. This experiment was performed for one axis motion to find differences between DOB and DOBFC. The desired output sequence used in simulations is generated by sampling a fifth order polynomial trajectory which is smooth to the third order derivative at the start and end points and accomplishes a sigmoid shaped transition. Fig. 9 shows the desired trajectory which has a 1 mm amplitude and a 0.25 sec transition time.

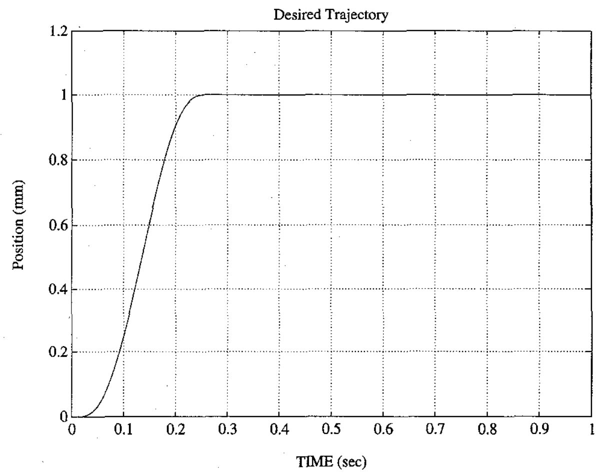
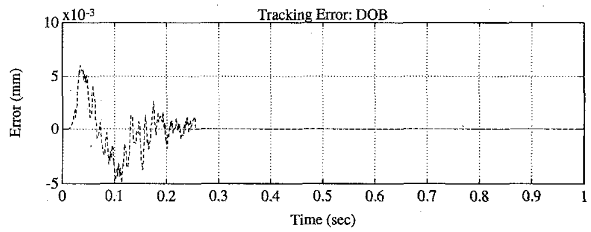
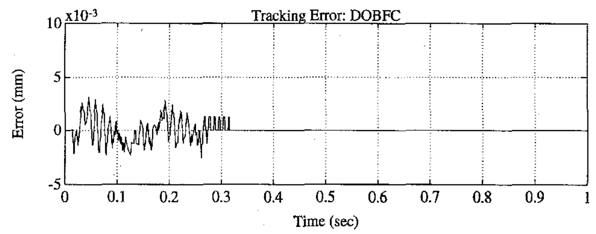


Fig. 9. Desired trajectory for single-axis experiments.



(a)



(b)

Fig. 10. Tracking errors: Single axis experiment (dashed line: DOB; solid line: DOBFC).

The dashed line in Fig. 10 shows the tracking error under DOB. While the error is somewhat large initially, it remains small after the estimation transient of the disturbance observer is over. Another plot with the solid line indicates the tracking performance of DOBFC. Notice that the maximum tracking error of DOBFC is 3 μm compared with that of DOB which is around 6 μm. The overall tracking error is about two encoder counts. The major improvement appears at the initial phase where stiction dominates. In DOB, a several sampling steps are required for the disturbance observer and PD-controller generates a control input required to break stiction. In DOBFC, the feedforward friction compensator provides a kick from the start resulting in the initial performance improvement as seen in Fig. 10.

VI. Conclusion

In this paper, the design of a robust digital controller has been presented. It is composed of four controller components: a friction compensator either in the feedforward or feedback loop, a disturbance observer in the velocity loop, a feedback controller in the position loop, and ZPETC as a feedforward controller.

The performance of the robust digital controller (DOB) has been examined through experiments under the various uncertainties and external disturbances. Its performance has been compared with two other digital tracking controllers: one with ZPETC-plus-PD controller (ZP), the other with ZPETC-plus-PD-plus-Friction Compensation controller (ZPFC). The robust controller has performed best among them. Since the performance of the disturbance observer is highly dependent on the choice of the Q-filter, it requires further research how to systematically determine the Q-filter in either the continuous-time or the discrete-time domain. In most applications, the performance obtained by DOB is considered to be adequate. However, the disturbance observer is a feedback compensator. The feedforward friction compensator overcome this limitation and further enhances the tracking performance. The presence of the disturbance observer assures superior performance even in the case that the feedforward friction compensator is not perfectly tuned.

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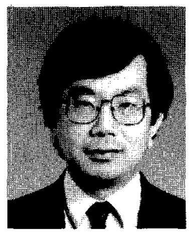

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