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Self-tuning PID Controller for Torque Tracking Control of Electrohydraulic Servo Systems

Lie Yu[1, 2], Jianbin Zheng[1, 2], Yang Wang[1, 2, *], Qiuzhi Song[3] and Enqi Zhan[1, 2]

[1]School of Information Engineering, Wuhan University of Technology, Wuhan, China
[2]Key Laboratory of Fiber Optic Sensing Technology and Information Processing, Ministry of Education, Wuhan University of Technology, Wuhan, China
[3]School of Mechatronical Engineering, Beijing Institute of Technology, Beijing, China
E-mail: powerflow@whut.edu.cn

Abstract

Electrohydraulic servo systems (EHSS) have been widely used in industry and play an important role in the engineering applications. Proportional-integral-differential (PID) controller is utilized extensively to control the EHSS, but the tracking performance is limited using this approach, due to the nonlinear dynamics of EHSS. In this paper, we propose a self-tuning PID controller to improve the tracking performance of EHSS which is used to drive the One-Degree-Of-Freedom (1 DOF) rotary system. The gains of the proposed controller are adjustable parameters which are designed to vary within a predetermined range to eliminate the problems caused by the conventional PID controller. The simulation results show that the proposed self-tuning PID controller gives better tracking performances, such as less tracking lag and less mean absolute error (MAE), when compared with the conventional PID controller.

Keywords*: Electrohydraulic Servo Systems; Tracking Performance; One-Degree-Of-Freedom Rotary System; Conventional PID Controller; Self-tuning PID Controller*

1. Introduction

Electrohydraulic servo systems (EHSS) have been widely used for their ability of high load efficiency, fast response, high tracking accuracy and big ratio of power to weight [1]. Typical applications of EHSS include turbine control, commercial aircraft, control of industrial robots, active suspension and automated manufacturing systems [2].

Based on the desired objectives, control strategies for EHSS are mainly classified as position control [3-6], velocity control [7] and force/torque control [8-9]. As to any one types of the control strategies, the controller selection will directly effect the tracking performance of EHSS. Generally, the conventional PID controller is most widely used due to its simple control structure, ease of design and low cost. However, large gains of the conventional PID controller result in undesirable oscillations, while small gains lead to big tracking lags and errors [10].

To overcome these drawbacks, we propose a self-tuning PID controller which takes use of adjustable PID gains. In addition, the adjustable PID gains vary within a predetermined range. The simulation results show that the proposed self-tuning PID controller provides less tracking lag and less mean absolute error (MAE) than the conventional PID controller.

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2. Modeling of the Electrohydraulic Servo Systems

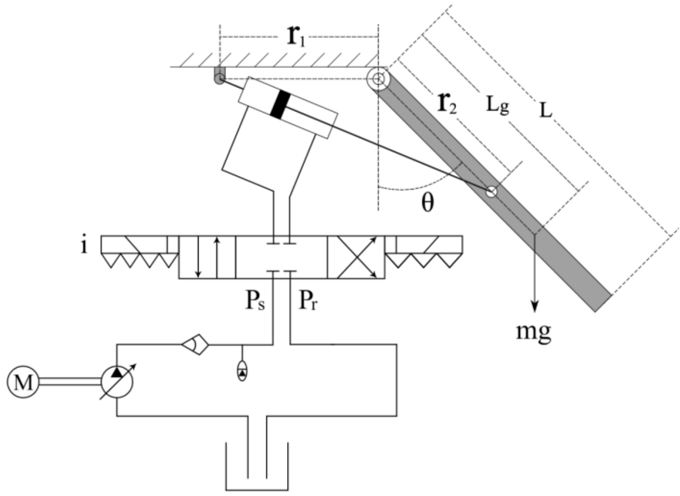

Figure 1. Schematic Diagram of the Electrohydraulic Servo System

As shown in Figure 1, the left part is the EHSS which mainly consists of a hydraulic and a servo valve, while the right part is the One-Degree-Of-Freedom (1 DOF) rotary system drive by the EHSS. Neglecting the frictional force, the dynamics of 1 DOF rotary system is considered as follows.

$$J\ddot{\theta} = T_L - mgL_g \sin(\theta) \tag{1}$$

where J is the rotary inertia of the load; θ is the rotary angle; T_L is the actuated torque; m is the load mass; g is the acceleration due to gravity; L_g is the position of the center of the load mass.

However, the actuated torque T_L can be acquired as

$$\begin{cases} T_L = (P_1 A_{p1} - P_2 A_{p2})H \\ H = \dfrac{r_1 r_2 \cos(\theta)}{\sqrt{r_1^2 + r_2^2 + 2r_1 r_2 \sin(\theta)}} \end{cases} \tag{2}$$

where P_1 is the head-side pressure; P_2 is the rod-side pressure; A_{p1} is the head-side area; A_{P2} is the rod-side area; r_1 and r_2 are the geometric length of the system as drawn in Figure 1.

The A_{p1} and A_{P2} can be calculated through the following formulas when the bore diameter D_1 and the rod diameter D_2 are acquired.

$$\begin{cases} A_{p1} = \dfrac{\pi D_1^2}{4} \\ A_{p2} = \dfrac{\pi (D_1^2 - D_2^2)}{4} \end{cases} \tag{3}$$

As the the rod diameter D_2 is even less than the bore diameter D_1, the Equation (3) can be simplified as

$$A_{p2} = \frac{\pi (D_1^2 - D_2^2)}{4} \approx \frac{\pi D_1^2}{4} = A_{p1} \tag{4}$$

Substituting the Equation (2) and Equation (4) into Equation. (1), the system model can be simplified as

$$J\ddot{\theta} = P_L A_{p1} H - mgL_g \sin(\theta) \tag{5}$$

where P_L=P_1-P_2 is the load pressure of the dynamic actuator.

As referred to [11-12], the external leakage of EHSS is extremely small and can be neglected. Then, the pressure dynamics of EHSS in both hydraulic chambers can be described as

$$\begin{cases} \dot{P}_1 = \dfrac{\beta}{V_1}(-A_{p1}v_p - C_t P_L + Q_1) \\ \dot{P}_2 = \dfrac{\beta}{V_2}(A_{p1}v_p + C_t P_L - Q_2) \end{cases} \tag{6}$$

where β is the effective bulk modulus in the hydraulic chambers; V_1 and V_2 are the hydraulic chambers volumes and can be computed as $V_1=V_0+A_{p1}x_p$, $V_2=V_0-A_{p1}x_p$; V_0 is the chamber volume on the conditions that $x_p=0$, $V_1=V_2=V_0$ while x_p is the hydraulic piston velocity; C_t is the coefficient of the total internal hydraulic leakage; Q_1 is the supplied flow rate of the forward chamber, while Q_2 is the return flow rate of the return chamber. The Q_1 and Q_2 correlate with the servo-valve spool displacement x_v. Their relationships can be stated as

$$\begin{cases} Q_1 = k_q x_v \left[s(x_v)\sqrt{P_s - P_1} + s(-x_v)\sqrt{P_1 - P_r} \right] \\ Q_2 = k_q x_v \left[s(x_v)\sqrt{P_2 - P_r} + s(-x_v)\sqrt{P_s - P_1} \right] \end{cases} \tag{7}$$

where P_s is the supply pressure of the fluid and P_r is the return pressure. The k_q is the valve discharge gain which can be described as

$$k_q = C_d w \sqrt{\frac{2}{\rho}} \tag{8}$$

where C_d is the discharge coefficient, w is the spool valve area gradient and ρ is the density of hydraulic oil.

For Equation. (7), the function $s(x_v)$ is defined as

$$s(x_v) = \begin{cases} 1, & x_v \geq 0 \\ 0, & x_v < 0 \end{cases} \tag{9}$$

However, due to the geometry structure, x_p and v_p can be calculated as

$$\begin{cases} x_p = \sqrt{r_1^2 + r_2^2 + 2r_1 r_2 \sin(\theta)} - L_0 - x_{p0} \\ v_p = \dfrac{2r_1 r_2 \dot{\theta}\cos(\theta)}{\sqrt{r_1^2 + r_2^2 + 2r_1 r_2 \sin(\theta)}} \end{cases} \tag{10}$$

where L_0 is the cylinder dead length and x_{p0} is the piston position when the volumes are equal on both cylinder sides.

Generally, the input current (i) to the servo valve is directly proportional to the servo-valve spool position x_v such that it can be described as

$$x_v = k_c i \tag{11}$$

where k_c is the positive electrical constant. Based on Equation (11), the function $s(x_v)$ can be equivalent as

$$s(x_v) = s(i) \tag{12}$$

Then, the Equation (7) can be rewritten as

$$\begin{cases} Q_1 = g_s R_1 i \\ Q_2 = g_s R_2 i \end{cases} \tag{13}$$

where g_s is an integral parameter which can be describe as

$$g_s = k_q k_c \tag{14}$$

Meanwhile, the R_1 and R_2 in Equation (13) are defined as

$$\begin{cases} R_1 = s(i)\sqrt{P_s - P_1} + s(-i)\sqrt{P_1 - P_r} \\ R_2 = s(i)\sqrt{P_2 - P_r} + s(-i)\sqrt{P_s - P_2} \end{cases} \tag{15}$$

Based on the Equation (6) and Equation (13), the derivative of hydraulic load pressure can be rewritten as

$$\dot{P}_L = \dot{P}_1 - \dot{P}_2 = (\frac{R_1}{V_1} + \frac{R_2}{V_2})\beta g_s i - (\frac{1}{V_1} + \frac{1}{V_2})(\beta C_t P_L + A_{p1} v_p) \tag{16}$$

Therefore, the time derivative of Equation (2) can be obtained as

$$\dot{T}_L = A_{p1}[(\frac{R_1}{V_1} + \frac{R_2}{V_2})\beta g_s i - (\frac{1}{V_1} + \frac{1}{V_2})(\beta C_t P_L + A_{p1} v_p)] \frac{r_1 r_2 \cos(\theta)}{\sqrt{r_1^2 + r_2^2 + 2 r_1 r_2 \sin(\theta)}} \tag{17}$$

Before the controller design, the assumption must be followed that P_1 and P_2 are both bounded by P_s and P_r, *i.e.* $0<P_r<P_1<P_s$ and $0<P_r<P_2<P_s$.

3. Controller Design

As the controller is designed to evaluate the torque tracking performance of EHSS, the actuated actual torque and the desired torque should be obtained before the controller design. In Eq. (13), the derivative of the actuated torque T_L is acquired. However, the desired torque T_d is not given to be a step or sinusoidal signal. In this paper, the system input is the rotary angle θ which is designed to be a sinusoidal signal.

$$\theta(t) = \sin(2\pi t) \tag{18}$$

Therefore, the desired torque T_d can be calculated using the Lagrange equation.

$$T_d(t) = (mL_g^2 + J)\ddot{\theta}(t) - mgL_g \sin(\theta(t)) \tag{19}$$

As described in [17], T_d can also be named as computed virtual torque. Then, the controller design can be made at a sampling frequency of $f_s=1000Hz$ to examine the behavior of the system.

3.1. Design of the Conventional PID Controller

The conventional PID controller can be constructed as follows.

$$i(t) = k_p e(t) + k_i \int e(t)dt + k_d \frac{de(t)}{dt} \tag{20}$$

where k_p, k_i and k_d are the fixed proportional, integral and differential gains, respectively. $e(t)=T_d(t)-T_L(t)$ is the error between the desired torque and actuated torque of the system.

3.2. Design of the Self-tuning PID Controller

The design of the proposed controller is due to its gains which can be varied in changeable operating situations (*i.e.*, the variation of torque error). When the torque error is large, the proportional gain can be selected to be a high value for better control effort. Similarly, when the torque error is small, the integral gain can be chosen to be a high value to overcome steady state error, and the differential gain can be also selected to be a high value for better system stability. The control law of the self-tuning PID controller can be written as:

$$\begin{cases} i(t) = k_p(t)e(t) + k_i(t)\int e(t)dt + k_d(t)\dfrac{de(t)}{dt} \\ k_p(t) = k_{p\max} f(t) + k_{p\min} g(t) \\ k_i(t) = k_{i\max} g(t) + k_{i\min} f(t) \\ k_d(t) = k_{d\max} g(t) + k_{d\min} f(t) \end{cases} \tag{21}$$

where

$$\begin{cases} f(t) = \dfrac{e^{ke(t)} - e^{-ke(t)}}{e^{ke(t)} + e^{-ke(t)}} \\ g(t) = 1 - f(t) \end{cases} \tag{22}$$

where k is a constant to determines the rate of $k_p(t)$, $k_i(t)$ and $k_d(t)$ varying between the maximum and minimum values. When the torque error $e(t)$ is large, the function $f(t)$ approaches 1 ($f(t)\rightarrow 1$) and the function $g(t)$ approaches zero ($g(t)\rightarrow 0$). Then, the proportional gain $k_p(t)$ approaches its maximum value $k_{p\max}$ ($k_p(t)\rightarrow k_{p\max}$) which is used to speed up the transient response and reduce the tracking lag. The integral gain $k_i(t)$ approaches its minimum value $k_{i\min}$ ($k_i(t)\rightarrow k_{i\min}$) which is used to eliminate the undesirable oscillations. The differential gain $k_d(t)$ approaches its minimum value $k_{d\min}$ ($k_d(t)\rightarrow k_{d\min}$) which is used to reduce the rise time and tracking lag.

Similarly, when the torque error $e(t)$ is small, the function $f(t)$ approaches zero ($f(t)\rightarrow 0$) and the function $g(t)$ approaches 1 ($g(t)\rightarrow 1$). Then, the proportional gain $k_p(t)$ approaches its minimum value $k_{p\min}$ ($k_p(t)\rightarrow k_{p\min}$) which is used to eliminate the oscillations. The integral gain $k_i(t)$ approaches its maximum value $k_{i\max}$ ($k_i(t)\rightarrow k_{i\max}$) which is used to reduce the steady-state error. The differential gain $k_d(t)$ approaches its maximum value $k_{d\max}$ ($k_d(t)\rightarrow k_{d\max}$) which is used to reduce the adjustment time and enhance the system stability.

4. Simulation Results

4.1. Selection of System Parameters

In order to demonstrate the effectiveness of this system, the EHSS is simulated with the following nominal parameters: J=0.96$kg\cdot m^2$, L_g=0.24m, r_1=0.04m, r_2=0.27m, P_s=2×10^6Pa, P_r=0.5×10^5Pa, L_0=0.1m, x_{p0}=0.08m, L_g=0.37m, m=7kg, β=2×10^7Pa, A_{p1}=1.77×10^{-4}m^2, V_0=1.15×10^{-4}m^3, C_t=8×10^{-12}$m^5N^{-1}s^{-1}$, g_s=3.97×10^{-8} $m^4s^{-1}N^{-1/2}$, k=0.2,

The gains of the conventional PID are chosen using Ziegler-Nichols method. Then, optimum gains of k_p=0.5, k_i=3 and k_d=0.7×10^{-3} are used. In addition, bigger gains of k_p=1, k_i=4 and k_d=1×10^{-3} are selected to test the tracking performance of the conventional PID. However, the gains of the self-tuning PID controller are chosen using the combination of the optimum gains and bigger gains that $k_{p\max}$=1, $k_{p\min}$=0.5, $k_{i\max}$=4, $k_{i\min}$=3, $k_{d\max}$=1×10^{-3} and $k_{d\min}$=0.7×10^{-3}.

4.2. Comparisons of the Conventional PID Controller and Self-tuning PID Controller

The system simulation is implemented by Matlab based on the introduced EHSS model and 1-DOF rotary system model. The computer simulations are conducted using two types of controllers, such as the conventional PID controller and the self-tuning PID controller. Figure 2, Figure 3 and Figure 4 show the the tracking lag and mean absolute error (MAE) using the two types of controllers.

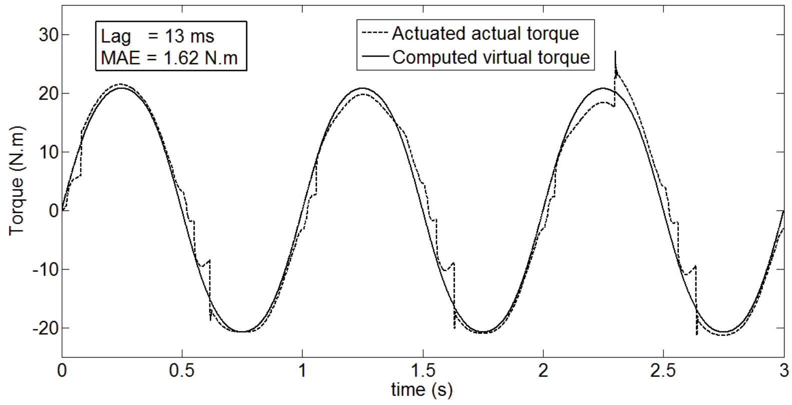

Figure 2. Tracking Performance of the Conventional PID Controller Using Optimum Gains

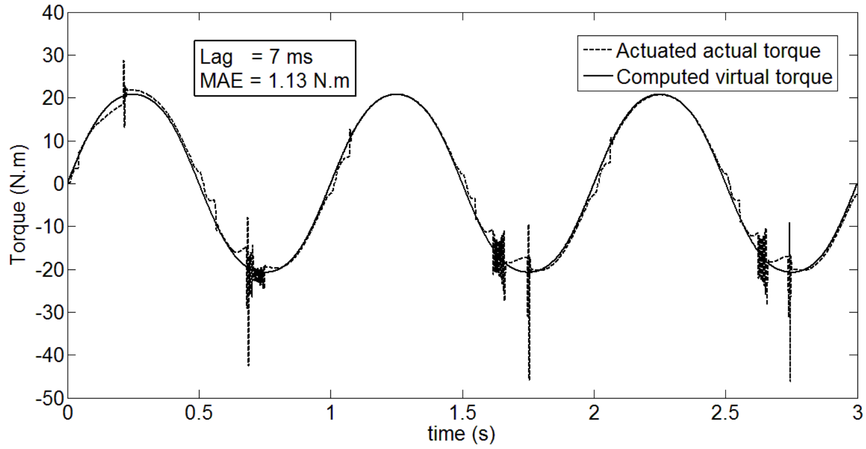

Figure 3. Tracking Performance of the Conventional PID Controller Using Bigger Gains

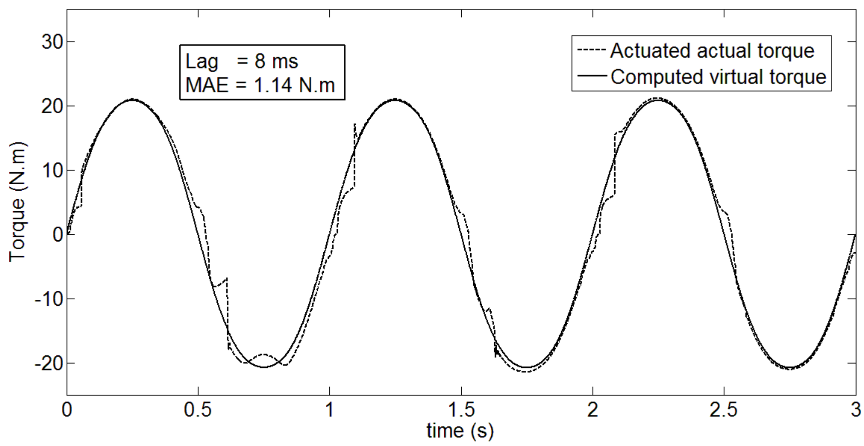

Figure 4. Tracking Performance of the Self-tuning PID Controller

The comparison between Figure 2 and Figure 4 describes that the tracking lag (8ms) of the self-tuning PID controller is less than that (13ms) of the conventional PID controller using optimum gains. Meanwhile, the self-tuning PID controller obtains less MAE (1.14N.m) than the conventional PID controller (MAE=1.62N.m).

The comparison between Figure 3 and Figure 4 states that the conventional PID controller using bigger gains obtains better tracking performance (Lag=7ms and MAE=1.13N.m). However, the conventional PID controller receives oscillations in the control process while the self-tuning PID controller doesn't.

To sum up, the self-tuning PID controller not only eliminate the oscillations caused by the conventional PID controller using bigger gains, but also acquired less tracking lag and errors when compared with the conventional PID controller using optimum gains.

5. Conclusion

This paper has presented a self-tuning PID controller for EHSS. The 1 DOF rotary system as a test platform has been examined through simulations using the proposed self-tuning PID. The gains of the proposed controller have been varied and tuned such that the EHSS receives better performance compared with the conventional PID controller in terms of trajectory tracking. It comes to a conclusion that the proposed self-tuning PID controller has given improved results when compared with the conventional PID controllers.

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Authors

Lie Yu, he is a Ph.D candidate from Wuhan University of technology, Wuhan, China. He received his B.E degree from Xidian University, in electronic information engineering, China, in 2009. He received his M.E degree from Wuhan University of Science and Technology, in control engineering, China, in 2011. His research interests mainly include the development of rehabilitation robots and its control application.

Jianbin Zheng, he is a professor at School of Information Engineering in Wuhan University of Technology, Wuhan, China. he received his B.E degree from University of Electronic Science and Technology of China, in communication and electronic system, China, in 1988, and the M.E degree from University of Electronic Science and Technology of China, in communication and electronic system, China, in 1991. His research interests focus on Information Processing, Pattern Recognition, Embedded System and Intelligent Control.

Yang Wang, he is presently working as an associate Professor in the School of Information Engineering, Wuhan University of Technology, Wuhan, China. He received his Ph.D degree in Wuhan University, Wuhan, China. His research areas are Robot Control and Pattern Recognition. *Corresponding author: Yang Wang, PhD, School of Information Engineering, Wuhan University of Technology, Jian Hu Campus, 122 Luo Shi Road, Hongshan District, Wuhan 430070, China, E-mail: powerflow@whut.edu.cn.