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Performance Comparison between Sliding Mode Control with PID Sliding Surface and PID Controller for an Electro-hydraulic Positioning System

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***Abstract*— In this paper, the position tracking performance of an electro-hydraulic hydraulic servo (EHS) system using sliding mode control (SMC) with proportional-integral-derivative (PID) sliding surface is presented. The dynamics of the EHS system in modelling process are developed by consider its nonlinearities incorporating a friction model. Then, SMC with PID scheme is derived from the developed dynamics equation and stability of the control system is theoretically proven by Lyapunov theorem. Finally, simulation work is demonstrated and the result shows the proposed controller can achieve better tracking performance compared with conventional PID controller with good accuracy for any desired trajectory.**

***Keywords*— sliding mode control, electro-hydraulic servo system, position tracking control, PID sliding surface.**

I. Introduction

Electro-hydraulic servo (EHS) system has emerged a great interest in various engineering applications due to its advantages as an actuator in providing high forces with compact design [1]. The sophisticated design of EHS system with the versatility of electronic and hydraulic components offers a great performance and improvement for wide range of applications such as aircrafts [2], manufacturing machines [3], fatigue testing [4] and automotive application [5]. It is established that the EHS system can be more well-known and crucial nowadays.

It is well-known that an EHS system is typically a complicated system suffers from nonlinearities, uncertainties and disturbances. These inconveniences may lead to degradation of control performance in force, pressure or position tracking of EHS system. Nonlinear flow/pressure characteristics, fluid compressibility, actuator friction and internal/external leakages are identified as the major sources of nonlinearity exist in the EHS system [6]. Recently, authors in [7] discussed the mathematical model of the EHS system by including the effect of compressibility, external leakage in actuator and internal leakage in the servo valve. Most of proportional flow valves suffer from hard nonlinearities such as dead-zone due to asymmetric overlap in the spool valve design [8, 9]. Besides, the friction phenomena are also affect the tracking performance and often considered as a nonlinear model in developing the dynamics equation of EHS system [10].

Numerous control strategies have been proposed and reported in the literature to encounter the difficulties in controlling the EHS system. The increasing numbers of works dealing with EHS system over the past decades involved a linear control, nonlinear control and intelligent control approaches such as neural network (NN) [8], self-tuning Fuzzy PID [11, 12], model reference adaptive control (MRAC) [13, 14] and generalized predictive control (GPC) [15]. It is found that, the nonlinear control strategy has been an efficient tool and extensively used for controlling the nonlinear system. In recent years, sliding mode control (SMC) method is gaining research interest in controlling the EHS system [9, 16-18].

SMC is recognized as a one of the most potential approach in nonlinear control field and has been proved to the problem of maintaining the stability for controlling many classes of model that are subjected to parameter variations and external disturbances [19]. SMC has widely implemented in many engineering applications including pneumatic systems [20], active suspension systems [21] and active magnetic bearing systems [22]. Current study has been published by author in [23] utilizing the second-order SMC for electromechanical plant. The experimental application is an extended work presented in [24] using the proportional-integral-derivative sliding surface. In [25], the sliding surface is augmented with an integral action and has improved the tracking performance compared with the conventional design.

In this paper, the position tracking control performance of EHS system is evaluated using SMC with PID sliding surface. The nonlinear dynamics model consists of servo valve and hydraulic actuator incorporating with friction model is derived. Based on the EHS model, the SMC control law is developed and the stability is analysed in the sense of Lyapunov theorem. Afterward, position tracking control is performed and comparison with the conventional PID controller is presented.

II. ELECTRO-HYDRAULIC SERVO SYSTEM MODEL

Dynamics equation of EHS system consist of servo valve and hydraulic actuator as illustrated in Fig. 1. The hydraulic actuator motion is controlled by modulating the hydraulic oil flow from the cylinder chambers using a servo valve. The mass is attached with a spring and damper that generates the counter force against the actuator.

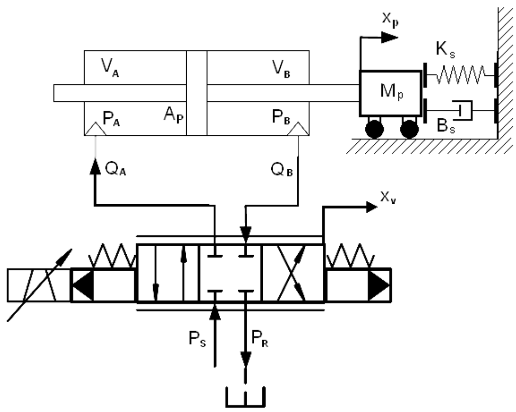

Fig. 1 Electro-hydraulic servo system configuration

A. Servo valve model

In the electronic design, the spool valve position in servo valve is driven by the torque motor. The electrical signal to drive the current to be used in coil as represented in (1).

$$V = \frac{dI}{dt} L_C + R_C I \tag{1}$$

The servo valve dynamics in (2) can be considered as second-order system relates the current drive from the torque motor and the spool valve position.

$$\frac{d^2 x_v}{dt^2} + 2\xi\omega_n \frac{dx_v}{dt} + \omega_n^2 = I\omega_n^2 \tag{2}$$

In mechanical design of servo valve, critical-centred is considered where the spool valve is unexposed from flow leakages and dead-zone problems for each port. The servo valve control the flows in each chambers in the actuator can models from the orifice equations. For the ideal orifice equation:

$$Q = Kx_v \sqrt{\Delta P_V} \tag{3}$$

The flow relations for each chambers are given in (4) and (5) by neglecting the internal leakages effects in servo valve [26].

$$Q_A = \begin{cases} K_A x_v \sqrt{P_S - P_A} & x_v \geq 0 \\ K_A x_v \sqrt{P_A - P_R} & x_v < 0 \end{cases} \tag{4}$$

$$Q_B = \begin{cases} -K_B x_v \sqrt{P_B - P_R} & x_v \geq 0 \\ -K_B x_v \sqrt{P_S - P_B} & x_v < 0 \end{cases} \tag{5}$$

where the coefficient gain is assumed to be $K = K_A = K_B$ for a symmetrical valve.

B. Hydraulic actuator model

The hydraulic actuator is modeled from the dynamics of volume of each chamber as follows:

$$V_A = V_{line} + A_p(x_s + x_p) \tag{6}$$

$$V_B = V_{line} + A_p(x_s - x_p) \tag{7}$$

where V_{line} is the volume of the pipeline and the zero position is located at the centre of the cylinder. From the flows and volume equations, pressure from each chamber can be determined by defining the relation between flow rate, bulk modulus, external leakages and volume rate.

$$P_A = \frac{\beta}{V_{line} + A_p(x_s + x_p)} \int \left(Q_A - q_{ab} - q_a - \frac{dV_A}{dt} \right) dt \tag{8}$$

$$P_B = \frac{\beta}{V_{line} + A_p(x_s - x_p)} \int \left(\frac{dV_B}{dt} - q_{ba} - q_b - Q_B \right) dt \tag{9}$$

The total force generated from the actuator can be expressed in (10) from the overall dynamics equation of spring, damper, moving mass and friction.

$$F_p = A_p(P_A - P_B) = M_p \frac{d^2 x_p}{dt^2} + B_s \frac{dx_p}{dt} + K_s x_p + F_f \tag{10}$$

C. Friction model

Lu-Gre friction model is preferred for representing the friction in EHS system since it captures most of the friction behaviour [7, 10]. The Lu-Gre friction model which combines the pre-sliding behaviour of the Dahl model with the steady state friction characteristics like Coulomb friction, viscous friction and the Stribeck effect can be stated as follows:

$$F_f = \sigma_0 z + \sigma_1 \frac{dz}{dt} + \alpha_2 \dot{x}_p \quad (11)$$

$\frac{dz}{dt}$ is the average of bristle deflection which can be represented as:

$$\frac{dz}{dt} = \dot{x}_p - \frac{\sigma_0 |\dot{x}_p|}{g(\dot{x}_p)} z \quad (12)$$

The Stribeck function in (13) is a function where expressing the steady-state friction behavior at a constant velocity.

$$g(\dot{x}_p) = \alpha_0 + \alpha_1 e^{-(\dot{x}_p / v_{sk})^2} \quad (13)$$

The supply pressure is generates from the pumps and drive to the servo valve. Generally, EHS system is equipped with a pressure regulator to regulate the maximum pressure operating in that system. The continuity dynamics between the pump and servo valve can be written as:

$$P_S = \frac{\beta}{V_t} \int (Q_{pump} - Q_L) dt \quad (14)$$

With this dynamics system model, some parameters may impossible to gather and the controller design is more complicated to design. Then, the simplified model can be used in controller design where the EHS system can be represent using third-order perturbed linear model including disturbances and uncertainties as in (15).

$$\dddot{x}_p(t) = -(A_n + \Delta A)\ddot{x}_p(t) - (B_n + \Delta B)\dot{x}_p(t) + (C_n + \Delta C)u(t) + d(t) \quad (15)$$

where d(t) consist of external load disturbance, nonlinear friction and leakage. An, Bn and Cn are the nominal system parameters. The bounded uncertainties ΔA, ΔB, ΔC are the uncertainties exist from the unmodeled dynamics.

$$\dddot{x}_p(t) = -A_n \ddot{x}_p(t) - B_n \dot{x}_p(t) + C_n u(t) + L \quad (16)$$

The lumped uncertainty is defined by:

$$L = -\Delta A \ddot{x}_p(t) - \Delta B \dot{x}_p(t) + \Delta C u(t) + d(t) \quad (17)$$

and the upper bound of the lumped uncertainty L_{max} is

$$L_{\max}(t) = \Delta A |\ddot{x}_p(t)| + \Delta B |\dot{x}_p(t)| + \Delta C |u(t)| + d(t) \quad (18)$$

III. Sliding Mode Control with PID Sliding Surface

SMC is a type of variable structure control (VSC) developed in the early of 60's in Russia [19]. It is established that the most crucial step in designing the SMC is the construction of the sliding surface which is expected to response the desired control specifications and performances. The states trajectories are forced to be reached and stayed on the sliding surface as depicted in Fig. 2.

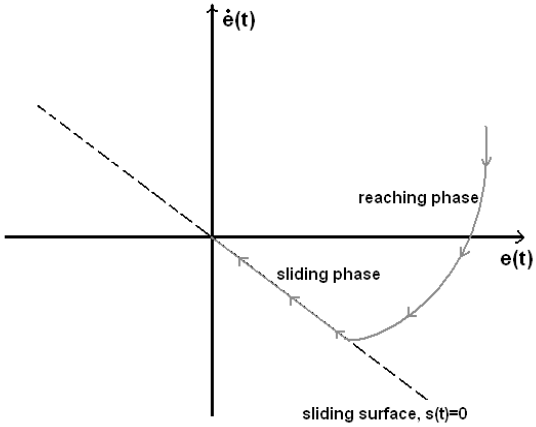

Fig. 2 Phase portrait of a sliding motion in sliding mode control

The PID sliding surface in the SMC design can be expressed in following equation where k_p, k_i and k_d are the PID parameters [23]. For a third-order system, the sliding surface can be defined as in (19).

$$s(t) = k_p e(t) + k_i \int_0^t e(\tau) d\tau + k_d \dot{e}(t) \quad (19)$$

The tracking error can be determined as a difference between the trajectory and actual position of the actuator.

$$e(t) = x_r(t) - x_p(t) \quad (20)$$

Since the linear model is a third-order model, the third derivative of the error is:

$$\dddot{e}(t) = \dddot{x}_r(t) - \dddot{x}_p(t) \quad (21)$$

The control signal in SMC design consists of equivalent control and switching control where the control action is corresponding with the sliding phase and reaching phase. The equivalent control is determined when s(t) = 0, while the switching control is described when s(t) ≠ 0. The common control structure of SMC can represented as:

$$u_{smc}(t) = u_{eq}(t) + u_{sw}(t) \quad (22)$$

In sliding mode phase, the tracking error will converge to equilibrium point and the sliding surface is supposed to be $s(t) = \dot{s}(t) = \ddot{s}(t) = 0$ where in this situation, the tracking error is trapped in the sliding surface. The equivalent control is obtained when $\ddot{s}(t) = 0$ and the second derivative of the sliding surface,

$$\ddot{s}(t) = k_p \ddot{e}(t) + k_i \dot{e}(t) + k_d \dddot{e}(t) \quad (23)$$

By substituting (21) into (23) and let $\ddot{s}(t) = 0$ without lumped uncertainty (L=0), the equivalent control can be defined as:

$$u_{eq}(t) = (k_d C_n)^{-1}(k_p \ddot{e}(t) + k_i \dot{e}(t) + k_d(\dddot{x}_r + A_n \ddot{x}_p + B_n \dot{x}_p)) \quad (24)$$

The switching control can be chosen as a sign function of sliding surface [23],

$$u_{sw}(t) = \lambda s(t) + k_s sign(\dot{s}(t)) \quad (25)$$

where $\lambda, k_s \in \Re^+$ and $sign(\dot{s}(t))$ denotes signum function as:

$$sign(\dot{s}(t)) = \begin{cases} 1, & \dot{s}(t) > 0 \\ 0, & \dot{s}(t) = 0 \\ -1, & \dot{s}(t) < 0 \end{cases} \tag{26}$$

To verify the stable convergence behavior of nonlinear controller, Lyapunov theorem as a well-known approach is used in stability analysis [10, 16, 18]. Lyapunov function can be chosen to prove the stability as:

$$V(t) = \frac{1}{2}s^2(t) + \frac{1}{2}\dot{s}^2(t) \tag{27}$$

with V(0) = 0 and V(t) > 0 for s(t) ≠ 0. To guarantee the trajectory move from reaching phase to sliding phase and ensure the stability, it is necessary to follow the reaching condition:

$$\dot{V}(t) < 0, \quad \text{for} \quad s(t) \neq 0, \dot{s}(t) \neq 0 \tag{28}$$

By substituting (21), (22) and (23) into (28),

$$\begin{aligned} \dot{V}(t) &= s(t)\dot{s}(t) + \dot{s}(t)\ddot{s}(t) \\ &= s(t)\dot{s}(t) - k_d C_n \lambda s(t)\dot{s}(t) - k_d C_n k_s |\dot{s}(t)| - k_d L(t))\dot{s}(t) \\ &\leq |\dot{s}(t)| [s(t) - k_d C_n \lambda s(t) - k_d C_n k_s - k_d L(t))] \\ &\leq |\dot{s}(t)| [|s(t)| - k_d C_n \lambda |s(t)| - k_d C_n k_s - k_d L(t))] \\ &\leq -|\dot{s}(t)| [|s(t)|(k_d C_n \lambda - 1) + k_d C_n k_s - k_d L_{\max}(t))] < 0 \end{aligned} \tag{29}$$

When the system is in the reaching phase where $s(t) \neq 0$, $\dot{s}(t) \neq 0$ the requirement for SMC parameters to guarantee the stability is $k_s > \frac{L_{\max}}{C_n}, \lambda > \frac{1}{k_d C_n}$ and $\dot{V}(t)$ is negative definite.

The discontinuous function in (25) caused a chattering in the control signal. To avoid the chattering effect that can be harm to the system, the hyperbolic tangent function with boundary layer φ can be proposed [23].

$$u_{sw}(t) = \lambda s(t) + k_s \tanh(\frac{\dot{s}}{\varphi}) \tag{31}$$

IV. Results and Discussion

In simulation study, the parameters used in nonlinear model of EHS system can be seen in Table 1. In the early design of SMC, the nominal values of EHS system can be identified using linearization of the EHS model or any identification techniques [8, 27]. The selection of sliding surface parameters based on the identified third-order model. λ=100 and k_s=10 is used in the numerical simulation where k_s is set to maximum voltage that can be operated by the system. To reduce the chattering effect, the φ parameter is set by sensitivity method until the controller achieves the adequate performance. The trade-off between the chattering and high-speed performance should made in tuning this parameter.

To simulate the friction model, the nonlinear Lu-Gre friction is implemented in the dynamics of the EHS system where parameters are stated in Table II.

TABLE I
Parameters of EHS System [7]

Hydraulic actuator and Servo valve parameters		
Description	***Symbol***	***Value***
Total mass	Mp	9 kg
Damping coefficient	Bs	2000 Ns/m
Spring stiffness	Ks	10 Nm
Total actuator displacement	Xs	0.1 m
Piston area	Ap	645×10^{-6} m^2
Servo valve damping ratio	ζ	0.48
Servo valve natural frequency	ω	534 rad/s
Servo valve coil resistance	Rc	100 Ω
Servo valve coil inductance	Lc	0.02 A
Servo valve gain	K	2.38×10^{-5} $m^{5/2}/kg^{1/2}$
Bulk modulus of hydraulic fluid	β	1.4×10^9 N/m^2
Pump pressure	P_S	2.1×10^7 Pa
Return pressure	P_R	0 Pa

TABLE II
Friction Parameters

Description	***Symbol***	***Value***
Stribeck velocity	v_{sk}	0.032 m/s
Coulomb friction	α_0	370 N
Stribeck friction	α_1	217 N
Viscous friction	α_2	2318 N/m/s
Bristles stiffness	σ_0	5.77×10^6 N/m
Bristles damping	σ_1	2.28×10^4 N/m/s

P, PD and PID controllers are used for performance comparison in the position tracking control. The parameters of each controller in Table III are taken from [7] which have been optimized by genetic algorithm.

TABLE III
P, PD and PID Controller Parameters

Controller	**Value of Parameter**		
	Kp	***Ki***	***Kd***
P	59.7117	0	0
PD	206.1246	0	1.1490
PID	221.0415	3.5260	1.2573

The results of position tracking performance are present concurrently for the SMC, P, PD and PID controller in Figure 3 and the control effort produced by SMC is shown in Figure 4. From the step input trajectory of x_r=0.05m, the results obtained with SMC controller is better compared to P, PD and PID controllers where the proposed controller produced more fast responses to reach the desired trajectory. In sliding surface design, the parameters are set to λ=1, k_s=10 and φ=15. The phase plot of SMC as can be seen in Figure 5 shows that the system reach the sliding phase and stay on that surface until meet the equilibrium point where the error and derivative of error approaching to zero. It is also can be seen that no chattering occur in sliding surface and its control effort during the position tracking control.

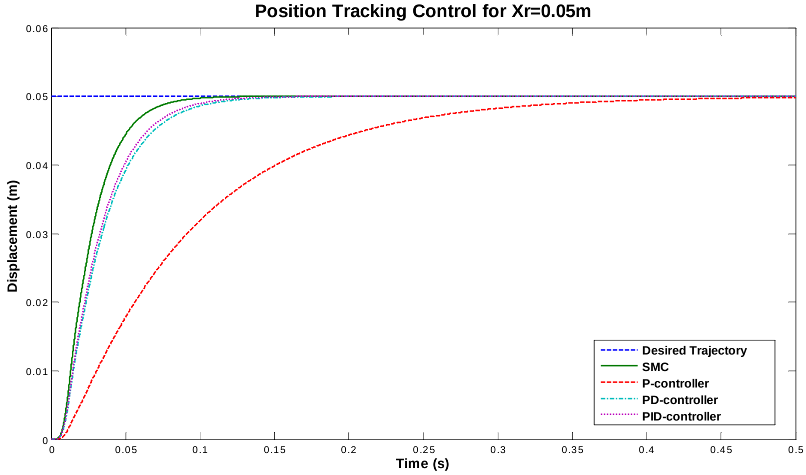

Fig. 3 Tracking performance of different controller for position control

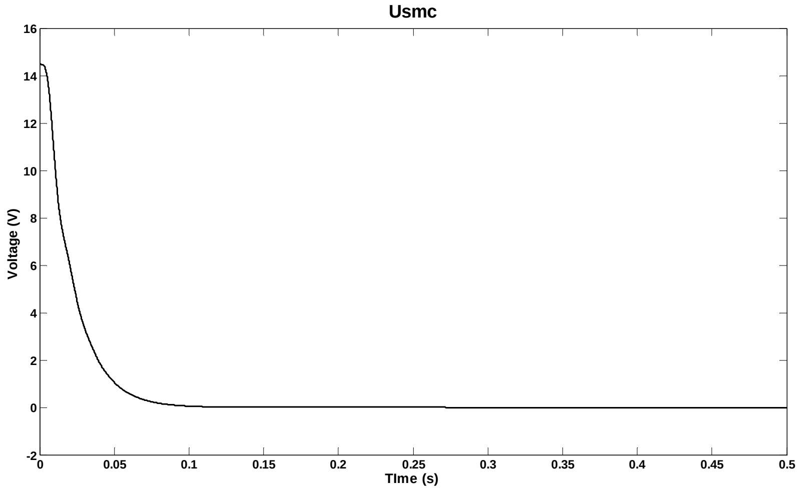

Fig. 4 Control effort of SMC with PID sliding surface

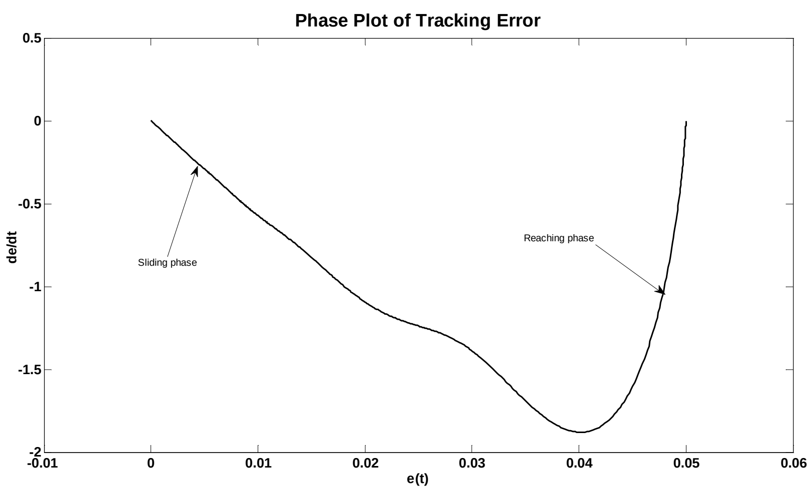

Fig. 5 The phase plot of SMC

TABLE IV
ERROR ANALYSIS

Control method	*SMC*	*P-controller*	*PD-controller*	*PID-controller*
Steady state error	$1.8917 x 10^{-5}$	$1.6640 x 10^{-4}$	$9.9614 x 10^{-11}$	$2.666 x 10^{-5}$
Mean-square-error	$8.2324 x 10^{-5}$	$2.7210 x 10^{-4}$	$1.1171 x 10^{-4}$	$1.0709 x 10^{-4}$

Based on the error analyses, control effort and observation on the tracking performance, the SMC provides more convinient and better performance in position tracking control and ensured that the control system in under stable condition. Is it found that equivalent control u_{eq} can be neglected in this simulation study. The assumption where the position tracking is not one of reaching sliding surface, but one of near or already on sliding surface can be made. In applications where stability in not an issue, the term u_{eq} often has a minimal effect in system response, so the equivalent control can be omitted [20]. Consequently, a simple robust control method without much control effort can be made with this proposed design.

V. CONCLUSIONS

In this study, the performance of SMC, P, PD and PID controllers are evaluated for position tracking control. The Lyapunov approach is used in theoretical analysis in developing the SMC with PID scheme and to ensure that the system is under stable condition. The numerical simulation study shows that the proposed controller provides better performance in tracking accuracy and time response. The control effort produced from SMC without the chattering effect also is practical to be used in real application. In conclusion, a simple control method without much control effort and better performance can be made with the SMC design based on PID sliding surface instead of using conventional PID controller.

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