We are IntechOpen,
the world's leading publisher of
Open Access books
Built by scientists, for scientists

4,800
Open access books available

122,000
International authors and editors

135M
Downloads

Our authors are among the

154
Countries delivered to

TOP 1%
most cited scientists

12.2%
Contributors from top 500 universities

CLARIVATE ANALYTICS
BOOK CITATION INDEX
INDEXED

WEB OF SCIENCE™

Selection of our books indexed in the Book Citation Index
in Web of Science™ Core Collection (BKCI)

Interested in publishing with us?
Contact book.department@intechopen.com

Numbers displayed above are based on latest data collected.
For more information visit www.intechopen.com

Robust Control of Electro-Hydraulic Actuator Systems Using the Adaptive Back-Stepping Control Scheme

Jong Shik Kim, Han Me Kim and Sung Hwan Park
School of Mechanical Engineering, Pusan National University
Republic of Korea

1. Introduction

Conventional hydraulic actuator (CHA) systems have been widely used as power units because they can generate very large power compared to their size. In general, a CHA system consists of an electric motor, a pump, a reservoir, various valves, hoses, which are used to transfer the working fluid and an actuator. CHA systems, however, have some problems such as environmental pollution caused by the leakage of the working fluid, maintenance load, heavy weight and limited installation space. These shortcomings can be overcome by compactly integrating the components of CHA systems and by applying a suitable control scheme for the electric motor. To overcome these shortcomings of CHA systems, electro-hydraulic actuator (EHA) systems have been developed, having merits such as smaller size, higher energy efficiency and faster response than existing CHA systems (Kokotovic, 1999). However, for the robust position control of EHA systems, system uncertainties such as the friction between the piston and cylinder and the pump leakage coefficient have to be considered.

To solve these system uncertainty problems of EHA systems and to achieve the robustness of EHA systems with system disturbance and bounded parameter uncertainties, Wang et. al. presented a sliding mode control and a variable structure filter based on the variable structure system (Wang, 2005). Perron et. al proposed a sliding mode control scheme for the robust position control of EHA systems showing volumetric capacity perturbation of the pump (Perron, 2005). However, these control methods have some chattering problem due to the variable structure control scheme. The chattering vibrates the system and may reduce the life cycle of the system. Jun et. al. presented a fuzzy logic self-tuning PID controller for regulating the BLDC motor of EHA systems which has nonlinear characteristics such as the saturation of the motor power and dead-zone due to the static friction (Jun, 2004). Chinniah et. al. used a robust extended Kalman filter, which can estimate the viscous friction and effective bulk modulus, to detect faults in EHA systems (Chinniah, 2006). Kaddissi et. al. applied a robust indirect adaptive back-stepping control (ABSC) scheme to EHA systems having perturbations of the viscous friction coefficient and the effective bulk modulus due to temperature variations (Kaddissi, 2006). However, in spite of the variation of the effective bulk modulus due to the temperature and pressure variations, Chinniah et. al. considered

only the case of constant effective bulk modulus and Kaddissi et. al. used EHA systems that are not controlled by an electric motor but by a servo valve.

In this chapter, an ABSC scheme was proposed for EHA systems to obtain the desired tracking performance and the robustness to system uncertainties. Firstly, to realize a stable back-stepping control (BSC) system with a closed loop structure and to select new state variables, EHA system dynamics are represented with state equations and error equations. Defining the Lyapunov control functions, we can design a BSC system, which can guarantee exponential stability for the nominal system without system uncertainties. However, the BSC system cannot achieve robustness to system uncertainties. To overcome the drawback of the BSC system, an ABSC scheme for EHA position control systems with classical discrete disturbance observer was proposed. To evaluate the tracking performance and robustness of the proposed EHA position control system, both BSC and ABSC schemes were evaluated by computer simulation and experiment.

2. System modeling of EHA system

Figure 1 shows the simplified schematic diagram of an EHA system that consists of an electric servo motor, bi-directional gear pump and actuator. The servo motor rotates the gear pump, which, in turn, generates the flow rate. The pressure generated by the flow rate changes the position of the piston rod. The movement direction of the piston is related to the rotational direction of the servo motor. The chamber volumes of the actuator depend on the cross sectional area and the displacement of the piston as follows

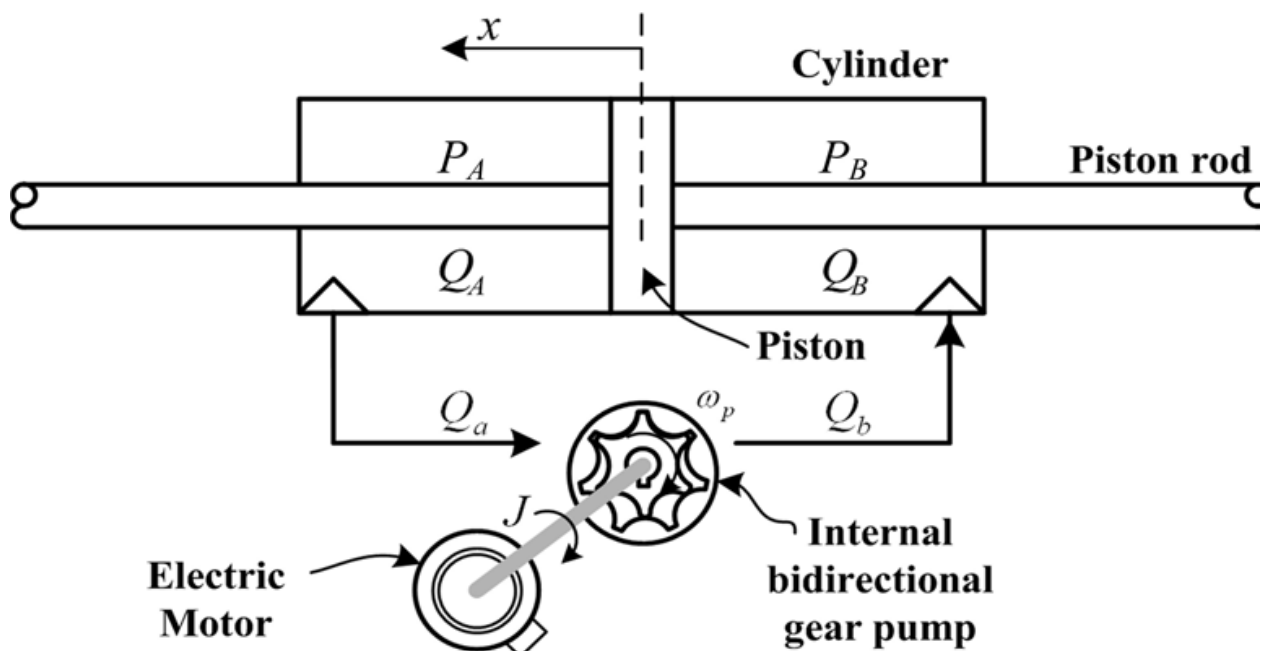$$\begin{cases} V_A(t) = V_{0A} + Ax(t) \\ V_B(t) = V_{0B} - Ax(t) \end{cases} \tag{1}$$

Fig. 1. Simplified schematic diagram of an EHA system

where V and V_0 are the chamber volume and the initial chamber volume, respectively, A and x are the pressure area of a double rod hydraulic cylinder and displacement of the piston, respectively, and subscripts 'A' and 'B' denote the chamber notations of the actuator. Considering the fluid compressibility and continuity principle for the actuator, the flow rate equations of both ports of the actuator can be represented as (Merritt, 1967)

$$\begin{cases} Q_A = A\dot{x} + \dfrac{V_{0A} + Ax}{\beta_e}\dot{P}_A + LP_A \\ Q_B = A\dot{x} - \dfrac{V_{0B} - Ax}{\beta_e}\dot{P}_B - LP_B \end{cases} \tag{2}$$

where Q is the flow rate in the actuator, β_e is the effective bulk modulus of the working fluid, and L and P are the actuator external leakage coefficient and the pressure in the chamber, respectively.

It is assumed that there is no fluid leakage of conduits because the conduits of EHA systems are very short and hard. Then, (2) can be expressed as

$$\begin{cases} Q_A = A\dot{x} + \dfrac{V_{0A} + Ax}{\beta_e}\dot{P}_A \\ Q_B = A\dot{x} - \dfrac{V_{0B} - Ax}{\beta_e}\dot{P}_B \end{cases} \tag{3}$$

The electric motor, directly connected to the hydraulic pump, changes the flow direction and adjusts the flow rate through the ports. In addition, the pressure generated by the continuous supply of flow in the actuator can produce a minute fluid leakage of the pump. Hence, the equations for the fluid leakage of the pump are expressed as

$$\begin{cases} Q_a = C_p \omega_p - L_f P_L \\ Q_b = -Q_a \end{cases} \tag{4}$$

where Q is the flow rate of the pump, whose subscripts a and b denote the ports of the pump, C_p is the volumetric capacity of the pump, ω_p is the rotational velocity of the electric motor, L_f is the leakage factor of the pump and the load pressure $P_L = P_A - P_B$. From (4), the inflow and outflow of the pump are expressed as functions of the rotational velocity ω_p. In addition, the actuator dynamic equation of EHA systems is expressed as

$$(P_A - P_B)A = M\ddot{x} + F_f + F_{ex} \tag{5}$$

where M and x are the mass and displacement of the piston, respectively, F_f is the friction force between the cylinder and piston and F_{ex} is the external disturbance force.
In order to substitute (3) into (5), the derivative of (5) is expressed as

$$(\dot{P}_A - \dot{P}_B)A = M\dddot{x} + \dot{F}_f + \dot{F}_{ex} \tag{6}$$

In addition, it is assumed that the conduits connected between the actuator ports and the pump ports are very short. Then, the flow rates in (3) and (4) can be represented as $Q_A = Q_a$

and $Q_B = Q_b$. Substituting (1) through (4) into (6), therefore, the dynamic equation of EHA systems can be represented as

$$\ddot{x} = -\frac{1}{M}\left\{\beta_e A^2\left(\frac{1}{V_A} + \frac{1}{V_B}\right)\dot{x} + \dot{F}_f + \dot{F}_{ex}\right\} + \frac{\beta_e A}{M}\left(\frac{1}{V_A} + \frac{1}{V_B}\right)\left(L_f P_L - C_p \omega_p\right)$$ (7)

To represent the characteristics of the friction F_f between the piston and cylinder, the LuGre friction model is considered. The LuGre friction model is based on bristles analysis, which is represented with the average deflection force of bristles stiffness. The deflection displacement equation of bristles z, which is actually unmeasurable by experiment, is expressed as (Choi, 2004)(Lee, 2004)

$$\frac{dz}{dt} = \dot{x} - \frac{\sigma_0 \,|\, \dot{x}\,|}{g(\dot{x})} z$$ (8)

where σ_0 is the bristles stiffness coefficient, z is the unmeasurable internal state and the nonlinear function $g(\dot{x})$ depends on the material property, grade of lubrication and temperature; that is

$$g(\dot{x}) = F_c + F_s e^{-(\dot{x}/v_{sv})^2}$$ (9)

where F_c, F_s and v_{sv} represent the Coulomb friction force, static friction force, and Stribeck velocity between the cylinder and piston, respectively.

If the relative velocity of the contact materials increases gradually, the friction force decreases instantaneously and then it increases gradually again; this effect is called the Stribeck effect and the relative velocity is the Stribeck velocity. This phenomenon depends on the material property, grade of lubrication and temperature. The friction force F_f can be represented with the average deflection z and the velocity of the piston $\dot{x}$ as follows

$$F_f = \sigma_0 z + \sigma_1 \dot{z} + \mu \dot{x}$$ (10)

where σ_1 and μ represent the bristles damping and viscous friction coefficients, respectively.

3. Controller design for the EHA position control systems

The EHA position control system consists of the inner loop for the angular velocity control of the servo motor/pump and the outer loop for the position control of the piston. For the velocity control of the motor in the inner loop, Kokotovic et al. applied an adaptive control scheme so that the EHA position control systems can have robustness (Kokotovic, 1999). Habibi et. al. presented that if the inner loop dynamics are stable, the control gains of the PID velocity controller in the inner loop can have relatively large values and then the disturbance effect can be sufficiently rejected (Habibi, 1999). The velocity controller in the inner loop is very important because it regulates the electric motor. However, the case of (Kokotovic, 1999) is very complicated and the case of (Habibi, 1999), although it is theoretically possible, has a physical limitation that increases the control gains of the inner loop controller. Therefore, it is desirable to handle the controller in the outer loop rather

than in the inner loop to improve the performance and robustness of EHA position control systems.

In this chapter, the BSC and ABSC schemes based on EHA system dynamics are considered as the position controller. Firstly, to design a BSC system, (7) is transformed to a general form (Slotine, 1999) as follows

$$\dddot{x} = f + bu \tag{11}$$

where

$$f = -\frac{1}{M}\left\{\beta_e A^2 \left(\frac{1}{V_A} + \frac{1}{V_B}\right)\dot{x} + \dot{F}_f + \dot{F}_{ex} - \beta_e L_f A \left(\frac{1}{V_A} + \frac{1}{V_B}\right)P_L\right\},$$

$$b = -\frac{\beta_e A(V_A + V_B)C_p}{MV_AV_B},$$

$$u = \omega_p$$

Now, let (11) represent state equations as follows

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = x_3 \\ \dot{x}_3 = f + bu \end{cases} \tag{12}$$

And, in order to design the BSC system, new state variables are defined as follows

$$z_1 = x_1 - x_d \tag{13}$$

$$z_2 = x_2 - \alpha_1(z_1) \tag{14}$$

$$z_3 = x_3 - \alpha_2(z_1, z_2) \tag{15}$$

where x_d is the desired position input, and α_1 and α_2 are the functions for new state variables, which can be obtained through the following BSC design procedure.

Step 1.

From (13), the state equation for z_1 can be described as

$$\dot{z}_1 = z_2 + \alpha_1(z_1) - \dot{x}_d \tag{16}$$

$\alpha_1(z_1)$ is the virtual control which should be selected to guarantee the stability of the control system through the Lyapunov control function(LCF) which is defined as

$$V_1(z_1) = \frac{1}{2}z_1^2 \tag{17}$$

Then,

$$\dot{V}_1(z_1) = z_1\dot{z}_1 = z_1[\alpha_1(z_1) - \dot{x}_d] + z_1z_2 \tag{18}$$

From (18), if $\alpha_1(z_1) = -k_1z_1 + \dot{x}_d$, (16) can be exponentially stable when $t \to \infty$. And $k_1(>0)$ is a design parameter.

Step 2.

From (14), the state equation for z_2 can be described as

$$\dot{z}_2 = z_3 + \alpha_2(z_1, z_2) - \dot{\alpha}_1(z_1) \tag{19}$$

where

$$\dot{\alpha}_1(z_1) = \frac{\partial \alpha_1}{\partial z_1}\dot{z}_1 + \frac{\partial \alpha_1}{\partial \dot{x}_d}\ddot{x}_d = -k_1\dot{z}_1 + \ddot{x}_d = -k_1(z_2 - k_1 z_1) + \ddot{x}_d$$

Since (19) includes the information of (16), the second LCF for obtaining the virtual control to guarantee the stability of the control system can be selected as

$$V_2(z_1, z_2) = V_1(z_1) + \frac{1}{2}z_2^2 \tag{20}$$

Then,

$$\dot{V}_2(z_1, z_2) = \dot{V}_1(z_1) + \dot{z}_2 z_2 = -k_1 z_1^2 + z_2 z_3 + z_2[z_1 + \alpha_2(z_1, z_2) - \dot{\alpha}_1(z_1)] \tag{21}$$

If the virtual control α_2 in the last term of (21) is defined as

$$\alpha_2(z_1, z_2) = -k_2 z_2 - z_1 + \dot{\alpha}_1(z_1)$$

where $k_2(>0)$ is a design parameter, then another expression of α_2 can be rearranged as

$$\alpha_2(z_1, z_2) = -(k_1 + k_2)z_2 - (1 - k_1^2)z_1 + \ddot{x}_d \tag{22}$$

Therefore

$$\dot{V}_2 = z_2 z_3 - k_1 z_1^2 - k_2 z_2^2 \tag{23}$$

Step 3.

From (15), the state equation for z_3 is described as

$$\dot{z}_3 = \dot{x}_3 - \dot{\alpha}_2(z_1, z_2) = f + bu - \dot{\alpha}_2(z_1, z_2) \tag{24}$$

where

$$\dot{\alpha}_2(z_1, z_2) = \frac{\partial \alpha_2}{\partial z_1}\dot{z}_1 + \frac{\partial \alpha_2}{\partial z_2}\dot{z}_2 + \frac{\partial \alpha_2}{\partial \dot{\alpha}_1}\ddot{\alpha}_1 = -\dot{z}_1 - k_2\dot{z}_2 + \ddot{\alpha}_1 \tag{25}$$

and

$$\ddot{\alpha}_1 = \frac{\partial \dot{\alpha}_1}{\partial z_1}\dot{z}_1 + \frac{\partial \dot{\alpha}_1}{\partial z_2}\dot{z}_2 + \frac{\partial \dot{\alpha}_1}{\partial \ddot{x}_d}\ddot{x}_d = k_1^2\dot{z}_1 - k_1\dot{z}_2 + \dddot{x}_d \tag{26}$$

Substituting (16), (19), and (26) into (25), (25) can be rearranged as

$$\dot{\alpha}_2(z_1, z_2) = (k_1^2 - 1)\dot{z}_1 - (k_1 + k_2)\dot{z}_2 + \dddot{x}_d \tag{27}$$

Since (24) uses the information of z_1 and z_2 due to the property of the design procedure of the back-stepping control, the third LCF for (15) can be defined as

$$V_3(z_1, z_2, z_3) = V_2(z_1, z_2) + \frac{1}{2}z_3^2 \tag{28}$$

Then,

$$\dot{V}_3(z_1, z_2, z_3) = \dot{V}_2 + z_3\dot{z}_3 = -k_1z_1^2 - k_2z_2^2 + z_3(z_2 + f + bu - \dot{\alpha}_2) \tag{29}$$

If the last term of (29) for satisfying the system stability is defined as

$$-k_3z_3 = z_2 + f + bu - \dot{\alpha}_2 \tag{30}$$

then the BSC law can be selected as

$$u = \frac{1}{b}(\dot{\alpha}_2 - k_3z_3 - z_2 - f) \tag{31}$$

From (31), if the information of f is assumed to be known, the negative semi-definite of $\dot{V}_3$ can be obtained by substituting (31) into (29) as

$$\dot{V}_3(z_1, z_2, z_3) = \dot{V}_2 + z_3\dot{z}_3 = -k_1z_1^2 - k_2z_2^2 - k_3z_3^2 \leq 0 \tag{32}$$

From (32), it is found that EHA position control systems using the BSC law of (31) can guarantee exponential stability.

If system uncertainties can be exactly known, the BSC law of (31) can achieve the desired tracking performance and the robustness to the system uncertainties of EHA systems. However, the BSC law of (31) will cause a tracking error and does not achieve the robustness to the system uncertainties because the value of f cannot be exactly known. To improve the tracking performance and the robustness to the system uncertainties, the value of f, in which system uncertainties are included, should be estimated.

Therefore, in this chapter, an ABSC scheme is proposed, which is the BSC scheme with the estimator of f. In order to design the ABSC system, the BSC law of (31) is modified as

$$u = \frac{1}{b}(\dot{\alpha}_2 - k_3z_3 - z_2 - \hat{f}) \tag{33}$$

where $\hat{f}$ is the estimator of the system uncertainties.
Substituting (33) into (12), (12) is modified as

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = x_3 \\ \dot{x}_3 = \tilde{f} + \dot{\alpha}_2 - k_3z_3 - z_2 \end{cases} \tag{34}$$

where $\tilde{f} = f - \hat{f}$.

From (13), (14), and (15), these equations are the error equations for the velocity, acceleration and jerk of the piston, which include $\alpha_1(z_1)$ and $\alpha_2(z_1, z_2)$ that guarantee the exponential stability of EHA position control systems. Substituting these equations into (34), the error dynamics can be represented as

$$\begin{cases} \dot{z}_1 = z_2 - k_1 z_1 \\ \dot{z}_2 = z_3 - k_2 z_2 - z_1 \\ \dot{z}_3 = \tilde{f} - k_3 z_3 - z_2 \end{cases} \tag{35}$$

From (35), the LCF is defined as

$$V_4 = \frac{1}{2} z_1^2 + \frac{1}{2} z_2^2 + \frac{1}{2} z_3^2 + \frac{1}{2\gamma} \tilde{f}^2 \tag{36}$$

where γ is a positive constant.

The derivative of (36) can be described as

$$\dot{V}_4 = z_1 \dot{z}_1 + z_2 \dot{z}_2 + z_3 \dot{z}_3 + \frac{1}{\gamma} \tilde{f} \dot{\tilde{f}} = -k_1 z_1^2 - k_2 z_2^2 - k_3 z_3^2 + \tilde{f}(z_3 - \frac{1}{\gamma} \dot{\tilde{f}}) \leq 0 \tag{37}$$

From (37), an estimation rule to guarantee the system stability can be obtained as

$$\dot{\hat{f}} = \dot{f} - \gamma z_3 \tag{38}$$

Equation (38) uses the information of z_3, which depends on the information of z_1 and z_2. Therefore, (38) closely relates to α_1 and α_2, which guarantee the stability of BSC systems. However, (38) cannot be used to the estimation rule because the value of f is unknown. On the other hand, if f is assumed as a lumped uncertainty, system uncertainty f can be estimated by $\hat{f} = -\gamma z_3$. However, since the value of f for the EHA system is changed according to the operating condition, it cannot be assumed as the lumped uncertainty. Therefore, to obtain the value of f, the classical discrete disturbance observer scheme was used. Assuming that the sampling rate of the control loop is very fast, the classical discrete disturbance observer expressed by the difference equation is induced from (31) as follows

$$f(k-1) = bu(k) - \dot{\alpha}_2(k) + k_3 z_3(k) + z_2(k) \tag{39}$$

To analyze the stability of the proposed control scheme, (38) is substituted into (37). Then,

$$\dot{V}_4 = -k_1 z_1^2 - k_2 z_2^2 - k_3 z_3^2 = -\mathbf{z}^T \mathbf{K} \mathbf{z} < 0 \tag{40}$$

where $\mathbf{K}$ is the diagonal matrix of k_1, k_2 and k_3, $\mathbf{z} = [z_1 \ z_2 \ z_3]^T$, and $\dot{V}_4 = 0$ if $\mathbf{z} = 0$. If $\mathbf{z}$ is bounded, (40) can be defined as

$$\Phi(t) = \mathbf{z}^T \mathbf{K} \mathbf{z} \geq 0 \tag{41}$$

Integrating (41) from 0 to t, the following result can be obtained

$$\int_0^t \Phi(\tau)d\tau = V_4(\mathbf{z}(0), \tilde{f}(0)) - V_4(t) \tag{42}$$

Applying Barbalat's Lemma(Krstic, 1995) to (42), we can obtain that $\Phi(t) \to 0$ as $t \to \infty$. Therefore,

$$\lim_{t \to \infty} \int_0^t \Phi(\tau)d\tau \le V_4(\mathbf{z}(0), \tilde{f}(0)) < \infty \tag{43}$$

4. Computer simulation

In order to evaluate the validity of the proposed control scheme for EHA position control systems, a sinusoidal reference input was considered as follows

$$x_d = \sin(0.25\pi t) + \sin(0.05\pi t) \text{ [cm]} \tag{44}$$

This sinusoidal reference input is suitable for evaluating the tracking performance and the robustness of EHA position control systems because it reflects the various changes in the magnitudes of the velocity and position of the piston. Table 1 shows the system parameters of the EHA system which are used to computer simulation. Figure 2 shows the block diagram of the EHA position control system.

| Notation	Description	Unit
V_0	Initial volume of the chamber	$3.712 \times 10^{-4} \text{ m}^3$
A	Pressure area of the piston	$5.58 \times 10^{-3} \text{ m}^2$
M	Piston mass	5 kg
β_e	Effective bulk modulus	$1.7 \times 10^3 \text{ MPa}$
L_f	Leakage factor of the pump	$3.16 \times 10^{-16} \text{ m}^3/\text{Pa}$
σ_0	Bristles stiffness coefficient	$5.77 \times 10^6 \text{ N/m}$
σ_1	Bristles damping coefficient	$2.28 \times 10^4 \text{ N/m/s}$
$\omega_{p\max}$	Maximum speed of the motor	178 rad/s
C_p	Volumetric capacity of the pump	$1.591 \times 10^{-6} \text{ m}^3/\text{rad}$
μ_0	Coulomb friction coefficient	370 N
μ_1	Static friction	217 N
μ_2	Viscous friction coefficient	2318 N/m/s
v_{sv}	Stribeck velocity	0.032 m/s

Table 1. System parameters of the EHA
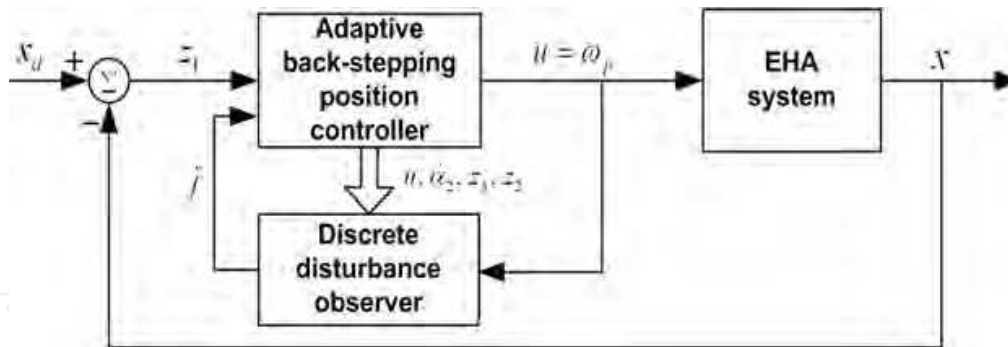
Fig. 2. Block diagram of the EHA position control system

Figure 3 shows the tracking errors of the BSC and ABSC systems for the sinusoidal reference. This result shows that the ABSC system has better tracking performance than the BSC system and has error repeatability precision of higher reliability than the BSC system. In addition, in both position and control schemes relatively large tracking errors occur at the nearly zero velocity regions. This is due to the effect of dynamic friction characteristics, which produce an instantaneous large force at the nearly zero velocity regions. For the transient response region of the initial operation of EHA position control systems, the ABSC system with the estimator for system uncertainties yields approximately 40% improvement compared with the BSC system without the estimator because the f in (31) including system uncertainties is estimated by (43), as shown in Fig. 4. Figure 4 shows the estimated value $\hat{f}$ for the system uncertainties of EHA systems obtained by the proposed adaptive rule. The estimated value plat a role in the consideration of nonlinearity and uncertainties included in EHA systems.
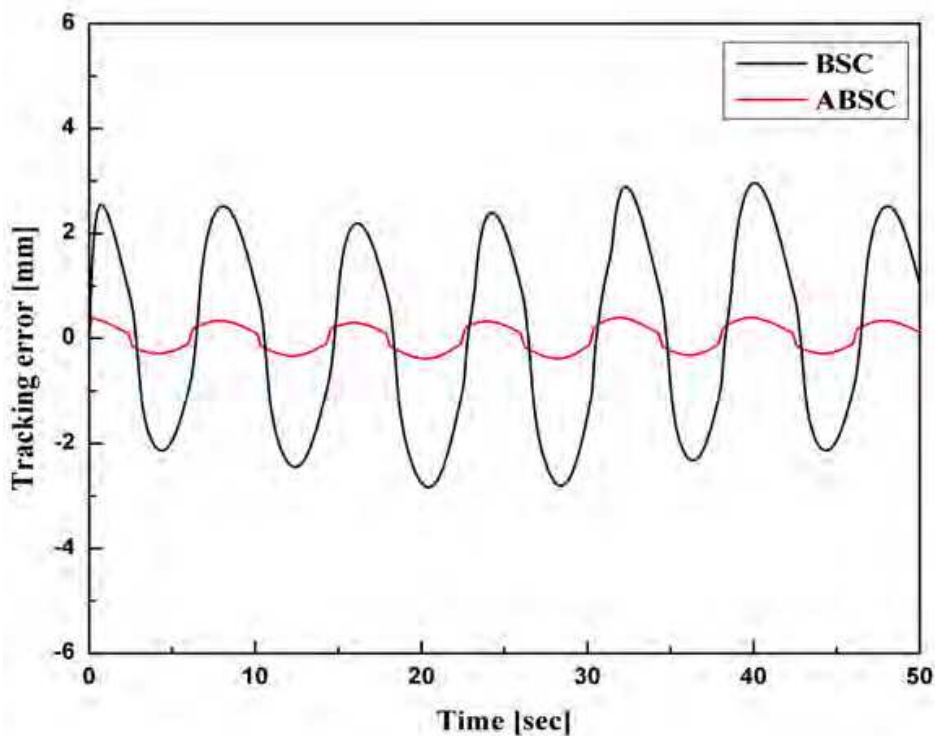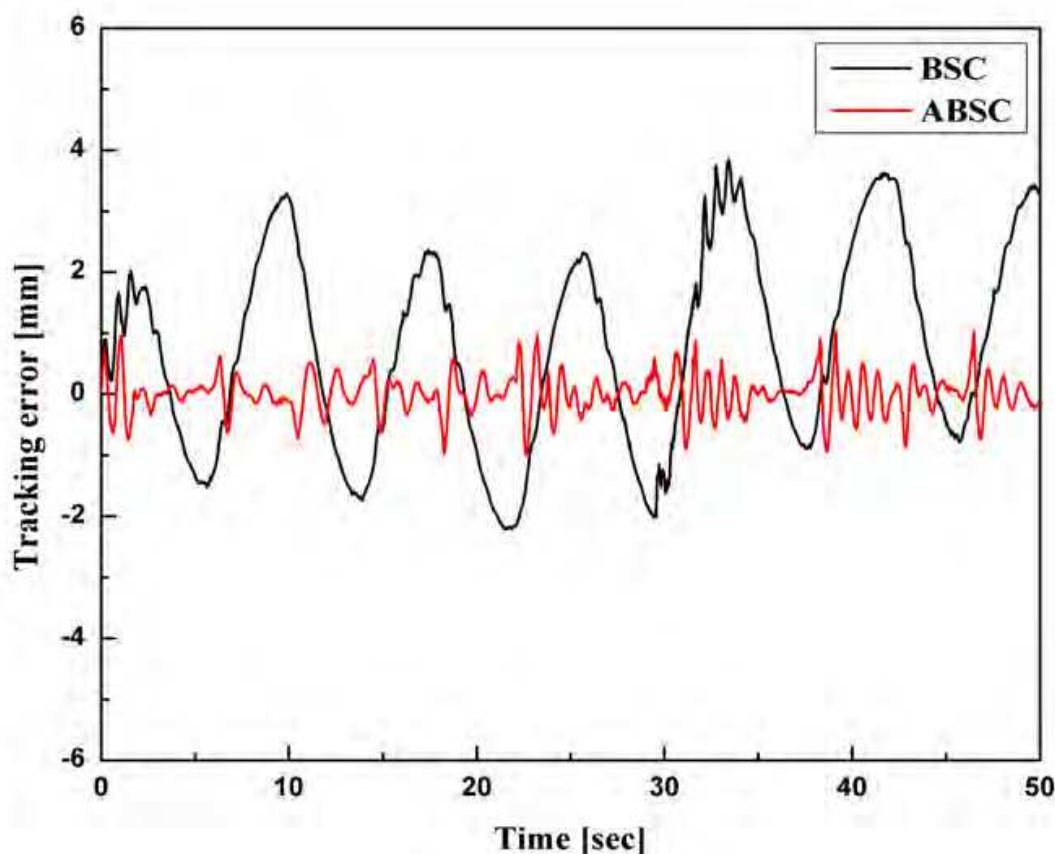

Fig. 3. Tracking errors of the BSC and ABSC systems for the sinusoidal reference input
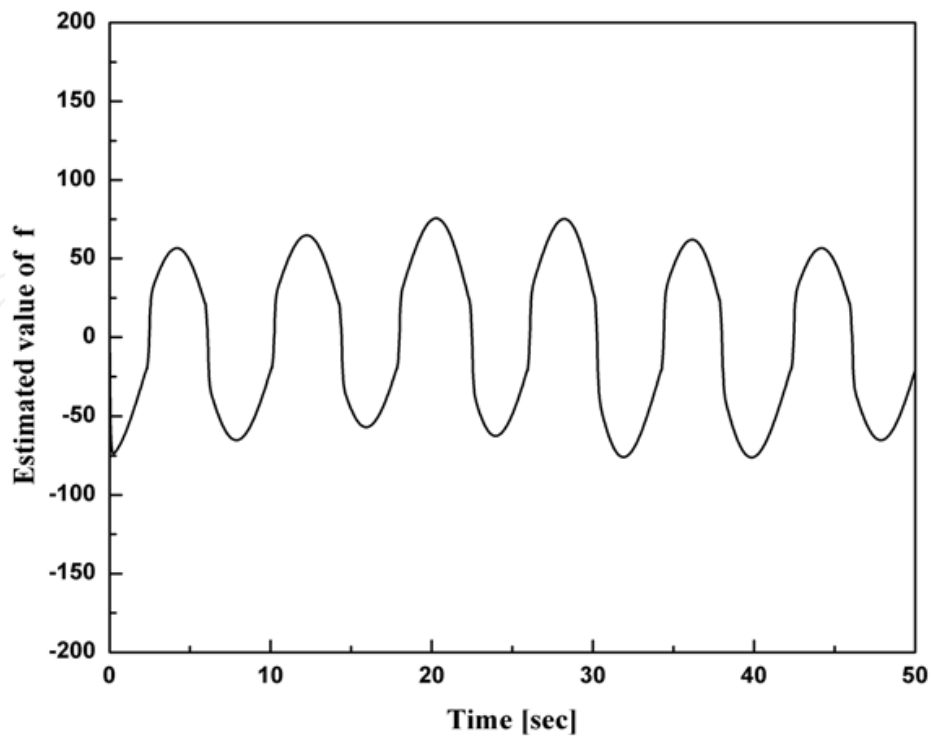
Fig. 4. Estimated value for the system uncertainties of the ABSC system for the sinusoidal reference input
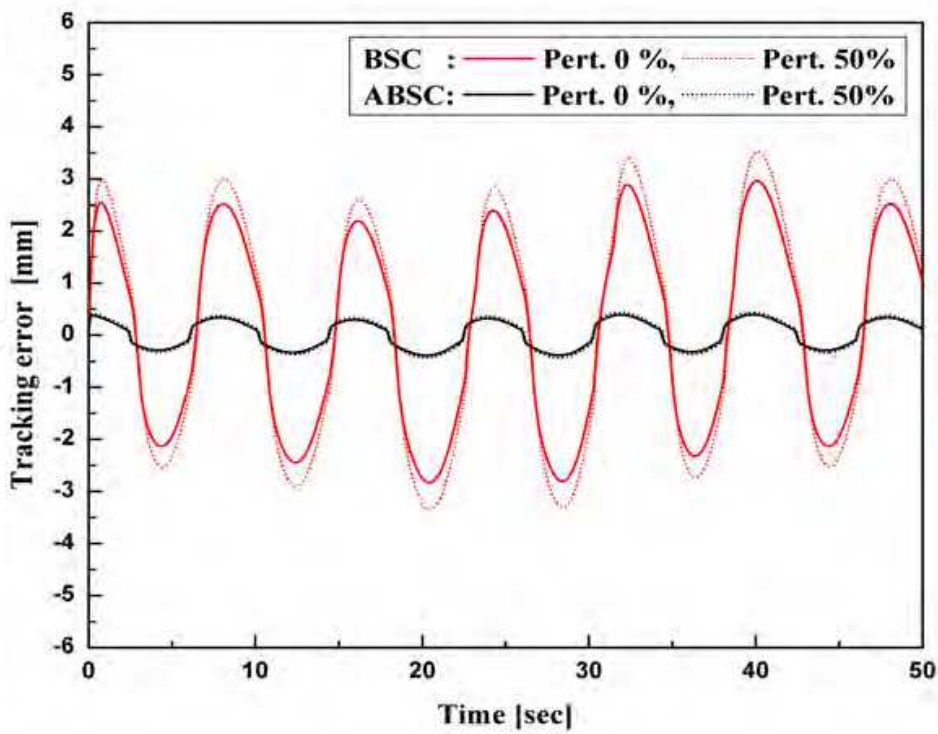
Fig. 5. Tracking errors of the BSC and ABSC systems with the perturbation of the system parameters

Figure 5 shows the tracking errors of the BSC and ABSC systems having perturbations of the system parameters such as the Coulomb friction, viscous friction and pump leakage coefficient in the EHA system for the sinusoidal reference input. It was assumed that the system parameters have a perturbation of 50%. From Fig. 5, it was found that the perturbations of the system parameters of the EHA system are closely related with the tracking performance of the EHA system. Table 2 shows the tracking RMS errors of the BSC and ABSC systems according to the perturbation of the system parameters. The variations of the tracking RMS errors due to the 50% perturbation of the system parameters for the BSC and ABSC systems are 17.6% and 3.02%, respectively. These results show that the proposed position control scheme has the desired robustness to system uncertainties such as the perturbation of the viscous friction, Coulomb friction and pump leakage coefficient.

Control scheme	Perturbation ratio	RMS value
BSC	0%	1.878 mm
	50%	2.209 mm
ABSC	0%	0.265 mm
	50%	0.273 mm

Table 2. Tracking RMS errors of the BSC and ABSC systems according to the perturbations of the system parameters

5. Experimental results and discussion

Figure 6 shows the experimental setup of the EHA system. To evaluate the effectiveness of the proposed control system, the PCM-3350(AMD Geode processor, 300MHz) was used. The control algorithms were programmed by Turbo-C++ language on MS-DOS, in order to directly handle the PCM-3718 as a data acquisition board. The PCM-3718 is a fully multifunctional card with PC/104 interface. In addition, to measure the position of the piston, an LVDT(linear variable differential transformer) sensor was used. The sampling rate was set to 1 kHz.

Fig. 6. Experimental setup of the EHA system

Figure 7 shows the tracking errors of the BSC and ABSC systems for the sinusoidal reference input, which was used in the computer simulation. The tracking error of the BSC system is relatively large when the direction of the piston is changed because the BSC system cannot compensate the friction of the EHA system. In addition, the tracking error of the BSC varies according to the direction of the piston because of the system uncertainties of the EHA system. However, the ABSC system has better tracking performance than the BSC system because the ABSC system can effectively compensate the system uncertainties as well as the nonlinear friction effects by using the estimated value $\hat{f}$, which is shown in Fig. 8.

Figure 9 shows the speed of the motor as the control input for the sinusoidal reference input. Figure 10 shows the tracking errors of the BSC and ABSC systems for the square wave type reference input. The characteristics of the transient responses of the BSC and the ABSC systems are almost same. In the BSC system, however, steady-state error occurs relatively large in the backward direction. This shows that the BSC system cannot compensate the system uncertainties of the EHA system. But we can show that the ABSC system can effectively compensate the system uncertainties regardless of the piston direction. Figure 11 shows the estimated value $\hat{f}$ for the system uncertainties of the ABSC system for the square wave type reference input. The estimated value $\hat{f}$ for the system uncertainties makes the desired tracking performance and robustness to the EHA system with system uncertainties. Figure 12 shows the speed of the motor as the control input for the square wave type reference input.

Fig. 7. Tracking errors of the BSC and ABSC systems for the sinusoidal reference input
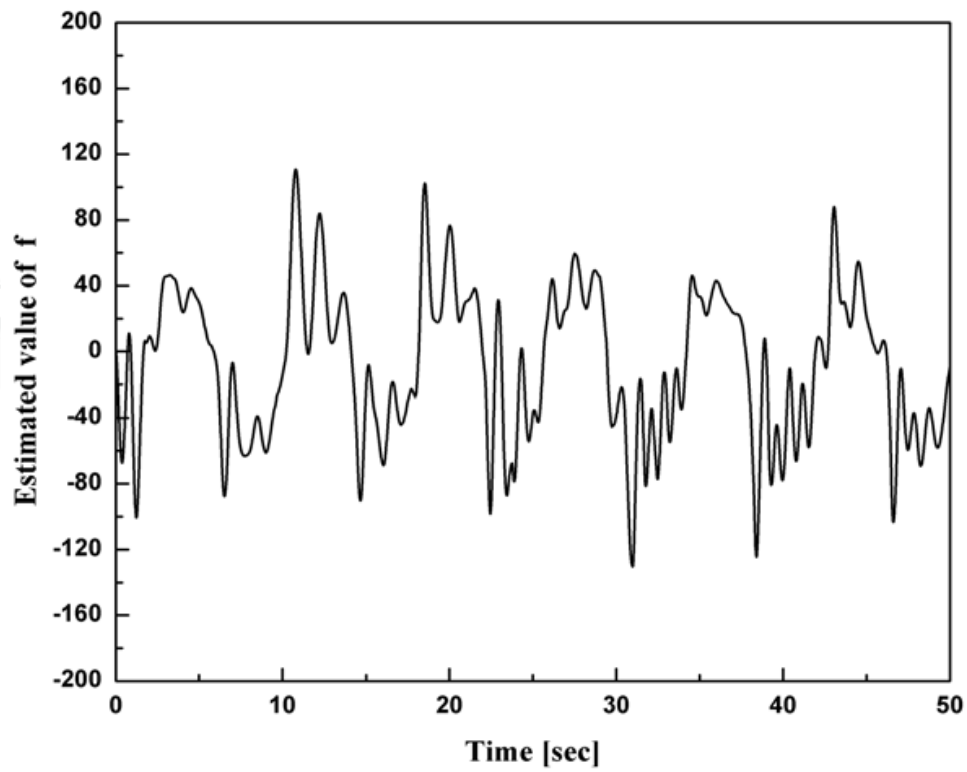
Fig. 8. Estimated value for the system uncertainties of the ABSC system for the sinusoidal reference input
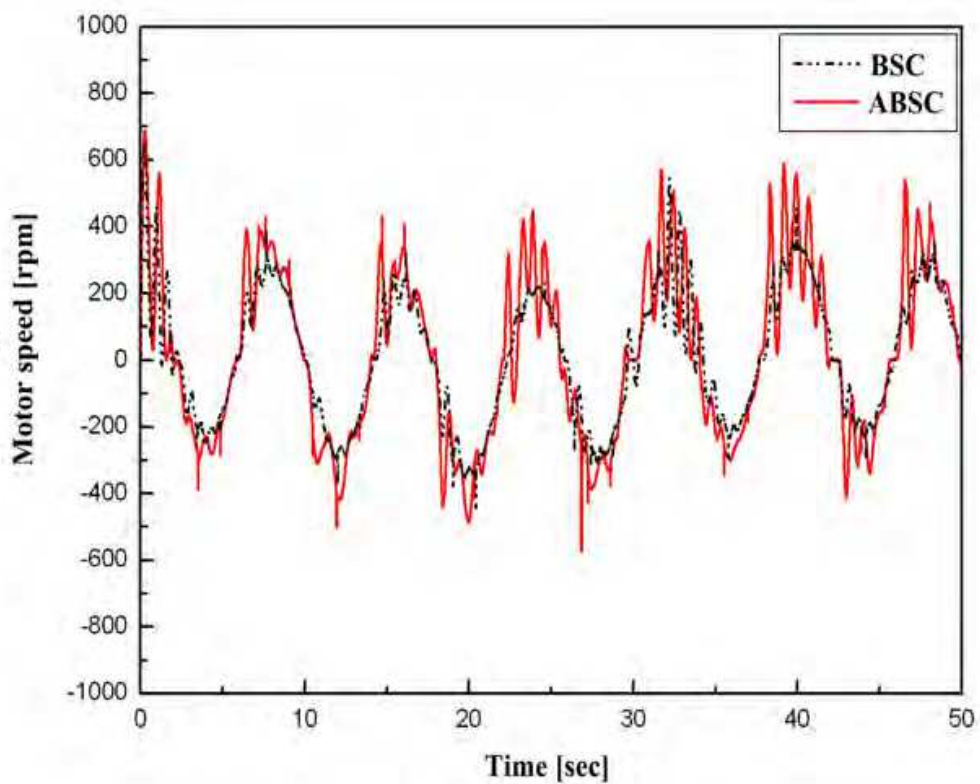

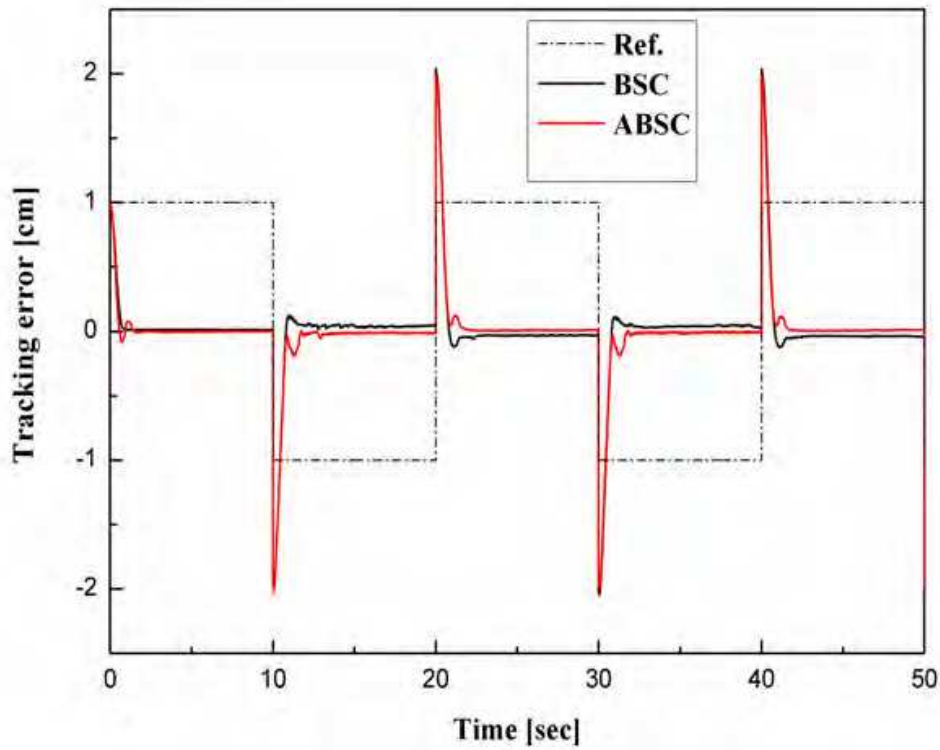Fig. 9. Speed of the motor as the control input for the sinusoidal reference input

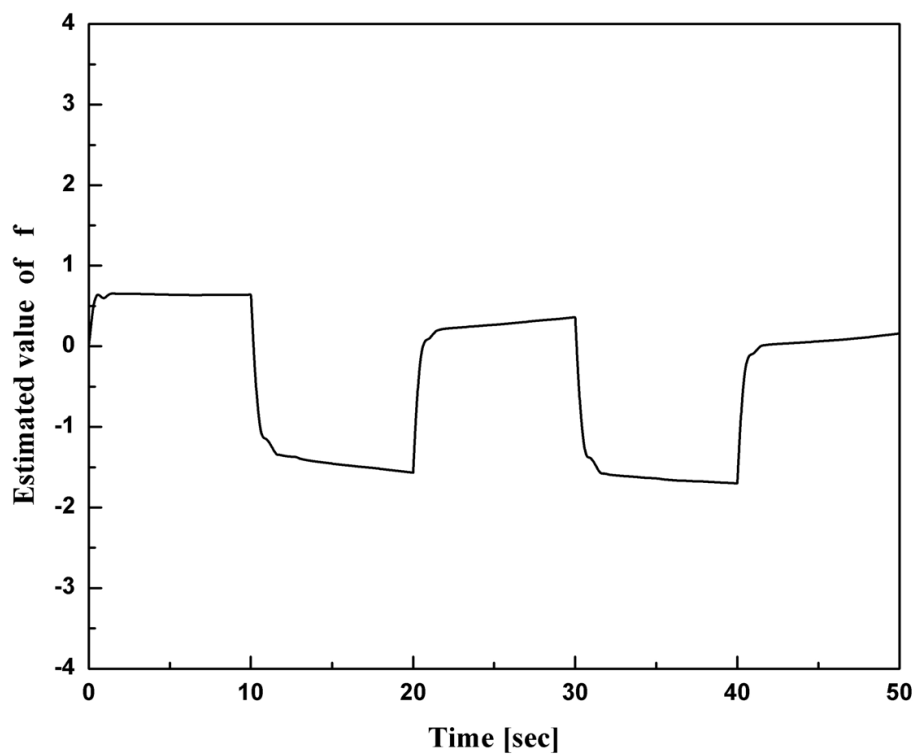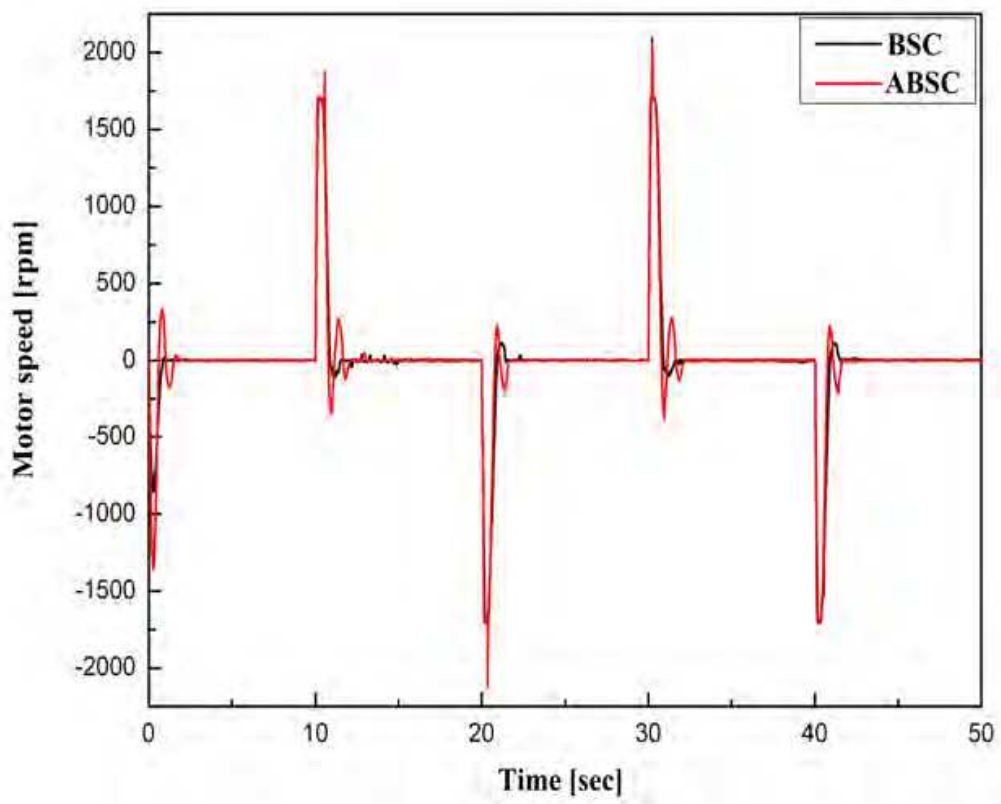Fig. 10. Tracking errors of the BSC and ABSC systems for the square wave type reference input

Fig. 11. Estimated value for the system uncertainties of the ABSC system for the square wave type reference input

Fig. 12. Speed of the motor as the control input for the square wave type reference input

Table 3 shows the tracking RMS errors of the BSC and ABSC systems for the sinusoidal reference input and the square wave type reference input at steady-state. From Table 3, it was found that using the ABSC system instead of the BSC system yields about 5 times improvement in the tracking performance of the EHA position control system.

Control system	Sinusoidal reference input	Square wave type reference input at steady state
BSC	1.762 mm	0.395 mm
ABSC	0.309 mm	0.114 mm

Table 3. Tracking RMS errors of the BSC and ABSC systems

6. Conclusion

A robust position control of EHA systems was proposed by using the ABSC scheme, which has robustness to system uncertainties such as the perturbation of viscous friction, Coulomb friction and pump leakage coefficient. Firstly, a stable BSC system based on the EHA system dynamics was derived. However, the BSC scheme had a drawback: it could not consider system uncertainties. To overcome the drawback of the BSC scheme, the ABSC scheme was proposed having error equations for the velocity, acceleration and jerk of the piston, respectively, which were induced by the BSC scheme. To evaluate the performance and robustness of the proposed EHA position control system, BSC and ABSC schemes were implemented in a computer simulation and experiment. It was found that the ABSC scheme can yield the desired tracking performance and the robustness to system uncertainties.

7. References

Y. Chinniah, R. Burton and S. Habibi (2006), Failure monitoring in a high performance hydrostatic actuation system using the extended kalman filter, *Int. J. Mechatronics 16(10)* , pp. 643-653.

J. J. Choi, J. S. Kim and S. I. Han (2004), Pre-sliding friction control using the sliding mode controller with hysteresis friction compensator, *KSME Int'l J. 18(10)*, pp. 1755-1762.

S. Habibi and A. Goldenberg (2000), Design of a new high-performance electro-hydraulic actuator, IEEE Trans. *Mechatronics 5(2)*, pp. 158-164.

L. Jun, F. Yongling, Z. Guiying, G. Bo and M. Jiming (2004), Research on fast response and high accuracy control of an airborne brushless DC motor, *Proc. 2004 IEEE Int. Conf. Robotics and Biomimetics*, Shenyang, China, pp. 807-810.

C. Kaddissi, J. P. Kenne and M. Saad (2006), Indirect adaptive control of an electro-hydraulic servo system based on nonlinear backstepping, *IEEE Int. Symposium Ind. Electron*, Montreal, Quebec, Canada, pp. 3147-3153.

V. V. Kokotovic, J. Grabowski, V. Amin and J. Lee (1999), Electro hydraulic power steering system, *Int. Congress & Exposition*, Detroit, Michigan, USA, pp. 1-4.

M. Krstic, I. Kanellakopoulos and P. Kokotovic (1995), Nonlinear and Adaptive Control Design, *Wiley Interscience*, New York, USA.

K. J. Lee, H. M. Kim and J. S. Kim (2004), Design of a chattering-free sliding mode controller with a friction compensator for motion control of a ball-screw system, *IMechE J. of Systems and Control Engineering*, 218, pp. 369-380.

H. E. Merritt (1967), Hydrostatic Control Systems, Wiley, New York, USA.

M. Perron, J. de Lafontaine and Y. Desjardins (2005), Sliding-mode control of a servomotor-pump in a position control application, *IEEE Conf. Electrical and Computer Eng*, Saskatoon, Canada, pp. 1287-1291.

J. J. Slotine and W. Li (1991), Applied Nonlinear Control, *Pearson Education*, New Jersey, USA.

S. Wang, R. Burton and S. Habibi (2005), Sliding mode controller and filter applied to a model of an electro-hydrostatic actuator system, *ASME Int. Mechanical Engineering Congress & Exposition*, Orlando, Florida, USA, pp. 1-10.
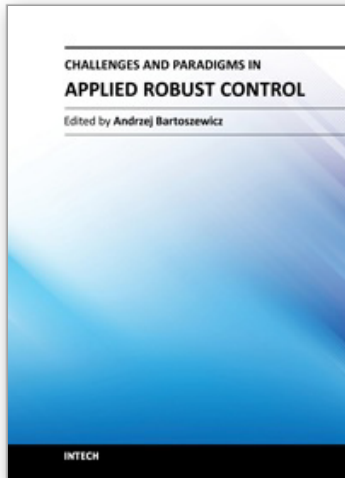
Challenges and Paradigms in Applied Robust Control

Edited by Prof. Andrzej Bartoszewicz

The main objective of this book is to present important challenges and paradigms in the field of applied robust control design and implementation. Book contains a broad range of well worked out, recent application studies which include but are not limited to H-infinity, sliding mode, robust PID and fault tolerant based control systems. The contributions enrich the current state of the art, and encourage new applications of robust control techniques in various engineering and non-engineering systems.

How to reference

In order to correctly reference this scholarly work, feel free to copy and paste the following:

INTECH
open science | open minds