

# Position Control of an Electro-hydraulic Servo System Based on Switching between Nonlinear and Linear Control

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**Abstract**—This paper proposes a position controller to handle three problems in the control of an electro-hydraulic servo system. The first two problems are about the nonlinearities and the uncertainties of the electro-hydraulic servo system, and the third one is the chattering problem caused by adopting sliding mode control. The proposed controller is called a Switch Controller, for its control output switches between the outputs of two controllers, a Nonlinear Controller and a conventional Linear Controller. The Nonlinear Controller is actually a nonlinear sliding mode controller and is focused to deal with nonlinearities and uncertainties. The Linear Controller is designed to improve the transient performance near the steady state. Experimental results show that the Switch Controller can improve the response performance of the electro-hydraulic servo system and effectively avoid chattering.

**Index Terms**—*Electro-hydraulic servo system, nonlinear control, sliding mode control, chattering avoidance.*

## I. INTRODUCTION

Electro-hydraulic servo systems are widely used in industrial applications. The nonlinearities of an electro-hydraulic servo system including fluid nonlinearity, asymmetrical mechanical characteristics, etc., cause the control performance varying with the moving direction and the position of the piston while using a traditional linear controller. The uncertainties of the hydraulic system including load, friction, etc., will also affect the control performance. These two aspects make a traditional linear controller be limited to fully exploit the dynamic capability of the hydraulic system.

For this reason, nonlinear controllers are adopted to improve the dynamic performance considering nonlinearities [1] or considering both nonlinearities and uncertainties [2]–[6]. Sliding mode control is a famous robust control method to deal with uncertainties and has already been adopted in the control of hydraulic servo systems [3]–[6], but it is also known for the chattering problem which may lead to instability of the system.

This paper proposes a Switch Controller which has two position controllers running concurrently, a Nonlinear Controller and a Linear Controller. If the system state is far to the steady state, the output of the Switch Controller is

that of the Nonlinear Controller. The Nonlinear Controller has a sliding mode nonlinear force tracking controller as its inner loop, and is focused to the deal with the nonlinearities and the uncertainties. Even we adopt a boundary layer for the Nonlinear Controller, chattering will still probably be excited when the system state is near the steady state. On this occasion, we switches the control output of the Switch Controller to be that of the Linear Controller. This switching strategy shows to be able to effectively avoid chattering.

The remaining of this paper is organized as follows. In section II, the simplified nonlinear model and the simplified linear model of the electro-hydraulic servo system are obtained. In section III, the Nonlinear Controller and the Linear Controller are designed based on the simplified nonlinear model and the simplified linear model respectively, and the Switch Controller is given subsequently. In section IV, experiments are carried out to verify the proposed Switch Controller. And section V comes the conclusion.

## II. MODEL OF THE ELECTRO-HYDRAULIC SERVO SYSTEM

### A. Simplified Nonlinear Model of the Electro-hydraulic Servo System

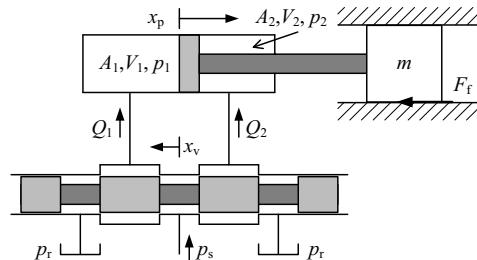


Fig. 1. Electro-hydraulic servo system controlled by a servo-valve.

The electro-hydraulic servo system as shown in Fig. 1 is a single-rod cylinder system that controlled by a servo-valve, driving the load with a mass of  $m$  moving forth and back.  $A_1$ ,  $V_1$  and  $p_1$  are the effective area, the volume and the pressure of the left chamber respectively; and  $A_2$ ,  $V_2$  and

$p_2$  are those of the right chamber.  $x_p$  is the position of the piston with its home position at the middle of the stroke.  $p_s$  is the supply pressure and  $p_r$  is the return pressure.  $x_v$  is the spoon position,  $Q_1$  is the flow into the left chamber, and  $Q_2$  is the flow into the right chamber.  $F_f$  is the friction parallel to the moving direction.

If the discharge coefficients of the valve orifices are equal, the pressure-flow equations of the servo valve is [7]

$$\begin{aligned} Q_1 &= \begin{cases} c_v x_v \sqrt{p_s - p_1} & \text{if } x_v \geq 0 \\ c_v x_v \sqrt{p_1 - p_r} & \text{if } x_v < 0 \end{cases}, \\ Q_2 &= \begin{cases} -c_v x_v \sqrt{p_2 - p_r} & \text{if } x_v \geq 0 \\ -c_v x_v \sqrt{p_s - p_2} & \text{if } x_v < 0 \end{cases}, \end{aligned} \quad (1)$$

where  $c_v$  is the discharge coefficient of the servo valve.

The dynamics of the servo valve can be modelled as a third-order linear system or a second-order linear system [7], [8]. If the response frequency of the servo valve is higher enough, the dynamics of the servo valve can be simplified as a proportional component, then (1) can be written as

$$\begin{aligned} Q_1 &= \begin{cases} k_q u \sqrt{p_s - p_1} & \text{if } u \geq 0 \\ k_q u \sqrt{p_1 - p_r} & \text{if } u < 0 \end{cases}, \\ Q_2 &= \begin{cases} -k_q u \sqrt{p_2 - p_r} & \text{if } u \geq 0 \\ -k_q u \sqrt{p_s - p_2} & \text{if } u < 0 \end{cases}, \end{aligned} \quad (2)$$

where,  $u \in [-1, 1]$  is the normalized control input of the servo valve, with  $|u| = 1$  corresponding to the nominal control input of the servo valve;  $k_q$  is the equivalent discharge coefficient that can be calculated based on the nominal pressure  $\Delta p_N$  and the nominal flow  $Q_N$  of the servo valve by the flowing equation

$$k_q = Q_N / \sqrt{\Delta p_N}. \quad (3)$$

If ignore the internal leakage and external leakage of the hydraulic system, the pressure dynamics in each chamber can be expressed as [7]

$$\begin{aligned} \dot{p}_1 &= \frac{\beta_e(p_1)}{V_1} (Q_1 - A_1 \dot{x}_p) \\ \dot{p}_2 &= \frac{\beta_e(p_2)}{V_2} (Q_2 + A_2 \dot{x}_p), \end{aligned} \quad (4)$$

where,  $V_1 = V_{10} + A_1 x_p$ ,  $V_2 = V_{20} - A_2 x_p$ ,  $V_{10}$  and  $V_{20}$  are the volumes of the left chamber and the right chamber respectively when  $x_p = 0$ ,  $\beta_e$  is the effective bulk modulus.

Considering (2), the pressure dynamics (4) in each chamber can be rewritten as

$$\begin{aligned} \dot{p}_1 &= \frac{\beta_e(p_1)}{V_1} [g(p_1) k_q u - A_1 \dot{x}_p] \\ \dot{p}_2 &= \frac{\beta_e(p_2)}{V_2} [-g(p_2) k_q u + A_2 \dot{x}_p], \end{aligned} \quad (5)$$

where,

$$\begin{aligned} g_1(p_1) &= \begin{cases} \sqrt{p_s - p_1} & \text{if } u \geq 0 \\ \sqrt{p_1 - p_r} & \text{if } u < 0 \end{cases} \\ g_2(p_2) &= \begin{cases} \sqrt{p_2 - p_r} & \text{if } u \geq 0 \\ \sqrt{p_s - p_2} & \text{if } u < 0 \end{cases}. \end{aligned} \quad (6)$$

The dynamics of the cylinder and the load is given by

$$m \ddot{x}_p = A_1 p_1 - A_2 p_2 - F_f, \quad (7)$$

where,  $F_f$  is modelled as the Stribeck friction curve[7]. The parameters of the Stribeck friction curve can be obtained through experiment. For a single-rod cylinder, however, the parameters depend on the moving direction, so two groups of parameters shall be identified.

In summary, (5) and (7) give the simplified nonlinear model of the electro-hydraulic servo system. The parameters of this model of the hydraulic servo system involved in this paper is given in Table I. Note that  $\sigma_v^+$ ,  $F_{c0}^+$ ,  $F_{s0}^+$  and  $c_s^+$  are for moving forward (moving to the right) and  $\sigma_v^-$ ,  $F_{c0}^-$ ,  $F_{s0}^-$  and  $c_s^-$  are for moving backward (moving to the left).

TABLE I  
PARAMETERS OF THE SIMPLIFIED NONLINEAR MODEL OF THE ELECTRO-HYDRAULIC SERVO SYSTEM

Parameters	Value	Parameters	Value
$A_1$	$1.9635 \times 10^{-3} \text{ m}^2$	$\sigma_v^+$	$1458.4 \text{ Ns/m}$
$A_2$	$1.2566 \times 10^{-3} \text{ m}^2$	$F_{c0}^+$	$13.8 \text{ N}$
$V_{10}$	$4.5872 \times 10^{-4} \text{ m}^3$	$F_{s0}^+$	$235.5 \text{ N}$
$V_{20}$	$2.7141 \times 10^{-4} \text{ m}^3$	$c_s^+$	$0.026$
$m$	$0 \sim 325 \text{ kg}$	$\sigma_v^-$	$1475.9 \text{ Ns/m}$
$p_s$	$7.0 \text{ MPa}$	$F_{c0}^-$	$13.8 \text{ N}$
$p_r$	$0.1 \text{ MPa}$	$F_{s0}^-$	$222.3 \text{ N}$
$k_q$	$3.7796 \times 10^{-7} \text{ m}^{7/2}/\text{kg}^{1/2}$	$c_s^-$	$0.035$
$\beta_e$	$\approx 900 \text{ MPa}$		

### B. Simplified Linear Model of the Electro-hydraulic Servo System

The linear model of the elector-hydraulic servo system can be obtained by linear approximation near a state point  $(u, x_p, p_1, p_2) = (u_0, x_{p0}, p_{10}, p_{20})$ . Then the transfer function from the control input  $u(t)$  to the position of the piston  $x_p(t)$  according to [7] is

$$G_h(s) = \frac{X_p(s)}{U(s)} = \frac{A_p K_Q}{m} \frac{1}{s^2 + 2\zeta_h \omega_h s + \omega_h^2} \frac{1}{s}, \quad (8)$$

where,  $A_p = A_1$ ,  $\zeta_h$  is the damping ratio,  $\omega_h$  is the natural frequency,  $K_Q$  is a gain with a number of parameters.

Base on experiment data, for linear approximation, set  $u_0 = 0$ ,  $x_{p0} = 0$ ,  $p_{10} \approx 3.0 \text{ MPa}$ ,  $p_{20} \approx 4.5 \text{ MPa}$ ,  $\sigma_v = (\sigma_v^+ + \sigma_v^-)/2$ , then the value of the parameters for the linear model can be obtained [7]. Take the dynamics of the servo valve as a first-order linear system, and according to its specifications the dynamics can be described as  $1/(0.0312s + 1)$ .

Then the transfer function from  $u(t)$  to  $x_p(t)$  for moving forward becomes

$$G_+(s) = \frac{0.4856}{5.4843e-8s^3 + 1.7968e-5s^2 + 3.2453e-3s + 1} \frac{1}{s}, \quad (9)$$

and that for moving backward becomes

$$G_-(s) = \frac{0.3883}{5.4843e-8s^3 + 1.7968e-5s^2 + 3.2453e-3s + 1} \frac{1}{s}. \quad (10)$$

Equations (9) and (10) give the simplified linear model of the electro-hydraulic servo system for the piston moving forward and backward respectively.

### III. CONTROLLER DESIGN FOR THE ELECTRO-HYDRAULIC SERVO SYSTEM

#### A. Nonlinear Controller Design

The nonlinear controller is a position controller with two control loops. The inner loop is a sliding mode robust controller for force tracking, transforming the nonlinear model into a linear form. This linear form is then used to design a position tracking controller for the outer loop.

For the inner loop force tracking controller, denote  $F_L$  as the driving force of the cylinder and it is given by

$$F_L = A_1 p_1 - A_2 p_2. \quad (11)$$

The goal of the force tracking controller is to have  $F_L$  approach the desired driving force  $F_{\text{des}}$  as closely as possible.

For simplicity, assume  $\beta_e(p_1) = \beta_e(p_2) = \beta_e$ , and differentiate both sides of (11) and substitute (5) in, the dynamics of  $F_L$  is obtained

$$\dot{F}_L = h_1(p_1, p_2, x_p, \dot{x}_p) + h_2(p_1, p_2, x_p)u, \quad (12)$$

where,

$$\begin{aligned} h_1 &= -\beta_e(A_1^2/V_1 + A_2^2/V_2)\dot{x}_p, \\ h_2 &= \beta_e k_q[A_1/V_1 g_1(p_1) + A_2/V_2 g_2(p_2)]. \end{aligned} \quad (13)$$

Set the estimated value of  $\beta_e$  be

$$\hat{\beta}_e = \sqrt{\beta_{e,\max}/\beta_{e,\min}}, \quad (14)$$

where  $\beta_{e,\min}$  and  $\beta_{e,\max}$  are respectively the minimum value and maximum value of  $\beta_e$ . The uncertainty bound of  $\beta_e$  can be represented by

$$B_{\beta_e} = \sqrt{\beta_{e,\max}/\beta_{e,\min}}. \quad (15)$$

Considering the uncertainty of  $\beta_e$  and  $k_q$ , the estimated value of  $h_1$  and  $h_2$  are

$$\begin{aligned} \hat{h}_1 &= -\hat{\beta}_e(A_1^2/V_1 + A_2^2/V_2)\dot{x}_p, \\ \hat{h}_2 &= \hat{\beta}_e \hat{k}_q(A_1 g_1/V_1 + A_2 g_2/V_2), \end{aligned} \quad (16)$$

where  $\hat{k}_q$  is the estimated value of  $k_q$ . The upper bound of  $h_1$  is derived as

$$\begin{aligned} U_{h_1} &= (\beta_{e,\max} - \beta_{e,\min})(A_1^2/V_1 + A_2^2/V_2)|\dot{x}_p| \\ &= \hat{\beta}_e(B_{\beta_e} - 1/B_{\beta_e})(A_1^2/V_1 + A_2^2/V_2)|\dot{x}_p|. \end{aligned} \quad (17)$$

The uncertainty bound of  $h_2$  can be represented by

$$B_{h_2} = B_{\beta_e} B_{k_q}, B_{h_2}^{-1} \leq \hat{h}_2 h_2^{-1} \leq B_{h_2}. \quad (18)$$

Suppose that  $F_{\text{des}}$  is differentiable, define the following sliding surface

$$S_F = F_L - F_{\text{des}} = A_1 p_1 - A_2 p_2 - F_{\text{des}}, \quad (19)$$

and differentiate  $S_F$  obtain

$$\begin{aligned} \dot{S}_F &= \dot{F}_L - \dot{F}_{\text{des}} \\ &= h_1(p_1, p_2, x_p, \dot{x}_p) + h_2(p_1, p_2, x_p)u - \dot{F}_{\text{des}}. \end{aligned} \quad (20)$$

The following control law satisfies the sliding condition [9]

$$u = \hat{h}_2^{-1}[v - Q \text{sign}(S_F)], \quad (21)$$

where,

$$\begin{aligned} v &= -[\hat{h}_1(p_1, p_2, x_p, \dot{x}_p) - \dot{F}_{\text{des}}] - K_F S_F, \\ Q &\geq B_{h_2} U_{h_1} + (B_{h_2} - 1)|v|, \end{aligned} \quad (22)$$

and  $K_F$  is a positive constant.

For reducing chattering, adopt the boundary method. Replace  $\text{sign}(S_F)$  in (21) by the following saturation function

$$\text{sat}(S_F) = \begin{cases} \text{sign}(S_F/\Phi_F) & |S_F/\Phi_F| > 1 \\ S_F/\Phi_F & |S_F/\Phi_F| \leq 1 \end{cases}, \quad (23)$$

where  $\Phi_F$  is the boundary width. The control law  $u$  becomes

$$u = \hat{h}_2^{-1}[v - Q \text{sat}(S_F)]. \quad (24)$$

For the outer loop position controller, set the desired driving force be [1]

$$F_{\text{des}} = \hat{m}\ddot{x}_{\text{pd}} - K_1(\dot{x}_p - \dot{x}_{\text{pd}}) - K_2(x_p - x_{\text{pd}}) + \hat{F}_\tau, \quad (25)$$

where,  $\hat{m}$  is the estimated value of  $m$ ,  $K_1$  and  $K_2$  are positive constants,  $x_{\text{pd}}$  is the expected position of the piston,  $\hat{F}_\tau$  is the estimated value of friction and other disturbing forces. From (7), (11) and (25) obtain the position error dynamics

$$\hat{m}\ddot{e}_p + K_1\dot{e}_p + K_2e_p = (F_L - F_{\text{des}}) - \delta F_{\text{dis}}, \quad (26)$$

where,

$$\delta F_{\text{dis}} = (F_\tau - \hat{F}_\tau) + (m - \hat{m})\ddot{x}_p, \quad (27)$$

and  $e_p = x_{\text{pd}} - x_p$  is the position error.

Define  $\omega_{\text{err}} = \sqrt{K_2/\hat{m}}$ ,  $2\omega_{\text{err}}\zeta_{\text{err}} = K_1/\hat{m}$ . Then (26) can be written in Laplace form

$$e_p(s) = \frac{1/\hat{m}}{s^2 + 2\omega_{\text{err}}\zeta_{\text{err}}s + \omega_{\text{err}}^2} \{[F_L(s) - F_{\text{des}}(s)] - \delta F_{\text{dis}}(s)\}. \quad (28)$$

So, the performance of the position error dynamics are determined by  $\omega_{\text{err}}$  and  $\zeta_{\text{err}}$ . We could set  $K_1$  and  $K_2$  by first choosing the appropriate  $\omega_{\text{err}}$  and  $\zeta_{\text{err}}$ .

In simulation and experiment we found that if  $\omega_{\text{err}}$  is not large enough, the smooth step response will have a large overshoot. To reduce this overshoot, add a pre-filter before

$x_{pd}$  (although it is hard to depress overshoot using this method). Using the following transfer function

$$x_{pdf}(s) = \frac{\lambda_{pref}}{s + \lambda_{pref}} x_{pd}(s), \quad (29)$$

where  $\lambda_{pref}$  is a positive constant. Correspondingly replace  $x_{pd}$  with  $x_{pdf}$  in (25),  $F_{des}$  becomes

$$F_{des} = \hat{m}\ddot{x}_{pdf} - K_1(\dot{x}_p - \dot{x}_{pdf}) - K_2(x_p - x_{pdf}) + \hat{F}_\tau. \quad (30)$$

In summary, (24) and (30) give the nonlinear position controller for the electro-hydraulic servo system.

### B. Linear Controller Design

For the stability of the closed-loop system, design the linear controller based on the transfer function (9) for moving forward, which has a larger open-loop gain relative to the transfer function (10). The efficiencies of  $s^4$  and  $s^3$  are very small, so (9) can be simplified as

$$G(s) = \frac{K_o}{\tau_o s + 1} \frac{1}{s} = \frac{0.4856}{0.0032453s + 1} \frac{1}{s}. \quad (31)$$

The feedback controller could be designed as

$$G_c(s) = \frac{K_c}{\tau_c s + 1}. \quad (32)$$

The closed-loop transfer function under the control of  $G_c(s)$  is

$$T(s) = \frac{K_c K_o}{\tau_c \tau_o s^3 + (\tau_c + \tau_o)s^2 + s + K_c K_o}. \quad (33)$$

The value of  $\tau_c \tau_o$  is so small that  $\tau_c \tau_o$  could be ignored, then  $T(s)$  could be written as a standard second-order system

$$T(s) = \frac{\omega_T^2}{s^2 + 2\zeta_T \omega_T s + \omega_T^2}, \quad (34)$$

where,  $\omega_T$  is the natural frequency, and  $\zeta_T$  is the damping ratio, and they are related with the parameters of the feedback controller by the following equations

$$\tau_c = \frac{1}{2\zeta_T \omega_T} - \tau_o, K_c = \frac{\omega_T}{2\zeta_T K_o}. \quad (35)$$

We could specify the controller parameters based on (35) to set the natural frequency and damping ratio.

### C. Switching Between the Nonlinear Controller and the Linear Controller (Design of the Switch Controller)

The Nonlinear Controller is actually a sliding mode controller. It has a critical drawback, the chattering problem. Although a boundary layer is used, it seems to be not enough to avoid chattering. So we have the control input switch between the Nonlinear Controller and the Linear Controller. If the system state is far to the steady state, the Nonlinear Controller takes action, if the system state is near the steady state, the Linear Controller takes action. Using  $|e_p|$  as a simple criterion to measure the distance between the system state and the steady state.

Denote the control output of the Nonlinear Controller as  $u_{NLC}$  and that of the Linear Controller as  $u_{LC}$ . The Nonlinear Controller and the Linear Controller are running concurrently, the control output of the Switch Controller is

$$u_{SWC} = \begin{cases} u_{NLC} & \text{if } |e_p| > c_{th}, \\ u_{LC} & \text{if } |e_p| \leq c_{th}, \end{cases} \quad (36)$$

where,  $c_{th}$  is a positive constant, it is the threshold for switching between the two controllers.

## IV. EXPERIMENT

To verify the proposed Switch Controller, we made an experiment system as shown in Fig. 2. The parameters of the system are shown in Table I. The control algorithm is implemented in a real-time control system. The sampling period of the control system is 996  $\mu$ sec. The test input (the desired position  $x_{pd}$ ) is a smooth step which is third-order differentiable.

For the Linear Controller, set  $\zeta_T = 1.0$ ; for the Nonlinear Controller, set  $\beta_e = 900$  MPa,  $B_{\beta_e} = 1.1$ ,  $\hat{k}_q = 3.779645e-7$ ,  $B_{k_q} = 1.05$ ,  $\Phi_F = 2000$ ,  $K_F = 2\pi \times 40$ ,  $\hat{m} = 225$ ,  $\zeta_{err} = 1.0$ ,  $\lambda_{pref} = 2\pi \times 6$ ,  $Q = B_{h_2} U_{h_1} + (B_{h_2} - 1)|v|$ .

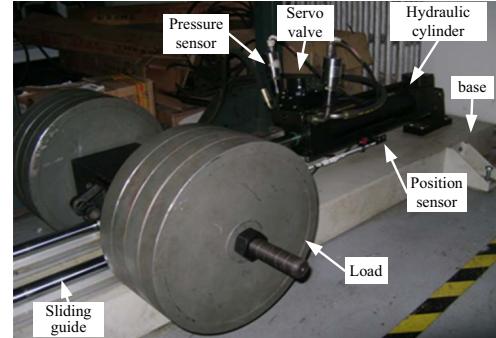


Fig. 2. An electro-hydraulic servo system for experiment.

### A. Smooth Step Response Using the Linear Controller and the Nonlinear Controller

Run a series of smooth steps to test the performance of the Linear Controller and the Nonlinear Controller with  $\omega_T$  and  $\omega_{err}$  varying from 6 Hz to 16 Hz. The smooth step responses for the two controllers are as shown in Fig. 3 and Fig. 4 respectively.

Fig. 3 is the smooth step responses using the Linear Controller. It shows that the response curves for moving forward and those for moving backward are not symmetrical, this mainly results from that the forward open-loop gain and the backward open-loop gain are not equal. It also shows that the rise time varies sensitively with  $\omega_T$ .

Fig. 4 is the smooth step responses using the Nonlinear Controller. It shows that the response curves for moving forward and those for moving backward are symmetrical. It

also shows that the rise time varies less sensitively with  $\omega_{\text{err}}$ . It should be noted that with  $\omega_{\text{err}}$  increasing the overshoot decreases. This is totally different from the Linear Controller. Consequently, we could set  $\omega_{\text{err}}$  be a large value to get a fast response and at the same time obtain a small overshoot.

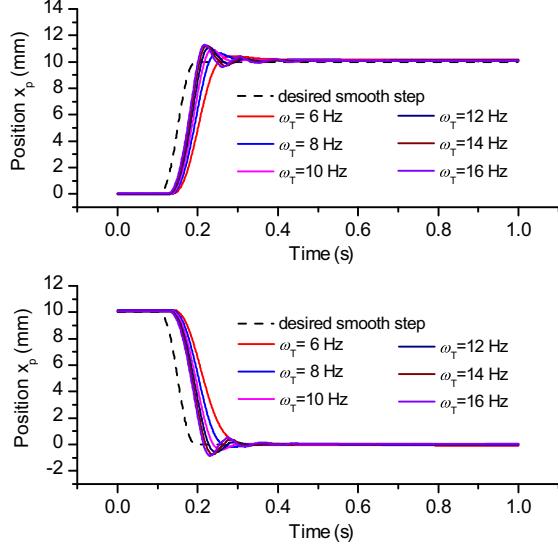


Fig. 3. Smooth step responses using the Linear Controller.

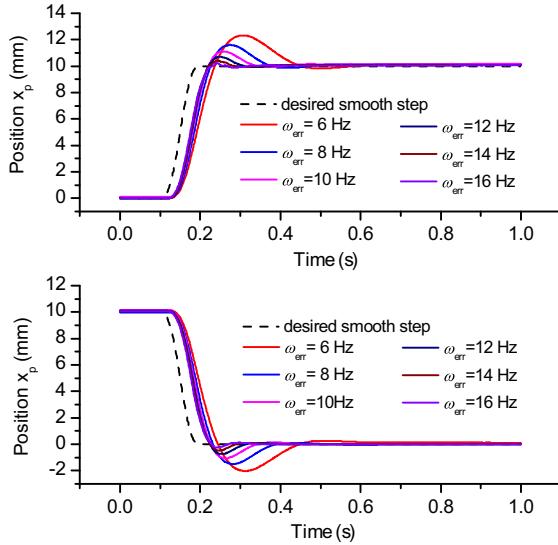


Fig. 4. Smooth step responses using the Nonlinear Controller.

### B. Smooth Step Response Using the Switch Controller

The smooth step responses while  $\omega_{\text{err}}$  varies from 6 Hz to 16 Hz are shown in Fig. 5. The transient performance of the Switch Controller and that of the Nonlinear Controller (shown in Fig. 4) are generally identical when the system state is far from the steady state. However, the switching

changes the transient performance while the system state is near the steady state, as shown in Fig. 6. Compared with the Nonlinear Controller, although the steady-state error becomes a little larger, the Switch Controller has a smaller overshoot and a shorter setting time.

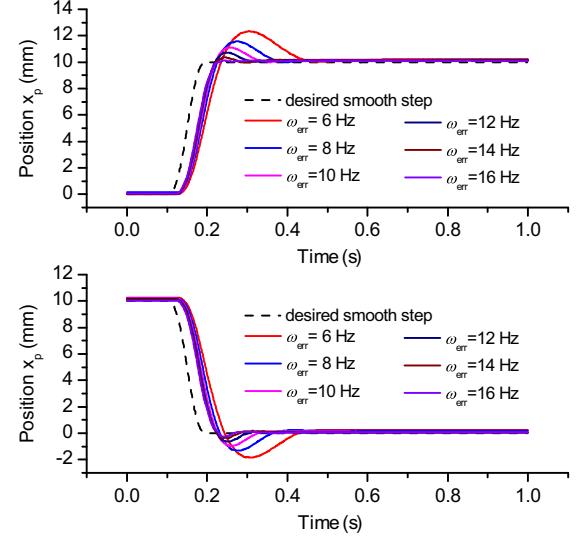


Fig. 5. Smooth step responses using the Switch Controller ( $\omega_T = 10\text{Hz}$ ,  $c_{\text{th}} = 0.1\text{mm}$ ).

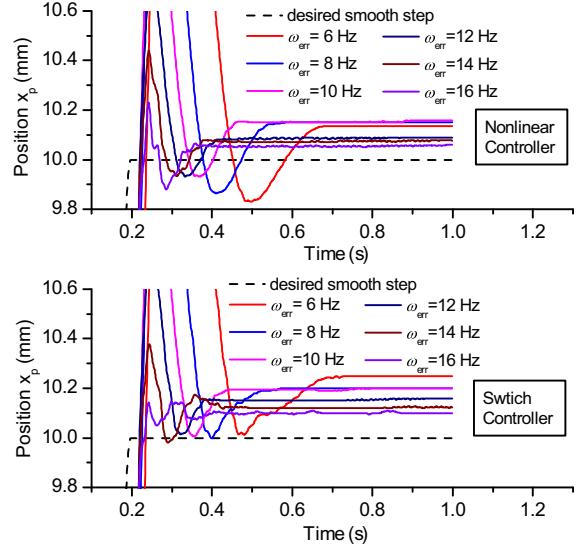


Fig. 6. Comparison of the transient performance near the steady state ( $\omega_T = 10\text{Hz}$ ,  $c_{\text{th}} = 0.1\text{mm}$ ).

### C. Chattering Avoidance of the Switch Controller

If  $\omega_{\text{err}}$  is a large value, even we adopt a saturation function as in (23), chattering will still probably be excited in experiment. As shown in Fig. 7, the control input  $u$  and the driving force  $F_L$  of the cylinder show being oscillating during the

steady state. This oscillation would probably cause instability of the system. The Switch Controller can effectively avoid chattering, as shown in Fig. 8.

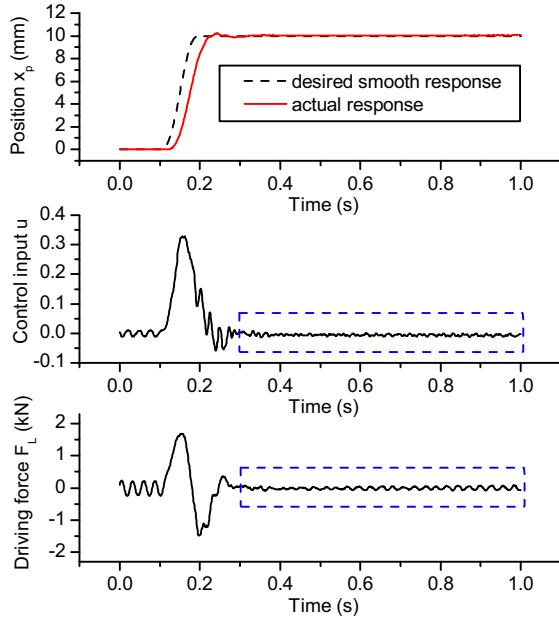


Fig. 7. Chattering happens during steady state while using the Nonlinear Controller ( $\omega_{err} = 16\text{Hz}$ ).

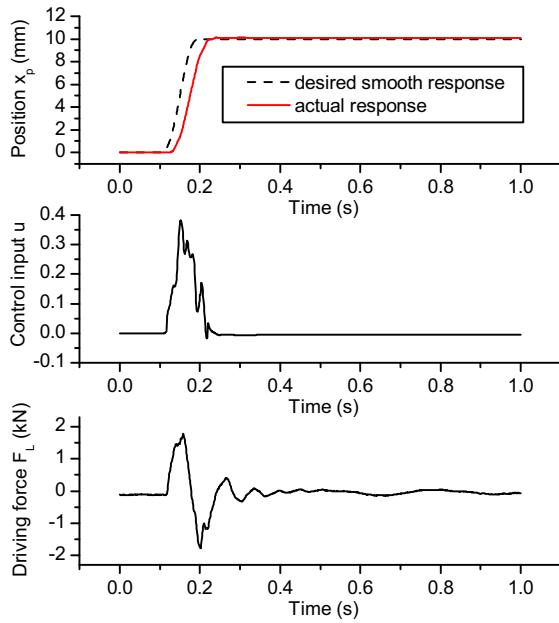


Fig. 8. Chattering is avoided by using the Switch Controller ( $\omega_{err} = 16\text{Hz}$ ,  $\omega_T = 10\text{Hz}$ ,  $c_{th} = 0.1\text{mm}$ ).

## V. CONCLUSION

In this paper we propose a Switch Controller, whose control output switches between the outputs of two controllers

that running concurrently, the Nonlinear Controller and the Linear Controller. The Nonlinear Controller is actually a sliding mode controller and is designed to handle the nonlinearities and the uncertainties of the electro-hydraulic servo system. We adopt a Linear Controller to improve the transient performance near the steady state to avoid chattering. We choose the absolute value of the position error  $|e_p|$  as a simple criterion to measure the distance between the system state and the steady state. If  $|e_p|$  is less than a threshold, the control input is switched to be the output of the Linear Controller. If  $|e_p|$  is not less than the threshold, the control input is switched to be the output of the Nonlinear Controller. Experimental tests were carried out to verify the proposed Switch Controller. The smooth step responses show that:

- 1) While the system state is far to the steady state, the response performance of the Switch Controller is identical to that of the Nonlinear Controller.
- 2) Compared with the Nonlinear Controller, although the Switch Controller has a little larger steady-state error, it has a lower overshoot and a shorter setting time. The most important is that the Switch Controller effectively avoid chattering.

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