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ER 브레이크 및 클러치를 이용한 이송 테이블의 위치추적제어

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Position Tracking Control of a Moving Table Using ER Brake/Clutch

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ABSTRACT

본 연구에서는 ER 브레이크와 ER클러치를 피드백작동기로 사용하여 이송테이블의 위치추적제어를 수행하였다. 이를 위해 먼저 아라빅 검(arabic gum)계통의 ER유체를 자체조성한 후 전기장에 대한 빙햄(Bingham)모델을 실험적으로 도출하였다. 빙햄모델에 근거하여 평판형의 ER브레이크와 실린더형의 ER클러치를 설계 제작하였으며, 계단입력(step input)전기장에 따른 출력토오크특성을 통하여 이들 작동기의 동적모델을 얻었다. 이들 작동기와 연계된 이송테이블시스템의 운동지배방정식을 유도한 후 위치추적제어를 위한 슬라이딩모드제어기를 설계하였다. 제어기 설계시 이송테이블의 부하질량 변화에 대한 시스템 불확실성과 마찰력을 고려하여 제어성능의 강건성을 보장하도록 하였다. 제안된 제어시스템의 제어영역(control bandwidth)을 주파수 영역에서 고찰한 후 여러 궤적에 대한 위치추적제어 실험을 수행하였다.

Key Words : Electro-Rheological Fluid (ER유체), ER Brake/Clutch (ER 브레이크/클러치), Position Control (위치제어), Moving Table (이송 테이블), Sliding Mode Control (슬라이딩 모드 제어)

1. INTRODUCTION

The development of positioning table systems is significantly important in various industries. High accuracy in the position control for electronic assembling is essential, and effective position control of moving tables is necessary for precise machining in machine tool systems. One of difficulties in achieving high accuracy in positioning tables is how to treat load and friction uncertainties. Since the positioning table system has inherently these uncertainties, the trajectory tracking accuracy may be often poor. This leads to the study of the feedback control of the positioning table system so that the tracking accuracy can be maintained within a desired limit value.

Currently, the most general control approach in positioning table systems is to apply a PID con-

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troller in association with d.c. servomotor. The PID controller may have intrinsically favorable tracking accuracy, but sometimes may cause poor tracking performance due to the presence of friction uncertainties. Use of higher PID gains can compensate for these uncertainties, but may result in serious problems such as system instability(1). Recently, in order to resolve this problem various types of modern controllers have been proposed. For instance, the Lyapunov-based adaptive control of a positioning table was developed through the incorporation of direct compensation for static and dynamic friction(2). On the other hand, most previous studies on the positioning table system adopted d.c. servomotors as feedback actuators. The d.c. servomotor can be easily incorporated with existing conventional feedback controllers, but it is unwelcome in terms of the cost.

This paper presents a new actuating mechanism to achieve the position control of table systems ; electro-rheological(ER) brake and ER clutch are adopted as feedback actuators, and the d.c. motor is used to generate a required motion. As well-known, the research interest in the ER brake or/and ER clutch has been growing because of the potential importance of such devices for a variety of torque coupling and tension control applications(3)~(7). Some advantages of such devices include continuous controllability (both directions) via controlling the intensity of electric fields, rapid response, low power consumption, and smooth operation without torque riffle.

In this work, a plate-type ER brake and a cylindrical ER clutch are manufactured on the basis of the level of the field-dependent yield shear stress of an arabic gum-based ER fluid. The positioning table system is then constructed using the ER brake/clutch actuators followed by the derivation of the governing equation of motion of this system. A sliding mode controller, which has inherent robustness to parameter uncertainties and external disturbances such as frictions, is formulated and experimentally implemented. Position tracking control responses to square and sinusoidal trajectories are presented in order to demonstrate the efficiency and feasibility of the proposed control method.

2. MODELING OF ER BRAKE/CLUTCH

A feedback position control system of a moving table proposed in this study is shown in Fig.1. Unlike conventional positioning table systems having the d.c. servomotor, the one proposed consists of the ER brake and the ER clutch as feedback actuators. The position of a moving table generated by the d.c. motor is to be controlled to meet a desired set-position or time-varying trajectory. This can be achieved by applying control input(electric field) to the ER brake and the ER clutch actuators. A plate-type ER brake actuator is devised as shown in Fig.2 (a). Two plates are

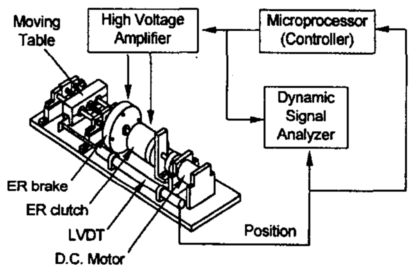
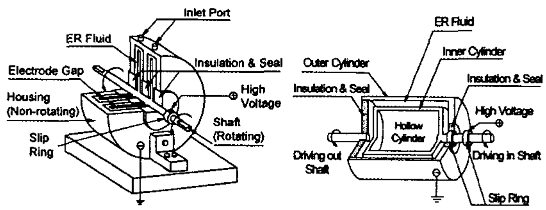


Fig. 1 Schematic diagram of the proposed positioning table system



(a) ER brake **(b) ER clutch**

Fig. 2 The layouts of the feedback actuators

enclosed by a housing coupled to the shaft and the electrode gap is fixed at 1.0mm. We clearly see from the layout that since the housing is fixed to a steel plate (ground), the rotational speed of the shaft can be stopped or reduced by controlling the intensity of electric fields. The braking torque consists of three components : torque from the field-dependent yield shear stress of the ER fluid, torque from the viscosity of the ER Fluid, and torque from the friction between the housing and the shaft. In general, the third component is hard to identify accurately, and hence it will be treated as an external disturbance in the formulation of a feedback controller. The second component can be considered as an equivalent viscous torque, which is proportional to angular velocity of the system. The equivalent viscous coefficient is easily obtained by integrating the viscous shear stress with respect to the total area containing the ER fluid. The first component is controllable braking torque, and hence it is treated as a control input in the synthesis of the feedback controller.

The controllable braking torque is obtained by integrating the yield shear stress with respect to the total area containing the ER fluid, and is given by

$$T_b = \frac{2}{3}\pi N(R_2^3 - R_1^3)\tau_y(E) \tag{1}$$

In the above, N is the number of the electrode gap (4, in this study), R_2 is the outside radius of the brake plate and R_1 is the radius of the insulation disk. $\tau_y(E)$ is the field-dependent yield shear stress of the ER fluid which is given by

$$\tau_y(E) = \alpha E^{\beta} \tag{2}$$

The ER fluid used in this work consists of 0.015 Pa · s silicone oil and arabic gum 30% by weight. The particle size is 26~88㎛. A couette type electroviscometer(Haake, VT-500) is employed to obtain the Bingham property of the ER fluid. The shear stress is measured by applying the electric field from 1 to 3kV/mm in 0.5kV/mm steps, while the rotational speed increases up to 600rpm. From the linear intercept at zero shear rate, the yield shear stress τ_y is found to be $67E^{1.2}$ Pa as the form of Eq.(2). Here the unit of E is kV/mm.

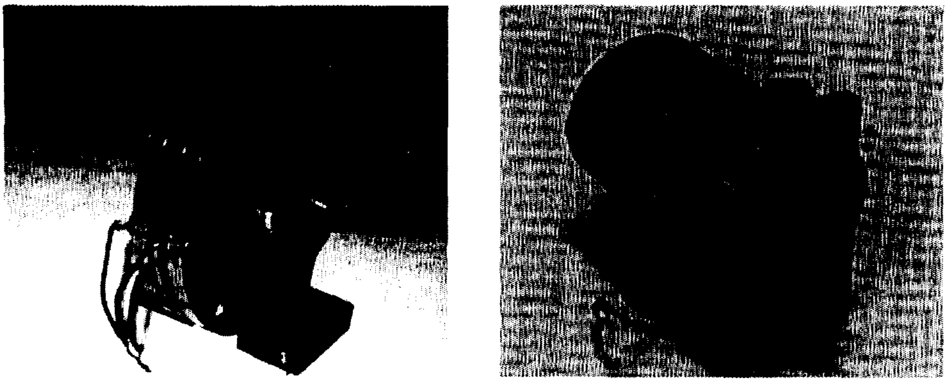
On the other hand, a cylindrical ER clutch is devised as shown in Fig.2 (b). The role of the ER clutch is to transmit the bi-directional torque generated from the d.c. motor to move the table. The torque of the ER clutch has also three components as those of the ER brake. The torque due to the friction is treated as an external disturbance, and the torque from the viscosity of the ER fluid is considered as the equivalent viscous torque in the system modeling. The torque owing to the field-dependent yield shear stress is adopted as a controllable torque, and it is given by

$$T_c = 2\pi R_3^2 L\tau_y(E) \tag{3}$$

where R_3 is the radius of the inner cylinder, and L is the length of the inner cylinder.

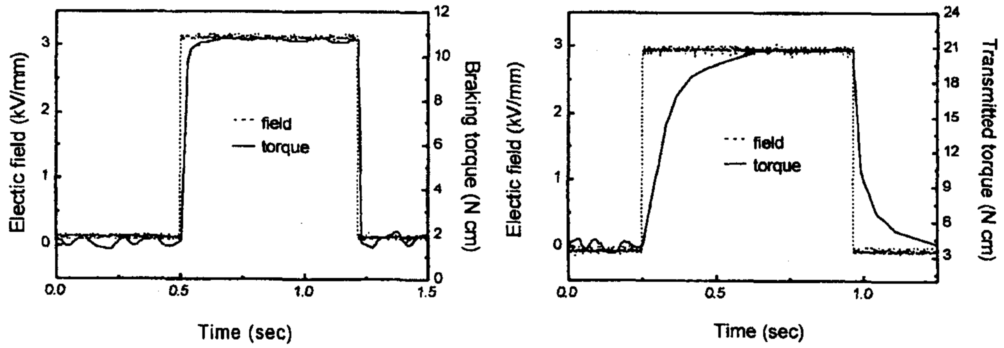
From the analysis of Eqs.(1)~(3), appropriate sizes of a plate ER brake and a cylindrical ER clutch are manufactured and their photographs are shown in Fig.3. For the ER brake, the outside radius of the brake plate $R_2=65mm$, and the radius of the insulation disk $R_1=30mm$. On the other hand, for the ER clutch, the radius of the inner cylinder $R_3=68mm$, and the length of the inner cylinder $L=77mm$.

In order to identify dynamic characteristics of the manufactured actuators, step responses are tested and presented in Fig.4. By applying the electric field of 3kV/mm, the braking torque of the ER brake actuator exponentially increases to a steady state value(10.7N · cm) without exhibiting overshoot. This implies that the ER brake actuator behaves like a first-order linear model with a time constant of 11msec. This response is fast enough to take account of the feedback con-



(a) ER brake (b) ER clutch

Fig. 3 The photographs of the feedback actuators



(a) ER brake (b) ER clutch

Fig. 4 Measured step responses of the actuators at steady speed of 100 rpm

trol system, and thus we can use the quasi- static relationship, Eq.(1), instead of using the first-order model. However, the ER clutch actuator exhibits relatively slow response with a time constant of 140msec to reach a steady state torque(20.5N · cm) by applying the electric field of 3kV/mm. This is mainly due to the inertia effect of the cylinders. Therefore, the governing model (3) of the ER clutch actuator needs to be modified by

$$T_c = 2\pi R_3^2 L \tau_y(E)\left(1 - e^{-\frac{t}{\tau}}\right) \tag{4}$$

where τ denotes the time constant. It is remarked that we may devise same types of brake and clutch actuators to have same time constants. This will be more convenient to formulate a closed-loop feedback control scheme.

3. FORMULATION OF CONTROL SYSTEM

Without considering d.c.motor dynamics, the governing equation of the rotational motion($\theta(t)$) of the proposed positioning table system can be expressed by

$$J_e\ddot{\theta}(t) + C_e\dot{\theta}(t) = \tau_c(t) + \tau_f(t) \tag{5}$$

where J_e is equivalent moment of inertia given by

$J+ml^2$. J represents the sum of moment of inertia of the ER brake, the ER clutch and the ball screw, and ml^2 is the moment of inertia due to the moving table with the mass of m and the ball screw lead of l. C_e is equivalent viscous coefficient from viscous torque of the ER brake and the ER clutch. $\tau_c(t)$ is control torque from the ER brake or/and ER clutch, and is $\tau_f(t)$ total frictional torque in this system. In practice, the table mass m varies due to the change of load (working piece), and hence the inertia J_e is subjected to be changed with a certain limit. Furthermore, it is known that it is very difficult to get an accurate model of the friction, and hence $\tau_f(t)$ is considered as an external disturbance. Consequently, the dynamic model (5) can be rewritten by

$$\begin{aligned} \dot{\theta}_1(t) &= \theta_2(t) \\ \dot{\theta}_2(t) &= -\frac{C_e}{J_e}\theta_2(t) + \frac{1}{J_e}\tau_c(t) + \frac{1}{J_e}\tau_f(t) \end{aligned} \tag{6}$$

where,

$\theta_1(t) = \theta(t),\ \theta_2(t) = \dot{\theta}(t),\ J_e = J_o + \Delta J,\ |\Delta J| \le \alpha < 0.5 J_o,\ |\tau_f(t)| \le \beta$

In the above equation, the moment of inertia is divided into the nominal part(J_o), which is known, and uncertain part(ΔJ), which is unknown, but bounded. The frictional torque is assumed to be unknown, but it is bounded.

Now, the ultimate goal of control task is to have tracking error between actual trajectory $\theta(t)$ and desired trajectory $\theta_d(t)$ to be zero. This allows one to achieve accurate positioning of the table in the translational motion through the relationship given by $x(t) = l\theta(t)$. Prior to designing a controller, so-called matching condition[8] is assumed as follows.

$$\frac{1}{J_o + \Delta J} = \frac{1}{J_o} + \frac{\gamma}{J_o} = \frac{1}{J_o}(1+\gamma) \tag{7}$$

Then, from the bound of ΔJ the γ has the following bound.

$$|\gamma| < \phi < 1 \tag{8}$$

The matching condition (7) with (8) physically implies that the input uncertain part ΔJ can not have arbitrarily large perturbation.

In order to formulate a sliding mode controller, we first define tracking error as follows :

$$e(t) = [e_1(t),\ e_2(t)]^T = [\theta_1(t) - \theta_{1d}(t),\ \theta_2(t) - \theta_{2d}(t)]^T \tag{9}$$

where $\theta_{1d}(t)$ and $\theta_{2d}(t)$ are desired angular displacement and velocity, respectively. The problem now is to design a sliding surface that guarantees stable sliding mode motion on the surface itself. Because there is only one control input $\tau_c(t)$, we construct one sliding surface for the system (6) :

$$s(t) = g e_1(t) + e_2(t) = 0,\ g > 0 \tag{10}$$

Then the following sliding condition is introduced to guarantee that the tracking variables $e_1(t)$ and $e_2(t)$ of the system are constrained to the sliding surface during the sliding mode motion :

$$s(t)\dot{s}(t) < 0 \tag{11}$$

Now, we propose a sliding mode controller given by

$$\begin{aligned} \tau_c(t) = \frac{-J_o}{1-\phi}\left\{g\left(|\theta_2(t)| + |\theta_{2d}(t)|\right) + |\dot{\theta}_{2d}(t)|\right\}\mathrm{sgn}(s(t)) \\ + C_e\theta_2(t) - k\ \mathrm{sgn}(s(t)),\ k > \beta \end{aligned} \tag{12}$$

Then, we can show that the uncertain system (6) with the proposed controller (12) satisfies the sliding condition (11) as follows :

$$\begin{aligned} s(t)\dot{s}(t) = &\left\{g\theta_2(t)s(t) - \left(\frac{1+\gamma}{1-\phi}\right)g|\theta_2(t)||s(t)|\right\} + \left\{-g\theta_{2d}(t) - \left(\frac{1+\gamma}{1-\phi}\right)g|\theta_{2d}(t)||s(t)|\right\} \\ &+ \left\{-\dot{\theta}_{2d}(t) - \left(\frac{1+\gamma}{1-\phi}\right)|\dot{\theta}_{2d}(t)||s(t)|\right\} + \left\{\frac{1}{J_o}(1+\gamma)\tau_f(t)s(t) - \frac{k}{J_o}(1+\gamma)|s(t)|\right\} < 0 \end{aligned} \tag{13}$$

To alleviate the chattering due to the signum function, we replace the signum function in Eq.(12) by the saturation function as

$$sat(s(t)) = \begin{cases} s(t)/\varepsilon, & |s(t)| \le \varepsilon \\ \mathrm{sgn}(s(t)), & |s(t)| > \varepsilon \end{cases} \tag{14}$$

where ε is the boundary layer thickness. Once the control torque $\tau_c(t)$ is achieved from Eq.(12), control electric fields to be applied to the ER brake and the ER clutch are determined from Eq.(1) and Eq.(4), respectively.

4. RESULTS AND DISCUSSION

In order to demonstrate the efficiency and practical feasibility of the proposed control system, an experimental apparatus is established as shown in Fig.5. The actual position of the moving table is measured by the LVDT(linear variable differential transformer) sensor and is fed back to the microprocessor via the A/D(analog/digital) converter. Then, the control electric field determined from the sliding mode controller is applied to the ER brake or ER clutch through the D/A (digital/analog) converter and the high voltage amplifier. The sampling frequency for the controller implementation is chosen to be 100Hz, and the program for the controller is written in Borland C language. The moving direction of table is controlled by simply using an electrical relay circuit which is devised to change the rotating direction of d.c. motor according to the sign of clutch control input determined from the proposed sliding mode controller.

It is remarked that the angular displacement $\theta(t)$ in the controller is calibrated with respect to the sensor displacement $x(t)$ by $x(t) = l\theta(t)$. The system parameters employed in this experimental work are as follows : $J_o = 0.012\mathrm{kg \cdot m^2}$, $C_e = 0.0523 Nm \cdot s$, $l = 0.381$ mm/rad and $m = 0.9kg$. On the other hand, the bounds of the uncertain parameters are prescribed as follows : $|\Delta J| \le 0.008\mathrm{kg \cdot m^2}$ and $|\tau_f| \le 0.025\mathrm{N \cdot m}$. It is noted that the boundedness of the moment of inertia ($|\Delta J|$) is determined by imposing a load mass of 1.5kg on the table.

In order to identify the frequency bandwidth of the position control system, the Bode plot is experimentally obtained as shown in Fig.6. This is very important for one to set a desired trajectory which should have a frequency within the sys-

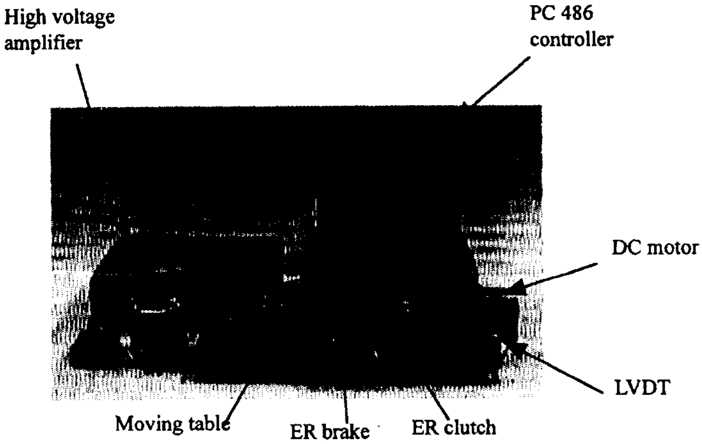


Fig. 5 Photograph of experimental configuration

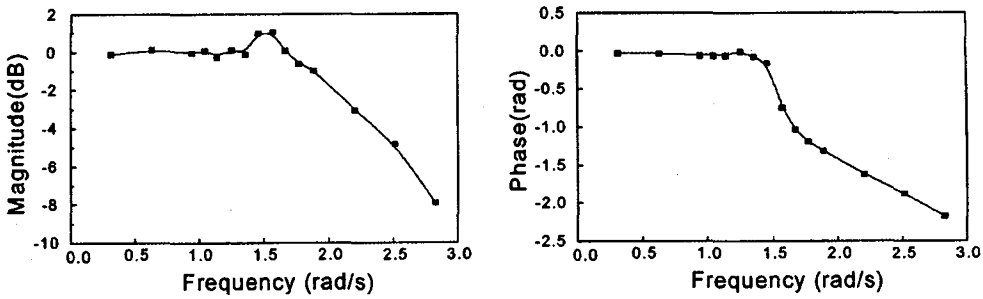
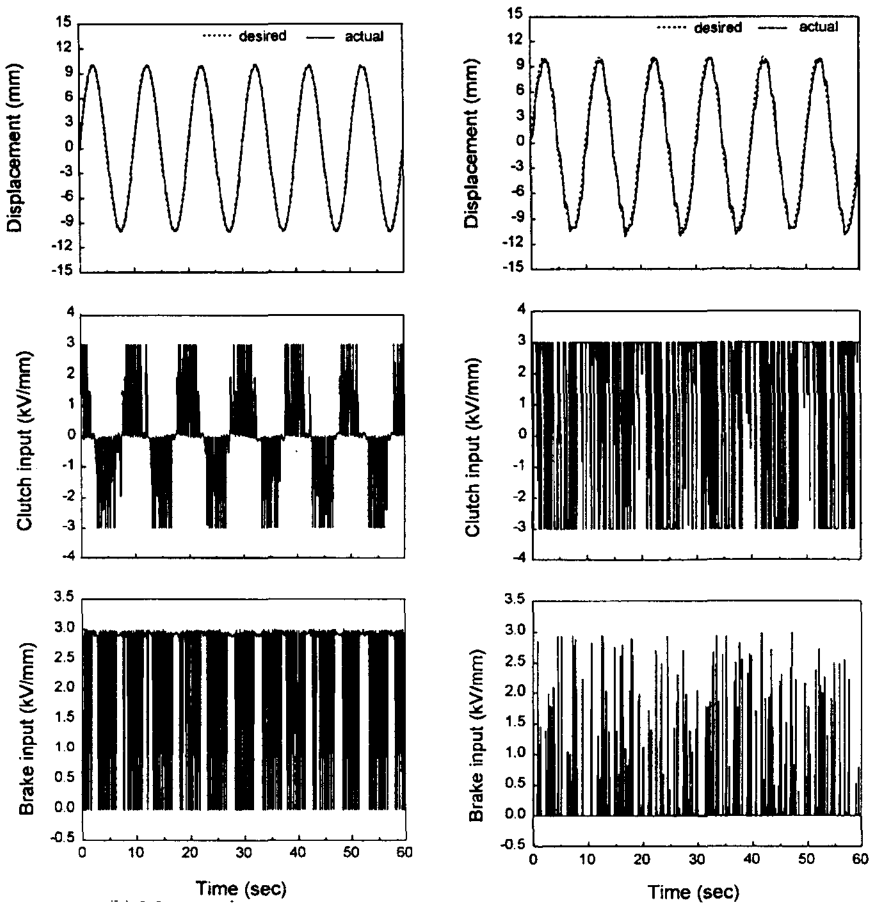


Fig. 6 Bode plot of the moving table system

tem bandwidth. It is clearly observed from Fig.6 that the proposed system exhibits a typical response of a second-order underdamped system, and its bandwidth frequency is identified by 1.6rad/sec. Thus, a maximum frequency of the desired trajectory to be tracked needs to be set within the bandwidth frequency. In this experiment, a desired trajectory which has a frequency of 0.1Hz is chosen.

Fig.7 presents the sinusoidal tracking control responses of the proposed control system with the load mass of 1.5 kg. The employed control parameters are as follows : $g=8.5$, $\phi=0.5$, $k=5.3$, and $\varepsilon=0.03$ It is clearly observed from the diplacement trajectories that the position tracking is successfully performed with favorable tracking accuracy in the sinusoidal trajectory. We see from the displacement history that control action is carried out in both directions with respect to neutral position(displacement is equal to zero). It is also seen from the control inputs that the electric fields are applied to the ER clutch or/and the ER brake in a switching manner within a maximum field of 3.0kV/mm. This can be expected from the characteristic of the sliding mode controller. Though high frequency noise induced from the LVDT used as a position sensor affects control input signal of the ER clutch and the ER brake to be different from the input signals of the simulation results as shown in Fig.7(b), tracking control performance is fairly maintained because the bandwidth of the moving table system is relatively smaller than the frequency of the noise generated from LVDT sensor and the proposed sliding mode controller can actively control the position of the moving table in the presence of the disturbances such as sensor noise. Also the presented tracking results indicate that the proposed control system associated with the sliding mode controller is very robust to the system uncertainties such as the change of the moment of inertia(due to the change of mass from 0.9kg to 2.4kg), and the existence of the unmodeled frictions. From the position tracking control responses, we also see that there exists favorable agreement between the simulated and experimental results showing the validity of the proposed control system model.

Fig.8 compares the measured tracking responses of a square trajectory between the proposed sliding mode controller and PID controller. The PID(proportional(k_p) - integral(k_i) - derivative(k_d)) controller was designed in the absence of the parameter uncertainty as well as external disturbance. The control gains of $k_p=3.5$, $k_i=0.9$ and $k_d=1.5$ were determined by Ziegler-Nichols tuning rule. It is clearly observed that in the PID control the tracking error is relatively large compared to the sliding mode controller. Moreover, we see that the tracking error in the PID control is changed at every cycle. This is arisen from that the magnitude of the unmodeled friction may be altered at every moment. In addition, though there exist



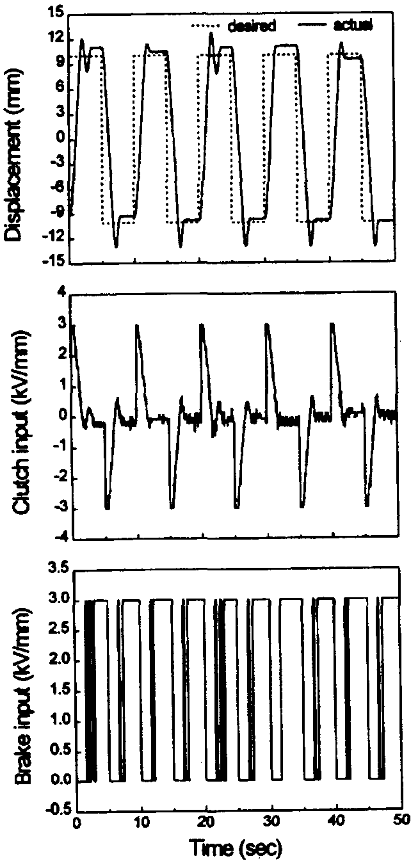
(a) Simulated

(b) Measured

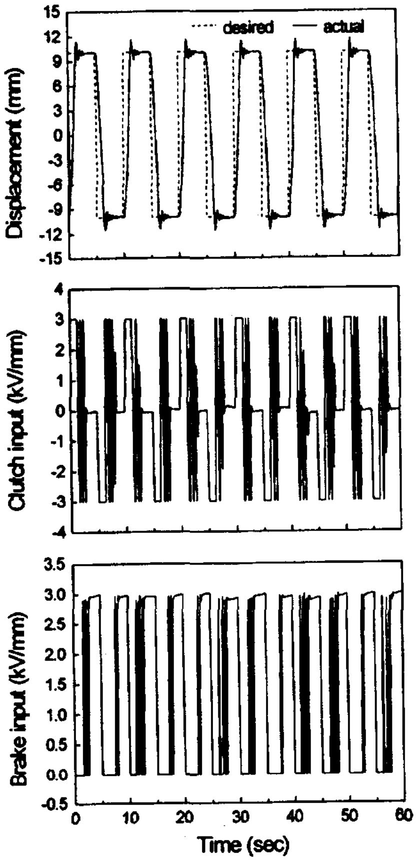
Fig. 7 Sinusoidal tracking responses using the sliding mode controller

some overshoot induced from actuating time delay of relay circuit and d.c. motor, it can be favorably compensated by the sliding mode controller as shown in Fig.8 (b). It is finally remarked that we may achieve more accurate and faster tracking control response in the sliding mode control by changing control parameters such as k and ε.

As observed in this experimental investigation, the proposed ER brake and clutch are unidirectional. Thus, we have to use the relay switching system to determine the rotational direction of the d.c. motor. This consequently leads to undesirable overshoot phenomenon. In order to resolve this problem, a bi-directional ER clutch system can be devised as shown in Fig.9. Without any electrical circuit for the control of d.c. motor's rotating direction, bi-directional type ER clutch can change its rotating direction in very simple manner as follows. A pair of ER clutches are set to rotate in both direction(clock-wise and counter-clock-wise direction) by using two spur gear and we just need to alternatively apply an electric field to clock-wise or counter-clock-wise rotating ER clutch for the control of rotating direction. Since ER fluid has very fast response characteristics, it is expected that the bi-direc-



(a) PID controller



(b) Sliding mode controller

Fig. 8 Measured tracking responses for a square trajectory

tional ER clutch can easily and quickly convert the actuating torque. So this can settle the time delay problem as observed in this study.

5. CONCLUSIONS

A feedback control method of the positioning table system was proposed by adopting the ER brake and the ER clutch actuators. After manufacturing these actuators on the basis of the field-dependent yield shear stress of the arabic gum-based ER fluid, a sliding mode controller was designed by considering the variation of the moment of inertia as the uncertain parameter, and the unmodeled friction as the external disturbance. The controller was then experimentally realized, and favorable position tracking responses were achieved in terms of the tracking accuracy and the robustness to the system uncertainties. The control results presented in this work are self-explanatory justifying that the proposed control method provides a feasibility of practical application. By employing ER brake and ER clutch actuators, we may achieve favorable position tracking accuracy within $\pm 5\%$ with a relatively low cost compared with conventional servomotor by simply using electrical relay circuit and d.c. motor. The durability of the tracking control

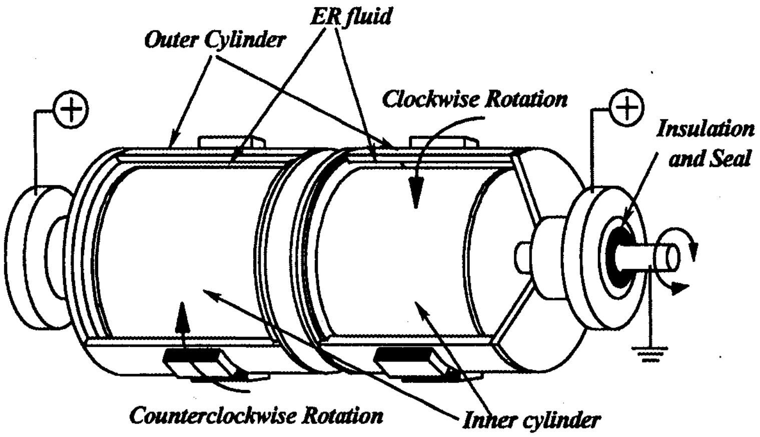


Fig. 9 The bi-directional ER clutch

system in the presence of many uncertainties such as temperature variation will be further investigated for adaptation of the proposed ER clutch actuator to practical robotic system in the near future.

6. ACKNOWLEDGMENT

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