ROBUST CONTROL OF HYDRAULIC ACTUATOR USING BACK-STEPPING SLIDING MODE CONTROLLER

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ABSTRACT: To develop an unmanned automatic excavator system the control performance of hydraulic actuators should be guaranteed. However, hydraulic actuators with single rod cylinder have inherently severe nonlinearities that significantly affect to the command following performance of end-effect. PID control system widely used in industries is not proper to compensate the nonlinearities and it is difficult to cancel the unexpected disturbance. In this paper, the robust control scheme to compensate an nonlinearity of hydraulic actuator and cancel an unexpected disturbance is proposed. The proposed control system is designed based on back-stepping controller with sliding model control system. To design the control system, the dynamics of hydraulic actuators is induced and the proposed controller is designed based on the dynamics. In order to evaluate the performance of proposed control system, pressure tracking control system is established and evaluated through experiments.

***Keywords**: Excavator, Hydraulic System, Back-stepping Controller, Sliding Mode Controller*

1. INTRODUCTION

Hydraulic excavators are universal machines working in large open areas due to their high throughput and adaptability to various tasks through replacement of their attachments. Even though the hardware performance of excavators has improved in general, their work efficiency mainly depends on the operator's skill levels. Therefore, an automatic system for excavator has been focused on. For robust tracking control of the hydraulic system in the excavator, control of the hydraulic single-rod cylinder using an electro hydraulic manipulator was studied[1]. A team of Lancaster university have been researched intelligent excavator (LUCIE) in [2] and developed a nonlinear simulation model using MATLAB/ SIMULINK in [3]. Huang et al. presented an impedance control study for a robotic excavator[4]. In order to establish the automatic excavator system, the tracking control system should be designed for the hydraulic actuator. Related researches of hydraulic actuator are to verify and analyze the uncertain factor of hydraulic system. Influence of a small variation of hydraulic oil flow into cylinders was discussed in [5]. Nonlinearities and parameter uncertainties have been discussed in [6][7]. In this sense, lots of hydraulic control systems have been proposed to guarantee the robustness in working environment which has large disturbance[8][9][10]. In order to improve control performance of the hydraulic system, these kinds of nonlinearities and uncertainties should be analyzed and considered to design a control system. In this paper, the bulk modulus is considered in terms of parameter uncertainty and the robust control system is designed to compensate it. The proposed control system is back-stepping control algorithm based on the sliding mode controller and the performance is represented with experiments.

The rest of the paper is organized as follows: The next section describes the configuration of the dynamics of the hydraulic system. In the following section, the back-stepping sliding mode control scheme for hydraulic cylinder system is presented. The performance of the developed controller is evaluated based on experiments. Section 5 concludes the paper.

2. DYNAMICS OF HYDRAULIC SYSTEM

Hydraulic actuator of excavator is operated through flow and pressure control of main control valve(MCV) which is manipulated by electrical joystick. Hydraulic actuator in excavator is usually used single rod hydraulic cylinder. In order to design the robust controller for this hydraulic actuator the back-stepping sliding mode controller is proposed in this paper. To design the back-stepping sliding mode controller the simple single rod hydraulic actuator system is considered as shown in Fig. 1. In Fig. 1 flows Q_1 and Q_2 can be induced by Bernoulli's equation as follows:

$$Q_1 = C_d A_1 \sqrt{\frac{2}{\rho}(p_s - p_1)} \qquad (1)$$

$$Q_2 = C_d A_2 \sqrt{\frac{2}{\rho}(p_2 - p_r)} \qquad (2)$$

where A_1 and A_2 are orifice area of inlet and outlet, x_v is distance of spool, P_s, P_r, P_1 and P_2 are supply, drain and cylinder pressures respectively and ρ is density of hydraulic fluid.

Under the assumption that the flows Q_1 and Q_2 are same the load flow Q_L can be represented as follows:

$$Q_L = \frac{Q_1 + Q_2}{2} = \alpha C_d w x_v \sqrt{\frac{1}{\rho}\left(p_s - \frac{x_v}{|x_v|} p_L\right)} \qquad (3)$$

where

$$\alpha = \frac{(1+\eta)}{2(1+\eta^2)}, \ p_L = p_1 - p_2 \qquad (4)$$

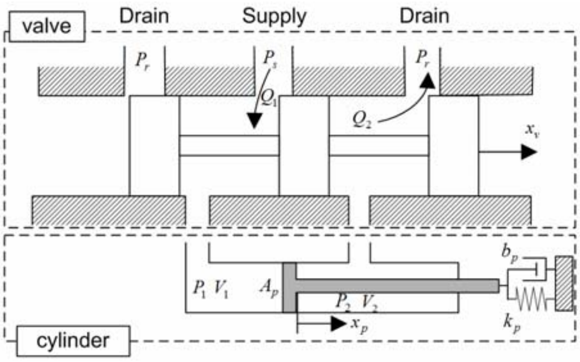

Fig. 1 Single rod hydraulic cylinder system

Linearized equation of Eq. (3) in neutral position, Q_L can be rewritten as follows:

$$Q_L = Q_{L1} + \frac{\partial Q_L}{\partial x_V}\Delta x_v + \frac{\partial Q_L}{\partial p_L}\Delta p_L + H.O.T \qquad (5)$$

Therefore, the ΔQ_L is induced as follows:

$$\Delta Q_L = Q_L - Q_{L1} = \frac{\partial Q_L}{\partial x_V}\Delta x_v + \frac{\partial Q_L}{\partial p_L}\Delta p_L \qquad (6)$$

where $\frac{\partial Q_L}{\partial x_V}$ and $\frac{\partial Q_L}{\partial p_L}$ are valve flow gain and valve flow-pressure coefficient can be written by

$$\frac{\partial Q_L}{\partial x_V} = K_q = \alpha C_d w \sqrt{\frac{1}{\rho}(p_s - p_{L0})},$$

$$\frac{\partial Q_L}{\partial p_L} = K_c = \frac{\alpha C_d w x_v \sqrt{1/\rho}}{2\sqrt{p_s - p_L}}.$$

Therefore, simplified load flow Q_L can be rewritten as follows:

$$\Delta Q_L = K_q \Delta x_v - K_c \Delta p_L \qquad (7)$$

Cylinder dynamics can be induced using inlet outlet flows Q_1, Q_2 and pressures P_1, P_2 in hydraulic cylinder as follows:

$$Q_1 - C_{ip}(p_1 - p_2) - C_{ep} p_1 = \frac{dV_1}{dt} + \frac{V_1}{\beta}\dot{p}_1 \qquad (8)$$

$$C_{ip}(p_1 - p_2) - C_{ep} p_2 - Q_2 = \frac{dV_2}{dt} + \frac{V_2}{\beta}\dot{p}_2 \qquad (9)$$

where β is compressibility of fluid, C_{ip} and C_{ep} are internal and external leakage coefficient.

From Eqs. (7), (8) and (9) the load flow can be rewritten by

$$Q_L = A_p \dot{x}_p + C_{tp} p_L + \frac{V_t}{4\beta}\dot{p}_L \qquad (10)$$

where V_t is total volume of cylinder and $C_{tp} = C_{ip} + \frac{1}{2}C_{ep}$.

Using Eqs. (7) and (10) the pressure dynamics can be induced as shown in Eq. (11)

$$\dot{p}_L = \frac{4\beta}{V_t}\left\{K_q x_v - (K_c + C_{tp})p_L - A_p \dot{x}_p\right\} \tag{11}$$

Also the cylinder dynamics can be represented as follows:

$$F = A_p p_L = M_p \ddot{x}_p + b_p \dot{x}_p + k_p x_p \tag{12}$$

$$\ddot{x}_p = \frac{1}{M_p}\left(A_p p_L - b_p \dot{x}_p - k_p x_p\right) \tag{13}$$

As the state variable $\vec{x} = \begin{bmatrix} x_1 & x_2 & x_3 \end{bmatrix}^T = \begin{bmatrix} x_p & \dot{x}_p & p_L \end{bmatrix}^T$ is defined, the total dynamics can be induced as follows:

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= \frac{A_p}{M_p}x_3 - \frac{k_p}{M_p}x_1 - \frac{b_p}{M_p}x_2 \\ \dot{x}_3 &= -\frac{4\beta(K_c + C_{tp})}{V_t}x_3 + \frac{4\beta K_q K_v}{V_t}u \end{aligned} \tag{15}$$

3. BACK-STEPPING SLIDING MODE CONTROLLER FOR HYDRAULIC SERVO SYSTEM

Hydraulic actuator of excavator is operated by the flow and pressure control of main control valve(MCV) which is manipulated by control electric joystick. In order to design the proposed back-stepping sliding mode controller the system dynamics is represented as follows:

$$\begin{aligned} \dot{x}_1 &= x_2 + f_1(x_1) \\ \dot{x}_2 &= x_3 + f_2(x_1, x_2) \\ &\vdots \\ \dot{x}_n &= f_n(x_1, \cdots, x_n) + u \end{aligned} \tag{16}$$

In order to induce the back stepping SMC, new state z_{s1} and z_{s2} is defined as follows:

$$z_{s1} = x_1 \tag{17}$$

$$z_{s2} = x_2 - \alpha_{s1} \tag{18}$$

where α_{si} is virtual control.

Derivative of z_{s1} can be written by

$$\dot{z}_{s1} = \dot{x}_1 = x_2 + f_1(x_1) \tag{19}$$

In Eq.(19), the x_2 is substituted for using Eq. (18) and the derivative of z_{s1} can be rewritten as follows:

$$\dot{z}_{s1} = z_{s2} + \alpha_{s1} + f_1(x_1) \tag{20}$$

In order to decide the virtual control α_{s1} which makes stabilize the x_1, the sliding surface is defined as follows:

$$s_1 = z_{s1} + \int_0^t z_{s1} dt \tag{21}$$

For the designed sliding surface, the dynamics of sliding surface can be written as Eq. (22) which is to satisfy the reaching condition of sliding motion

$$\dot{s}_1 = \dot{z}_{s1} + z_{s1} = z_{s2} + \alpha_{s1} + f_1(x_1) + z_{s1} \tag{22}$$

To guarantee the sliding motion and satisfy the reaching condition, the virtual control α_{s1} is selected as follows:

$$\alpha_{s1} = -z_{s1} - f_1(x_1) - D_1 s_1 - K_1 \,\mathrm{sgn}(s_1) \tag{23}$$

Through virtual control α_{s1}, derivative sliding surface s_1 can be rewritten by

$$\dot{s}_1 = z_{s2} - D_1 s_1 - K_1 sng(s_1) \tag{24}$$

Also, the derivative of z_{s1} can be rewritten by

$$\dot{z}_{s1} = -z_{s1} + z_{s2} - D_1 s_1 - K_1 \,\mathrm{sgn}(s_1) \tag{25}$$

Recursively, define new state $z_{s3} = x_3 - \alpha_{s2}$. Using the state z_{s3}, the derivative of z_{s2} can be written as follows:

$$\begin{aligned} \dot{z}_{s2} &= \dot{x}_2 - \dot{\alpha}_{s1} = x_3 + f_2(x_1, x_2) - \dot{\alpha}_{s1} \\ &= x_3 + \bar{f}_2(x_1, x_2) \\ &= z_{s3} + \alpha_{s2} + \bar{f}_2(x_1, x_2) \end{aligned} \tag{26}$$

The second sliding surface is selected by

$$s_2 = s_1 + z_{s2} \tag{27}$$

Using Eqs.(24) and (26), its derivative can be induced as follows:

$$\begin{aligned} \dot{s}_2 &= \dot{s}_1 + \dot{z}_{s2} \\ &= z_{s2} - D_1 s_1 - K_1 \operatorname{sgn}(s_1) \\ &\quad + z_{s2}\{z_{s3} + \alpha_{s2} + \bar{f}_2(x_1, x_2)\} \end{aligned} \tag{28}$$

From Eq.(28), the second virtual control law can be selected to satisfy the reaching condition of sliding mode control system.

$$\begin{aligned} \alpha_{s2} &= -1 - z_{s2} - f_2(x_1, x_2) + D_1 s_1 \\ &\quad + K_1 \operatorname{sgn}(s_1) - D_2 s_2 - K_2 \operatorname{sgn}(s_2) \end{aligned} \tag{29}$$

By virtue of Eq. (29), derivatives of s_2 and z_{s2} can be rewritten as follows:

$$\dot{s}_2 = z_{s3} - D_2 s_2 - K_2 \operatorname{sgn}(s_2) \tag{30}$$

$$\begin{aligned} \dot{z}_{s2} &= -1 - z_{s2} + z_{s3} + D_1 s_1 + K_1 \operatorname{sgn}(s_1) \\ &\quad - D_2 s_2 - K_2 \operatorname{sgn}(s_2) \end{aligned} \tag{31}$$

As the same manner, the derivative of i[th] state z_{si} and sliding surface can be written by

$$\begin{aligned} \dot{z}_{si} &= x_i - \alpha_{si-1} = x_{i+1} + f_i(x_1, x_2) - \alpha_{s1-1} \\ &= x_{i+1} + \bar{f}_i(x_1, x_2) \\ &= z_{si+1} + \alpha_{si} + \bar{f}_i(x_1, x_2) \end{aligned} \tag{32}$$

$$s_i = s_{i-1} + z_{si} \tag{33}$$

From the sliding surface and a reaching condition for sliding motion in the i[th] step, the virtual control law can be also selected by

$$\begin{aligned} \alpha_{si} &= -z_{si} - f_i + D_{i-1} s_{i-1} + K_{i-1} \operatorname{sgn}(s_{i-1}) \\ &\quad - D_i s_i - K_i \operatorname{sgn}(s_i) \end{aligned} \tag{34}$$

At the final step, the state z_{sn} and sliding surface s_n are defined as follows:

$$\dot{z}_{sn} = x_n - \alpha_{sn-1} = f_n + u - \alpha_{sn-1} \tag{35}$$

$$s_n = s_{n-1} + z_{sn} \tag{36}$$

Using the reaching condition of n[th] sliding surface, the final control law can be selected as follows:

$$u = \dot{s}_{n-1} - D_n s_n - K_n \operatorname{sgn}(s_n) - \bar{f}_n \tag{37}$$

In order to evaluate the stability of the designed back stepping SMC, the Lyapunov function is defined as shown in Eq.(38) and its derivative is induced as Eq.(39).

$$V_n = \frac{1}{2} s_n^2 \tag{38}$$

$$\dot{V}_n = -D_n s_n^2 - K_n s_n \operatorname{sgn}(s_n) \le 0 \tag{39}$$

At the last step, since the sets of sliding surface include the previous sliding surface, the stability of proposed control scheme is guaranteed by the last Lyapunov function V_n. Also, asymptotically stability is satisfied based on the Lasalle's lemma because the set of sliding surface is invariant set during the sliding motion is executed according to the sliding surface.

4. EXPERIMENTS AND DISCUSS

In hydraulic system there are lots of parameter uncertainties such as fluid density, temperature and bulk modulus. Especially, the bulk modulus has large amount of variation in low pressure range and the related research result is represented in Eq. (37) and Fig. 2[11].

$$\beta = \frac{\beta_0 + \beta_1 p}{1 + x\left(\frac{p_0}{p}\right)^{1/k_b}\left(\frac{\beta_0}{k_b p} - 1\right)} \tag{37}$$

As shown in Fig. 2 the large parameter uncertainty observed in low pressure range in term of bulk modulus. Therefore, the pressure control system in low pressure is established to evaluate the proposed controller. The considered operating pressure range is 1.5~3.5[MPa]. Fig. 3 shows the schematic of the experiment and the main specifications of equipment are as shown in Table 1.

To evaluate the performance of proposed control scheme, a step responses of conventional SMC and proposed back-stepping SMC were compared at 1.5, 2.5 and 3.5 MPa reference signals. Fig. 4 shows the step response for conventional SMC and Fig. 5 shows the response for the proposed control system.

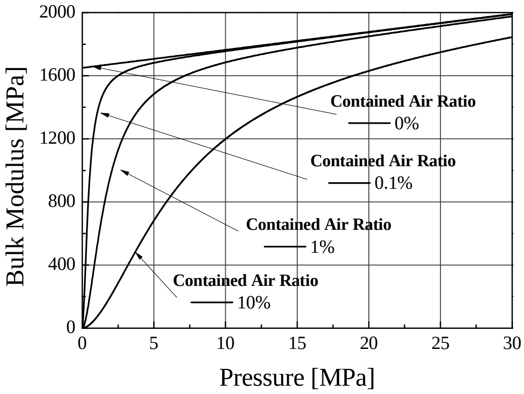

Fig. 2 Influence of entrained air volume with respect to bulk modulus

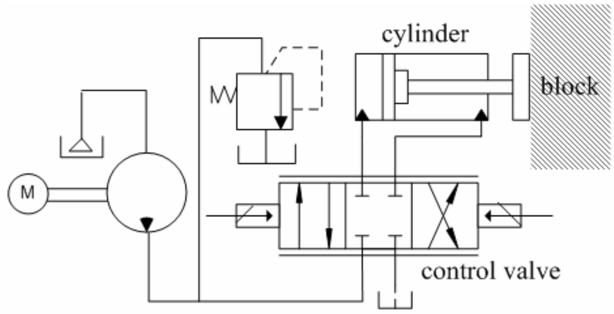

Fig. 3 System configuration

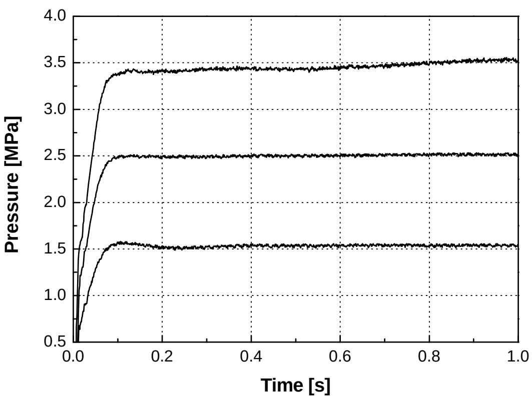

Fig. 4 Step response for the conventional SMC

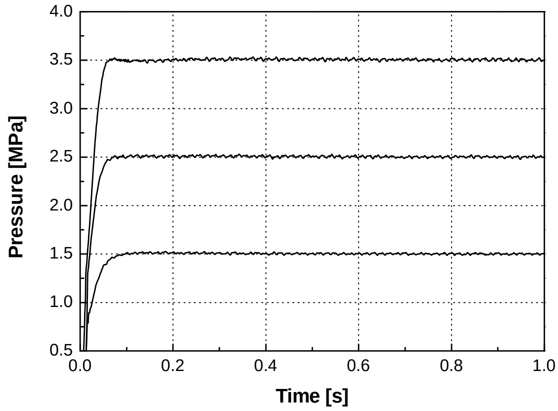

Fig. 5 Step response for the Back-Stepping SMC

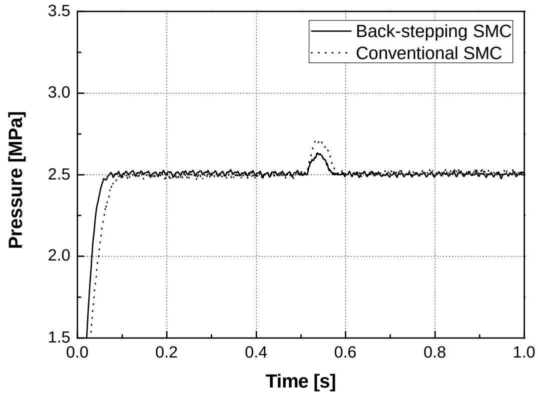

Fig. 6 Performance of disturbance cancellation

Table 1 Specifications of instruments used in the experiment

Instruments	Specification	Manufacturer	Model
Hydraulic Cylinder	4.0MPa Max	TAIYO	35Z-1
Hydraulic Pump	31.5MPa Max	UCD	A10V28
Pressure Sensor	35MPa Max	Omega	PX951
ServoAMP	±7.5 mA	MOOG	J121-001
Servo Valve	1.4~28MPa	MOOG	J076-103
Controller	1kHz sampling		SimulLink

In experiments control gains tuned at the 2.5 MPa reference signal and the step response with 1.5 and 3.5 MPa of the reference signal were analyzed using the same gain. As shown in experimental result it is found that the step response for the conventional SMC has overshoot in low pressure and late settling time in high pressure. In the same manner the control gain tuning for the proposed controller was done and the other step responses were observed as shown in Fig. 5. As compared with Figs. 4 and 5, it is found that the proposed control system has better performance. Also, to evaluate the performance of the disturbance cancellation, an impact applied to the end of hydraulic cylinder at 5 second. Fig. 6 shows the response of the performance of disturbance cancellation. The superior response of the proposed controller can be observed.

5. CONCLUSION

Control system for single rod hydraulic actuator was proposed in this paper. The proposed control system is designed based on the back-stepping controller with SMC. In design procedure of the proposed controller, the SMC control scheme was applied to back-stepping control system. To evaluate the performance of the proposed control system, step responses were compared with a conventional SMC in low pressure area. Also, the performance of disturbance cancellation was compared applying the impact at the end of hydraulic cylinder. Through the experiments it is found that the proposed control has a better performance against parameter uncertainty and unexpected disturbance.

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