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A Modified Sliding Mode Control for Accurate Position Control of a Hydraulic Cylinder

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ABSTRACT

The electrohydraulic servo systems (EHSS) are used extensively in industry such as flight, ships engineering, machines of injection molding, robotics, and steel and aluminum factory's equipment. It has the advantages of high power to weight ratio, linear movement, and fast response. In order to overcome the problem of nonlinearity of the controlled system, a sliding mode controller with error (ESMC) is proposed in such a way that the error signal modifies the ordinary switching mode control action. Pulse width modulation (PWM) technique is used to convert the control action into a digital signal capable of driving the solenoid of an electrohydraulic servo valve to control the oil flow entering the hydraulic cylinder so that its position could be controlled. A dynamic model of an electrohydraulic control system is simulated using MATLAB/Simulink software where the control parameters are optimized for experimental testing of PID and sliding mode controllers. The two controllers are then fine-tuned experimentally and their performance is compared under square and sinusoidal reference inputs. The experiments showed that the proposed ESMC gives less steady state error, less overshoot, and less settling time compared to both PID and ordinary switching sliding mode controllers.

Keywords: Position control, pulse width modulation, PWM, hydraulic cylinder, PID control, SMC.

1. INTRODUCTION

The electro hydraulic servo systems (EHSS) are widely used in industry and machinery applications due to their high performance in position and force control. They have high power to weight ratio, good dynamic performance and the ability to tolerate abrupt and aggressive loadings[1]. However, hydraulic servo systems have nonlinear behavior which has the characteristics of parameter accuracy structured uncertainty. The control of these systems has been a serious challenge to the control community[2]. There are numerous control design strategies based on classical control, advanced control and intelligent control systems. The control of hydraulic systems could be classified as position, velocity or force control. Essa et al., [3], used different control methods to control the position of a hydraulic press machine emulator as an actual industrial application. It has been shown that the algorithm of particle swarm optimization, PSO, control technique is more efficient than other used control strategies in terms of settling time, overshoot and transient response for trajectory position control of a hydraulic cylinder. Several present controllers achieve moderate bandwidth with fixed gain by oversizing the cylinder diameter. Whilst large diameter cylinder increases the effective stiffness of the fluid column in the cylinder, it also requires more costly system components and higher flow rates to move with the required speed. G. Sohl et al., [4] simulated and implemented a nonlinear tracking control law for a hydraulic servo system based on the analysis of the nonlinear system equations according to Lyapunov criteria. Good stable trajectory of force tracking was achieved. They extended this control law to force and position control of a hydraulic cylinder and excellent performance was achieved experimentally. Ayman A. Aly [5] presented an adaptive control method based on comparing the performance of the Model Reference (MR) response with the nonlinear model response to feed an adaptation signal to the PID control system to eliminate the error in between. It was found that the proposed MR-PID control strategy provided consistent performance in terms of rise time and settling time regardless of the nonlinearities. Shuzhong Zhang et al., [6] used a conventional (PID) controller plus a feed forward control (FFC) to control the position hydraulic cylinder. The simulated results showed 20-87% reduction in the root mean square tracking error. Essa et al., [7], showed that a hybrid MPC- PID controller gives good position regulation.

The problem of chattering is studied by many researchers, [8-11]. Two ways are proposed to reduce it. The first way

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uses the boundary layer control and the second way uses the dynamic sliding mode control DSMC. The dynamic sliding mode control based on dynamic switching function is proposed for a hydraulic system. H. Yanada et al., [12], used a SMC combined with observer to reduce the problem of chattering in hydraulic servo systems. O. Cerman et al., [13] proposed a new method for the design of a fuzzy SMC for nonlinear system with unknown dynamics. The prime conception of the proposed method depends on the proposal of the fuzzy self-tuning technique for adaptation of the sliding mode parameters and switching gains.

The sliding mode control, SMC, is widely spread technique due to its high robustness and easy implementation. It can overcome system uncertainties due to switching control or variable structure control.

The present work uses SMC modified by error term to control the position of a double acting hydraulic cylinder.

2. SIMULATION MODEL OF A HYDRAULIC SYSTEM

Fig. 1 shows a schematic of the hydraulic cylinder with the electrically driven 4/3 directional control valve. All notations will be shown in their respective positions.

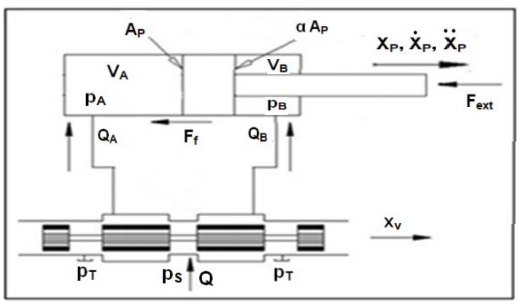


Figure. 1: Schematic of the servo valve connected to a hydraulic cylinder

2.1. Servo-Valve Model

The dynamic behavior of the valve is determined by the following second order differential equation, [14]:

$$\frac{d^2x_v(t)}{dt} + 2\xi_v\omega_v\frac{dx_v(t)}{dt} + \omega_v^2\,x_v(t) = \omega_v^2\,k_v ut(t) \qquad (1)$$

The flow through the valve is treated as flow through sharp edge orifice.

$$Q = c_v\pi d_v\,x_v\sqrt{\frac{2\sqrt{|p_u - p_d|}}{\rho}} \qquad (2)$$

p_u may take the value of the supply pressure or chamber pressure according to the flow direction. Also, p_d may take the value of the chamber pressure or atmospheric pressure according to the flow direction.

2.2. Model of a Hydraulic Cylinder.

2.2.1. Continuity equation

The pressures inside the left and right chambers are determined from the continuity equation ,[15]:

$$\dot{E}_A = \frac{\dot{E}}{V_A}\left(Q_A - A_P\dot{X}_p + C_{Li}(p_A - p_B)\right) \qquad (3)$$

$$\dot{E}_B = \frac{\dot{E}}{V_B}\left(Q_B + \alpha A_P\dot{X}_p + C_{Li}(p_B - p_A)\right) \qquad (4)$$

2.2.2. Equation of motion of the piston.

Applying Newton's second law to the all piston forces, [14]:

$$m_t\ddot{X}_p + F_f(\dot{X}_p) = A_P(P_A - \alpha P_B) - F_{ext} \qquad (5)$$

$$m_t = m_p + m_{A.fl} + m_{B.fl} \qquad (6)$$

Where:

$m_{A.fl}$ Mass of the fluid in the cylinder chambers

$m_{B.fl}$ Mass of the fluid in the pipes.

However, the mass of fluid in the pipes can be ignored compared with the piston mass [14].

2.2.3. Friction model

Friction model is based on the work of M. Jelali et al., [14] whose used Stribeck-curve, shown in Fig. 2.

$$F_f(\dot{X}_p) = F_v(\dot{X}_p) + F_c(\dot{X}_p) + F_s(\dot{X}_p) \qquad (7)$$

$$F_f(\dot{X}_p) = \sigma\dot{X}_p + sign(\dot{X}_p)[F_{co} + F_{so}\,e^{\left(-\frac{|\dot{X}_p|}{C_s}\right)}] \qquad (8)$$

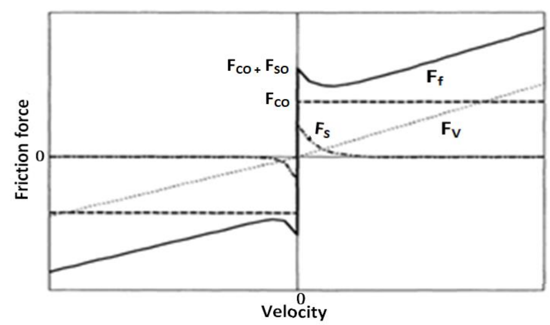


Figure. 2: The force of friction depends on the velocity (Stribeck-curve)

2.3. Controller Design

The controller is a part of the feedback control system, shown in Fig. 3, which controls the position of the valve spool so that the position of the piston cylinder is precisely tracked according to the input reference.

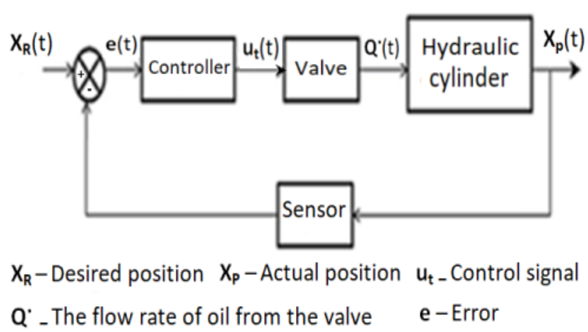


Figure. 3: The block diagram of a hydraulic system with feedback control

2.3.1. Sliding mode controller

SMC controller has been introduced in the early 60's whose fundamental concept is extracted from variable structure control developers [16], The most pivotal stage in the establishment of SMC control is the structure of sliding surface which is anticipated to be responsive to the desired control criterion. The control signal that reaches to the sliding surface is expected to be staying on the surface and slides to the origin point which is the desired position as depicted in Fig. 4.

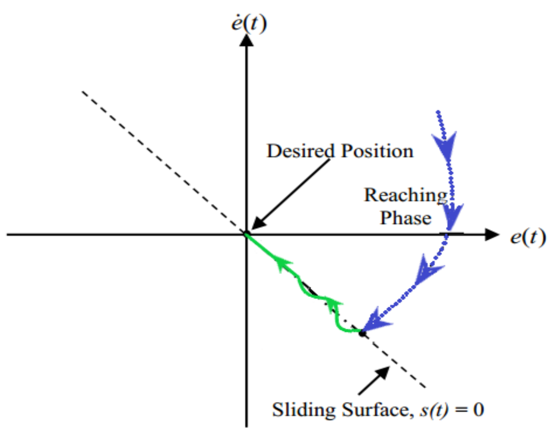


Figure. 4: The structure in the sliding mode control design

The sliding surface for the SMC is determined by the following equations:

$$\mathrm{s(t)} = W_S * \mathrm{e(t)} + \dot{\mathrm{e}}\mathrm{(t)} \qquad (9)$$

$$\mathrm{e(t)} = \mathrm{X}_{p,ref}\mathrm{(t)} - \mathrm{X}_p\mathrm{(t)} \qquad (10)$$

$$\dot{\mathrm{e}}\mathrm{(t)} = \dot{X}_{p,ref}\mathrm{(t)} - \dot{\mathrm{X}}_p\mathrm{(t)} \qquad (11)$$

Where X_p is the piston displacement, $\mathrm{X}_{p,ref}$ is the input reference position and $\mathrm{W_s}$ is a constant depends on the system natural frequency, ω_v and damping ratio,ξ. The dots on the parameters indicate their derivatives.

The Switching Sliding Mode Control (SSMC) can be obtained by applying the sign function to the sliding surface, [17],[18] as:

$$ut(t) = K_{us}\, sign(s) \qquad (12)$$

Where K_{us} is a constant with positive value and *sign*(s) represents the signum function as follows:

$$sign\big(s(t)\big)\begin{cases} 1 & ; s(t) > 0 \\ 0 & ; s(t) = 0 \\ -1 & ; s(t) < 0 \end{cases} \qquad (13)$$

2.3.2. Sliding mode control with error.

The Sliding Mode Control with Error (ESMC) is a proposed controller which takes the error effect directly on the control signal as follows:

$$ut(t) = K_{ue}\ e(t)\ sign(s) \qquad (14)$$

2.4. Simulation of the Hydraulic Model

The model equations have been simulated with MATLAB/Simulink to study in detail the dynamic behavior of the control system. Furthermore, the controller's parameters are tuned before running the experiment. The simulation is composed of four modules as shown in Fig. 5. The first module is the main program module. The second module is the controller module, which simulates three controllers; PID, SSMC, and ESMC. The third module is responsible to create the desired reference signal which takes three input forms; square, step and sinusoidal signals. The fourth module is responsible for recording data.

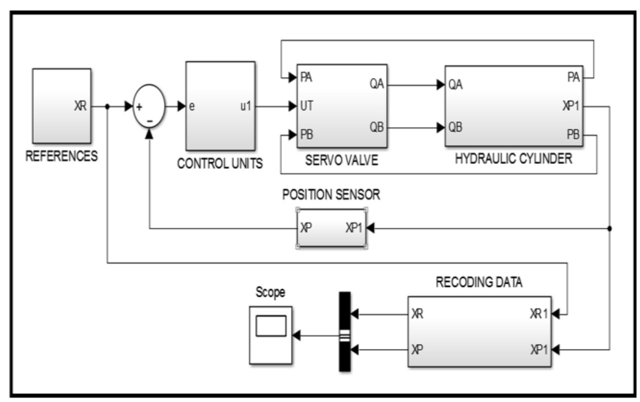


Figure. 5: MATLAB/Simulink program for control units

3. EXPERIMENTAL SETUP

Fig. 6 shows a schematic diagram of the experimental test rig while Fig.7 shows a photograph for the real rig. It consists of a double acting cylinder of 38 mm bore and 200 mm stroke controlled by a proportional 4/3 directional control valve operates by two solenoids The working fluid is a hydraulic oil whose pressure is controlled by a pressure control valve. The cylinder rod displacement is measured by a linear position sensor with linearity better than 0.5%. A proportional 4/3 directional control valve with two solenoids is used to control the cylinder motion. The solenoids operate by a PWM electric signal according to the controller command. The control circuit is shown in Fig. 8. A frequency clock operates between frequencies 50- 500 Hz, which drives the counter of an 8-bit binary plus the digital to analogue converter, which adds the 8-binary levels to produce a saw-tooth signal. A comparator is comparing the control action with the voltage levels of the saw tooth signal to provide the final variable duty signal. Next, the power circuit is designed to provide the solenoids of the directional valve with the desired power for their operations, which is mainly constructed of three-stage transistors. An analysis of the values of various stages of the PWM method is shown in Fig. 9. The supply pressure is set by using a relief valve and pressure regulator at 25 bar gauge pressure. The LabVIEW software program is utilized to control the directional valve by switching the input and output signals through the interface data acquisition card (NI-USB6009). The operating frequency of the valve is selected at 100 Hz to be inside the permissible bandwidth to ensure the best operation of the valve. A LabVIEW software is used to acquire data of the cylinder piston position, then makes the necessary calculation for the controller and finally provides the PWM circuit with the necessary voltage value to drive the valve solenoids to control the oil mass flow rates coming into and out of the cylinder in order to control the piston position as required.

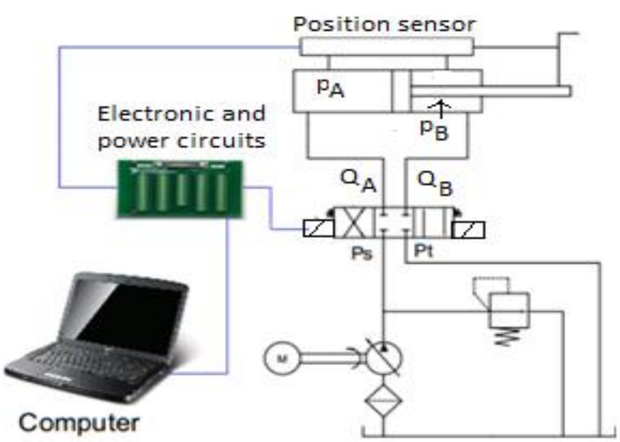
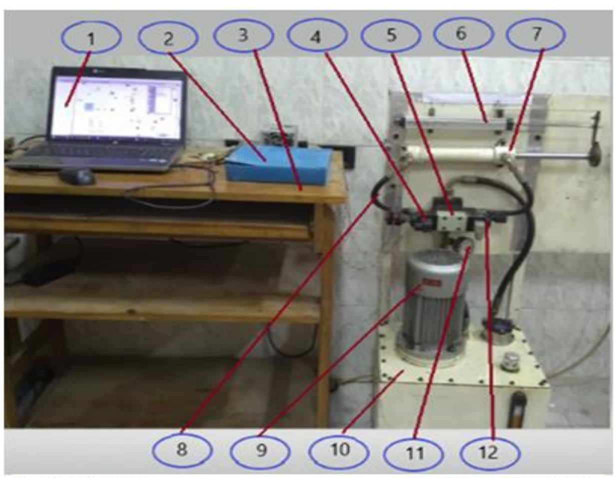


Figure. 6: Schematic diagram of Electrohydraulic system



1- Computer	2- Electronic circuit
3- Desk	4- Solenoid 1
5- Proportional	6- Position sensor
7- Hydraulic cylinder	8- Pipes
9- Motor connected with pump	10- Tank
11- Relief valve	12- Solenoid 2

Figure. 7: Photograph of the test Rig

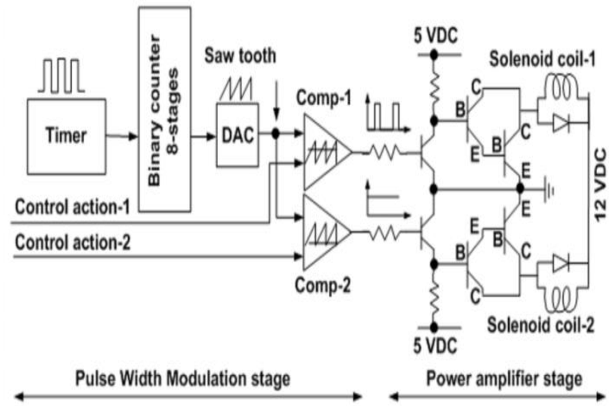


Figure. 8: The power amplifier circuit of PWM

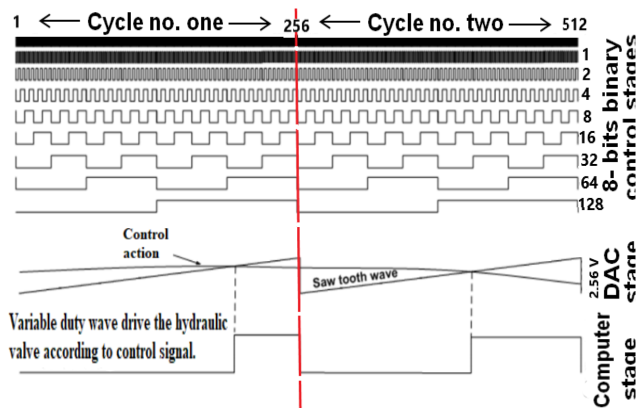


Figure. 9: Analysis of the Pulse Width Modulation

4. RESULTS AND DISCUSSION

The simulation model is tested against the experiment for open-loop control of cylinder position as a function of inlet oil flow rate which is controlled by the valve duty. The test is carried out for duties of 100%, 80%, and 60%. Fig. 10 shows one case of 80% duty where the model results agree well with the experiment.

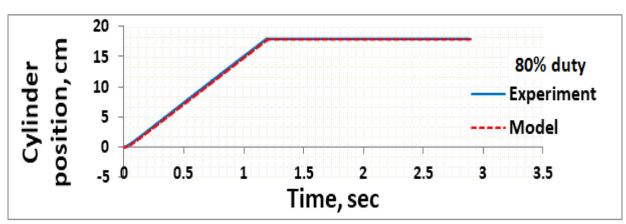


Figure. 10: Open-loop cylinder position at 80% valve duty for simulation model and experimental systems

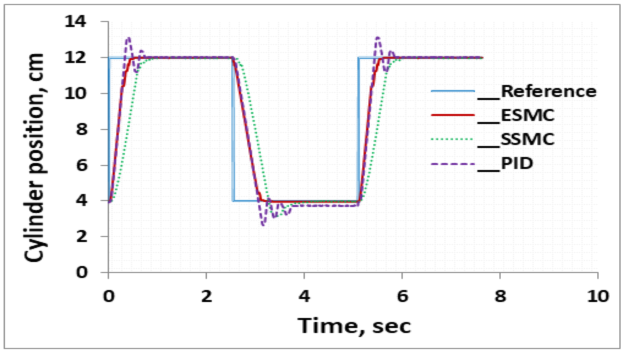


Figure. 11: Experimental Position for square wave reference input with ESMC, SSMC PID controller

The model is then used to optimize a PID controller parameters using Ziegler Nichols method. The obtained parameters are then used in the experiment for further fine tuning of the PID controller. The optimized parameters are Kp=48, Ki=5.12 and Kd=0.04875. Also, the parameters W_S & K_{us} for SSMC and W_{Se} & K_{ue} of ESMC have been optimized first using the simulation model followed by experimental fine tuning based on minimization of mean absolute error, MAE, and first overshoot. Using the model results as a tool for preliminary controller tuning saves a lot of experimental efforts where the experiment always starts with optimized controller parameters. An experimental comparison between the PID, SSMC, and ESMC controllers are shown in Fig. 11 for a square wave input using the optimized parameters. A summary of the time specifications is shown in Table 1. It can be seen that the PID controller and ESMC attain the same rise time while the SSMC is much slower. The PID has the highest overshoot while the ESMC has the lowest MAE and steady state error. Furthermore, the PID suffers from steady state error in the backward stroke.

Table 1: Time response specifications for ESMC, SSMC and PID controllers

		PID	SSMC	ESMC
MAE (cm)		0.8695	1.3361	0.7021
For Forward	Rise time (sec)	0.34	1.00	0.53
	Settling time (sec)	0.74	1.00	0.53
	Overshoot %	14	0.0	0.0
	ess (cm)	0.04	0.004	0.0
For Backward	Rise time (sec)	0.53	0.71	0.57
	Settling time (sec)	1.23	1.21	0.57
	Overshoot %	17	10.5	0.5
	ess (cm)	0.28	0.04	0.012

Figs. 12-14 show the experimental time response of the hydraulic cylinder when subjected to sinusoidal input at 0.2, 0.3 and 0.4 Hz. It is evident that the ESMC has the best tracking capability amongst the three controllers. Also, it has the minimum overshoot. The MAE, however, increases as the input frequency increases.

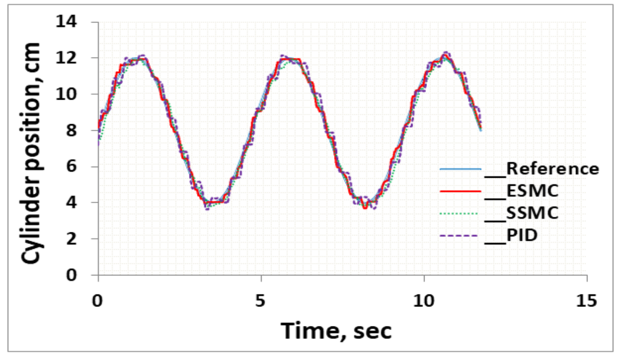


Figure. 12: Experimental sinusoidal position control at 0.2 Hz

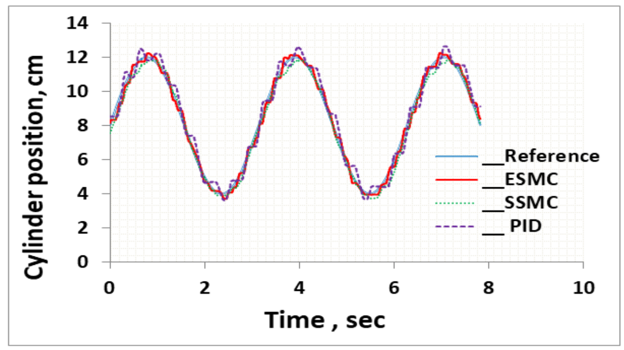


Figure 13: Experimental sinusoidal position control at 0.3 Hz

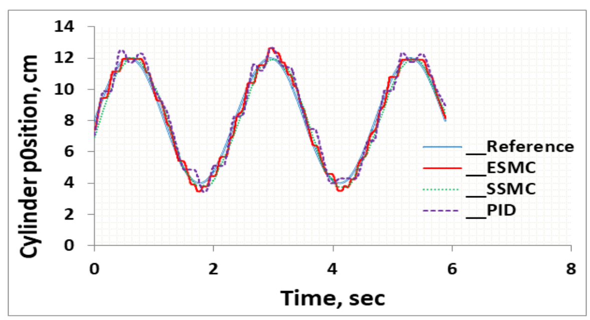
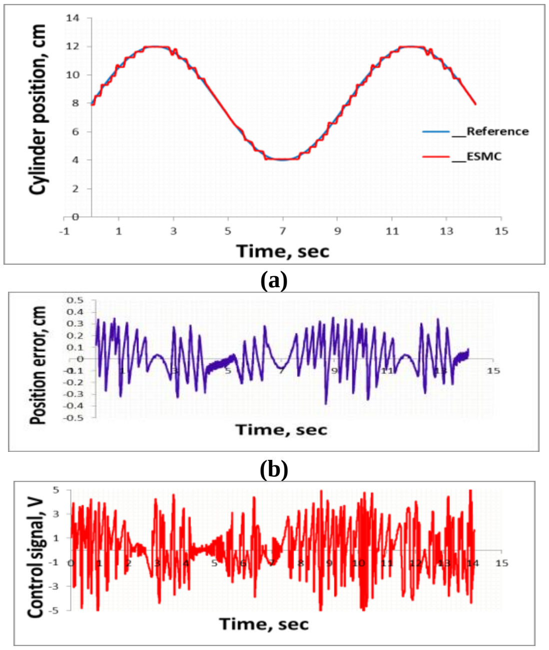


Figure. 14: Experimental sinusoidal position control at 0.4 Hz

Table 2: Summary of the MAE for the PID controller as compared to the SSMC and ESMC controller.

Table 2. MAE			
Input frequency, Hz	MAE, cm		
	PID	SSMC	ESMC
0.1	0.32	0.24	0.11
0.2	0.32	0.24	0.17
0.3	0.37	0.24	0.19
0.4	0.47	0.39	0.3

The detailed time response of the cylinder position control system using ESMC is shown in Fig. 15 for a sinusoidal input of 0.1 Hz. Good position tracking is shown in (a) even when the cylinder piston changes its direction. The instantaneous error is shown in (b). The corresponding control signal responds to the error signal as shown in (c). The maximum error does not exceed 4.5% from the total piston stroke.



(a) (b) (c)

Figure. 15: Experimental Position control for sinusoidal reference input of 0.1 Hz with ESMC controller; (a) Time response (b) Error, (c) control signal

5. CONCLUSION

A modified sliding mode control scheme is experimentally applied to a hydraulic cylinder position control and compared to the well-known PID control. The concluded points can be summarized as follows:

1- ESMC has a superior position tracking over the SSMC and PID control schemes.
2- Maximum absolute error in position does not exceed 4.5% from the total piston stroke for sinusoidal reference input of 0.1 Hz.
3- The traditional PID controller responds differently in the forward and return strokes. The dry friction seems to be responsible of the unexpected behavior of such controllers.

NOMENCLATURE

Symbol	Description
A_P	Piston cross sectional area, m^2
C_{Li}	Internal leakage coefficient
C_v	Oil flow coefficient = 0.65
$\dot{E}$	Bulk modulus of oil
e(t)	Error
F_c	Coulomb friction, N
F_{co}	Coulomb friction parameter
F_f	Friction force, N
F_{ext}	External force, N
F_S	Static friction force, N
F_{so}	Static friction parameter
F_v	Viscous friction force, N
K_p	Proportional gain
K_d	Derivative gain
K_i	Integral gain
k_v	Servo valve spool position gain
K_{ue}	Constants for ESMC
K_{us}	Constants for SSMC
m_P	Piston mass, kg
m_t	Total mass, kg
p_A	Left chamber pressure, Pa
p_B	Right chamber pressure, Pa
p_d	Downstream pressure, Pa
p_u	Upstream pressure, Pa
Q	Oil volume low rate, m^3/s
Q_A	Left chamber oil volume flow rate, m^3/s
Q_B	Right chamber oil volume flow rate, m^3/s
Q_{Li}	Internal flow leakage, m^3/s
V_A	Chamber A volume, m^3
V_B	Chamber B volume, m^3
u_t	Control signal
W_S	Constant for SSMC
W_{Se}	Constant for ESMC
d_v	The diameter of the valve orifice, 0.005m
X_p	Cylinder rod displacement, m
$\dot{X}_p$	Piston velocity, m/s
$\ddot{X}_p$	Piston acceleration, m/s^2
x_v	Open spool valve displacement, m
α	Area ratio, $\alpha = A_B/A_P$
ξ	Damping ratio of the servo valve
ρ	Oil density, 850 kg/m^3
σ	Viscous friction. parameter
ω_v	Servo valve natural frequency, rad/s

ABBREVIATIONS

EHSS	Electrohydraulic Servo Systems
ESMC	Sliding Mode Control with Error
MAE	Mean Absolute Error
SMC	Sliding Mode Control
SSMC	Switching Sliding Mode Control
PWM	Pulse Width Modulation

CREDIT AUTHORSHIP CONTRIBUTION STATEMENT:

E. S. Mohammed: Software, Resources, Investigation, Data Curation, Writing - Original Draft, **M. Hammam**: Validation, Writing – Review & Editing, **G.A.Z.Mousa:** Methodology, Formal analysis, Writing - Review & Editing, **A. M. Abdul Aziz:** Conceptualization, Investigation, Resources, supervision.

DECLARATION OF COMPETING INTEREST

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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