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A switched energy saving position controller for variable-pressure electro-hydraulic servo systems

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ABSTRACT

The electro-hydraulic servo system (EHSS) demonstrates a relatively low level of efficiency compared to other available actuation methods. The objective of this paper is to increase this efficiency by introducing a variable supply pressure into the system and controlling this pressure during the task of position tracking. For this purpose, an EHSS structure with controllable supply pressure is proposed and its dynamic model is derived from the basic laws of physics. A switching control structure is then proposed to control both the supply pressure and the cylinder position at the same time, in a way that reduces the overall energy consumption of the system. The stability of the proposed switching control system is guaranteed by proof, and its performance is verified by experimental testing.

1. Introduction

The Electro-hydraulic Servo System (EHSS) is widely used in industrial and machinery settings for high performance position tracking applications. The EHSS has proven to be a promising choice for various mobile and high-performance applications due to its high power to weight ratio, good dynamic performance and its ability to tolerate abrupt and aggressive loadings [1]. This type of system is able to generate very high forces, and it has a very high power to weight ratio compared to its electrical counterparts. This characteristic makes the EHSS ideal for various high-performance applications, and it is widely utilized in aircraft control [2], machine tools and manufacturing [3], excavating [4] and automotive industries [5].

The problem of EHSS position tracking control has been addressed in many papers, and numerous publications have proposed Linear, Non-linear, Adaptive, and Robust control methods to improve the position tracking performance of the EHSS [6–10] and have proven to be successful in their specific purpose.

The common problem among the control implementations in the literature is that little attention has been devoted to the actual workings of an EHSS and the actual source of its energy. This lack of attention has caused the proposed implementations to suffer from low efficiencies during their operation [3,11]. The EHSS usually consists of a double-acting cylinder actuator, which is driven by a proportional directional control valve connected to a hydraulic pressure unit. The pressure unit provides constant fluid pressure to the system. An overall schematic of such a system is depicted in Fig. 1. When the valve opens in either of its directions, the supply fluid pressure causes the fluid to flow to/from the chambers of the cylinder and perform work on the environment. Most of the energy loss takes place in the constant pressure unit. This is due to the mechanism that most of the available units use to maintain constant pressure, which usually consists of a fixed-displacement pump, a fluid accumulator and a pressure relief valve in parallel, as in [8]. The pumped fluid accumulates until the pressure relief valve is activated and the excessive fluid is discharged to the fluid tank. This type of system suffers from excessive energy consumption and produces a significant amount of heat, resulting from high-pressure discharge of the hydraulic fluid to the tank during low-demand tasks. This power loss is proportional to both the fluid flow rate to the tank and the fluid pressure at the location of the pump [1]. The pressure drop across the proportional directional control valve is another source of power loss in these kinds of systems.

Certain efforts have been made in the literature in order to improve the overall efficiency of the EHSS. Among the earliest of efforts is the utilization of the patented load-sensing unloading valve at the location of the pump, which maintains the system pressure mechanically using an on–off valve at a level that

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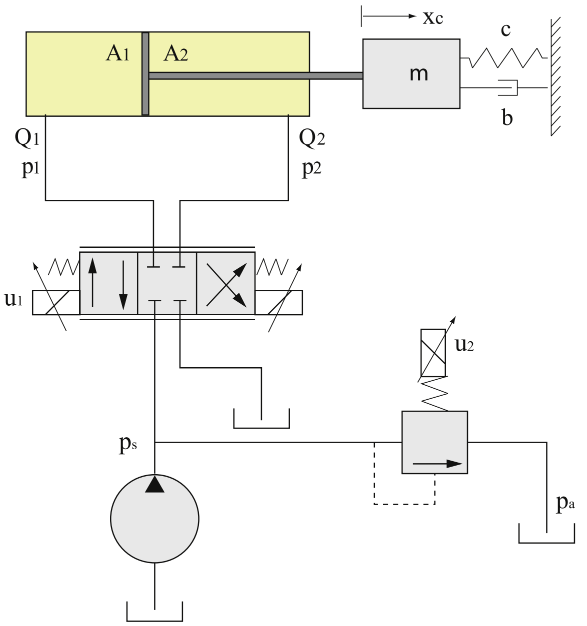

Fig. 1. Schematic diagram of the electro-hydraulic system.

matches the highest pressure demand from the unknown external load. Electrical adaptations of this method are also available in [3]. This method, though effective for position control tasks, does not guarantee that it is the optimal way to save energy through unloading. Another approach has used a variable displacement pump with built-in constant pressure supply control [6]. Furthermore, a valveless EHSS setting, which controls the actuator position and pressure with a specially designed pump and without any control valves has been used in [12]. Despite the significant effect of the last two methods on overall efficiency, they suffer from high equipment and maintenance costs resulting from utilizing a relatively complex variable-displacement pump and built-in pressure compensation. Furthermore, the overall achievable closed-loop crossover frequency for these systems is much less than their valve-actuated counterparts due to their higher inertias [12]. A more recent trend has used independent valve metering to save energy. A five-valve plus accumulator configuration has been used by Lu et al. [13]. These methods have a significant energy-saving capability at the cost of a highly complex system and controller, which limits the application domain of the configuration. More recently, Tivay et al. [11] have proposed a variable-pressure EHSS and a modified switching PID controller similar to [14] capable of saving energy. However, no energy optimality and no stability guarantees have been considered in its controller design.

The purpose of this paper is to introduce a complete hydraulic control system capable of tracking a demanded position trajectory, paying special attention to minimizing the overall energy consumption. In this paper, a hydraulic supply with controllable pressure is introduced. For this purpose, a proportional relief valve is used in parallel with the hydraulic line at the location of the pump, as it is shown schematically in Fig. 1. Using this setup, the ability to control the system pressure at the location of the pump is achieved. Thus, the plant becomes a multi input single output (MISO) system in which the inputs are directional and pressure control valve solenoid currents and the output is the cylinder position. Using these inputs, two degrees of freedom are available to design a controller with two conflicting objectives: minimizing the tracking error and saving energy. However, this relaxation comes at the cost of a new MISO plant with a particular switched behaviour for which the design of a suitable controller and its stability analysis become challenging [15].

To address the problem at hand, the existing EHSS models are extended to account for the variable supply pressure and its impact on different subsystems. Then, a switching plant model is derived from the non-linear state equations, using the notion of hybrid linear systems. For each of the switching modes of the plant, a separate LQ-optimal controller is designed that uses the two plant inputs to realize position tracking performance and reduce the overall energy consumption at the same time. A switching law is then designed to switch the appropriate controller in its place, according to the current state of the plant. Thus, the overall control structure consists of several controllers switched in place by a specially designed law. This law decides certain changes in control structure, based on the current state of the plant. The particular control structure of this paper needs a systematic way of tuning its parameters to satisfy the efficiency and performance requirements. The LQ-optimization problem approach for controller design ensures that demanded positions are followed acceptably in each mode of the plant, minimizing the overall energy consumption of the system at the same time.

The rest of the paper is structured as follows: Section 2 describes the detailed derivation of model equations for the variable-pressure MISO electro-hydraulic plant. The controller design and its related issues are presented in Section 3. Section 4 verifies the controller validity by simulation, and the experimental results are presented in Section 5. A conclusion then summarizes the main points of the paper.

2. EHSS plant model

2.1. Nonlinear dynamic model

Consider the non-linear dynamic model of the proposed electro-hydraulic system shown in Fig. 1, which consists of a hydraulic cylinder, a proportional directional control valve and a proportional pressure control valve. The dynamics of these hydraulic elements have been thoroughly described in [1,8] for a constant supply pressure setting. The objective in this section is to extend these EHSS plant models to obtain a unified set of state equations describing the EHSS with variable supply pressure.

The movement dynamics of the spool inside the proportional directional control valve, shown in Fig. 2a, can be described by a single-DOF damped and forced vibration model of the spool, which yields the following second-order differential equation:

$$\ddot{x}_d + 2\zeta_d\omega_d\dot{x}_d + \omega_d^2 x_d = k_d\omega_d^2 u_1 \tag{1}$$

in which x_d is the spool position, k_d is the directional valve gain, ω_d is natural frequency, ζ_d is damping ratio and u_1 is the input voltage to the directional control valve.

Equations of flow through the valve orifices come from orifice flow equations explained in [1]

$$Q_1 = C_{d1}\omega_1 x_d\sqrt{\frac{2}{\rho}\Delta P_1} \tag{2}$$

$$Q_2 = C_{d1}\omega_1 x_d\sqrt{\frac{2}{\rho}\Delta P_2} \tag{3}$$

where Q_1 and Q_2 are flows through the cylinder connections, C_{d1} is the directional valve discharge coefficient, ω_1 is the directional valve orifice area gradient and ΔP_1 and ΔP_2 are pressure differences across forward and return valve orifices.

In order to obtain a mathematical description of instantaneous pressure inside the forward and return cylinder chambers, the fluid flow balance equations are considered for two control volumes at

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