IOSR Journal of Engineering (IOSRJEN) **www.iosrjen.org**
ISSN (e): 2250-3021, ISSN (p): 2278-8719
Vol. 10, Issue 5, May 2020, ||Series -I|| PP 01-07

Dynamic surface control associates adapting external torque algorithm for electro-hydraulic velocity servo system

Bui Lap Hien[1] Ha Thi Hoai Thu[1], Vo Quang Vinh[2]
[1]Department ofcontrol technology and automation,Hanoi University of Industry, [2]Department ofcontrol and automation, Electric Power University, Vietnam
Corresponding Author: Vo Quang Vinh, *Department ofcontrol and automation, Electric Power University, Vietnam*
Received 28 April 2020; Accepted 11 May 2020

Abstract: The nonlinear model of the electro-valve driven hydraulic system is presented. Base on mathematical model, a speed control of an electro-hydraulic system is investigated in the paper based on Dynamic Surface Control (DSC). DSC is a systematic controller caculated by combining backstep-ping technique and sliding mode control... In order to regulate the system's state to the desired speed under the changes of external torque, an adaptive control is proposed in this paper. The system stability is proven, and a number of simulations are given to illustrate the effectiveness of the controller.
Keywords: Hydraulic system, Dynamic surface control, Adative control, Backstepping-sliding mode control, Lyapunov's stability

I. INTRODUCTION

Hydraulic systems are broadly used in different applications because of the capacity in handling large inertia and loads. Typically like processing of plastics, industrial robots, lunch vehicles, flight simulators, floating cranes, and various military applications [1]. Depending on desired control objectives, the reseaches about this system are divided into numerous feature like position, velocity or torque system. Many researches focus on position and tracking problems of the hydraulic systems in the quest of seeking appropriate control schemes for better control performance [2]-[4].With an idea of exploiting simplicity of pulse-width-modulated on/off valve accompanied with a feedback linearization control to stabilize the supply pressure, the authors successfully construct a nonlinear position control for a hydraulic system [5].This paper considers about velocity control schemes for hydraulically driven process. For successful closed loop velocity control, development of suitable controller which could reflect such characteristics is very significant, although the dynamics of electro hydraulic servo system is highly nonlinear [6]. Although the using the feedback linearization controller successfully achieves the desired objectives theoretically, another controller is designed using the backstepping approach. It avoids adverse effects of the uncertainties or non-modelled dynamics. The authors Krstic et al. in their book [7] introduced the fundamental concept of backstepping method. The backstepping control method is also presented in [8÷10] where this technique is explained in detail for regulating and tracking problemDynamic surface control (DSC) is a outstanding schemes that developed from backstepping technique and multi sliding surface control. In the bool [11], J. Karl Hedrick et al. introduced basically and using DSC for uncertain nonlinear systems in generally. This method was also applied in numerous system as electrical motors[12], power system[13]...This paper proposes a new controller for electro-hydraulic velocity servo system base on dynamic surface control. To increase the performance of system with external torque, an adaptive algorithms was added to adjust this parameters. The effifcient designed controller is validated by the appropriate simulation of nonlinear mathematical model of the system

II. SYSTEM DESCRIPTION AND DYNAMIC MODEL

Two stages servo valve consists of three main parts:the electrical torque motor, the hydraulic amplifier, andthe valve spool assembly. The dynamics of the valvespool with no noticeable decline in accuracy in a widerange of frequencies can be described through the firstorder transfer function between the valve opening area A_{sv} and control input u.

$$\tau.\dot{A}_{sv} + A_{sv} = K_{sv}.u \tag{1}$$

where K_{sv} is the servo valve gain and τ_{sv} is the servo valvetime-constant. The constants mentioned can bedetermined for by certaintests. Due to the fact that the input of the valve is anelectric current but the interface card output is in the formof an electric voltage, it is in common to use a current tovoltage converter.

For an ideal critical center, the servo valve with amatched and symmetric orifice the input/output flow rate from the servo valve through the orifices (assuming negligible leakage) can be expressed in the following form:

$$Q_L = C_d A_{sv} \sqrt{\frac{P_s - p_L sign\left(A_{sv}\right)}{\rho}} \qquad (2)$$

where $p_L = p_{C1} - p_{C2}$ is a load pressure or pressure difference between both chambers, $p_S = p_{C1} + p_{C2}$ is thesupply pressure and Q_L is the load flow. Assuming noexternal leakage, Q_L can be considered as the averageflow in each path $Q_L = \frac{Q_{C1} + Q_{C2}}{2}$, Q_{C1} and Q_{C2} are flow rates to and from the servo valve.

$$\frac{V_0}{2\beta} \dot{p}_L = C_d A_{sv} \sqrt{\frac{p_s - p_L sign\left(A_{sv}\right)}{\rho}} - D_m \dot{\Theta} - C_L p_L \qquad (3)$$

where β and V_0 are, respectively, the fluid bulk modulusand the oil under compression in one chamber of theactuator. D_m and C_L represent the actuator volumetricdisplacement and total leakage coefficient, respectively.By applying Newton’s second law for the rotary motion of ahydraulic actuator and neglecting the Coulomb’sfrictional torque:

$$J_T \ddot{\Theta} = D_m p_L - B.\dot{\Theta} + T_L \qquad (4)$$

Combining(1),(2), (3) and (4),the third-order nonlinear system that describe the system dynamics can be derived as:

$$\ddot{\Theta} = -a_1 \dot{\Theta} + a_2 p_L + a_3 \qquad (5)$$

$$\dot{p}_L = -a_4 \dot{\Theta} - a_5 p_L + a_6 \left(\sqrt{P_s - p_L}\right) A_{sv} \qquad (6)$$

$$\dot{A}_{sv} = -a_7 A_{sv} + a_8 u \qquad (7)$$

where $a_1 = \frac{B}{J_t}$, $a_2 = \frac{D_m}{J_t}$, $a_3 = \frac{T_L}{J_t}$, $a_4 = \frac{2\beta D_m}{V_0}$ $a_5 = \frac{2\beta}{V_0} C_L$, $a_6 = \frac{2\beta}{V_0\sqrt{\rho}} C_d$, $a_7 = \frac{1}{\tau_{sv}}$ and $a_8 = \frac{K_{sv}}{\tau_{sv}}$

III. SYSTEM DESCRIPTION AND DYNAMIC MODEL

A.Dynamic surface control

Dynamic Surface Control (DSC) is constructed based on integrator backstepping (IB) technique and multiple sliding mode (MSS) method. So that, this controller is inherited all the advantages of both above mechanisms. Beside, with an important addition in the design: the low-pass filter, this controller also brought significant effect in diminishing error in calculating and minimizing the amount of computation by avoiding the “explosion of terms” phenomenon. Let consider the following to express the DSC approach for the nonlinear system:

$$\begin{cases} \dot{x}_1 = x_2 + f(x_1) \\ \dot{x}_2 = u \end{cases} \qquad (8)$$

Where the function $f\left(x\right)$ is non-Lipschitz nonlinearity and completely known.

Defining the first error valuable: $S_1 = x_1 - x_{1r}$ (9)

Firstly, we determind the virtual control signal $\bar{x}_{2r}$ so that if x_2 tracks this value, x_1 will aprroach the reference.

Proposing Lyapunov candidate function for S_1 as:

$$V_1 = \frac{1}{2} S_1^{\,2} \qquad (10)$$

Differentiating (10) gives:

$$\dot{V}_1 = S_1 \dot{S}_1 = S_1 \left(x_2 + f\left(x\right) - \dot{x}_{1r}\right) \qquad (11)$$

Choosing $\bar{x}_{2r} = -f\left(x\right) + \dot{x}_{1r} - k_1 S_1$, where k_1 is a positive gain, thus $V_1 = -k_1 S_1^{\,2} \le 0$ or x_1 will be driven to x_{1r} by x_{2r}.

Signal $\bar{x}_{2r}$ determined above is a virtual signal. At this step, a low pass filter is added, x_{2r} track to $\bar{x}_{2r}$ through this filter as:

$$\begin{cases} \tau \dot{x}_{2r} + x_{2r} = \bar{x}_{2r} \\ x_{2r}\left(0\right) = \bar{x}_{2r}\left(0\right) \end{cases} \qquad (12)$$

Defining sliding surface:

$$S_2 = x_2 - x_{2r} \tag{13}$$

The control signal u will drive S_2 to zero, thus, u is chosen as:

$$u = \dot{x}_{2r} - k_2 S_2 = \frac{\bar{x}_{2r} - x_{2r}}{\tau} - k_2 S_2 \tag{14}$$

The stability of system will be consider by Lyapunov cadidate function following:

$$V = \frac{S_1^{\,2} + S_2^{\,2}}{2} \tag{15}$$

We will discuss specifictly about the stability of DSC for roll-to-roll system in the next section.

B. Dynamic surface control for electro-hydraulic velocity system

The controller aims to keep the velocity at desired values, web speed is controlled through tracking angular velocity at references. This section will express the DSC controller formular for the electro-hydraulic system, then we will demonstrate the stability of system with that controller. The outstanding characteristics will be also discussed in this section and illustrated through simulation results in the final section. Firstly, define some error variables that are considered for design controller as following:

$$x_1 = \dot{\Theta} - \dot{\Theta}_d \tag{16}$$

$$x_2 = p_L - \alpha_1 \tag{17}$$

$$x_3 = A_{sv} - \alpha_2 \tag{18}$$

Where $\dot{\Theta}_d$ is desired desired rewind velocity. α_1, α_2 are the control virtual signals. The conventional ideal is using virtual control signals α_{1d}, α_{2d} generated through backstepping technique in order to $x_1 \to 0$ and $x_2 \to 0$, respectively. But in this research, instead of using directly virtual signal α_{1d}, α_{2d} we use α_1, α_2 that are filtered signal after let α_{1d}, α_{2d} through low-pass filter that was defined as:

$$\begin{cases} \tau\dot{\alpha} + \alpha = \alpha_d \\ \alpha(0) = \alpha_d(0) \end{cases} \tag{19}$$

This addition helping decrease significantly amount of computation from caculating derivative of virtual control, that generate “explotion of term” phenomenon. Rewrite the mathmetical system by these error variables as:

$$\dot{x}_1 = -a_1\dot{\Theta} + a_2 p_L + a_3 - \ddot{\Theta}_d \tag{20}$$

$$\dot{x}_2 = -a_4\dot{\Theta} - a_5 p_L + a_6\left(\sqrt{P_s - p_L}\right)A_{sv} - \dot{\alpha}_1 \tag{21}$$

$$\dot{x}_3 = -a_7 A_{sv} + a_8 u - \dot{\alpha}_2 \tag{22}$$

Choose the first Lyapunov candidate funtion as:

$$V_1 = \frac{1}{2}x_1^{\,2} \tag{23}$$

Take time derivative (23) and using (20), we obtain:

$$\dot{V}_1 = x_1\dot{x}_1 = x_1\left(-a_1\dot{\Theta} + a_2 p_L + a_3 - \ddot{\Theta}_d\right) \tag{24}$$

From equation (24), the first control virtual signal can be chosen as:

$$\alpha_{1d} = \frac{1}{a_2}\left(a_1\dot{\Theta} - a_3 + \ddot{\Theta}_d - k_1.\mathrm{sign}\left(x_1\right)\right) \tag{25}$$

Where k_1 is a positive gain. Replace the signal (25) to (24), we obtain:

$$\dot{V}_1 = -k_1 x_1.\mathrm{sign}\left(x_1\right) \le 0 \tag{26}$$

The derivative of Lyapunov funtion is negative definite as above demonstrate, signal x_1 will aprroach to zero as Lyapunov stability theory.

Next, we will determine the second control virtual signal. The desired value of pressure difference between both chambers is α_1 which is filtered signal after let α_{1d} through the low-pass filter. Choose the second Lyapunov candidate funtion as:

$$V_2 = \frac{1}{2}x_2^{\,2} \tag{27}$$

Take time derivative(27)and using(21), we obtain:

$$\dot{V}_2 = x_2\dot{x}_2 = x_2\left(-a_4\dot{\Theta} - a_5 p_L + a_6\left(\sqrt{P_s - p_L}\right)A_{sv} - \dot{\alpha}_1\right) \tag{28}$$

From (28), choose the desired value of A_{sv} as:

$$A_{svd} = \alpha_{2d} = \frac{a_4\dot{\Theta} + a_5 p_L + \dot{\alpha}_1 - k_2.\text{sign}(x_2)}{a_6\left(\sqrt{P_s - p_L}\right)} \tag{29}$$

Where k_2 is a positive gan. Replace the fomular above to (28), we obtain:

$$\dot{V}_2 = -k_2 x_2.\text{sign}(x_2) \le 0 \tag{30}$$

The derivative of the second Lyapunov funtion is negative definite as demonstrate, signal x_2 will aprroach to zero as Lyapunov stability theory.

The fomular (29) is the second control signal, after letting this signal through the low-pass filter, we obtain the desired value of valve opening area α_2. Choose the third Lyapunov candidate funtion as:

$$V_3 = \frac{1}{2}x_3^2 \tag{31}$$

Take time derivative (31) and using (22), we obtain:

$$\dot{V}_3 = x_3\dot{x}_3 = x_3\left(-a_7 A_{sv} + a_8 u - \dot{\alpha}_2\right) \tag{32}$$

Choose the control signal as:

$$u = \frac{1}{a_8}\left(a_7 A_{sv} + \dot{\alpha}_2 - k_3\text{sign}(x_3)\right) \tag{33}$$

Where k_3 is a positve gain.

With the control signal as choosen, the derivative of third Lyapunov candidate function can be rewritten as:

$$\dot{V}_3 = -k_3 x_3.\text{sign}(x_3) \le 0 \tag{34}$$

The derivative of Lyapunov funtion is negative definite as above demonstrate, signal x_3 will aprroach to zero as Lyapunov stability theory.

To demonstrate the stability of all states, we propose the total Lyapunov candidate function as:

$$V = \frac{1}{2}x_1^2 + \frac{1}{2}x_2^2 + \frac{1}{2}x_3^2 \tag{35}$$

Take time derivative of (35) and use the control signals (25), (29), (33), we obtain:

$$\dot{V} = -k_1 x_1.\text{sign}(x_1) - k_2 x_2.\text{sign}(x_2) - k_3 x_3.\text{sign}(x_3) \tag{36}$$

From above formular, we can infer that derivative of total Lyapunov candidate function is negative definite. The system will be stable as Lyapunov stability theory.

C. Adaptive external torque for electro-hydraulic velocity system

In fact, when the system is operating, the external torque is an uncertainty. Although the DSC controller proposed in previous section has adjustable disturbances function, is is necessary to associate with a adaptive external torque algorithm if this parameter is changed significantly. Base on the previous designed controller, we can rewrite the virtual control signal (25) as:

$$\hat{\alpha}_{1d} = \frac{1}{a_2}\left(a_1\dot{\Theta} - \frac{\hat{T}_L}{J_t} + \ddot{\Theta}_d - k_1.\text{sign}(x_1)\right) \tag{37}$$

Where $\hat{T}_L$ is the adapted value of the torqe. Propose the Lyapunov candidate function as:

$$V = \frac{1}{2}x_1^2 + \frac{1}{2}x_2^2 + \frac{1}{2}x_3^2 + \frac{1}{2}\lambda\left(\hat{T}_L - T_L\right) \tag{38}$$

Take time derivative of (38) and use the control signals (29), (33), (37) we obtain:

$$\dot{V} = -k_1 x_1.\text{sign}(x_1) - k_2 x_2.\text{sign}(x_2) - k_3 x_3.\text{sign}(x_3) + \left(\hat{T}_L - T_L\right)\left(\lambda\dot{\hat{T}}_L - \frac{x_1}{J_t}\right) \tag{39}$$

From (39), to guarantee that the derivative of Lyapunov candidate function is negative definite, we choose the adapted law as:

$$\dot{\hat{T}}_L = \frac{\lambda x_1}{J_t} \tag{40}$$

The next section will express some simulation results with proposed algorithm with the uncertain torque element considered.

IV. SIMULATION RESULT

In this section, the performance of the proposed controller will be evaluated through a numerical simulations. In order to test the quality of the designed controller, there will be three cases and the simulation results of the presented controller is compared to the reference. To evaluated the outstanding efficient, we will also compare with performances of PID controller. The parameters of system is shown in table below:

Table 1: Physical parameters

$J_t = 3.4\times10^{-3}\,kgm^2$	$D_m = 0.75\times10^{-6}\,m^3/rad$
$B = 1.25\times10^{-6}\,Nms/rad$	$C_L = 9.5\times10^{-12}\,m^5/Ns$
$\beta = 0.35\times10^{9}\,Pa$	$V_0 = 2.75\times10^{-5}\,m^3$
$C_d = 0.65$	$T_L = 0.7Nm$
$K_{sv} = 4.23\times10^{-7}\,m^2/V$	$\tau_{sv} = 0.001s$

In the first case, the reference is 40 (rd/s) and the system will be operate from initial state. The figure 1 is shown the response of proposed controller in this case.. The figure 2 is expressed the perfomance when the desired value changes suddenly. In last case, an unknown torque will be added to system when it's operating, the details will be shown in figure3.

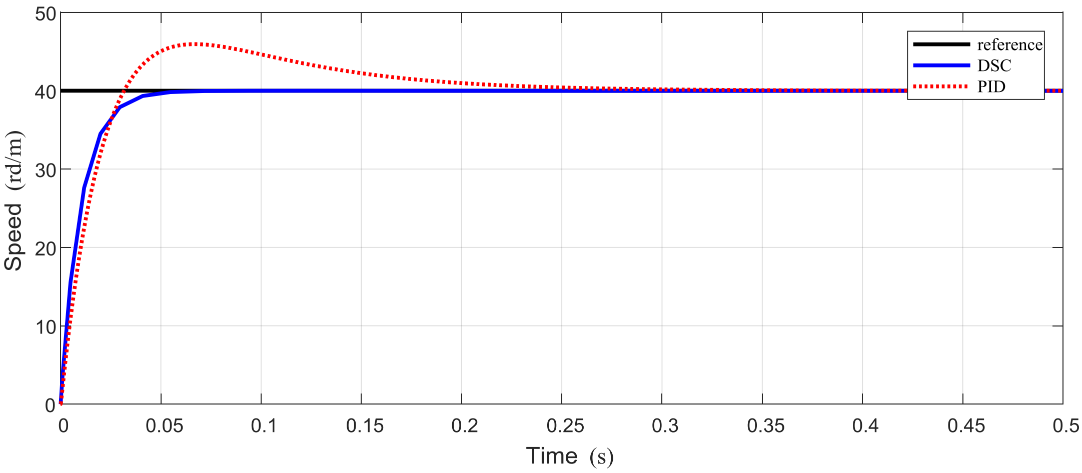

Figure 1: The velocity responses with constant reference

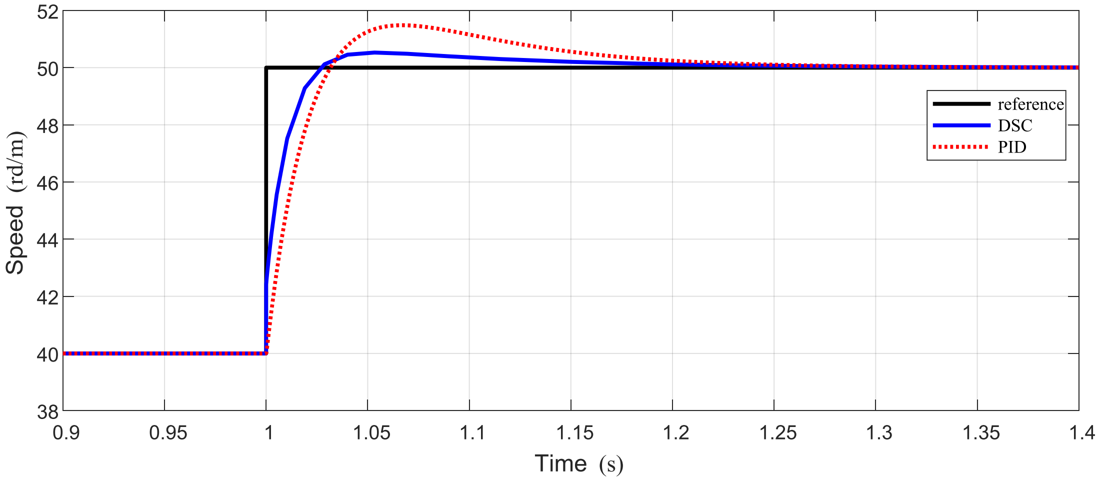

Figure 2: The velocity responses with suddenly changed reference

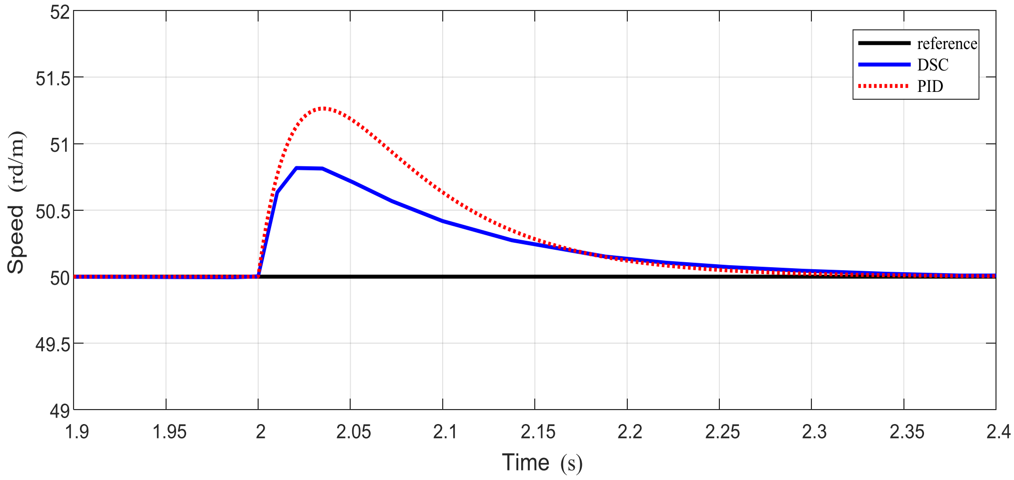

Figure 3: The velocity responses with suddenly external torque added

As the simulation results, the responses of velocity and the are shown in fig.1-3 coresponding to three cases. It can be seen that the velocity responses can track the reference and converge to the desired value in a short time in all case but the response of new controller is faster than PID algorithm. The overshoot is signigicantly diferrent in the performance of the controller between the proposed controller and conventional one in all case.Once the external torque is suddenly added, both the adaptive DSC and PID controller ensure the settling time is small, but PID's overshoot is bigger than DSC. The simulations showed that the outstanding effecient of new algorithm compare with conventional controller. Especially when unknown disturbance.

IV. CONCLUSION

In the paper, an adaptive dynamic surface control for electro-hydraulic system is introduced. Hydraulic nonlinearities are taken care of using the systematic approach. Uncertain external torqueis handled thanks to the adaptive algorithm that was associated with DSC controller. The system is proven to be stable in Lyapunov's sense. Several numerical studies are included to show the effectiveness of the proposed control.

ACKNOWLEDGEMENTS

The authors would like to thank the Electric Power University (EPU) and Hanoi University of Industry for supporting this work.

REFERENCES

[1] M. B. Rannow and P. Y. Li, "On/off valve based position control of a hydraulic cylinder," in *ASME 2007 International Mechanical Engineering Congress and Exposition*, 2007, pp. 141–149.

[2] A. A. Aly, "Model reference PID control of an electro-hydraulic drive," *Int. J. Intell. Syst. Appl.*, vol. 4, no. 11, p. 24, 2012.

[3] D. Lovrec, V. Tic, and T. Tasner, "Simulation-aided determination of an efficiency field as a basis for maximum efficiency controller design," *Int. J. Simul. Model.*, vol. 14, no. 4, pp. 669–682, 2015.

[4] F. Najafi, M. Fathi, and M. Saadat, "Dynamic modelling of servo pneumatic actuators with cushioning," *Int. J. Adv. Manuf. Technol.*, vol. 42, no. 7–8, pp. 757–765, 2009.

[5] C. C. Soon, R. Ghazali, H. I. Jaafar, S. Y. S. Hussien, S. M. Rozali, and M. Z. A. Rashid, "Position tracking optimization for an electro-hydraulic actuator system," *J. Telecommun. Electron. Comput. Eng.*, vol. 8, no. 7, pp. 1–6, 2016.

[6] H. M. Kim, S. H. Park, J. H. Song, and J. S. Kim, "Robust position control of electro-hydraulic actuator systems using the adaptive back-stepping control scheme," *Proc. Inst. Mech. Eng. Part I J. Syst. Control Eng.*, vol. 224, no. 6, pp. 737–746, 2010.

[7] M. Krstic, P. V Kokotovic, and I. Kanellakopoulos, *Nonlinear and adaptive control design*. John Wiley & Sons, Inc., 1995.

[8] S. J. Lee and T.-C. Tsao, "Nonlinear backstepping control of an electrohydraulic material testing system," in *Proceedings of the 2002 American Control Conference (IEEE Cat. No. CH37301)*, 2002, vol. 6, pp. 4825–4830.

[9] C. Kaddissi, J.-P. Kenne, and M. Saad, "Identification and real-time control of an electrohydraulic servo system based on nonlinear backstepping," *IEEE/ASME Trans. Mechatronics*, vol. 12, no. 1, pp. 12–22, 2007.

[10] M. Jovanovic, “Nonlinear control of an electrohydraulic velocity servosystem,” in *Proceedings of the 2002 American Control Conference (IEEE Cat. No. CH37301)*, 2002, vol. 1, pp. 588–593.
[11] B. Song and J. K. Hedrick, *Dynamic surface control of uncertain nonlinear systems: an LMI approach*. Springer Science & Business Media, 2011.
[12] X. S. Luo, B. H. Wang, and J. Q. Fang, “Robust adaptive dynamic surface control of chaos in permanent magnet synchronous motor,” *Phys. Lett. A*, vol. 363, no. 1–2, pp. 71–77, 2007.
[13] S. Mehraeen, S. Jagannathan, and M. L. Crow, “Power system stabilization using adaptive neural network-based dynamic surface control,” *IEEE Trans. Power Syst.*, vol. 26, no. 2, pp. 669–680, 2010.

Bui Lap Hien. "Dynamic surface control associates adapting external torque algorithm for electro-hydraulic velocity servo system." *IOSR Journal of Engineering (IOSRJEN)*, 10(5), 2020, pp. 01-07.