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VIRTUAL MODELING AND CONTROLLING OF AN ELECTRO-HYDRAULIC ACTUATOR

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Abstract. Instead of experiment, this paper builds a virtual model of the electro-hydraulic actuator (EHA) thought an Amesim software to evaluate the control response. The main feature of the EHA is to use the closed-loop circuit to reduce the size and oil volume as well as to eliminate the pressure loss caused by the orifice area of the valves. Firstly, the mathematical model of the EHA is established. Secondly, based on this model, an adaptive fuzzy sliding mode controller (AFSMC) is then designed to control the accurate position of the piston. In this control strategy, the system parameters are considered unknown, and they are lumped into two unknown time varying functions. An approximate technique is used to express one of the unknown functions as a finite combination of the basis function. In addition, a fuzzy logic inference mechanism is utilized for realizing a hitting control law to remove completely the chattering problem from the conventional sliding mode control. Then, the Lyapunov stability theorem is utilized to find the adaptive laws for updating the coefficients in the approximate series and turning the fuzzy parameter.
Keywords. Electro-hydraulic actuator, Sliding mode control, Fuzzy controller, Virtual model.

1 INTRODUCTION

Currently, hydrostatic transmission is used widely in the modern industry due to high power, low inertia, reliability and flexibility in changing the transmission ratio as well as high automation. The hydraulic system can be classified including: open-loop and closed-loop circuit. The former is operated through valve controlled system. As known, the pressure drop and leakage are always occurred at the control valves, indicating that with this transmission, the amount of the energy is wasted at the control valves. The latter can be considered as hydraulic transmission without the control valve because the hydraulic actuator is controlled directly by operation of the pump as presented by Cundiff [1]. Hence, closed-loop circuit can offer higher transmission efficiency to obtain high force or torque of the actuator. Based on the merit of the closed-loop circuit, a hydraulic actuator called electro-hydraulic actuator (EHA) was proposed by Altare et al. [2]. The main feature of the EHA is that the power is shifted from the high speed of the electric motor to the high force of the hydraulic cylinder, and the EHAs are considered as force or position generators. Up to now, the EHA has been developed as the commercial products in [3].

In addition, the hydraulic transmission as well as the EHA has strongly nonlinear characteristic and uncertainties. Furthermore, it is not easy to obtain an accurate dynamic model of the system. Moreover, in realistic application, the parameters of this system are difficult to obtain accurately. Hence, it is a challenge for applying the conventional control algorithms to control the position of the actuator. As well known, the sliding model control algorithm is one of useful approaches for solving the nonlinear systems. But the drawback of this control method is to need an accurate dynamic model of the system. In order to solve these disadvantages, some control strategies have been proposed. for example, Guan et al. [4] designed adaptive time-varying sliding control for hydraulic servo system. Shuangxia et al. [5] proposed and experimented successfully an adaptive sliding mode controller for electro-hydraulic system. Richardson et al. [6] used self-tuning control for a low friction pneumatic actuator under the influence of gravity. Acarman et al. [7] proposed a feedback-linearization control strategy with consideration of various status of the chamber pressure in the system model. In addition, Fuzzy control technique is also considered as a good tool for the nonlinear structures such as Earth mitigation structure with MR damper studied by Xu et al. [8] and Tang et al. [9]. Or a robust integral of the signal of the error controller and adaptive controller are synthesized via the backstep method for motion control of a hydraulic rotary actuator as studied by Jao et al. [10]

In this paper, a control algorithm for controlling the position of the EHA is designed. In this controller structure, the system uncertainties are lumped into two unknown time-varying functions. The boundary of one of the unknown functions is not available. An approximate technique is used to express the unknown function as a finite combination of basis function. Moreover, a fuzzy logic inference mechanism is utilized for realizing a hitting control law to remove completely the chattering problem from the conventional sliding mode control. Thereafter, the virtual model is built to assess the control performance of the EHA. The remainder of this paper is organized as follows. Section 2 presents modelling of the EHA. Based on the dynamic model, an adaptive fuzzy sliding mode controller will be designed in section 3. Virtual model and simulation result are presented in section 4. Finally, some conclusions are given in section 5

2 MODELING OF ELECTRO-HYDRAULIC ACTUATOR

As shown in [11] the electro-hydraulic actuator is described in Fig. 1. The main feature of the actuator is to use a closed-loop hydraulic circuit without the directional control valve. Hence, pressure loss caused by the orifice area of valves is reduced. However, due to the symmetric of the hydraulic cylinder, two plot-operated check valves are used to supply the supplement volume of the oil from tank or discharge the oil volume to tank. In addition, a relief valve is used to limit pressure in the system through two check valves without spring. The flow rate and direction of fluid flow of the bidirectional pump are adjusted through an AC servo controlled by an adaptive fuzzy sliding model controller.

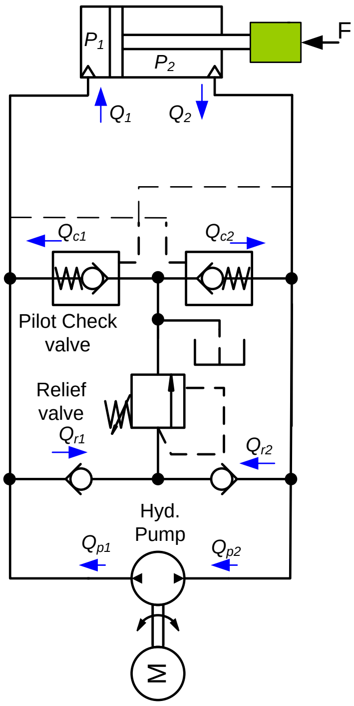


Figure 1: Schematic diagram of an electro-hydraulic actuator

By applying the second Newton's Law: the dynamic of the piston is expressed as follows:

$$M\ddot{x} + c\dot{x} = A_p P_1 - (A_p - a)P_2 - F \tag{1}$$

in which M is the mass in kg; c is the damping coefficient in N.s/m; A_p is the effective area of the piston, a is the area of the rod, both of the areas are in m^2; P_1 and P_2 are pressure in two chambers in Pa as shown in Fig. 1; F is the external loaded force acting on the cylinder in N and x the displacement of the piston

Through basic principles of the hydraulic transmission, the pressure in working chambers is obtained

$$\dot{P}_1 = \frac{\beta}{V_{01} + A_p x}\left(Q_1 - A_p \dot{x} - \frac{1}{R_i}(P_1 - P_2)\right)$$
$$\dot{P}_2 = \frac{\beta}{V_{02} - (A_p - a)x}\left((A_p - a)\dot{x} + \frac{1}{R_i}(P_1 - P_2) - Q_2\right) \tag{2}$$

where, V_{o1} and V_{o2} are the original volumes of two chambers; R_i is the resistance to the internal leakage of the cylinder (i=1,2); Q_1 is the flow rate entering into the chamber 1, Q_2 is the flow rate leaving away the (other) chamber, and they are determined as follows:

$$Q_1 = Q_{p1} + Q_{c1} - Q_{r1}$$
$$Q_2 = \frac{A_p - A_r}{A_p} Q_1 = -Q_{p2} - Q_{c2} + Q_{r1} \tag{3}$$

in which as presented in Fig. 1 the flow rate through the check pilot-operated check valves 1 and 2 is denoted by Q_{c1} and Q_{c2} while the low rate passing the check valves without spring to return the tank is Q_{r1} and Q_{r2}. The flow rate supplied by the bidirectional pump (Q_{p1} and Q_{p2}) is calculated as follows:

$$Q_{p1} = -Q_{p2} = \eta_v D\omega \tag{4}$$

with η_v is volumetric efficiency of the pump, D is the displacement of the pump, and ω is the angular velocity of the pump shaft.

By letting $y_1 = x;\ y_2 = \dot{x};\ y_3 = \ddot{x}$, Eqs. (1-4) are expressed in form of the state space as follows:

$$\begin{cases} \dot{y}_1 = y_2 \\ \dot{y}_2 = y_3 \\ \dot{y}_3 = f(y,t) + g(y,t)\omega + d(t) \end{cases} \tag{5}$$

in which,

$$f(y,t) = -\frac{\beta A}{M(V_{01} + A_p y_1)}\left(Ay_2 + \frac{1}{R_i}(P_1 - P_2)\right) - \frac{\beta(A_p - a)}{M(V_{02} - (A_p - a)y_1)}\left((A_p - a)y_2 + \frac{1}{R_i}(P_1 - P_2)\right) - \frac{c}{M}y_2$$

$$g(y,t) = \left(\frac{\beta\eta_v D A_p}{V_{01} + A_p y_1} + \frac{\beta\eta_v D(A_p - a)}{V_{02} - (A_p - a)y_1}\right)$$

$d(t) = -\dot{F}/\mathrm{M}$ is the external disturbance

y = [y_1, y_2, y_3] is the state vector

It can be observed that Eqs. (5) reveals that the system state can be adjusted through the speed of the bidirectional pump which is driven by an AC servo motor. Next work of this paper is to design a control strategy to control the speed of the pump shaft so that the actual position of the cylinder tracks is as close as possible to the desirable trajectory.

3 DESIGN OF AFSMC

The purpose is to obtain the control law for the AC servo and the adaptive laws for updating the coefficients of the approximate series and turning the fuzzy parameter. The overall scheme of the controller is shown in Fig. 2.

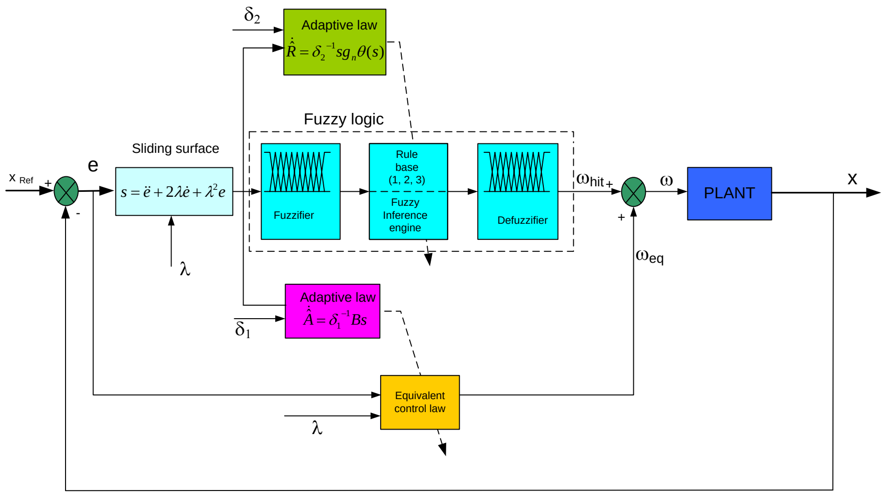


Figure 2: Block diagram of the controller

In order to design the controller, some following assumptions are considered

Assumption 1: *f(y,t)* is the unknown time varying function with the unknown variation bound but it is continuous. Therefore, *f(y,t)* can be approximated by the finite linear combination of the basis function as

$$f(t) = A^T B + \varepsilon \tag{6}$$

where, $A = [a_o, a_1, b_1 \ldots, a_n, b_n]^T$ is the parameter vector, $B = [1, \cos(\omega_1 t), \sin(\omega_1 t), \ldots, \cos(n\omega_1 t), \sin(n\omega_1 t)]^T$ is the basis function vector, $\omega_i = \dfrac{2\pi}{T}$ is the frequency of the basis function, T is the simulation time interval, and ε is the approximate error.

Assumption 2: *g(y,t)* is the unknown function but whose bound is known and estimated as

$$0 < g_{\min} \le g(t) \le g_{\max} \tag{7}$$

By supposing $g(t)=g_n \Delta g$; where, g_n is the known nominal value and Δg is uncertainty satisfying

$$0 < \beta_{\min} = \frac{g_{\min}}{g_n} \le \Delta g \le \frac{g_{\max}}{g_n} = \beta_{\max} \tag{8}$$

Assumption 3: The pressures, p_1, and, p_2, are bounded satisfying $0<p_1,\ p_2 \le P_{max}$, where, P_{max} is the maximum pressure of the pump

Derivation of the AFSMC begins from the definition of the sliding surface as

$$s = \left(\frac{d}{dt} + \lambda\right)^2 e = \ddot{e} + 2\lambda\dot{e} + \lambda^2 e \tag{9}$$

where, λ>0 is the convergent rate of the error on the sliding surface. Here, the error e is different between the zero reference and the real response x_{ref} of the mass

$$e = x_{ref} - x \tag{10}$$

By substituting Eq. (10) into (9) then taking time derivative of s, the dynamic of s is obtained as follows.

$$\dot{s} = -(\dot{y}_3 + \dddot{z}_c) + 2\lambda\ddot{e} + \lambda^2\dot{e} \tag{11}$$

Next, substituting Eq. (6) into (11), the dynamic of the signals can be rewritten as follows.

$$\dot{s}(t) = -A^T B - g_n\omega - L(t) + 2\lambda\ddot{e} + \lambda^2\dot{e} \tag{12}$$

where, $L(t) = g_n(\Delta g - 1)\omega + \varepsilon$ is a lumped function

Without regarding the lumped function *L(t),* to obtain solution of $\dot{s} = 0$, the control action named the equivalent action (u_{eq}) is determined as follows.

$$\omega_{eq} = \frac{1}{g_n}\left[-\hat{A}^T B + 2\lambda\ddot{e} + \lambda^2\dot{e}\right] \tag{13}$$

where, $\hat{A}$ is the estimation parameter vector of the vector *A*. In order to preserve the sliding condition $\left(\dot{V} = \frac{1}{2}\dot{s}^2 < 0\right)$, an auxiliary control action referred as the hitting control action *(u_{hit})* must be added as

$$\omega_{hit} = \eta g_n^{-1} sign(s) \tag{14}$$

where, η >0 is the hitting control gain.

Then, the overall control law is calculated as

$$\omega = \omega_{eq} + \omega_{hit} = \frac{1}{g_n}\left[-\hat{A}^T B + 2\lambda\ddot{e} + \lambda^2\dot{e}\right] + \eta g_n^{-1} sign(s) \tag{15}$$

By substituting Eq. (15) into Eq. (12), the dynamic of sliding surface *s* is rewritten as

$$\dot{s} = \tilde{A}^T B - \eta\,\text{sgn}(s) - L(t) \tag{16}$$

in which $\tilde{A} = \hat{A} - A$ is the estimated error

The Lyapunov function candidate is chosen as

$$V = \frac{1}{2}s^2 + \frac{1}{2}\delta_1\tilde{A}^2 \tag{17}$$

Talking time derivative of *V*, we have

$$\dot{V} = \tilde{A}\left(sB + \delta_1\dot{\hat{A}}\right) - \eta\ \text{s sgn(s)} - sL(t) \tag{18}$$

The adaptive law is selected as

$$\dot{\hat{A}} = -\delta_1^{-1} sB \tag{19}$$

where, δ_1 is the positive constant, $\dot{\tilde{A}} = \dot{\hat{A}}$

Hence, the Eq. (18) remains as follows.

$$\dot{V} = -\eta\ \text{s sgn(s)} - sL(t) \le -\eta\ |\text{s}| + |s||L(t)| = -|\text{s}|\left(\eta - |L(t)|\right) \tag{20}$$

In order to satisfy the stability condition $\left(\dot{V} < 0\right)$, the positive constant η must be satisfied below condition

$$\eta > |L(t)| \tag{21}$$

Eq. (21) reveals that the value of η depends on the upper bound of the function *L(t)*.

As defined in condition (8), It can be seen that although the bound of Δg is estimated but in practical application it is difficult to obtain precisely. Besides, as mentioned in assumption 1, ε is an unknown value. Therefore, the bound of *L(t)* is also difficult to obtain accurately. If the bound of *L(t)* is chosen too large, the hitting control action will cause serious chattering phenomenon, this phenomenon will excite unstable system dynamic, by contrast the bound is chosen too small, the stability condition cannot satisfy. Thus, we consider that the bound of the function *L(t)* is unknown.

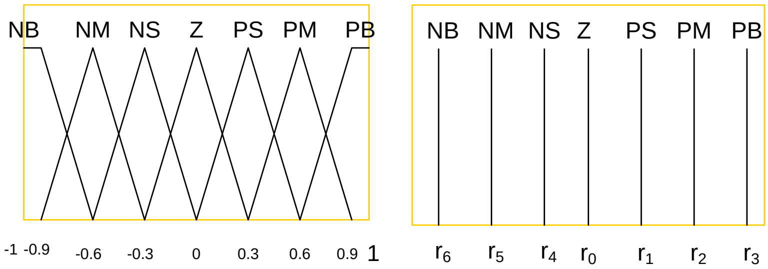
To reduce the influence of the chattering phenomenon, a saturation function is used as

$$\omega_{hit} = \eta\ sat\left(\frac{\text{s}}{\psi}\right) \tag{22}$$

where, ψ is the thickness of the boundary layer.

However, likewise above problem, the positive constant η in Eq. (22) also depends on the bound of *L(t)*. Hence, the stability inside the boundary layer cannot be guaranteed.

For this reason, in this study, a fuzzy logic algorithm is employed to determine the hitting control action (u_{hit}). Here, the sliding surface s is the input linguistic variable of the fuzzy logic, and the hitting control action is the output linguistic variable. Seven linguistic states of the linguistic input and output variable are negative big (NB), negative medium (NM), negative small (NS), zero (Z), positive small (PS), positive medium (PM), and positive big (PB). The membership function for the linguistic input and output variable is shown in Fig. 3.



(a) The input variable (b) The out variable

Figure 3: The membership functions for the input variable S and the output variable u_{hit}

According to the hitting control action given by Eq. (14), the basis laws of the fuzzy system are constructed as follows.

Law 1: If s is the Z then uhit is the Z
Law 2: If s is the PS then uhit is the PS
Law 3: If s is the PM then uhit is the PM
Law 4: If s is the PB then uhit is the PB
Law 5: If s is the NS then uhit is the NS
Law 6: If s is the NM then uhit is the NM
Law 7: If s is the NB then uhit is the NB

Then, the resulting discrete the output variable can be obtained by using the center-average method as

$$\omega_{hit} = \frac{\sum_{i=0}^{6} r_i \mu_i(s)}{\sum_{i=0}^{6} \mu_i} \tag{23}$$

where, $0 \le \mu_i \le 1$ i=0,1,…,6 are the firing strengths of rules 1, 2,…,7. r_0, $r_1, r_2, r_3, r_4, r_5, r_6$ are the center of the membership function Z, PS, PM, PB, NS, NM, and NB of the output variable, respectively. Here, we choose as follows.

$$r_o = 0,\ r_1 = r,\ r_2 = 2r,\ r_3 = 3r,\ r_4 = -r,\ r_5 = -2r,\ r_6 = -3r \tag{24}$$

in which, r is called to be the fuzzy parameter. Due to $\sum_{i=0}^{6} \mu_i = 1$, hence, the Eq. (14) can be rewritten as follows.

$$\omega_{hit} = \sum_{i=0}^{6} r_i \mu_i(s) = r\theta(s) \tag{25}$$

with $\theta(s) = \left[\left(\mu_1(s) + 2\mu_2(s) + 3\mu_3(s) \right) - \left(\mu_4(s) - 2\mu_5(s) - 3\mu_6(s) \right) \right]$

As seen that, when s>0 leads to $\theta(s) = \left(\mu_1(s) + 2\mu_2(s) + 3\mu_3(s) \right) > 0$, oppositely when s<0 leads to $\theta(s) = -\left(\mu_4(s) + 2\mu_5(s) + 3\mu_6(s) \right) < 0$. Therefore, $s\theta(s) > 0$

By replacing u_{hit} in Eq. (14) by Eq.(25), time derivative of the sliding surface is rewritten as

$$\dot{s} = \tilde{A}^T B - g_n r s \theta(s) - L(t) \tag{26}$$

In order to satisfy the sliding condition, the fuzzy parameter r must satisfy following condition

$$r > \frac{|L(t)|}{g_n |\theta(s)|} \tag{27}$$

According to Eq. (27), there exists an optimal value r^* to satisfy the sliding condition

$$r^* = \frac{|L(t)|}{g_n |\theta(s)|} + \beta \tag{28}$$

where, β is the positive constant.

However, the optimal value of r^* is very difficult to determine precisely, so we use a simple adaptive algorithm to estimate the optimal value of r^*. The control law of the AFSMC is presented as

$$\omega = \omega_{eq} + \omega_h = \omega_{eq} + \hat{r}\theta(s) \tag{29}$$

here, $\hat{r}$ is the estimative value of the optimal value r^*

By substituting Eq.(13) into Eq. (29), then it is rearranged as follows:

$$\omega = \omega_{eq} + \omega_h = \frac{1}{g_n}\left[-\hat{A}^T B + 2\lambda\ddot{e} + \lambda^2\dot{e}\right] + \hat{r}\theta(s) \tag{30}$$

Dynamic of s is rewritten as follows:

$$\dot{s} = -A^T B + \hat{A}^T B - g_n\hat{r}\theta(s) - L(t) + g_n r^*\theta(s) - g_n r^*\theta(s) \tag{31}$$

Let us put $\tilde{r} = r^* - \hat{r}$, by rearranging Eq. (31), it can be obtained as

$$\dot{s} = \tilde{A}^T B + g_n\tilde{r}\theta(s) - L(t) - g_n r^* s\theta(s) \tag{32}$$

Now, we definite a Lyapunow candidate function as

$$V = \frac{1}{2}s^2 + \frac{1}{2}\delta_1\tilde{A}^2 + \frac{1}{2}\delta_2\tilde{r}^2 \tag{33}$$

Taking time derivative of V, and using Eq. (33) it obtains as follows:

$$\dot{V} = \tilde{A}\left(sB + \delta_1\dot{\tilde{A}}\right) + \tilde{r}\left(sg_n\theta(s) + \delta_2\dot{\tilde{r}}\right) - s\theta(s)\left(\frac{L(t)}{\theta(s)} + g_n r^*\right) \tag{34}$$

where, $\dot{\tilde{A}} = \dot{\hat{A}}$ and $\dot{\tilde{r}} = -\dot{\hat{r}}$, if the adaptive laws are design as

$$\begin{aligned} \dot{\hat{A}} &= -\delta_1^{-1} sB \\ \dot{\hat{r}} &= \delta_2^{-1} s g_n \theta(s) \end{aligned} \tag{35}$$

with δ_1 and δ_2 are the positive constants

Due to $s\theta(s) > 0$, Eq. (34) is rewritten as follows:

$$\begin{aligned} \dot{V} &= -s\theta(s)\left(\frac{L(t)}{\theta(s)} + g_n r^*\right) \le |s||\theta(s)|\left(\frac{|L(t)|}{|\theta(s)|} - g_n r^*\right) \\ &= |s||\theta(s)|\left(\frac{L(t)}{\theta(s)} - \frac{|L(t)|}{|\theta(s)|} - \beta g_n\right) < 0 \end{aligned} \tag{36}$$

Therefore, the control system is stable according to the Lyapunov stability theory. Based on Barbarlet's lemma [12], the error will asymptotically convergent to zero.

4 VIRTUAL MODEL AND SIMULATION RESULT

4.1 Virtual model

For the purpose of control performance verification of the electro-hydraulic actuator, a virtual model of the EHA is built by using Amesim software as shown in Fig. 4. In this model, two pressure sensors are used to monitor the pressure at two ports of the hydraulic cylinder. The position of the mass is measured by a position sensor, and the signal from this sensor is also sent to the controller to generate control signal for electrical motor. The virtual model of the EHA is embedded in environment of MATLAB/Simulink in which the fuzzy sliding model controller is built as shown in Fig. 5. The parameters of the EHA are listed in Table 1.

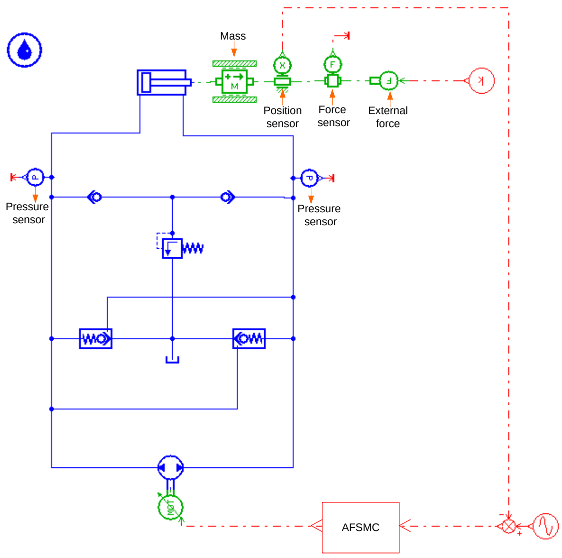


Figure 4: The virtual model of EHA by using Amesim software

Table 1: Setting parameter for the EHA

Components	Parameters	values
Bidirectional pump	Rotational speed	1000 [rpm]
	Displacement	24 [cc/rev]
Hydraulic cylinder	rod	10 [mm]
	piston	20 [mm]
	Stroke of length	100 [mm]
Hydraulic oil	Effective bulk	1.5x10^9 [Pa]
	Specific gravity	0.87
Relief valve	Cracking pressure	50 [Pa]

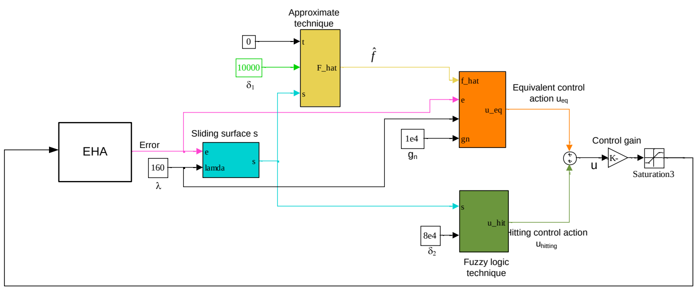


Figure 5: The overall control scheme

4.2 Simulation result

The desirable trajectory is s sinusoidal signal with the amplitude of 30 mm and frequency of 0.1 Hz. The external force shown in Fig. 6 is also sinusoidal form with amplitude of 10 kN and the same frequency as the desirable signal, meaning that when piston extends, the external load is the pushed force meanwhile it becomes the pulled force as the piston retracts. At the ends of the piston stroke, the load force is equal to zero. By using the proposed controller, the position of the mass is shown in Fig. 7, the detailed annotation of the types of responses is presented in upper-right corner of the figure. It can be observed that the actual position of the piston follows smoothly and closely the reference with state error of the control system which is bounded within 1 mm as shown in Fig. 8. In addition, at the beginning, the position of the mass can track the reference. The pressure at two ports of the hydraulic cylinder is obtained as in Fig. 9, seeing notations of the types of lines in upper-right corner of the figure. It is evident that with this condition load, there is a phase difference between the pressure P_1 and P_2, indicating that $P_1>0$, $P_2=0$ for the extension of the piston, reversely $P_1 = 0$, $P_2 > 0$ for the retraction of which. Besides, the pressure P_2 is always larger than the pressure P_1 due to the effective area of the chamber 2 is always smaller than that of the chamber 1.

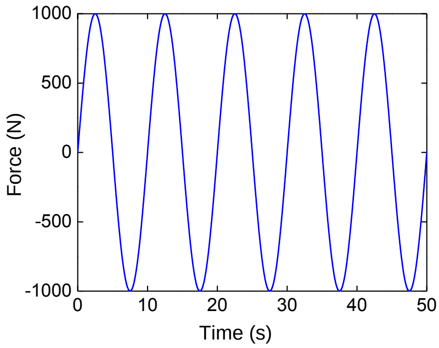


Figure 6: Load force versus time.

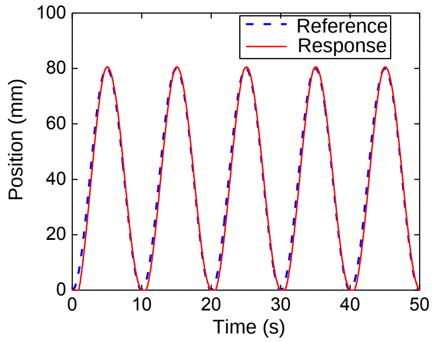


Figure 7: Comparison between the reference and the actual position of the mass

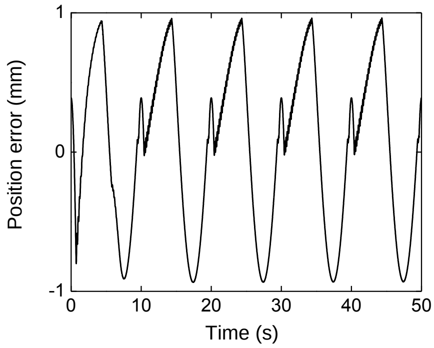


Figure 8: Error between the actual and reference position of the mass

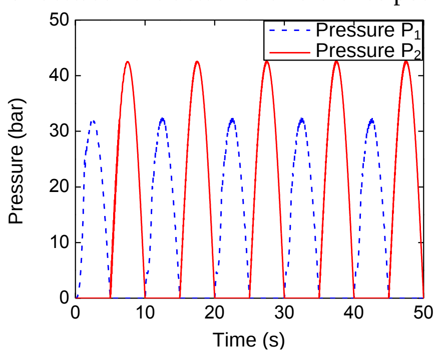


Figure 9: The pressure state of the ports of the hydraulic

5 CONCLUSIONS

In this study, an adaptive fuzzy sliding mode controller was designed and successfully employed in nonlinear EHA with uncertainties. This control strategy used the approximate technique to express the unknown function as a finite combination of the basis function and the fuzzy logic technique to determine the hitting control action. The control structure was designed by selecting a special Lyapunov function meanwhile all uncertain terms were adapted by selecting another Lyapunov function. Then, the virtual model of the EHA was built to simulate the control response of the EHA. The simulation results confirmed that the adaptive FSMC can attain accurate position response.

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MÔ HÌNH ẢO VÀ ĐIỀU KHIỂN CƠ CẤU CHẤP HÀNH ĐIỆN THỦY LỰC

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Tóm tắt. Thay vì thực nghiệm, bài báo này xây dựng mô hình ảo của một cơ cấu chấp hành điện thủy lực (EHA) thông qua phần mềm Amesim để đánh giá đáp ứng điều khiển của hệ. Đặc điểm chính của EHA là sử dụng mạch thủy lực vòng kính để giảm kích thước cũng như thể tích dầu và loại bỏ hiện tượng mất áp cục bộ được gây ra tại van phân phối. Đầu tiên, mô hình toán của hệ được xây dựng. Sau đó, một bộ điều khiển trượt mờ thích nghi được thiết kế để điều khiển vị trí của piston. Luật thích nghi của bộ điều khiển được xác định thông qua tiêu chuẩn Lyapunov.

Từ khóa. Cơ cấu chấp hành điện thủy lực, Điều khiển trượt, Điều khiển mờ, Mô hình ảo

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