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ACTIVE DISTURBANCE REJECTION CONTROL (ADRC) OF ELECTRO-HYDRAULIC SERVO SYSTEM

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LADRC (Linear Active Disturbance Rejection Control) is a control method based on real-time estimation and compensation of internal and external disturbance of the system. The basic idea is to set up a dynamic and real-time interference observer to estimate all kinds of interference to the system, and use these estimates to adjust the output of the controller in real time, so as to suppress the ability of the system to be affected by interference and realize the accurate control of the system.

Keywords: electro-hydraulic servo, force control, modeling, simulation, ADRC

INTRODUCTION

The core concept of active disturbance rejection control(ADRC) is to actively observe and estimate various disturbances to the system, and incorporate these disturbances into the controller for real-time compensation [1]. Compared with the traditional PID control method, the active disturbance rejection control can improve the control precision of the system by arranging the transition process and disturbance compensation.

Linear ADRC are mainly based on Track differentiator (TD), Liner extend state observe, (LESO) and Liner state error feedbackcontrol law (LESF) [2], the structure of which is shown in Figure 1.

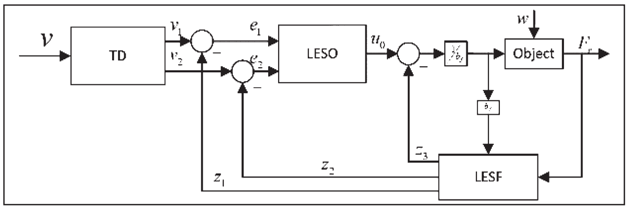

Figure1 LADRC basic structure

System description and modeling

Its hydraulic principle diagram is shown in Figure 2, in which: 1 is the oil tank; 2 is a quantitative pump; 3 is the motor; 4 is the relief valve; 5 is the servo valve; 6 is a hydraulic cylinder; 7 is the load; 8 Power the sensor.

The force balance equation can be expressed as:

$$p_1 A_1 - p_2 A_2 = m\ddot{x}_p + B_p \dot{x}_p + K_S x_p + f \qquad (1)$$

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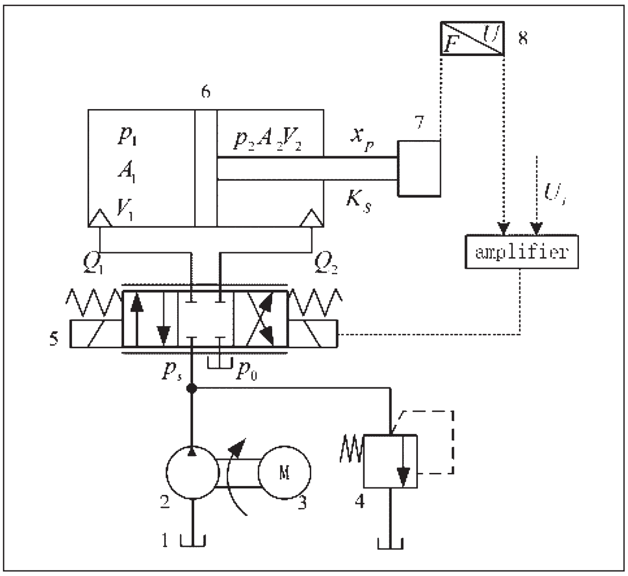

Figure 2 Hydraulic principle diagram

Where, p_1 is Hydraulic cylinder without rod cavity pressure; p_2 is Hydraulic cylinder has rod cavity pressure; A_1 is Effective area of rodless chamber of hydraulic cylinder; A_2 is Hydraulic cylinder has effective area of rod cavity; m is Equivalent load mass of moving parts; B_p is Equivalent load damping coefficient; x_p is Displacement of piston rod; K_s is Stiffness coefficient of the loaded spring; f is Unmodeled interference forces including friction and external unknown interference forces. Regardless of external leakage factors, hydraulic cylinder flow continuity equation:

$$\begin{cases} Q_1 = A_1\dot{x}_p + C_{ip}\left(p_1 - p_2\right) + \beta_e^{-1} V_1 \dot{p}_1 \\ Q_2 = A_2\dot{x}_p + C_{ip}\left(p_1 - p_2\right) - \beta_e^{-1} V_2 \dot{p}_2 \end{cases} \qquad (2)$$

Where, Q_1 is Flow into the rodless chamber of the hydraulic cylinder; Q_2 is Flow out of the rod cavity of the hydraulic cylinder; V_1 is Volume of rodless chamber of hydraulic cylinder, $V_1 = V_{O1} + A_1 x_p$; V_2 is The hydraulic cylinder has the volume of the rod cavity, $V_2 = V_{O2} + A_2 x_p$; V_{O1} is Initial volume of rodless chamber of hydraulic cylinder; V_{O2} is Hydraulic cylinder has initial volume of rod cavity; C_{ip} is Hydraulic cylinder leakage coefficient; β_e is Hydraulic oil elastic modulus.

At the same time, Q_1, Q_2,can also be expressed as a function of the servo valve spool displacement x_v:

$$\begin{cases} Q_1 = C_d w\sqrt{2/\rho}x_v\left[s(x_v)\sqrt{p_s - p_1} + s(-x_v)\sqrt{p_1 - p_0}\right] \\ Q_2 = C_d w\sqrt{2/\rho}x_v\left[s(x_v)\sqrt{p_2 - p_0} + s(-x_v)\sqrt{p_s - p_2}\right] \end{cases} \tag{3}$$

Where, C_d is Valve port flow coefficient; w is Valve port area gradient; ρ is Hydraulic oil density; p_s is Fuel supply pressure; p_0 is Oil return pressure; $s(x_v)$ is Symbolic function of spool displacement.

$$s(x_v) = \begin{cases} 1 & x_v \geq 0 \\ 0 & x_v < 0 \end{cases} \tag{4}$$

Both the amplifier and the force sensor can be regarded as proportional links, respectively:

$$i = K_q u \tag{5}$$

$$u_f = K_F F \tag{6}$$

Where, i is Amplifier output current; u is amplifier input voltage; K_q is flow gain of the amplifier; u_f is voltage signal from force sensor feedback; K_F is gain of the force sensor; F is Output of force sensor feedback. Ignoring the elastic deformation of the hydraulic cylinder piston and the force sensor, the loading part is simplified into a pure spring mechanism. Therefore, the force applied to the spring is proportional to the displacement of the hydraulic cylinder, that is

$$F = K_S x_P \tag{7}$$

Formula(1) can be written as:

$$\ddot{F} = -\frac{B_P}{m}\dot{F} - \frac{K_S}{m}F + \frac{K_S}{m}\left(p_1 A_1 - p_2 A_2 - f\right) \tag{8}$$

Without considering leakage, formula (2) can be written as:

$$\begin{cases} \dot{p}_1 = \dfrac{\beta_e}{K_S V_{O1} + A_1 F}\left(K_S Q_1 - A_1 \dot{F}\right) \\ \dot{p}_2 = \dfrac{\beta_e}{K_S V_{O2} - A_2 F}\left(A_2 \dot{F} - K_S Q_2\right) \end{cases} \tag{9}$$

Ignore the non-linear factor of the servo valve dead zone, and approximate the servo valve flow and drive current as proportional links:

$$\begin{cases} Q_1 = KK_p u\left[s(\dot{x}_p) + \alpha s(-\dot{x}_p)\right] \\ Q_2 = KK_p u\left[\dfrac{1}{\alpha}(\dot{x}_p) + s(-\dot{x}_p)\right] \end{cases} \tag{10}$$

Where, K is servo valve flow and drive current gain; α is the ratio of the effective area of the two chambers of the hydraulic cylinder, $\alpha = A_1/A_2$; $s(\dot{x}_p)$ is symbolic function of piston rod displacement of hydraulic cylinder:

$$s(\dot{x}_p) = \begin{cases} 1 & x_p \geq 0 \\ 0 & x_p < 0 \end{cases} \tag{11}$$

Differential equation (8) and substitute equation (9) and equation (10) into the

system model can be obtained as [3]:

$$\dddot{F} = a_3 \ddot{F} + a_2 g_1(F)\dot{F} + a_2 \dot{f} - a_2 g_2(F) u \tag{12}$$

Where, $a_3 = -B_p/m$; $a_2 = -K_S/m$;

$$g_1(F) = \frac{\beta_e A_1^2}{K_S V_{O1} + A_1 F} + \frac{\beta_e A_2^2}{K_S V_{O2} - A_2 F} + 1;$$

$$g_2(F) = \frac{\beta_e A_1 K_S K K_P R_1}{K_S V_{O1} + A_1 F} + \frac{\beta_e A_2 K_S K K_P R_2}{K_S V_{O2} - A_2 F};$$

$$R_1 = \left[s(\dot{x}_p) + \alpha s(-\dot{x}_p)\right];$$

$$R_2 = \left[\frac{1}{\alpha}s(\dot{x}_p) + s(-\dot{x}_p)\right].$$

Active disturbance rejection controller design

The principle of tracking differentiator is as follows: Input a signal $v(t)$ to TD, it will output two signals v_1 and v_2, among them, v_1 track the input signal $v(t)$; $v_2 = \dot{v}_1$, which v_2 is the approximate differential of $v(t)$. For discrete systems:

$$\begin{cases} x_1(k+1) = x_1(k) + h x_2(k) \\ x_2(k+1) = x_2(k) - r u(k), \left|u(k)\right| \leq r \end{cases} \tag{13}$$

Where, k is Sampling time.

The structure for establishing the fastest discrete differential tracker is as follows:

$$\begin{cases} u = fhan\left[v(k), v_1(k), v_2(k), r, h\right] \\ v_1(k+1) = v_1(k) + v_2(k)\cdot T \\ v_2(k+1) = v_2(k) + u\cdot T \end{cases} \tag{14}$$

Where, $v_1(k)$ is the tracking signal of the input signal; $v_2(k)$ is Differential signal of the input signal; T is sampling period; r is speed factor; h is filter factor; $fhan(v_1, v_2, r, h)$ is the fastest control synthesis function, its formula is as follows[4]:

$$\begin{cases} d = rh \\ d_0 = dh \\ y = v_1 - v + v_2 h \\ a_0 = \sqrt{d^2 + 8r\left|y\right|} \\ a = \begin{cases} v_2 + \dfrac{a_0 - d}{2} sign(y), \left|y\right| > d_0 \\ v_2 + \dfrac{y}{h}, \left|y\right| \leq d_0 \end{cases} \\ u = \begin{cases} -r sign(a), \left|a\right| > d \\ -r\dfrac{a}{d}, \left|a\right| \leq d \end{cases} \end{cases} \tag{15}$$

According to the dynamic model of electro-hydraulic servo system described in equation (8), the term of order 3 is regarded as disturbance, the relative order of the system is taken as 2, and equation (8) is rewritten as:

$$\ddot{F} = w(\cdot) + b_f u \tag{16}$$

Where: $w(\cdot)=\dfrac{\dddot{F}-a_2 g_1(F)\dot{F}-a_2\dot{f}+a_2 g_2(F)u-a_3 b_f u}{a_3}$

is the total disturbance of the system including external interference and unknown error of the servo system. The state space equation of the system can be obtained by the dynamics equation of the system, take the state variable $x = [x_1, x_2, x_3]^T$, defined by equation (1) force balance equation of the state variable x_1 = F, $x_2 = \dot{F}$, $x_3 = w(\cdot)$, then the state space expression of the system is as follows:

$$\begin{cases}\dot{x}=\boldsymbol{A}x+\boldsymbol{B}u+\boldsymbol{E}h\\ F=\boldsymbol{C}x\end{cases} \tag{17}$$

Where: $\boldsymbol{A}=\begin{pmatrix}0&1&0\\0&0&1\\0&0&0\end{pmatrix}, \boldsymbol{B}=\begin{pmatrix}0\\b_f\\0\end{pmatrix}, \boldsymbol{E}=\begin{pmatrix}0\\0\\1\end{pmatrix}, \boldsymbol{C}=\begin{pmatrix}1\\0\\0\end{pmatrix}^T$,

$h = w(\cdot)$.

According to the state space expression of the system in equation (17), the extended state of the observer is introduced, then the original system is changed from the original second-order system to the third-order system. The design of the extended state observer can be expressed as:

$$\begin{cases}\dot{\boldsymbol{z}}=\boldsymbol{A}z+b_f\boldsymbol{B}u+\boldsymbol{L}\left(F-z_1\right)\\ F=\boldsymbol{C}\boldsymbol{z}\end{cases} \tag{18}$$

Where: $\dot{\boldsymbol{z}}$ is the observer vector, $z_i \to x_i (i = 1,2,3)$; $\boldsymbol{L} = (\beta_1, \beta_2, \beta_3)$ is the observer error feedback gain matrix. The observation error matrix can be obtained from formula (17) and formula (18) as follows[5]:

$$\dot{\boldsymbol{e}}=\boldsymbol{A}_e\boldsymbol{e}+\boldsymbol{E}h \tag{19}$$

Where: $\boldsymbol{A}_e=\boldsymbol{A}-\boldsymbol{LC}=\begin{pmatrix}-\beta_1&1&0\\-\beta_2&0&1\\-\beta_3&0&0\end{pmatrix}$

The design of linear state error feedback control law can be simplified to PD combination. For the second-order system, the PD controller available for the LADRC is:

$$u_0=k_p\left(v-z_1\right)-k_d z_2 \tag{20}$$

Where: v is system input value; k_p is scale factor; k_d is differential coefficient.

Modeling and simulation analysis of active disturbance rejection control system

In order to verify the strong robustness of the auto-disturbance rejection control algorithm to unknown disturbance, build a simulink model as Figure 3.

A white noise signal with PSD of 0,1 is applied to the controlled model as a random load disturbance, the amplitude of the input step signal is set to 1, the step time is 1s, and the simulation duration is 10s. The simulation results are shown in Figure 4.

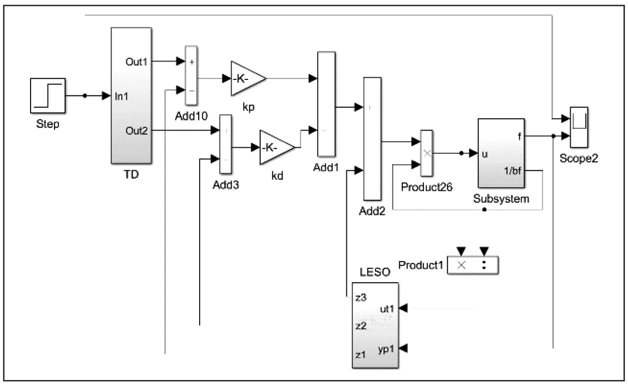

Figure 3 Simulink model

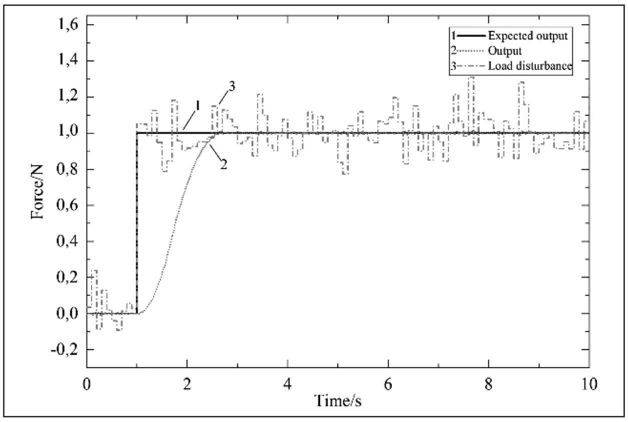

Figure 4 Active disturbance rejection step response

CONCLUSION

This paper takes electro-hydraulic servo force control system as the research object, establishes the linear active disturbance rejection controller and expounds the design process of the controller, and then uses MATLAB simulation platform to carry out experiments. The results show that the active disturbance rejection control can estimate and cancel the unknown disturbance in real time through the observer, and has better robustness and fast response characteristics for the existing disturbance in the system, and has strong adaptability, and can be applied to the control of the system under different working conditions.

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Note: The responsible translator for English language is T. X. Zhang - North China University of Science and Technology, China.