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A practical nonlinear robust control approach of electro-hydraulic load simulator

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Abstract This paper studies a nonlinear robust control algorithm of the electro-hydraulic load simulator (EHLS). The tracking performance of the EHLS is mainly limited by the actuator's motion disturbance, flow nonlinearity, and friction, etc. The developed controller is developed based on the nonlinear motion loading model. The problems of the actuator's disturbance and flow nonlinearity are considered. To address the friction problem, the friction model of the loading motor is identified experimentally. The friction disturbance is compensated using the obtained friction model. Therefore, this paper considers the main three factors comprehensively. The developed algorithm is easy to apply since the controller can be obtained just with one step back-stepping design. The stability of the developed algorithm is proven via Lyapunov analysis. Both co-simulation and experiments are performed to verify the effectiveness of this method.

1. Introduction

Load simulator is crucial equipment in hardware-in-the-loop (HIL) experiments, which is widely used in aviation and aerospace fields. Its main function is to generate the torque/force to simulate the aero-dynamic load acting on the actuator system, so that the whole flight control system, which includes the performance of the flight control algorithm and the reliability of the actuator system, can be verified under the real flight condition in the laboratory. The designer of the actuator system, by means of the load simulator, can foresee and detect potential problems related to the flight control algorithm and the actuator mechanics. A load simulator offers a much more efficient development platform in terms of time and cost.[1] According to the type of the energy source, a load simulator can be classified into three types: electro-hydraulic load simulator (EHLS), electric load simulator (ELS), and pneumatic load simulator (PLS). Compared to ELS and PLS, EHLS has many advantages such as: durability, high power to weight ratio, controllability, accuracy, and reliability.[2–4] In view of these advantages, EHLS have found a wide range of applications in aircraft and missile industries,[5,6] automotive industry,[7] robotics and fault tolerant fields.[8]

Aside from the common problems such as parameter uncertainties and nonlinear characteristics which all hydraulic servo systems possess, the most crucial problem for an EHLS is the external disturbance caused by the actuator's active operation. Improvement of tracking performance of EHLS has been of

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great interest from both academic and industrial perspectives, and extensive research has been done to resolve these problems. A common and natural idea is to carry out a feed-forward compensation using actuator's velocity signal. Liu,[9] Jacazio and Balossini[5] exploited actuator velocity to improve the tracking performance of an EHLS. Jiao et al.[10] proposed to make use of the actuator valve's input to decouple the actuator's motion disturbance. Based on this study, the velocity gap between the actuator and the loading system was extracted to make a further compensation[11,12] and the dual-loop scheme was developed.[13,14] Li et al.[15] developed the double-valve method in which a pressure servo valve was paralleled with a flow valve for the EHLS. The pressure valve was mainly responsible for tracking load instruction and the paralleled flow valve was responsible for releasing the disturbance flow caused by the actuator's exercise. The robustness against actuator disturbance was improved partially because the pressure valve was less sensitive to flow variation than the flow servo valve. Li[16] proposed a control scheme for an EHLS that was composed by a constant compensator, an inner-loop controller, and an outer-loop controller. The function of the constant compensator and the inner controller was to suppress the motion disturbance and the outer-loop controller was designed to improve tracking performance of the loading system. Su et al.[17,18] developed a novel load structure, of which another set position servo system was introduced to connect in series to an EHLS for releasing the disturbance flow. One deficiency of this method is the mechanical structure was too complicated and the cost was high. In Ref. [3], a hybrid cylinder was investigated as the actuator of an EHLS and the grey predictor-fuzzy PID controller was applied. So far, various techniques such as the quantitative feedback theory,[6,19,20] variable structure control,[21] the fuzzy technology,[22] neural networks,[23] and the H_∞ mixed sensitivity theory,[24] all have been implemented for EHLS. The central issue of these studies is to apply a certain robust algorithm for loading systems. However, the disturbance boundary was hard to determine and the disturbance strength was very serious in some occasions.

For most papers concerning on hydraulic position servo control,[25–27] it is very common to focus the efforts on the problem of nonlinear and parametric uncertainty. For most studies about load simulator, however, they usually give the center stage to suppress the actuator's motion disturbance while deemphasizing the nonlinear nature and the parameter uncertainty problem. Although a few research has taken the nonlinear and parametric uncertainty problem into consideration for force/pressure systems,[7,28–33] the loaded objectives in these studies have no active exercise. Alleyne et al.[34] has detailed the reason of limitation when using a simple control for a hydraulic force servo system. The EHLS is actually a motion force/torque hydraulic servo system with serious external disturbance; hence, it is necessary to resort to some advanced control technologies to improve the tracking performance.

Seminal works in the field of adaptive robust control for uncertain nonlinear systems were done in Refs. [35–37] which treated uncertain nonlinearity and parameters in a systematic way for hydraulic position servo systems. In this work, the adaptive robust torque control for an EHLS is studied. The principal contribution of this work is that the EHLS is addressed as a nonlinear motion loading system rather than a linear torque servo system with external disturbance. This result in the developed nonlinear controller is decoupling against the actuator's motion disturbance. In additional to the actuator's disturbance, friction and flow nonlinearity are also addressed with the developed controller. The rest of the paper is organized as follows. Section 2 gives a brief introduction about the EHLS. The nonlinear robust controller based on load flow planning is developed in Section 3. Both the design procedure and the stability analysis are presented. Section 4 presents co-simulation and experiment results. Finally, conclusions are drawn in Section 5.

2. System description

In general, an HIL experiment is mainly composed by two sets of servo systems which are the actuator and the EHLS system. The schematic diagram and the oil line principle of the EHLS are described by Fig. 1. As shown in Fig. 1(a), the left part denotes the position actuator system which is equipped with a servo valve, a hydraulic swing motor, and an angular encoder. The signal from the angle encoder is fed back to the actuator controller to achieve servo angle control. The right part is the EHLS which is composed by a valve controlled hydraulic swing motor, an angular encoder, a torque sensor, and an inertia disk to simulate the inertia of the control surface. The EHLS exerts torque to simulate the air dynamic load acting on the control surface of the actuator system in a flight process. Obviously, the actuator's active exercise will impact the torque tracking performance of the EHLS.

Unlike a common force/torque servo system in which a loaded objective has no active motion, the control objective of the EHLS, i.e., the output of the torque sensor (see Fig. 1), is governed by the angle difference between the ends of the torque sensor. Therefore, torque output can be expressed as

$$T_L = K_S(\varphi_L - \varphi_A) \tag{1}$$

where T_L is the torque output of the loading system (N·m), K_S is the stiffness of the torque sensor (N·m/rad), φ_L and φ_A are the angular displacements of the EHLS and the actuator system, respectively (rad).

According to the oil principle shown in Fig. 1(b), the flow continuity equation of the loading motor can be established. For simplification, we assume that the servo valve is matched symmetrically with ideal zero opening and zero lapping and the spool of the valve radial-clearance leakage and the external leakage of the load motor are both negligible, as well as stable oil source pressure, zero return pressure, and constant oil elastic modulus. Based on these assumptions, the load flow equation can be given[2]

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

where D_L is the displacement of the hydraulic motor (m^3/rad), $P_L = P_1 - P_2(N/m^2)$ is the pressure difference between the two chambers of the loading motor, P_1 and P_2 are the pressure in forward and return chamber, respectively. Q_L is the load flow rate (m^3/s), $\dot{\varphi}_L$ is the angular position of the loading system (rad/s), V is the total control volume of the EHLS system (m^3), β_e is the effective bulk modulus (N/m^2), and C_t is the coefficient of the total internal leakage of the loading motor due to pressure ($m^5/(N{\cdot}s)$).