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Adaptive Control of a Single Rod Hydraulic Cylinder - Load System under Unknown Nonlinear Friction

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Abstract : A discrete time model reference adaptive control has been applied in order to compensate the nonlinear friction characteristics in a hydraulic proportional position control system. As nonlinear friction, static and coulomb friction forces are considered and modeled as deadzone and external disturbance respectively. The model reference adaptive control system consists of a cascade combination of the dead zone, external disturbance and linear dynamic block. For adaptive control experiment, the DSP(Digital Signal Processor) board has been interfaced the hydraulic proportional position control system. The experimental results show that the MRAC(Model Reference Adaptive Control) for compensation of static and coulomb friction are very effective.

Key words : Hydraulic proportional position control system, Digital signal processor, Model reference adaptive control, Static friction, Coulomb friction

1. Introduction

The nonlinear friction characteristics play an important role in the performance determination of motion control system, and cause positioning errors and chattering[1]. Among the previous researches of the adaptive control methods for compensation of friction, Ohkawa[2] studied the friction elimination method through the compensation signal of static and coulomb frictions. Canudas[3],[4] investigated the adaptive friction compensation in DC motor and robot manipulator. Also, Tamura[5] studied the adaptive position control of DC servo motor system with coulomb friction. Friedland[6] studied the adaptive friction compensation using observer. In this paper, we executed an experiment for a hydraulic proportional position control system with Ohkawa's method[2].

The controlled plant consists of hydraulic cylinder and mass-friction load system which is driven by a hydraulic proportional position control valve. Static and coulomb frictions show nonlinear characteristics. So they are modeled by

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dead zone and external disturbance respectively.

The model reference adaptive control system (MRACS) in this study consists of a cascade combination of a nonlinear block that contains the dead zone and the disturbance term and a linear dynamic block which has been derived through linearization of hydraulic cylinder load system. The displacement and the pressure on the input and output port of the hydraulic cylinder are measured for realization of the model reference adaptive control with nonlinear friction characteristics. In order to embody adaptive control algorithm and process the measured data, digital signal processing(DSP) board are used.

2. Adaptive friction compensation of the hydraulic system

A schematic diagram of the hydraulic proportional position control system is shown in Fig. 1. The friction characteristic model used is shown in Fig. 2. Among the representative dynamic characteristics of hydraulic proportional position control system, extracting the nonlinear friction characteristic, that is static, coulomb friction and representing the remaining part as linear dynamic block, then the block diagram of the overall controlled system can be expressed as Fig. 3.

In Fig. 3 the linear dynamic block represents a open-loop transfer function, which consists of proportional directional control valve, hydraulic cylinder, mass load and viscous friction load. It has been modeled by 2nd-order system which means a linearized model. In order to construct discrete-time model reference adaptive control system(MRACS), z-transformation has been applied to this mathematical model.

$$\frac{Y(s)}{M(s)} = G(s) = \frac{K}{s(1+\tau s)} \tag{1}$$

$$\frac{Y(z^{-1})}{M(z^{-1})} = Z\left(\frac{1-e^{-Ts}}{s} \frac{K}{s(1+\tau s)} \right) = \frac{z^{-1}(b_0 + b_1 z^{-1})}{1 - a_1 z^{-1} - a_2 z^{-2}} \tag{2}$$

When the velocity is zero, the output of the deadzone element. i.e., m(k) can be expressed as eq.(3).

$$m(k) = \begin{cases} u(k) - f_{s+} & \text{for} \quad u(k) \geq f_{s+} \\ 0 & \text{for} \quad -f_{s-} < u(k) < f_{s+} \\ u(k) + f_{s-} & \text{for} \quad u(k) \leq -f_{s-} \end{cases} \tag{3}$$

In eq.(3), m(k) is the input value to linear system considering static friction force(fs). For controlled system shown in Fig. 3, the adaptive compensation process of nonlinear friction is as follows. Static friction is modeled to deadzone and to compensate it, the adaptive compensation signal $\sigma_s(k)$ and the input m(k) to linear block are derived as following with $\hat{f}_{s1}(k) = f_{s+}$, $\hat{f}_{s2}(k) = f_{s+} + f_{s-}$.

$$\sigma_s(k) = \hat{f}_{s1}(k) - \hat{f}_{s2}(k)\, p(k) \tag{4}$$

$$p(k) = \begin{cases} 0 & \text{for} \quad u(k) \geq 0 \\ 1 & \text{for} \quad u(k) < 0 \end{cases}$$

$$m(k) = u(k) + \{ \hat{f}_{s1}(k) - f_{s+} \} - \{ \hat{f}_{s2}(k) - f_{s+} - f_{s-} \} p(k) \tag{5}$$

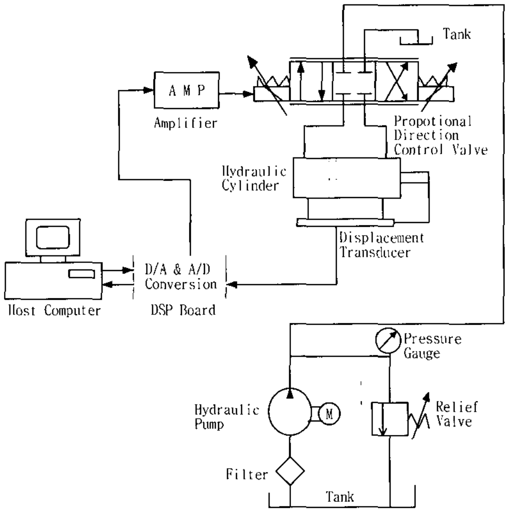


Fig. 1 Schematic diagram of the hydraulic proportional position control system

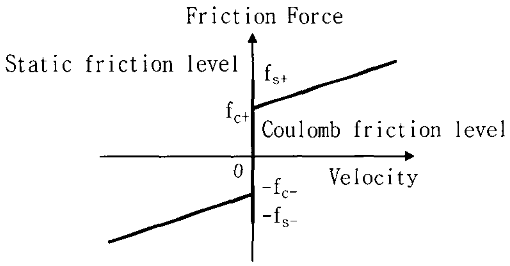


Fig. 2 Friction model

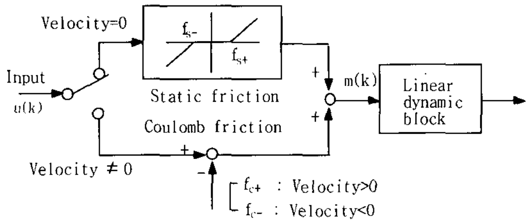


Fig. 3 Block diagram of control system

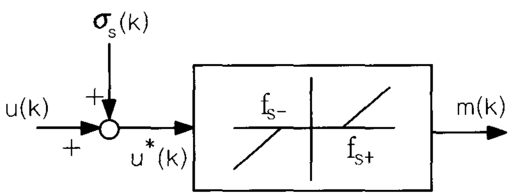


Fig. 4 Feedforward compensation of fs when velocity is zero

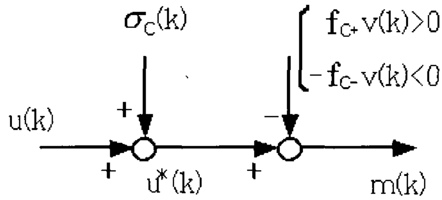


Fig. 5 Feedforward compensation of fc when velocity is not zero

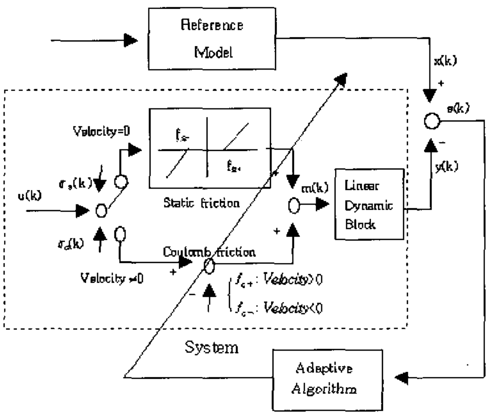


Fig. 6 Model reference adaptive control system

The process of compensating coulomb friction is similar to that of compensating static friction. The adaptive compensation signal $\sigma_c(k)$ and the input m(k) to linear block are represented as following with $\hat{f}_{s1}(k) = f_{s+}$, $\hat{f}_{s2}(k) = f_{s+} + f_{s-}$. The compensation process is shown in Fig. 5.

$$\sigma_c(k) = \hat{f}_{c1}(k) - \hat{f}_{c2}(k)q(k)$$

$$q(k) = \begin{cases} 0 & \text{for} \quad v(k) > 0 \\ 1 & \text{for} \quad v(k) < 0 \end{cases}$$

$$m(k) = u(k) + \{\hat{f}_{(k)} - f_{c+}\} - \{\hat{f}_{c2}(k) - f_{c+} - f_{c-}\}q(k) \qquad (6)$$

Finally, considering both the static and Coulomb friction characteristics, the input m(k) to the linear block becomes:

$$m(k) = u(k) + [\{\hat{f}_{k)} - f_{s+}\}$$

$$-\{ \hat{f}_{k)-f_{s+}} - f_{s-} \} p(k)] \; s(k)$$
$$+ \; [\{ \hat{f}_{k)-f_{c+}} \} - \{ \hat{f}_{k)} - f_{c+} - f_{c-} \} q(k)] \; s_1(k) \quad (7)$$

where

$$s(k) = \begin{cases} 1 & \text{for } v(k) = 0 \\ 0 & \text{for } v(k) \neq 0 \end{cases}, \; s_1(k) = 1 - s(k) \qquad (8)$$

$\hat{f}_{k)}$, $\hat{f}_{k)}$: friction compensation parameter

3. Model reference adaptive control system

Model Reference Adaptive Control System (MRACS) consists of a cascade combination of a nonlinear block containing the deadzone and disturbance term and a linear dynamic block. The MRACS is shown in Fig. 6. Reference model equation is constructed based on eq.(2) in section 2 and reference model parameters are arrayed in Table 1.

Table 1 Reference model parameters

Load \ Parameter	m = 1.5 kg	m = 15 kg
a_1	1.7261	1.9685
a_2	-0.7261	-0.9685
b_0	0.0222	0.2174
b_1	0.0045	0.0049

Let the linear dynamic block in Fig.6 be represented by:

$$A(z^{-1})y(k) = z^{-d}B(z^{-1})\,m(k)$$
$$A(z^{-1}) = 1 - a_1 z^{-1} - \cdots - a_n z^{-n},$$
$$B(z^{-1}) = b_1 z^{-1} + \cdots + b_m z^{-m}, \;\; b_1 \neq 0 \qquad (9)$$

Substituting eq.(7) into eq.(9), we can calculate output and control input like this.

$$y(k+1) = \sum_{i=1}^{n} a_i y(k+1-i) + \sum_{i=1}^{m} b_i \; [u(k+1-i)$$
$$+ \; \{ \hat{f}_{s1}(k+1-i) - f_{s+} \} \; s(k+1-i)$$
$$- \; \{ \hat{f}_{s2}(k+1-i) - f_{s+} - f_{s-} \} \; p^*(k+1-i)$$
$$+ \; \{ \hat{f}_{c1}(k+1-i) - f_{c+} \} \; s_1(k+1-i)$$
$$- \; \{ \hat{f}_{c2}(k+1-i) - f_{c+} - f_{c-} \} \; q^*(k+1-i)] \quad (10)$$

where $p^*(k) = p(k)\,s(k)$, $q^*(k) = q(k)\,s_1(k)$

And from eq. (10), input u(k) is:

$$u(k) = \sum_{i=0}^{n} \alpha_i y(k+1-i)$$
$$+ \sum_{i=2}^{m} \{ \beta_i u^*(k+1-i) + \gamma_i s(k+1-i)$$
$$+ \tau_i p^*(k+1-i) + \zeta_i s_1(k+1-i)$$
$$+ \mu_i q^*(k+1-i) - \{ \hat{f}_{s1}(k) - f_{s+} \} \; s(k)$$
$$+ \; \{ \hat{f}_{s2}(k) - f_{s+} - f_{s-} \} \; p^*(k)$$
$$- \; \{ \hat{f}_{c1}(k) - f_{c+} \} \; s_1(k)$$
$$+ \; \{ \hat{f}_{c2}(k) - f_{c+} - f_{c-} \} \; q^*(k) \qquad (11)$$

When the parameters in eq.(11) are unknown, estimator of u(k-1), i.e., $\hat{u}(k-1)$, can be obtained from :

$$\hat{u}(k-1) = \sum_{i=0}^{n} \hat{\alpha}_i(k)\, y(k-i)$$
$$+ \sum_{i=2}^{m} \{ \hat{\beta}_i(k)\, u^*(k-i) + \hat{\gamma}_i(k)\, s(k-i)$$
$$+ \; \hat{\tau}_i(k)\, p^*(k-i) + \hat{\zeta}_i(k)\, s_1(k-i)$$
$$+ \; \hat{\mu}_i(k)\, q^*(k-i) \} \qquad (12)$$

where $\hat{\alpha}_i(k)$, $\hat{\beta}_i(k)$, $\hat{\gamma}_i(k)$, $\hat{\tau}_i(k)$, $\hat{\zeta}_i(k)$ and $\hat{\mu}_i(k)$ are adjustable parameters that correspond to unknown parameters α_i , β_i, γ_i, τ_i, ζ_i , μ_i respectively. And with eq.(11), including reference model output $x(k)$, the new input $u(k)$ is defined as follows.

$$u(k)=\hat{\alpha}_0(k)\,x(k+1)+\sum_{i=1}^{n}\hat{\alpha}_i(k)\,y(k+1-i)$$
$$+\sum_{i=2}^{m}\{\hat{\beta}_i(k)u^*(k+1-i)+\hat{\gamma}_i(k)\,s(k+1-i)$$
$$+\hat{\tau}_i(k)\,p^*(k+1-i)+\hat{\zeta}_i(k)\,s_1(k+1-i)$$
$$+\hat{\mu}_i(k)\,q^*(k+1-i)\} \quad (13)$$

The process that the output error of the total MRAC system converges to zero becomes as following. Setting the input error signal $\eta(k)$ as eq. (14) using eq.(12) and eq.(13), then the error equation can be written as eq. (15).

$$\eta(k)=\hat{u}(k)-u(k) \quad (14)$$

$$\eta(k-1)=\Phi(k)\;W(k) \quad (15)$$

Considering eq.(12), we know that information of unknown friction is included in $\phi_3(k)\sim\phi_{10}(k)$. Setting the output error signal as eq.(16), input error equation is derived as eq.(17) using eq.(12) and (13).

$$e(k)=x(k)-y(k) \quad (16)$$

$$\eta(k)=-\hat{\alpha}_0(k)\,e(k+1)$$
$$+\{\Phi(k+1)-\Phi(k)\}W(k+1) \quad (17)$$

Since acccording to Goodwin's key technical lemma[7] $\eta(k)\rightarrow 0$ and $\Phi(k)\rightarrow$ constant vector, the output error converges to zero. In parameter estimation the following projection algorithm is utilized.

$$\Phi(k)=\Phi(k-1)$$
$$-\frac{W(k-1)}{c+W(k-1)^T W(k-1)}\,W(k-1)^T\Phi(k-1) \quad (18)$$

where $0.95\le c\le 0.99$.

4. Experimental method and results

4.1 Experimental method

As it is shown in Fig. 7, the hydraulic proportional position control system utilized in experiment consists of single-rod hydraulic cylinder, proportional directional control valve, friction-mass-load, digital signal processor, and personal computer. In order to realize adaptive real-time control and analyze the response characteristics of hydraulic system, rapid and effective computing systems are required. Not only the calculation speed but also the performance of I/O equipment and data transferring need to be good. Recently as the performance of DSP chip becomes more excellent and the realization of adaptive control system becomes more flexible. The DSP board offers rapid calculation speed, containing a TMS320C31 chip of Texas Instrument, and it offers versatile interface with ADC, DAC, digital I/O, encoder interface, and PWM. In designing controller, selecting sampling time is important. Considering Isermann theory[8], when settling time is Ts, the desirable range of sampling time is 0.067 Ts 〈 T 〈 0.2 Ts. In experiment, we set the sampling time 0.1 sec.

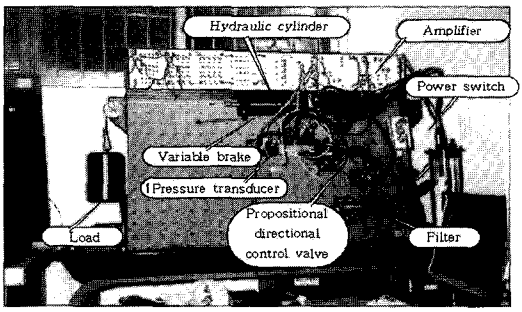
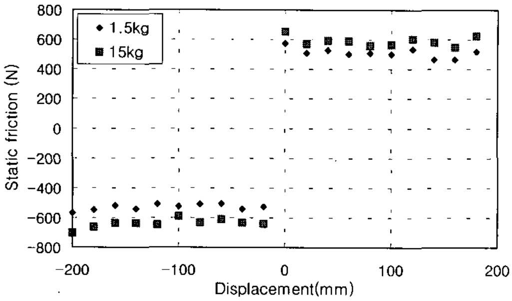


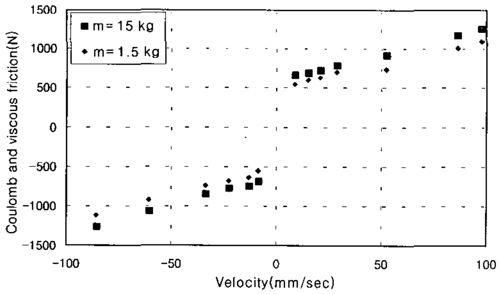
Fig. 7 Photograph of experimental equipment

4.2 Results and discussion

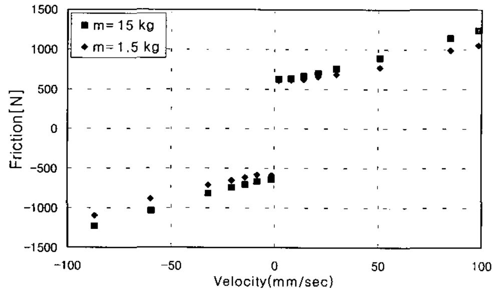
In order to acquire unknown friction information using parameter adaptation algorithm, we must set initial values for parameters consisting of parameter adaptation algorithm. Frictions existing in hydraulic cylinder-load system are measured for initial parameter setting of parameter adaptation algorithm. If we use



(a) Static friction force



(b) Coulomb and viscous friction force



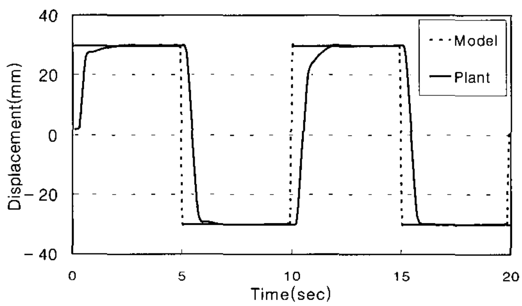
(c) Static, coulomb and viscous friction force

Fig. 8 Friction force behavior

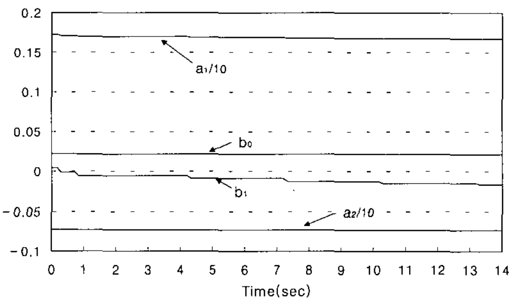
the measured friction forces as initial parameter values, it is out of range of adaptive control under unknown friction. Initial parameter values are set with endowing 40% error to the measured friction forces. Fig. 8 illustrates the behavior of friction forces. Especially Fig. 8(a) represents static friction force according to cylinder displacements. Static friction force becomes large at cylinder end position. Fig. 8(b) represents Coulomb and viscous friction force according to velocity. Fig. 8(c) shows static, Coulomb and viscosity friction force, i.e., total friction behavior.

From these observations we can conclude that static friction force appears large at cylinder stroke ends and static friction force is predominant under the velocity of 8mm/sec, and above it Coulomb and viscous friction force which is dominant. Through these analysis and design of adaptive friction compensator for the unknown friction force existing in hydraulic proportional position control system we have constructed MRAC system, the experimental results of which are as follows. In order to test model tracking performance of plant, square inputs are applied as a reference model output. In view of displacement, tracking performance is shown in Fig. 9(a). Parameter convergence behavior is also shown in Fig. 9(b).

When the mass load is 15 kg, The experimental results for model reference adaptive control are shown in Fig. 9 under various friction condition. Fig. 10(a) shows the result without friction compensation. The position error becomes large. Fig. 10(b) shows those of with static friction compensation. The position error appears small compared with those of no friction compensation. Fig. 10(c) shows those of with Coulomb friction compensation. While the mass load is in motion, the position error becomes small compared with those of static friction compensation. Fig. 10(d) shows those of both the static and Coulomb friction compensation.



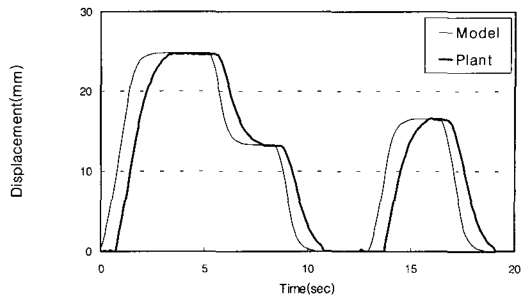
(a) Displacement behavior



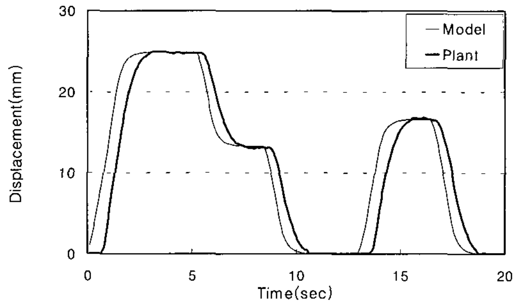
(b) On-line parameter estimation

Fig. 9 Adaptive control result for square wave : Static and Coulomb friction compensation (m=15kg)

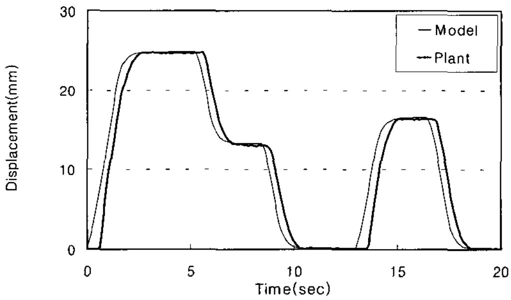
The position error decreases while the mass load is stationary or near stationary through static friction compensation and the position errors also decreases while the mass is in motion through Coulomb friction compensation, which eventually lead to improvement in overall system performance. Fig. 11 shows



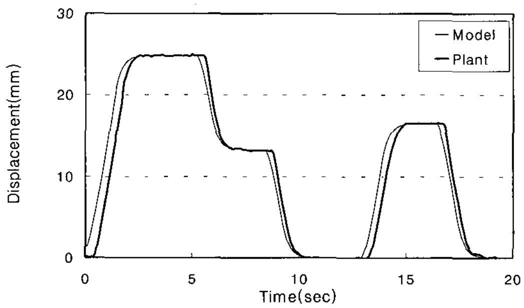
(a) Without friction compensation



(b) Static friction compensation



(c) Coulomb friction compensation



(d) Static friction and Coulomb friction compensation

Fig. 10 Adaptive Control results with different friction conditions (m=15kg)

the experimental results through both the static and Coulomb friction compensation when the mass load is 1.5kg. It shows the same tendancy as the result of Fig. 10(d). Experimental results of friction forces and adaptively estimated results of friction forces are shown in Fig.12 which coincide nearly for 15kg mass load.

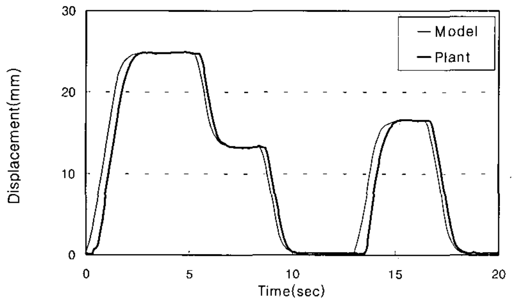


Fig. 11 Adaptive control result : static friction and coulomb friction compensation(m=1.5kg)

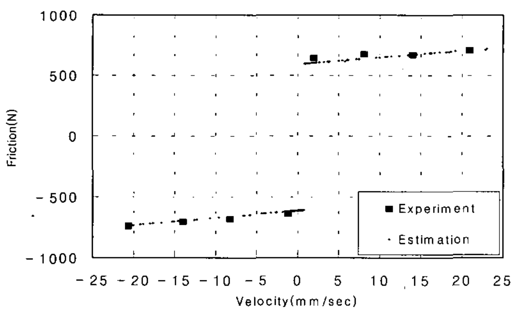


Fig. 12 A comparison of experimental result with estimated friction force.

5. Conclusion

In this paper the adaptive friction compensator has been designed in order to compensate the unknown nonlinear friction in a cylinder-load system of hydraulic proportional position control system and position control performance has been improved through Model Reference Adaptive Control. Static, Coulomb and viscosity frictions produced on the extending and retracting stages of hydraulic cylinder have been analyzed through experiments and these values has been utilized for setting initial values in parameter estimation.

The results of model reference adaptive control through the design of adaptive friction compensator are summarized as follows. In case of only deadzone compensation due to static friction the steady state errors are improved, but the position errors are not improved when the mass load is in motion. In case of only Coulomb friction compensation, the position errors are improved when the mass load is in motion.

In case of both the static and Coulomb friction compensation, the position error decreases while the mass load is stationary or near stationary through static friction compensation and the position errors also decreases while the mass is in motion through Coulomb friction compensation, which eventually lead to improvement in overall system performance. Moreover the parameter converges to a constant values.

In view of the resultant response of MRAC, the delay appearing at initial stage is due not only to the static friction in load system but also to the deadzone characteristic of hydraulic proportional control valve in itself, the latter case of which demands further research.

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