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Position tracking control of electro-hydraulic single-rod actuator based on an extended disturbance observer

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ABSTRACT

This paper presents a nonlinear cascade controller based on an extended disturbance observer to track desired position trajectory for electro-hydraulic single-rod actuators in the presence of both external disturbances and parameter uncertainties. The proposed extended disturbance observer accounts for external perturbations and parameter uncertainties separately. In addition, the outer position tracking loop uses sliding mode control to compensate for disturbance estimation error with desired cylinder load pressure as control output; the inner pressure control loop is designed using the backstepping technique. The stability of the overall closed-loop system is proved based on Lyapunov theory. The controller performance is verified through simulations and experiments. The results show that the proposed nonlinear cascade controller, together with the extended disturbance observer, provide excellent tracking performance in the presence of parameter uncertainties and external disturbances such as hysteresis and friction.

1. Introduction

Electro-hydraulic systems are widely used in many industrial and mobile applications, e.g., robot manipulators [1], hydraulic excavators [2], and tunnel boring machines [3] because of their high power-to-weight ratio compared with electric drives [4,5]. However, the dynamic behaviors of electro-hydraulic servo systems suffer from strong nonlinearities, such as square-root relationship between pressure and flow, temperature and pressure dependent oil properties and friction. Furthermore, industrial applications are likely to be affected by external disturbances and system parameter variations, such as the damping coefficient, the time-varying internal leakage coefficient, or supply pressure drops. These are great challenges for controller design of electro-hydraulic servo systems.

To obtain better dynamic performance, various control methods have been used. Local linearization of the nonlinear dynamics about a nominal operating condition allows the use of techniques such as pole placement [6] and adaptive control [7]. However, these controllers cannot guarantee satisfactory performance in all working points and are likely to fail if plant properties change drastically. The feedback linearization method was used in [8–10]. However, this method is based on cancelling nonlinear terms and does not account for system uncertainties. So, another control approach, sliding mode control (SMC) was applied to electro-hydraulic control systems in [11–14]. In SMC, trajectories are forced to reach a desired sliding manifold in finite time and then stay on this manifold for all future time, and dynamics on the sliding surface are independent of matched uncertainties and disturbances. However, chattering in the control signal, which is inherent in SMC, can easily excite high frequency modes and degrade system performance. Adaptive control has been proven to be a valid method to system uncertainties, therefore several nonlinear adaptive controllers were proposed for electro-hydraulic control systems in the literature. In [15,16], a nonlinear adaptive control scheme based on the backstepping method was proposed to the force control of electro-hydraulic systems. Yao et al. [17–20] proposed the nonlinear adaptive robust controller (ARC) for trajectory tracking of hydraulic actuators in the presence of uncertain nonlinearities and parameter uncertainties. A sliding mode adaptive controller was proposed in [21,22] to compensate for nonlinear uncertain parameters due to variations of the original control volumes. A novel adaptive controller based on the backstepping technique was proposed in [23].

Load disturbance or unmodeled load force would significantly degrade position tracking performance because force available to the system is diminished [24]. To obtain better tracking performance, disturbance compensation is needed. However, direct measurement of disturbances is not always possible in practice,

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therefore disturbance observers for disturbance rejection is critical and several disturbance observers have been adopted to solve this problem so far. In [25,26], disturbance was estimated and compensated according to the observer proposed in [27] for position tracking. A high-pass disturbance observer was designed for position tracking of electro-hydraulic actuators in [28] and the disturbances within the observer bandwidth can be cancelled. In [29,30], a disturbance observer was used to reject low-frequency disturbances and high-frequency noises in an electro-hydraulic servo system. An integral sliding mode disturbance compensator was proposed in [31] for load pressure control of hydraulic drives.

Motivated by [20,30], an extended disturbance observer is proposed to estimate uncertain parameters and external disturbances simultaneously. Unlike the disturbance observer used in [30], the proposed extended disturbance observer deals with parameter uncertainties and external disturbances separately. Compared with [20], parameter and disturbance updates proposed in this paper are driven by the state estimation error while that proposed in [20] are driven by the tracking error, in addition, alternative disturbance observers proposed in [28,32] could be used in this control scheme. Based on the proposed extended disturbance observer, a nonlinear cascade controller is developed for position tracking of an electro-hydraulic single-rod actuator. It comprises a position tracking outer loop and a load pressure control inner loop which provides the hydraulic actuator the characteristic of a force generator. SMC is also used to compensate for disturbance estimation errors. Stability of the closed-loop system consisting of the extended disturbance observer, the nonlinear controller and the plant model is proved by means of Lyapunov theory.

To verify the performance of the proposed controller, it is applied to an electro-hydraulic test bench. The test bench, as shown in Fig. 1, consists of a single-rod hydraulic actuator as a driving cylinder and a twin-tube shock absorber as a load force generator. The damping valves are integrated into the shock absorber. For a specific speed of the shock absorber cylinder rod, fluid is displaced through the damping valves at a specific flow rate. The damping valve flow resistance produces the pressure difference across the damping orifice which generates a damping force resisting the shock absorber rod motion. Therefore, the damping force–velocity characteristic of the shock absorber is closely related to the pressure-flow characteristic of the damping valve, besides, the damping coefficients in the compression stroke and the rebound stroke are different because of asymmetric orifice configurations. When the solenoid valve in the shock absorber switches on/off, the damping orifice equivalent flow area also changes, which varies the shock absorber damping coefficients. What's even worse is that the realistic force–velocity characteristic of a shock absorber exhibits huge hysteresis because of many factors [34–37] such as effective compliance of the damping fluid, the tube, and the entrained air; significant friction force also exists in the driving cylinder and the shock absorber. The performance of the proposed controller is verified through simulations and experiments.

The rest of the paper is organized as follows. Detailed nonlinear mathematical model is presented in Section 2. In Section 3, the extended disturbance observer and the nonlinear cascade controller are given. In Section 4, simulation results are presented. Then, experimental setup and results are discussed in Section 5. Finally, conclusions are shown in Section 6.

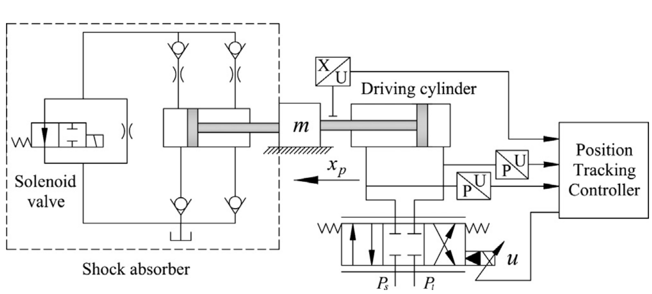

Fig. 1. Schematic diagram of the hydraulic system.

2. Mathematical modeling

The electro-hydraulic single-rod driving cylinder is shown in Fig. 2. The goal is to have the cylinder rod track any smooth motion trajectory as closely as possible. In the following, the nonlinear mathematical model will be derived.

The dynamics of the driving cylinder can be given by

$$m\ddot{x}_p = P_1A_1 - P_2A_2 - F_l \tag{1}$$

where m is the mass of the load, x_p is the displacement of the cylinder rod, P_1 and P_2 are the pressures in the cylinder forward and return chamber, respectively, A_1 and A_2 are the ram areas of the forward and return chamber, respectively, F_l is the total load force of the driving cylinder.

Neglecting the external leakage flow, the pressure dynamics of the two actuator chambers can be written as [4]

$$\begin{aligned} \frac{V_{01}+A_1x_p}{\beta_e}\dot{P}_1 &= Q_1 - A_1\dot{x}_p - C_t(P_1-P_2) \\ \frac{V_{02}-A_2x_p}{\beta_e}\dot{P}_2 &= -Q_2 + A_2\dot{x}_p + C_t(P_1-P_2) \end{aligned} \tag{2}$$

where V_{01} and V_{02} are the initial control volumes of the two actuator chambers, β_e is the effective hydraulic fluid bulk modulus, C_t is the total internal leakage coefficient of the cylinder, Q_1 is the supply flow rate to the forward chamber, and Q_2 is the return flow rate from the return chamber. Q_1 and Q_2 can be modeled by [4]

$$\begin{aligned} Q_1 &= k_{q1}x_v[s_g(x_v)\sqrt{P_s-P_1} + s_g(-x_v)\sqrt{P_1-P_t}] \\ Q_2 &= k_{q2}x_v[s_g(x_v)\sqrt{P_2-P_t} + s_g(-x_v)\sqrt{P_s-P_2}] \end{aligned} \tag{3}$$

Define function

$$s_g(\bullet) = \begin{cases} 1, & \text{if } \bullet \geqslant 0 \\ 0, & \text{if } \bullet < 0 \end{cases} \tag{4}$$

where k_q is the servo valve flow gain coefficient, x_v is the servo valve spool displacement, P_s is the pump supply pressure, and P_t is the tank pressure.

Because the desired closed loop dynamics is significantly slower than the servo valve dynamics, the servo valve dynamics can be neglected without a significant reduction on the model accuracy. For simplicity, we use the following approximation:

$$x_v = k_xu \tag{5}$$

where k_x is a positive constant and u is the control input voltage. Hence, combining (3)–(5), we get:

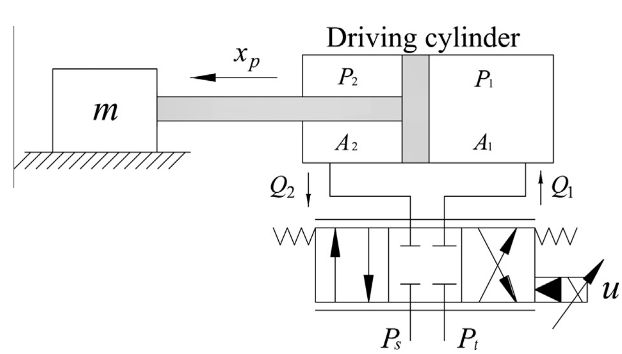

Fig. 2. Electro-hydraulic single-rod actuator configuration.