Delay Compensation Observer with Sliding Mode Controller for Rotary Electro-hydraulic Servo System

ZAKARYA Omar[1,2], *KHALID Hussein*[1], *WANG Xingsong*[1*], *ORELAJA Olusyi Adwale*[1]

1. School of Mechanical Engineering, Southeast University, Nanjing 211189, P.R. China;
2. Faculty of Engineering, Sana'a University, Sanaa, Yemen

(Received 25 April 2021; revised 10 June 2021; accepted 15 July 2021)

Abstract: The hip's lower limb exoskeleton essential and most important function is to support human's payload as well as to enhance and assist human's motion. It utilizes an electro-hydraulic servo manipulator which is required to achieve precise trajectory tracking and positioning operations. Nevertheless, these tasks require precise and robust control, which is very difficult to attain due to the inherent nonlinear dynamic behavior of the electro-hydraulic system caused by flow-pressure characteristics and fluid volume control variations of the servo valve. The sliding mode controller (SMC) is a widely used nonlinear robust controller, yet uncertainties and delay in the output degrade the closed-loop system performance and cause system instability. This work proposes a robust controller scheme that counts for the output delay and the inherent parameter uncertainties. Namely, a sliding mode controller enhanced by time-delay compensating observer for a typical electro-hydraulic servo system is adapted. SMC is utilized for its robustness against servo system parameters' uncertainty whereas a time-delay observer estimates the variable states of the controller (velocity and acceleration). The main contribution of this paper is improving on the closed loop performance of the electro hydraulic servo system and mitigating the delay time effects. Simulation results prove the robustness of this controller, which forces the position to track the desired path regardless of the changes of the amount of transport delay of the system's states. The performance of the proposed controller is validated by repeating the simulation analysis while varying the amount of delay time.

Key words: sliding mode controller; rotary electro-hydraulic servo system; delay compensating observer; transport delay

CLC number: TH7 **Document code:** A **Article ID:** 1005-1120(2021)S-0049-08

0 Introduction

Electro-hydraulic servo systems have a great indispensable role in many applications where large torque load and inertia have to be controlled with high degree of accuracy and efficient performance. Because of the nonlinear dynamic behavior of electro-hydraulic systems, such as flow-pressure characteristics and fluid volume control variations of the servo valve, the control of these systems becomes very difficult in return. Electro hydraulic models contain huge uncertainties and unmodeled dynamics[1]. With this regard, the sliding mode controller (SMC) has attracted considerable attention as it solves the problem of stability, modelling uncertainty, disturbances and it has a fast-transient response as well[2-3]. SMC is often combined with other controller schemes to further improve the system performance and to eliminate the most critical drawback of SMC which is called chattering caused by the high-frequency switching SMC, switching imperfection, delays in switching and small-time constants of actuators[4-6].

It is well known that time delay is a widely ex-

*Corresponding author, E-mail address: xswang@seu.edu.cn.

How to cite this article: ZAKARYA Omar, KHALID Hussein, WANG Xingsong, et al. Delay compensation observer with sliding mode controller for rotary electro-hydraulic servo system[J]. Transactions of Nanjing University of Aeronautics and Astronautics, 2021, 38(S): 49-56.

http://dx.doi.org/10.16356/j.1005-1120.2021.S.007

isting phenomenon in many industrial processes, such as electro hydraulic systems, electric circuits, etc.[7]. Time-delay and uncertainties result in the system instability and closed loop performance degradation. Therefore, time delay systems have drawn significant research interest in the area of developing new compensation techniques in the past few years. The delay compensation techniques guarantee that the controller designed for the system with no delay can be used for the control of the time delayed system given that the full state is measurable and delays are determined[8-10]. In Ref.[11], a robust control scheme for time delayed uncertain systems based on disturbance observer technique was proposed. It solves the linear matrix inequalities (LMIs) for the design parameters of the disturbance observer. Similarly, in Ref.[12], a robust control scheme for discrete time delayed uncertain systems based on disturbance observer technique was proposed.

In this paper, a robust nonlinear control technique is adapted for the control of electro hydraulic servo system. A time delay estimation with SMC (TDE-SMC) that counts for the time delay and the system uncertainties is proposed to ensure good performance and stable system. The response to the varying time delay is numerically tested for some given values of transport delay introduced to the system output.

1 Problem Formulation

1.1 System dynamics and modeling

The rotary electro-hydraulic system of the hip's exoskeleton shown in Fig.1 is discussed in this paper. It is mainly composed of a double-ended hydraulic curved piston rod, a center-mounted rotating shaft, an angular displacement sensor, a guide sleeve, two rigid links and an electro-hydraulic servo-proportional valve. In addition, it has some indispensable sealing components to prevent oil's external leakage as well as internal leakage between the two chambers[13-15].

The dynamical model for the cylinder can be described via Newton's Law by the following equation[12,16]

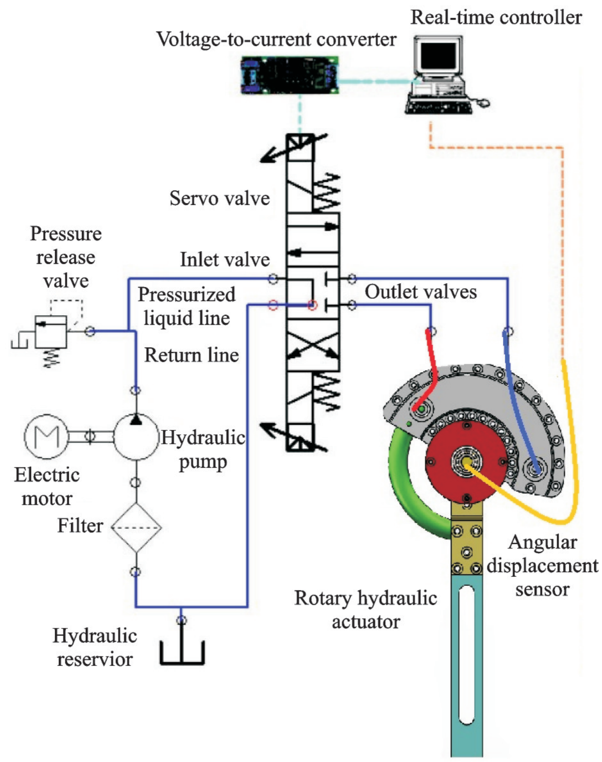

Fig.1 Schematic diagram of electro-hydraulic servo system

$$J\ddot{\theta}=\left(P_1\Omega_1-\Omega_2P_2\right)R-bR\dot{\theta}-T_{\mathrm{L}} \tag{1}$$

where R is the shortest distance between the center of rotation and the attached load's centroid of the mass; m the mass of the load and actuator; b the damping coefficient, and $\dot{\theta}$ the angular displacement. Ω_1 and Ω_2 are the effective ram areas of the cylinder.

The ratio between the effective area at both ends of the asymmetric cylinder is[14-17]

$$\eta=\frac{\Omega_2}{\Omega_1} \tag{2}$$

Hence, $P_{\mathrm{L}}=P_1-\eta P_2$.

The dynamical model for the cylinder becomes

$$J\ddot{\theta}=P_{\mathrm{L}}\Omega_1R-bR\dot{\theta}-T_{\mathrm{L}} \tag{3}$$

For an ideal electro hydraulic servo valve with symmetric and matched orifice, the load pressure P_{L} and flow Q_{L} can be expressed as

$$Q_{\mathrm{L}}=C_{\mathrm{d}}wx_{\mathrm{v}}\sqrt{\frac{P_{\mathrm{s}}-\operatorname{sgn}(x_{\mathrm{v}})P_{\mathrm{L}}}{\rho}} \tag{4}$$

where C_{d} is the valve discharge coefficient, w the spool valve area gradient, P_{s} the supply pressure and ρ the fluid density.

The dynamics of the spool valve is described as[18]

$$\tau_v \dot{\boldsymbol{x}}_v = -\boldsymbol{x}_v + K_v \boldsymbol{u} \tag{5}$$

where $\boldsymbol{x}_v$ is the spool valve displacement related to the current input $\boldsymbol{u}=\boldsymbol{i}$, and τ_v and K_v are the time constant and gain of the servo valve, respectively. The natural frequency of the servo valve is greater than that of the circular hydraulic cylinder, so that the spool valve dynamics is neglected and we have

$$\boldsymbol{x}_v = K_v \boldsymbol{u} \tag{6}$$

Substituting Eq.(6) to Eq.(4) and choosing

$$\boldsymbol{u}_0 = \sqrt{\boldsymbol{P}_s - \operatorname{sgn}(\boldsymbol{x}_v)\boldsymbol{P}_L}\, \boldsymbol{u} \tag{7}$$

The system states are selected as

$$[\boldsymbol{x}_1 \quad \boldsymbol{x}_2 \quad \boldsymbol{x}_3]^T = [\boldsymbol{\theta} \quad \dot{\boldsymbol{\theta}} \quad \boldsymbol{P}_L]^T \tag{8}$$

and the state space representation of the system is shown as

$$\begin{cases} \dot{\boldsymbol{x}}_1 = \boldsymbol{x}_2 \\ \dot{\boldsymbol{x}}_2 = -\phi_1 \boldsymbol{x}_2 + \phi_2 \boldsymbol{x}_3 \\ \dot{\boldsymbol{x}}_3 = -\psi_1 \boldsymbol{x}_2 - \psi_2 \boldsymbol{x}_3 + H\boldsymbol{u}_0 - C_{ts}\boldsymbol{P}_s + T_L \end{cases} \tag{9}$$

where $\phi_1 = \dfrac{bR}{J}$, $\phi_2 = \dfrac{\Omega_1 R}{J}$, $\psi_1 = \dfrac{2(1+\eta^2)\beta_e \Omega_1 R}{V_t}$, $\psi_2 = \dfrac{2(1+\eta^2)\beta_e C_{tp}}{V_t}$, $H = \dfrac{2(1+\eta^2)\beta_e C_d w K_v}{V_t\sqrt{\rho}}$.

The actual manipulation (control) signal of the system is defined by

$$\operatorname{sgn}(\boldsymbol{u}_0) = \operatorname{sgn}(\boldsymbol{u}) \tag{10}$$

$$\boldsymbol{u} = \frac{\boldsymbol{u}_0}{\sqrt{\boldsymbol{P}_s - \operatorname{sgn}(\boldsymbol{u}_0)\boldsymbol{x}_3}} \tag{11}$$

The corresponding errors of the system states and their relative derivations are represented by the difference by the desired and the actual states as represented in Eqs.(12, 13).

$$\dot{\boldsymbol{e}}_2 = \ddot{\boldsymbol{x}}_2 - \dddot{\boldsymbol{\theta}}_d = (-\phi_1 \dot{\boldsymbol{x}}_2 + \phi_2 \dot{\boldsymbol{x}}_3) - \dddot{\boldsymbol{x}}_d \tag{12}$$

$$\dot{\boldsymbol{e}}_2 = (\phi_1^2 - \phi_2\psi_1)\boldsymbol{x}_2 - (\phi_1\phi_2 + \phi_2\psi_2)\boldsymbol{x}_3 + (\phi_2 H)\boldsymbol{u}_0 - \dddot{\boldsymbol{\theta}}_d \tag{13}$$

1.2 Design of sliding mode controller

SMC is a nonlinear controller that provides a robust control action for nonlinear systems exhibiting uncertainties in its parameters. The sliding surface is firstly defined as[18]

$$\boldsymbol{S}_0 = \boldsymbol{e}_1 + \lambda_1 \boldsymbol{e} \tag{14}$$

But when differentiating the control input, u will not appear in the derived function because the relative degree between u and $\boldsymbol{S}$ is greater than one. Then, the sliding surface is chosen to be the following function with a relative degree equal to one between u and $\boldsymbol{S}$[19].

$$\boldsymbol{S} = \boldsymbol{e}_2 + \lambda_2 \boldsymbol{e}_1 + \lambda_1 \boldsymbol{e} \tag{15}$$

The control input u will appear when differentiating the sliding surface function. Furthermore, λ_2 and λ_1 are positive constants to drive the system states towards the sliding surface and remain there, hence ensuring the stability of the system. When the switching function reaches its zero level, the states will approach the origin as well[20-21].

The sliding mode controller law ensures that the states are driven to zero (towards the sliding surface), which is determined such that both reaching phase and sliding phase are $\boldsymbol{S}\dot{\boldsymbol{S}}<0$.

$$\dot{\boldsymbol{S}} = \dot{\boldsymbol{e}}_2 + \lambda_2 \boldsymbol{e}_2 + \lambda_1 \boldsymbol{e}_1 \tag{16}$$

The equivalent control is selected by the necessary condition of sliding mode to let $\dot{\boldsymbol{S}} = \boldsymbol{0}$ with nominal value.

$$\boldsymbol{u}_{eq} = (\phi_2 H)^{-1}(-(\phi_1^2 - \phi_2\psi_1)\boldsymbol{x}_2 + (\phi_1\phi_2 + \phi_2\psi_2)\boldsymbol{x}_3 + \dddot{\boldsymbol{\theta}}_d - \lambda_2 \boldsymbol{e}_2 - \lambda_1 \boldsymbol{e}_1) \tag{17}$$

The control rule is designed to meet the sliding condition regardless the estimation of errors. Hence, a discontinuous term is added to Eq.(17).

$$\boldsymbol{u} = \boldsymbol{u}_{eq} + \boldsymbol{u}_{sw} \tag{18}$$

where $\boldsymbol{u}_{sw} = H^{-1}K\operatorname{sgn}(\boldsymbol{S})$.

Achieving trajectories tracking with zero error requires all system states to be forced to converge to $\boldsymbol{S}$ in finite time and remain on $\boldsymbol{S}$ afterwards. For the system stability, the Lyapunov function is[22]

$$\boldsymbol{V} = 0.5\boldsymbol{S}^2 \tag{19}$$

The reaching condition of SMC is obtained where Π is a positive design parameter. Hence

$$\dot{\boldsymbol{V}} = 0.5\frac{\mathrm{d}}{\mathrm{d}t}\boldsymbol{S}^2 \leqslant -\Pi|\boldsymbol{S}| \tag{20}$$

The gain K of the discontinuous part of the controller rule is determined to meet the following condition

$$\begin{cases} K \geqslant \beta(F+\Pi) + (\beta-1)H|u| + D \\ \beta = \sqrt{H_{max}/H_{min}} \geqslant 1 \quad H^{-1}H \leqslant \beta \end{cases} \tag{21}$$

If the discontinuous switching gain K is met, reaching condition will be achieved and the tracking will be done within the specified error bounds of β and F.

1.3 Time delay estimation (TDE) with SMC

As elaborated in the previous section, the time delay of the electro hydraulic servo system degrades the system performance and excites undesirable noise that cause the system instability[22-24]. In this paper, the aim is to design a time-delay observer that estimates the states and ensure the system stability as well as ensure the system stability over the delay range[8].

The delayed output observer is designed as

$$\dot{\hat{\boldsymbol{x}}}(t)=\boldsymbol{A}\hat{\boldsymbol{x}}(t)+\boldsymbol{H}\boldsymbol{u}(t)+\boldsymbol{L}\left[\bar{\boldsymbol{y}}(t)-\frac{1}{2}(\hat{\boldsymbol{x}}(t-\Delta_1)+\hat{\boldsymbol{x}}(t-\Delta_2))\right] \tag{22}$$

$\boldsymbol{A}$ must be Hurwitz and K is selected to make A-KC Hurwitz as well[25].

$$\text{Let } \delta(t)=\bar{\boldsymbol{y}}(t)-\frac{1}{2}(\hat{\boldsymbol{x}}(t-\Delta_1)+\hat{\boldsymbol{x}}(t-\Delta_2)) \tag{23}$$

$$\bar{\boldsymbol{y}}=\boldsymbol{x}(t-\Delta)=\boldsymbol{C}(t-\Delta),\boldsymbol{C}=[1\ 0\ 0] \tag{24}$$

$$\begin{cases}\dot{\hat{\boldsymbol{x}}}_1=\hat{\boldsymbol{x}}_2+\boldsymbol{L}\,\delta(t)\\ \dot{\hat{\boldsymbol{x}}}_2=-\phi_1\hat{\boldsymbol{x}}_2+\phi_2\hat{\boldsymbol{x}}_3+\boldsymbol{L}\,\delta(t)\\ \dot{\hat{\boldsymbol{x}}}_3=-\psi_1\hat{\boldsymbol{x}}_2-\psi_2\hat{\boldsymbol{x}}_3+\boldsymbol{H}\boldsymbol{u}_0+\boldsymbol{L}\,\delta(t)\end{cases} \tag{25}$$

$$\boldsymbol{L}=[L_1\quad L_2\quad L_3]^{\mathrm{T}} \tag{26}$$

$$\begin{cases}\hat{\boldsymbol{e}}=\hat{\boldsymbol{x}}_1-\boldsymbol{\theta}_{\mathrm{d}}\\ \hat{\boldsymbol{e}}_1=\hat{\boldsymbol{x}}_2-\dot{\boldsymbol{\theta}}_{\mathrm{d}}\\ \hat{\boldsymbol{e}}_2=\dot{\hat{\boldsymbol{x}}}_2-\ddot{\boldsymbol{\theta}}_{\mathrm{d}}\end{cases} \tag{27}$$

The tracking error between the delayed output observer and the desired trajectories is

$$\dot{\hat{\boldsymbol{e}}}_2=\ddot{\hat{\boldsymbol{x}}}_2-\dddot{\boldsymbol{\theta}}_{\mathrm{d}}=-\phi_1\dot{\hat{\boldsymbol{x}}}_2+\phi_2\dot{\hat{\boldsymbol{x}}}_3-\dddot{\boldsymbol{\theta}}_{\mathrm{d}}+L\delta(t) \tag{28}$$

$$\dot{\hat{\boldsymbol{e}}}_2=(\phi_1^2-\phi_2\psi_1)\hat{\boldsymbol{x}}_2-(\phi_1\phi_2+\phi_2\psi_2)\hat{\boldsymbol{x}}_3+(\phi_2H)\boldsymbol{u}_0-\dddot{\boldsymbol{\theta}}_{\mathrm{d}}+L\delta(t) \tag{29}$$

The observing variable is

$$\hat{\boldsymbol{S}}=\hat{\boldsymbol{e}}_2+\lambda_1\hat{\boldsymbol{e}}_1+\lambda_2\hat{\boldsymbol{e}} \tag{30}$$

where λ_1 and λ_2 are positive constants. The Lyapunov function is selected as

$$V=0.5\hat{\boldsymbol{S}}^2 \tag{31}$$

$$\begin{aligned}\dot{V}=&\hat{\boldsymbol{S}}(\dot{\hat{\boldsymbol{e}}}_2+\lambda_1\hat{\boldsymbol{e}}_2+\lambda_2\hat{\boldsymbol{e}}_1)=\\&\hat{\boldsymbol{S}}((\phi_1^2-\phi_2\psi_1)\hat{\boldsymbol{x}}_2-(\phi_1\phi_2+\phi_2\psi_2)\hat{\boldsymbol{x}}_3+\\&(\phi_2H)\boldsymbol{u}_0-\dddot{\boldsymbol{\theta}}_{\mathrm{d}}+L\,\delta(t)+\lambda_1\hat{\boldsymbol{e}}_2+\lambda_2\hat{\boldsymbol{e}}_1)\end{aligned} \tag{32}$$

We select the controller as

$$\boldsymbol{u}=\boldsymbol{u}_{\mathrm{eq}}+\boldsymbol{u}_{\mathrm{sw}} \tag{33}$$

where $\boldsymbol{u}_{\mathrm{sw}}=\boldsymbol{b}^{-1}K\mathrm{sgn}(\boldsymbol{S})$.

$$\boldsymbol{u}_{\mathrm{eq}}=(\phi_2H)^{-1}(-(\phi_1^2-\phi_2\psi_1)\hat{\boldsymbol{x}}_2+(\phi_1\phi_2+\phi_2\psi_2)\hat{\boldsymbol{x}}_3+\dddot{\boldsymbol{\theta}}_{\mathrm{d}}-\lambda_2\boldsymbol{e}_2-\lambda_1\boldsymbol{e}_1) \tag{34}$$

$$\dot{V}=-K\hat{S}\,\mathrm{sgn}(\hat{S})=-K|\hat{S}|\leqslant 0 \tag{35}$$

Therefore, there exists t_s, for $t\geqslant t_s$, we have

$$\hat{\boldsymbol{S}}=\hat{\boldsymbol{e}}_2+\lambda_1\hat{\boldsymbol{e}}_1+\lambda_2\hat{\boldsymbol{e}} \tag{36}$$

i.e., $\hat{\boldsymbol{e}}_2,\hat{\boldsymbol{e}}_1,\hat{\boldsymbol{e}}\rightarrow\boldsymbol{0}$ as $t\rightarrow 0$.

2 Results and Analysis

The simulation of this technique is carried on MATLAB Simulink Table 1 lists all parameters used to represent the linearized model of the rotary electro hydraulic servo system. This defined model is then tested for some delay that is introduced to its states. The system is tested at five different delay times(0.2 s, 0.5 s, 0.8 s, 1 s, 1.5 s).

Table 1 Electro hydraulic servo system parameters

Symbol	Value	Unit
Ω_1	2.4×10^{-4}	m^2
Ω_2	4.9×10^{-4}	m^2
b	850	$\mathrm{N/(m\cdot s^{-1})}$
β_e	7.45×10^{8}	Pa
K_v	1.26×10^{-4}	m/V
w	1.2×10^{-2}	m^2
C_i	8×10^{-13}	$\mathrm{m^3/(Pa\cdot s)}$
V_1	2.09×10^{-5}	m^3
V_2	6.96×10^{-5}	m^3
λ_1	35 000	N/A
λ_2	16 000	N/A
ρ	850	$\mathrm{kg/m^3}$
C_d	0.8	N/A
K_q	0.342	$\mathrm{m^2/s}$
P_s	5×10^{6}	Pa
K	1.1	N/A
R	0.06	m
J	0.06	$\mathrm{kg\cdot m^2}$

Fig. 2 shows the actual state variables and the observer estimated states namely $(\boldsymbol{x}_1,\hat{\boldsymbol{x}}_1)$. The estimated states mimic and preserve the dynamical characteristics of the actual states. These estimated states are used as inputs for the sliding mode controller because in real systems some of the states are difficult to measure and the level noise raises the risk of triggering unwanted oscillations. The desired, actual trajectories and the estimated delay compensated trajectories are depicted when there is no delay at the system states.

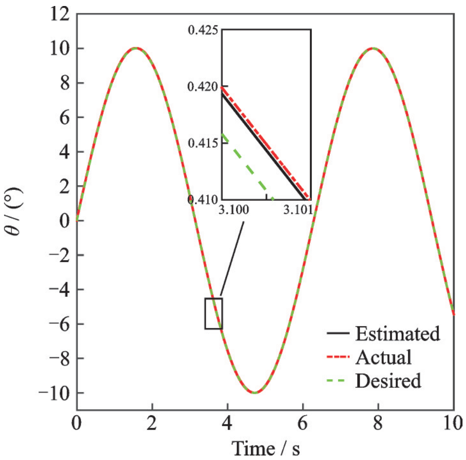

Fig.2 Output trajectories with delay-free system states

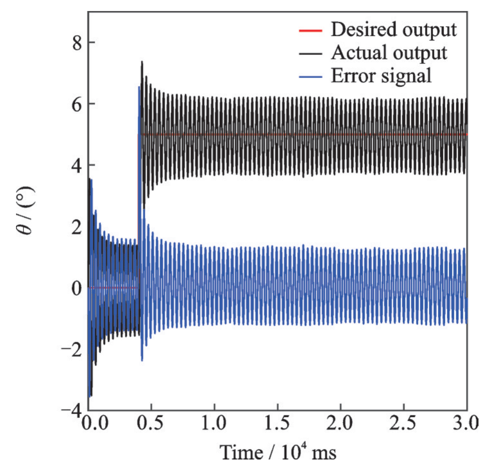

Fig.4 Estimated trajectory V_s actual response (Delay time response is not compensated)

Fig.3 shows the system trajectory tracking when the introduced delay is alleviated. Regardless of the delay of the output states the system response is maintained to be close to the desired one. This indicates that the delay effects have been eliminated. The system steady state errors are kept noticeably low and the response transient response are maintained within acceptable ranges. On the other hand, Fig. 4 depicts the system response while this delay exists for a step command positioning task.

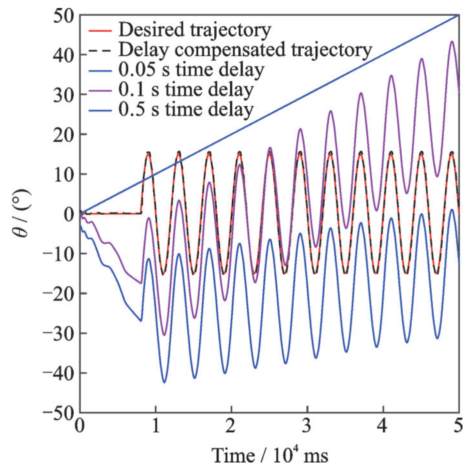

Fig.5 System response when delay exists and is compensated

Fig. 5 shows a comparison when the delay exists and when compensated. When delay exists, the system's response become instable and its motion profile can no longer be expected. Fig. 6 shows the error between the desired output state and the estimated delay compensated trajectories for different delay periods. It can be clear that the error margin is low regardless of the delay period. These signals are

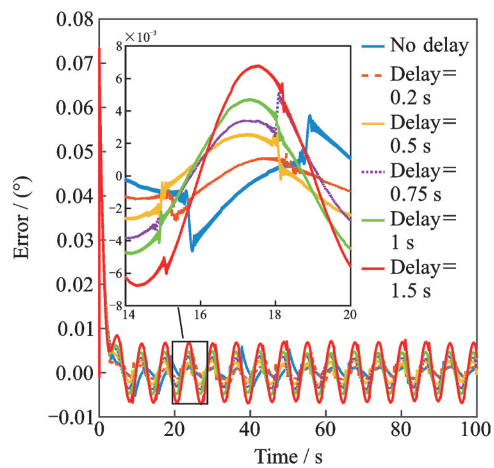

Fig.6 Error difference between the desired input trajectory and the estimated delay compensated output for a variety of delay ranges

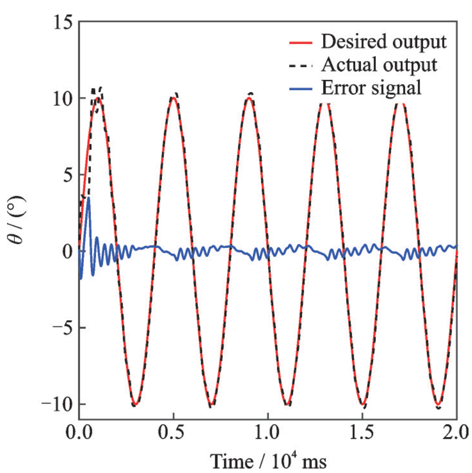

Fig.3 Estimated trajectory V_s actual response (Delay is compensated)

fed to the sliding mode controller and the system stability is maintained. Similarly, Fig.7 shows that the larger the delay, the larger the error. However, these transient and steady state errors are within the

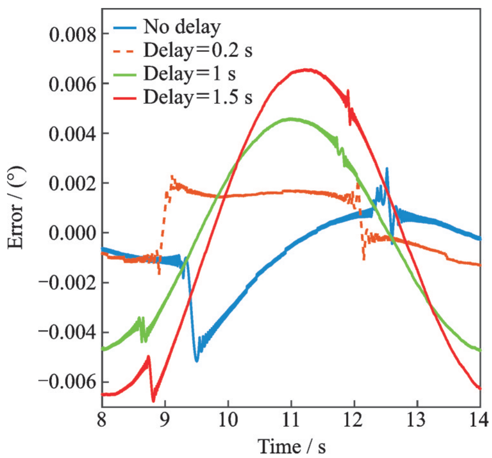

Fig.7 Magnitude of error signal for different delays

acceptable range with which a precise and smooth tracking are achieved.

3 Conclusions

A delay compensating observer for the SMC-controlled rotary electro-hydraulic servo systems with uncertain dynamics is considered. It estimates the states of the controller by which mitigating delay time of the system states becomes possible. A tiny delay in the output of an electro-hydraulic servo system that is controlled by the classical SMC could drive the system to be unstable and fire undesired attenuation. However, the designed technique is robust to the delay occurring in the output and stable over a wide operating range. Furthermore, the obtained numerical simulation results show desirable performance in terms of transient and steady response. It also shows a smooth and accurate tracking performance. The tracking error is kept noticeably low with less steady state error and settling time regardless of the transport delay occurring in the system states. Improving this new control technique that counts for the output delay phenomenon problem is the main contribution of this work. This technique is tested and validated by simulation in MATLAB.

References

[1] AHMED S, MAHDI S M, RIDHA T M M, et al. Sliding mode control for electro-hydraulic servo system[J]. Engineering Journal, 2015, 15(3): 1-10.

[2] GUO K, WEI J, FANG J, et al. Position tracking control of electro-hydraulic single-rod actuator based on an extended disturbance observer[J]. Mechatronics, 2015, 27: 47-56.

[3] UTKIN V, LEE H. Chattering problem in sliding mode control systems[C]//Proceedings of International Workshop on Variable Structure Systems. [S.l.]: [s.n.], 2006: 346-350.

[4] KACHROO P, TOMIZUKA M. Chattering reduction and error convergence in the sliding-mode control of a class of nonlinear systems[J]. IEEE Transactions on Automatic Control, 1996, 41:1063-1068.

[5] MA L, YANG Y, ZHANG J. Design and simulation of trajectory tracking controller based on fuzzy sliding mode control for AGV[C]//Proceedings of 2018 International Symposium in Sensing and Instrumentation. Shanghai, China:[s.n.], 2018: 1-5.

[6] TRAN D T, JEONG K, JUN G, et al. Adaptive gain back-stepping sliding mode control for electrohydraulic servo system with uncertainties[C]//Proceedings of 2017 14th International Conference on Ubiquitous Robots and Ambient Intelligence. Jeju, South Korea:[s.n.], 2017: 534-539.

[7] DIAGNE M, BEKIARIS-LIBERIS N, KRSTIC M. Compensation of input delay that depends on delayed input[J]. Automatica, 2017, 85: 362-373.

[8] GONZÁLEZ SORRIBES A, GARCÍA GIL P. A novel observer-predictor control for uncertain systems with unknown time-varying input and output delays[J]. International Journal Control, 2019, 7179: 1-11.

[9] CHEN J, YANG C, LIEN C, et al. New delay-dependent non-fragile H_∞ observer-based control for continuous time-delay systems[J]. Information Sciences, 2008, 178(24): 4699-4706.

[10] CHEN M, CHEN W H. Disturbance-observer-based robust control for time delay uncertain systems[J]. International Journal of Control, Automation and Systems, 2010, 8(2): 445-453.

[11] WU B, CHEN M, CHEN X. Observer-based bounded control for discrete time-delay uncertain nonlinear systems[J]. Discrete Dynamics in Nature and Society, 2015(5): 1-16.

[12] YANG M, MA K, SHI Y, et al. Modeling and position tracking control of a novel circular hydraulic actuator with uncertain parameters[J]. IEEE Access, 2019, 7: 181022-181031.

[13] YANG M, ZHU Q, XI R, et al. Design of the power-assisted hip exoskeleton robot with hydraulic servo ro-

tary drive[C]//Proceedings of the 23rd International Conference on Mechatronics and Machine Vision in Practice. [S.l.]:[s.n.], 2017.

[14] YUAN H B, NA H C, KIM Y B. System identification and robust position control for electro-hydraulic servo system using hybrid model predictive control[J]. Journal of Vibration and Control. 2018, 24(18): 4145-4159.

[15] LIN F. The design and simulation of electro-hydraulic velocity control system[C]//Proceedings of International Conference on Computer and Computing Technologies in Agriculture. [S.l.]:[s.n.], 2011: 568-574.

[16] WALTERS R B. Hydraulic and electric-hydraulic control systems[M]. [S.l.]:Springer, 2000.

[17] KONAMI S, NISHIUMI T. Hydraulic control systems: Theory and practice[M]. [S.l.]: World Scientific Publishing Company, 2016.

[18] MARTINEZ D I, RUBIO J D J, VARGAS T M. Stabilization of robots with a regulator containing the sigmoid mapping[J]. IEEE Access, 2020, 8: 89479-89488.

[19] WANG F, LIU Z, CHEN C L P, et al. Robust adaptive visual tracking control for uncertain robotic systems with unknown dead-zone inputs[J]. Journal of Franklin Institute, 2019, 356(12): 6255-6279.

[20] RUBIO J D J. Sliding mode control of robotic arms with deadzone[J]. IET Control Theory and Applications, 2017, 11(8): 1214-1221.

[21] CHEN H M, RENN J C, SU J P. Sliding mode control with varying boundary layers for an electro-hydraulic position servo system[J]. The International Journal of Advanced Manufacturing Technology, 2005, 26(1/2): 117-123.

[22] JEROUANE M, LAMNABHI-LAGARRIGUE F. A new robust sliding mode controller for a hydraulic actuator[C]//Proceedings of the 40th IEEE Conference on Decision and Control. [S.l.]: IEEE, 2001, 1: 908-913.

[23] JEROUANE M, SEPEHRI N, LAMNABHI-LAGARRIGUE F. Dynamic analysis of variable structure force control of hydraulic actuators via the reaching law approach[J]. International Journal of Control, 2004, 77(14): 1260-1268.

[24] LIN Y, SHI Y, BURTON R. Modeling and robust discrete-time sliding-mode control design for a fluid power electrohydraulic actuator (EHA) system[J]. IEEE/ASME Transactions on Mechatronics, 2013, 18(1): 1-10.

[25] REN C, HE S. Sliding mode control for a class of nonlinear positive Markov jumping systems with uncertainties in a finite-time interval[J]. International Journal of Control and Automation Systems, 2019, 17(7): 1634-1641.

Authors Mr. **ZAKARYA Omar** received his B.S. degree in mechatronics engineering from UTeM, Malaysia, in 2014. In 2021, he was awarded the M. S. degree from School of Mechanical Engineering, Southeast University, China. His major research interests include design and control of mechatronic systems, servo systems, robust control, and robots.

Prof. **WANG Xingsong** received the B.S. and M.S. degrees from Zhejiang University, Hangzhou, China, in 1988 and 1991, respectively, and the Ph. D. degree from Southeast University, Nanjing, China, in 2000, all in Mechanical Engineering. He was a visiting scientist with the School of Mechanical Engineering, Concordia University, Montreal, QC, Canada, from June 2000 to December 2000 and September 2001 to March 2002, and the School of Mechanical Engineering, Purdue University, West Lafayette, IN, USA, from September 2007 to March 2008. He is currently a full professor with the School of Mechanical Engineering and the Head of the Department of Mechatronics, Southeast University. His current research interests include control theory with application in precision CNC machine tools, advanced mechatronics with applications in biomedical engineering, and intelligent manufacturing.

Author contributions Mr. **ZAKARYA Omar** developed both the system model and the controllers' algorithms, ran the related simulation, and obtained and interpreted the results. Mr. **KHALID Hussein** contributed to the first draft of the manuscript. Prof. **WANG Xingsong** contributed to ideas and improvement of the whole work. Mr. **ORELAJA Olusyi Adwale** contributed to the writing and revising the manuscript. All authors commented on the manuscript draft and approved the submission.

Competing interests The authors declare no competing interests.

(Production Editor: WANG Jing)

旋转电液伺服系统的延迟补偿观察器与滑模控制器

ZAKARYA Omar[1,2]，KHALID Hussein[1]，王兴松[1]，
ORELAJA Olusyi Adwale[1]
（1.东南大学机械工程学院，南京 211189，中国；2.萨那大学工程学院，萨那，也门）

摘要：臂部的下肢外骨骼必不可少且最重要的功能是支持人类的有效载荷以及增强和辅助人类的运动。它采用电液伺服机械手，实现精确的轨迹跟踪和定位操作。然而，这些任务需要精确和稳健的控制，由于伺服阀的流量-压力特性和流体体积控制变化导致电动液压系统固有的非线性动态行为，因此很难实现。滑模控制器(SMC)是一种广泛使用的非线性鲁棒控制器，但输出中的不确定性和延迟会降低闭环系统的性能并导致系统不稳定。该工作提出了一种稳健的控制器方案，该方案考虑了输出延迟和固有参数的不确定性，即采用了典型电液伺服系统的时滞补偿观测器增强的滑模控制器。SMC因其对伺服系统参数不确定性的鲁棒性而被利用，而时延观测器则估计控制器的可变状态(速度和加速度)。本文的主要贡献是改进电液伺服系统的闭环性能并减轻延迟时间影响。仿真结果证明了该控制器的鲁棒性，无论系统状态的传输延迟量如何变化，它都会迫使位置跟踪所需的路径。通过在改变延迟时间量的同时重复仿真分析来验证所提出控制器的性能。

关键词：滑模控制器；旋转电液伺服系统；延迟补偿观察器；传输延迟