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Nonlinear adaptive torque control of electro-hydraulic load system with external active motion disturbance

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ABSTRACT

This paper develops a high performance nonlinear adaptive control method for electro-hydraulic load simulator (EHLS). The tracking performance of EHLS is mainly affected by the following factors: actuator's active motion disturbance, flow nonlinear and parametric uncertainties, etc. Most previous studies on EHLS pay too much attention on actuator's active motion disturbance, while deemphasize the other two factors. This paper concerns EHLS as a motion loading system. Besides actuator's motion disturbance, both the nonlinear characteristics and parametric uncertainties of the loading system are addressed by the present controller. First, the nonlinear model of EHLS is developed, and then a Lyapunov-based control algorithm augmented with parameters update law is developed using back-stepping design method. The stability of the developed control algorithm is proven via Lyapunov analysis. Both the co-simulation and experiment are performed to validate the effectiveness of the developed algorithm.

1. Introduction

Load simulator is a crucial equipment in the hardware-in-the-loop (HIL) experiments widely used in aerospace engineering. The main function of load simulator is to generate torque/force to simulate aero-dynamic load acting on the actuator system of aircraft, so that the whole flight control system, including the flight control algorithm and reliability of actuator system, can be valued on ground. It offers an effect way to foresee and detect the potential problems related to the flight control algorithm and actuator mechanics system. And it offers a much more efficient development stage in terms of time and cost [1]. According to energy source, load simulator can be classified as three types: electro-hydraulic load simulator (EHLS), electric load simulator (ELS), and pneumatic load simulator (PLS). Compared with ELS and PLS, EHLS has advantages in durability, power-to-weight ratio, accuracy and reliability [2–4]. It has wide applications in aircraft and missile [5,6], automotive industry [7], robotics [8] and fault tolerant field [9], etc.

How to improve the torque/force tracking performance of EHLS under external disturbance caused by actuator's active motion has been of great interest in both academia and industries. One traditional method is to implement feed-forward compensation using actuator's velocity signal. For example, Liu [10], Jacazio and Balossini [5] used actuator velocity to suppress actuator's motion disturbance. Alternatively, Jiao et al. [11] proposed to utilize actuator valve input to decouple actuator's motion disturbance effectively. The velocity gap between the actuator and loading system was extracted for further compensation [12] and the dual-loop scheme was developed [13,14]. Liu et al. [15] developed the double-valve method of which a pressure servo valve was paralleled with a flow valve for loading system. The pressure valve was mainly responsible for tracking load command and the paralleled flow valve released the disturbance flow caused by actuator's motion. The robustness against actuator disturbance was improved partially because the pressure valve is less sensitive to flow variation than the flow servo valve. Li [16] proposed a control scheme composed of a constant compensator, an inner loop control and an outer loop controller. The constant compensator and inner controller suppressed the motion disturbance, and the outer loop controller ensured the torque tracking performance. Su and Wang [17,18] developed a novel load structure, in which another set position servo system was connected in series to EHLS for releasing the disturbance flow. The mechanical structure of the system is very complicated, and the cost is high. In [3], a hybrid cylinder was used to develop the loading system, and the grey predictor-fuzzy PID control algorithm was implemented. Alleyne and Liu [19] found that, for the hydraulic force/torque system, excellent tracking performance cannot be guaranteed just use a simple PID control algorithm. EHLS is a special hydraulic force/torque servo system. It performance is not only limited by flow nonlinearity, parametric uncertainties, but also actuator's active motion disturbance. So far, massive control methods, such as quantitative feedback theory [6,20,21], variable structure control [22], fuzzy

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technology [23], neural networks [24], and H_∞ mixed sensitivity theory [25]. have been investigated for EHLS. One common feature of these studies is that the actuator motion is just regarded as external disturbance, and some robust algorithm is implemented to eliminate the disturbance. In fact, actuator's motion does not always hinder EHLS from achieving torque tracking. For example, actuator's motion is helpful when the direction of actuator's velocity and loading torque/force is opposite.

For most papers concerning on hydraulic position servo control such as [26–28], it is common to pay attentions to the points of nonlinear and parametric uncertainty. However, researchers of EHLS have pay more attentions on actuator's motion disturbance in load simulator, while deemphasize the flow nonlinear and parametric uncertainties. Although a few researches consider the nonlinear property and parametric uncertainties for force/pressure system such as [7,29–32], the loaded objectives in these studies have no active motion.

The principal contribution of this paper is to model EHLS as the motion loading system instead of ordinary torque/force servo systems with external disturbance. Based on the developed motion loading model, actuator's motion is utilized to help the torque tracking rather than considered as the external disturbance to eliminate. Moreover, the parameter uncertainty and flow nonlinear are also considered in this study to make a further improvement on control performance.

The rest of the paper is organized as follows. Section 2 develops the nonlinear motion loading model for EHLS. Section 3 develops the nonlinear controller augmented with parameter update law by using the back-stepping design method. Section 4 presents the co-simulation and experimental results. Finally, conclusions are drawn in Section 5.

2. Architecture and Model of EHLS

2.1. System architecture

In general, the HIL experiment is mainly composed by two sets of servo system. i.e., the actuator and loading system. The schematic structure of the EHLS and the orifice configuration are illustrated in Fig. 1. As shown in Fig. 1(a), the left part presents the actuator system equipped with a servo valve, hydraulic swing motor and angular encoder. φ_a and φ_l denote the angular displacement of the actuator and loading motor, respectively. Actuator's angle is feedback to actuator controller, constituting angle control. The right part denotes the loading system composed of a valve controlled hydraulic swing motor, angular encoder, torque sensor, and an inertia disk to simulate the inertia of control surface. The loading motor produces torque to simulate the load acting on the control surface of actuator system in flight process. Obviously, the torque tracking performance is subject to actuator's active motion.

2.2. Motion loading model of EHLS

The dynamics of load shaft in loading swing motor can be described by

$$D_L P_L = J_l \ddot{\varphi}_l + T_f + T \tag{1}$$

where D_L is displacement of hydraulic motor (m^3/rad), $P_L = P_1 - P_2$ is pressure difference between two chambers of loading motor (N/m^2), J_L is moment inertia of load shaft (kg m^2), $\ddot{\varphi}_l$ is rotor angular acceleration of loading system (rad/s^2), T_f is friction torque (N m) and T is reaction torque by actuator system (N m).

According to Fig. 1(b), the flow continuity equation of loading motor can be established. For simplicity, it is assumed that: the servo valve is matched symmetrically with ideal zero opening and zero lapping; The spool of the valve radial-clearance leakage and external leakage of load motor is negligible; The supply pressure is stable; Compared with the total contained volume of the loading motor, the change of both chamber volumes is small and negligible.

Based on these assumptions, the load flow equation is [2]

$$Q_L = D_L \dot{\varphi}_l + \frac{V_t}{4\beta_e} \dot{P}_L + C_t P_L \tag{2}$$

where Q_L is load flow rate (m^3/s) and $\dot{\varphi}_l$ is angular velocity of loading system (rad/s), β_e is effective bulk modulus (N/m^2), C_t is total internal leakage coefficient of loading motor (m^5/N s), and V_t is total compressed volume of EHLS system (m^3).

The load flow Q_L governed by the spool displacement is [2]

$$Q_L = C_d \omega x_v \sqrt{(P_s - \mathrm{sgn}(x_v) P_L)/\rho} \tag{3}$$

Moog D765, a closed-center nozzle-flapper valve, is used in this study. Its response time is about 2 ms and the natural frequency is higher than 150 Hz [33]. Because the dynamics of the servo valve is much higher than that of whole system, the spool dynamics is ignored, i.e.,

$$x_v = k_{xv} u \tag{4}$$

where C_d is flow coefficient of EHLS valve, ω is area gradient of EHLS valve (m), ρ is oil density (kg/m^3), x_v is spool displacement (m), P_s is supply pressure (Pa). k_{xv} is valve gain (m/V), and u is control output (V).

$\mathrm{sgn}(\cdot)$ denotes the discontinuous sign function

$$\mathrm{sgn}(\cdot) = \begin{cases} 1 & \text{if } \cdot > 0 \\ 0 & \text{if } \cdot = 0 \\ -1 & \text{if } \cdot < 0 \end{cases} \tag{5}$$

The torque output of EHLS is determined by the angle difference between two ends of torque sensor. Therefore, the torque output is expressed as

$$T = K_s(\varphi_l - \varphi_a) \tag{6}$$

where T is torque output (N m), K_s is stiffness of the torque sensor (N m/rad). φ_l and φ_a are angular displacement of loading and actuator system, respectively (rad).

Select the torque output T, angle velocity $\dot{\varphi}_l$, and the load pressure P_L as the state variables for the EHLS system, i.e., $x = [x_1, x_2, x_3]^T \triangleq [T, \dot{\varphi}_l, P_L]^T$. By combining from Eqs. (1)–(6), the system state form is obtained as

$$\begin{cases} \dot{x}_1 = K_s(x_2 - \dot{\varphi}_a) \\ \dot{x}_2 = \frac{1}{J_L}(D_L x_3 - T_f - x_1) \\ \dot{x}_3 = -\theta_1 x_2 - \theta_2 x_3 + \theta_3 g(x_3, x_v) k_{xv} u \end{cases} \tag{7}$$

where $\theta_1 = 4\beta_e D_L/V_t$, $\theta_2 = 4\beta_e C_t/V_t$, $\theta_3 = 4\beta_e C_d \omega/(V_t\sqrt{\rho})$, and $g(x_3, x_v) = \sqrt{P_s - \mathrm{sgn}(x_v) x_3}$.

Given the desired torque reference $T_d(t)$, the purpose of this paper is to synthesize a control output u, so that the output $y = x_1$ tracks $T_d(t)$ as closely as possible in spite of actuator's active motion disturbance, parameter uncertainty and flow nonlinearity.

3. Nonlinear adaptive controller for motion loading system

In this section, the standard recursive back-stepping design technology [34–36] is applied to system Eq. (7). The following assumptions are made for subsequent study: