Nonlinear backstepping-sliding mode control of electro-hydraulic systems

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Tung Lam Nguyen[1] and Hong Quang Nguyen[2], Manh Cuong Nguyen[1], Van Manh Tran[1], Danh Huy Nguyen[1], and Anh Duc Nguyen[3]

[1] Hanoi University of Science and Technology, Vietnam
[2] Thai Nguyen University of Technology, Vietnam
[3] Thai Nguyen University, Vietnam
quang.nguyenhong@tnut.edu.vn

Abstract. A speed control of an electro-hydraulic system is investigated in the paper. The nonlinear model of the electro-valve driven hydraulic system is developed. A systematic control approach using backstepping-sliding mode control is designed with can regulate the system to the desired speed. The system stability is proven, and a number of simulations are given to illustrate the effectiveness of the control.
Keywords: Electro-hydraulic, backstepping, sliding mode control.

1 Introduction

Thanks to the ability of handling heavy tasks, hydraulic systems are widely employed in various applications [1]. Many researches focus on position and tracking problems of the hydraulic systems in the quest of seeking suitable control schemes for better control performances [2]–[4]. In [5], the authors consider the position control of the hydraulic actuator taking into account system dead-zone, control input saturation, discharge coefficient and friction. A PID control is designed and reinforced by particle swarm optimization, the proposed control is validated through numerical simulations and experiments. 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 [6]. In [7], the authors suggest a fuzzy logic controller whose parameters are tuned using particle swarm optimization. The analogous technique of using optimization can be found in [8]. Other nonlinear control approaches to the hydraulic system can be listed in [9]–[11].

The mentioned researches either need accurate system information or relatively comprehensive understandings about the system. In actual situation, hydraulic system parameters are not easy to identify correctly. The paper proposes a backstepping based position control of the hydraulic system backed with sliding mode control for the sake of system robustness. The system stability and performance are proven analytically and validated numerically.

2 System description and dynamical modeling

Two stages servo valve consists of three main parts: the electrical torque motor, the hydraulic amplifier, and the valve spool assembly. The dynamics of the valve spool with no noticeable decline in accuracy in a wide range of frequencies can be described through the first order 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 valve time constant. The constants mentioned can be determined for by certain tests. Due to the fact that the input of the valve is an electric current but the interface card output is in the form of an electric voltage, it is in common to use a current to voltage converter.

For an ideal critical center, the servo valve with a matched 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:

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

where $p_L = p_{C1} - p_{C2}$ is a load pressure or pressure difference between both chambers, $p_S = p_{C1} + p_{C2}$ is the supply pressure and Q_L is the load flow. Assuming no external leakage, QL can be considered as the average flow 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.

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

where β and V_0 are, respectively, the fluid bulk modulus and the oil under compression in one chamber of the actuator. D_m and C_L represent the actuator volumetric displacement and total leakage coefficient, respectively. By applying Newton's second law for the rotary motion of a hydraulic actuator and neglecting the Coulomb's frictional torque:

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

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

$$\begin{aligned} \ddot{\Theta} &= -a_1 \dot{\Theta} + a_2 p_L + a_3 \\ \dot{p}_L &= -a_4 \dot{\Theta} - a_5 p_L + a_6 \left(\sqrt{P_s - p_L}\right) A_{sv} \\ \dot{A}_{sv} &= -a_7 A_{sv} + a_8 u \end{aligned} \tag{5}$$

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 $a_8 = \frac{K_{sv}}{\tau_{sv}}$

3. Nonlinear control design

This section addresses the problem of designing a controller which provides asymptotic stability of the operating point of interest. Assuming that the full state information is available, the underlying technique for solving this problem is SMC-backstepping. The first and second tracking error is defined as:

$$\begin{aligned} e_1 &= \dot{\Theta} - \dot{\Theta}_d \\ e_2 &= p_L - p_{Ld} \\ e_3 &= A_{sv} - A_{svd} \end{aligned} \tag{6}$$

Define the first candidate of control Lyapunov candidate function

$$V_1 = \frac{1}{2} e_1^2 \tag{7}$$

Then p_{Ld} is chosen as virtual control to drive e_1 to zero so that $\dot{\Theta}$ can track $\dot{\Theta}_d$.

$$p_{Ld} = \frac{1}{a_2}\left(a_1\dot{\Theta} + \ddot{\Theta}_d - a_3 - \xi_1 e_1\right) \tag{8}$$

$$\dot{V}_1 = e_1\dot{e}_1 = e_1\left(-a_1\dot{\Theta} + a_2 e_2 + a_2 p_{Ld} + a_3 - \ddot{\Theta}_d\right) \tag{9}$$

With such selection, we achieve the following form of augmented Lyapunov function:

$$\dot{V}_1 = a_2 e_1 e_2 - \xi_1 e^2 \tag{10}$$

The dynamics error e_1 can be described

$$\dot{e}_1 = \ddot{\Theta} - \ddot{\Theta}_d = -a_1\dot{\Theta} + a_2 p_L + a_3 - \ddot{\Theta}_d \tag{11}$$

Then p_{Ld} is chosen as virtual control to drive e_1 to zero, by substituting (6) into (11):

$$\dot{e}_1 = -a_1\dot{\Theta} + a_2\left(e_2 + p_{Ld}\right) + a_3 - \ddot{\Theta}_d \tag{12}$$

We obtain the virtual signal p_{Ld} as

$$p_{Ld} = \frac{1}{a_2}\left(a_1\dot{\Theta} + \ddot{\Theta}_d - a_3 - \xi_1 e_1\right) \tag{13}$$

Step 2: The virtual control error e_2 can be described by

$$\dot{e}_2 = \dot{p}_L - \dot{p}_{Ld} \tag{14}$$

By substituting (6) into (14):

$$\dot{e}_2 = -a_4\dot{\Theta} - a_5 p_L + \left(a_6\sqrt{P_s - p_L}\right) A_{sv} - \dot{p}_{Ld} \tag{15}$$

Define the second Lyapunov candidate function:

$$V_2 = V_1 + \frac{1}{2}e_2^2 \tag{16}$$

Differentiating V_2 gets:

$$\dot{V}_2 = \dot{V}_1 + e_2\dot{e}_2 \tag{17}$$

$$\dot{V}_2 = -\xi_1 e_1^2 + e_2(-\xi_2 e_2) + a_6\sqrt{P_s - x_2}\,e_2 e_3 \tag{18}$$

$$\dot{V}_2 = a_2 e_1 e_2 - \xi_1 e_1^2 + e_2\left[-a_4\dot{\Theta} - a_5 p_L + a_6\sqrt{P_s - p_L}\,A_{sv} - \dot{p}_{Ld}\right] \tag{19}$$

From (6), it is straightforward to show that:

$$\dot{V}_2 = a_2 e_1 e_2 - \xi_1 e_1^2 + e_2\left[-a_4\dot{\Theta} - a_5 p_L + a_6\sqrt{P_s - p_L}\,A_{svd} + a_6\sqrt{P_s - p_L}\,e_3 - \dot{p}_{Ld}\right] \tag{20}$$

Similarly, we define the sliding surface as:

$$s = \lambda(A_{sv} - A_{svr}) \tag{21}$$

Differentiating s yields

$$\dot{s} = \lambda(-a_7 A_{sv} + a_8 u - \dot{A}_{svr}) \tag{22}$$

The signal control can be computed as:

$$u = u_s + u_e \tag{23}$$

here u_e is the equivalent control and u_s is the switching control. By differentiating s with respect to time t in (6), letting $\dot{s}$ = 0, and substituting (6) into it, the equivalent control is formulated as

$$u_e = \frac{1}{a_8\lambda}\left(a_7 A_{sv} + \dot{A}_{svr}\right) \tag{24}$$

$$u_s = -\frac{k}{a_8\lambda}s - \frac{\eta}{a_8\lambda}sat\left(s\right) \tag{25}$$

Where *k* and *η* are positive constants. Define the Lyapunov candidate function as

$$V_s = \frac{1}{2}s^2 \tag{26}$$

Differentiating both sides of (26) gives:

$$\dot{V}_s = s\dot{s} \tag{27}$$

Then $$\frac{dV_s}{dt} = s[\text{-ks-}\eta sat(s)] = -ks^2 - \eta\left|s\right| \le 0\,. \tag{28}$$

(28) ensures the sliding mode is reachable in finite time. In the subsequent time interval, the system trajectory moves along the sliding surface and converges to the coordinate origin constructed by the intermediate variable s and its derivative.

4. Simulation studies

In this section, numerical simulation results are presented to verify the accuracy of the proposed control approach for the hydraulic rotary actuator. The results are shown in two cases: the system working without any external noises and the system affected by the disturbance. Different reference speed forms are set for the system to validate the robustness of the designed algorithm.

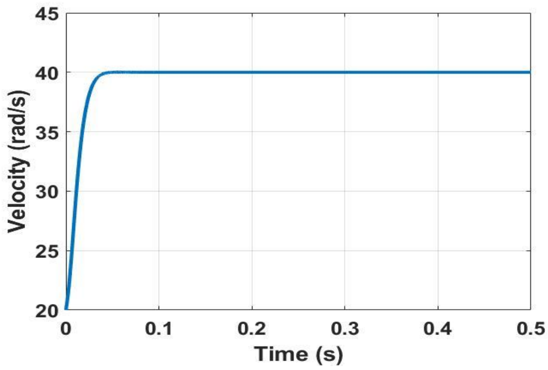

Fig .1 Actual velocity with fixed reference

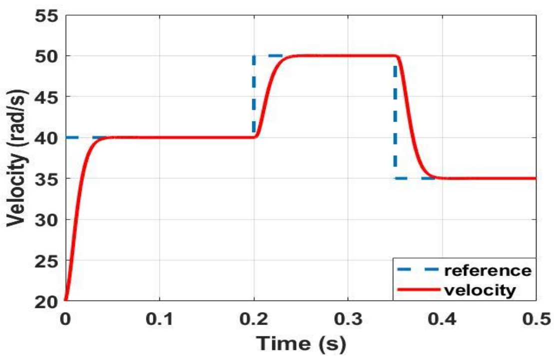

Fig. 2 Actual velocity with variable reference

Under the ideal condition, assuming that all system's parameters are clearly known, the performance of the electrohydraulic velocity servo system is demonstrated in Fig.1 and Fig.2. The reference speed is chosen as $\dot{\Theta} = 40(rad/s)$ and the actual value can be seen in Fig.1. For the variation of the reference velocity with time, the proposed controller still ensures the high tracking quality of the actual velocity in Fig.2. The speed response is in an acceptable range while guarantees the stability of the system while guarantees the stability of the system.

Commonly, in actual application, the system is always affected by the unknown disturbances, so that in this case, at the time 0.2s, the simulation condition is under an external noise. The purpose of this simulation is to verify the robustness of the controller.

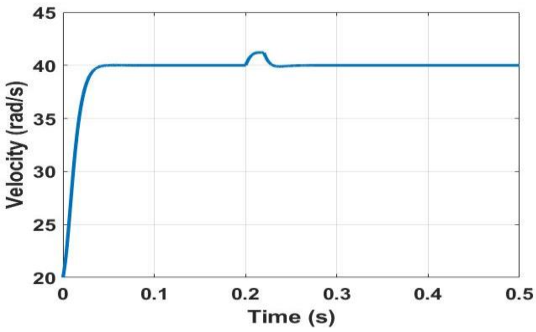

Fig. 3 Actual velocity with noise

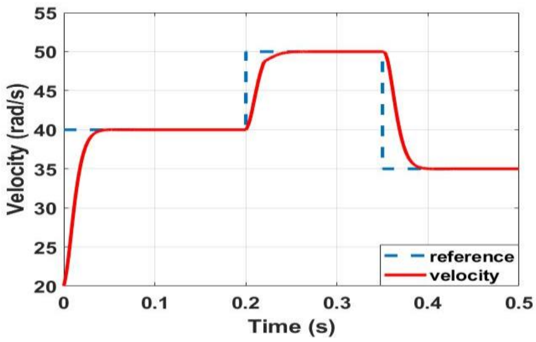

Fig. 4 Velocity response with system disturbance

It can be seen that in Fig. 3 and Fig. 4, the proposed controller has the ability to reject external noise after a short period of time.

Table 1

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

5. Conclusions

In the paper, a backstepping-sliding mode-based control for electro-hydraulic system is introduced. Hydraulic nonlinearities are taken care of using the systematic approach. System uncertainties and disturbances are handled thanks to the existence of sliding mode control. The system is proven to be stable in Lyapunov's sense. Several numerical studies are included to show the effectiveness of the proposed control.

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