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Robustness analysis of a PI controller for a hydraulic actuator

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ABSTRACT

In this work the authors address the problem of robustness of the classic PI controller implemented in a Hydraulic Servo-Actuator (HSA), by presenting a strategy based on the definition of a linear model of the system and the identification of its parameters for different working points. The variation of these parameters is considered as a measure of parametric uncertainty of the linear model. These uncertainties along with the definition of a nominal plant are used to analyze the robustness of the system implementing the Small Gain Theorem. Theoretical and experimental results show that a PI controller can provide robustness to the HSA.

1. Introduction

The dynamics of hydraulic systems are highly nonlinear due to the physical properties present in the system such as fluid compressibility, pressure losses, transient and turbulent flow conditions and non-linear friction characteristics in the hydraulic actuator.

These nonlinear characteristics demand the development of advanced control strategies when high performance response is required (Bobrow & Lum, 1995; Chen, Renn, & Su, 2005; Hwang, 1996; Kim & Tsao, 2000; Mohanty & Yao, 2011; Zhao & Virvalo, 1995; Zulfatman & Rahmat, 2009). Even though the nonlinear characteristics of the system cannot be neglected, attempts are also found in the literature to develop control strategies based on linear models of the HSA (Karpenko & Sepehri, 2005; Kim, 1997; Kim & Tsao, 2000; La Hera et al., 2008; Laval, 1996; Niksefat & Sepehri, 2001; Rahmat, Rozali, Wahab, Zulfatman, & Kamaruzaman, 2010).

Depending on the consideration taken, the simplified linear models have the general form of a third (Jelali & Kroll, 2002) or fourth order (Watton, 1989) type-I system. Nevertheless, in both models it can be distinguished two parts: one associated to the control input and the other associated to the effect of external forces. In general, for the design of the controller the latter one is discarded and only the first part is considered (Sepehri, Corbet, & Lawrence, 1995).

In this work, it is demonstrated that the later assumption leads to non-zero steady state error with classic P-controller, and that it is necessary to appeal to a PI-controller. However, it is controversial to use a PI controller with a type-I system, since the open loop system will have two poles at the origin and thus reducing the stability of the closed loop.

In this work the robustness of a HSA with a classic PI controller is assessed.

The approach presented in this work implements the linearized model of the system presented in Jelali and Kroll (2002), whose parameters are experimentally identified for different working points of the system (i.e. different positions and external load conditions). The variability of these parameters is considered as a measure of parametric uncertainties present in the linear model and is used to provide boundaries that will define the robustness of the system in terms of its stability and insensibility to external perturbations using the Small Gain theorem.

The main contributions of this work are the following:

- P-controller cannot provide zero steady state error, and it can only be achieved with a PI-controller.
- Classical PI controller can provide robustness to the HSA.
- Classical PD and PID controllers may lead to instability issues.

The rest of this work is organized as follows. Firstly, the linearized model of the HSA is derived and analyzed. Then, the experimental identification procedure of the system is completely

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Nomenclature

P_L	load pressure
P_A, P_B	pressure at chambers A and B
Q_A, Q_B	oil flow through control Port A and Port B
α	ratio of the effective surfaces at both sides of the piston
A_p	effective surface of the piston
c_{vi}	discharge coefficient of the orifices
x_v	servo-valve's input signal and servo-valve's spool position
K_{xA}, K_{xB}	flow sensibility constant regarding the spool position
K_{P_A}, K_{P_B}	flow sensibility constant regarding the pressure at the cylinder chambers
V_{A0}, V_{B0}	initial volume at the chambers
m	total mass. It consists of the piston mass, the mass of hydraulic fluid in the cylinder chambers and the external load
β_A and β_B	bulk modulus of the fluid at chambers A and B of the cylinder

detailed and 33 sets of parameters and their corresponding transfer functions are found. From the experimental identification a nominal plant is defined and some considerations regarding the stability and steady state error with classical P, PI, PD and PID controllers are presented. Following this, the robustness of the system is analyzed implementing the Small Gain Theorem. The real HSA is submitted to several experiments in order to prove the performance of the PI controller and finally, the conclusions and discussions are stated.

2. Theoretical model of HSA

The complete model of a HSA derives from complex equations that depend on many parameters that cannot be always accurately obtained and the rigorousness of the model is lost. However, considering that the dynamics of the system are governed by the slower dynamics (i.e. the dynamics of the piston), some of the dynamics derived from the internal components of the servo-valve can be neglected (Jelali & Kroll, 2002; Watton, 1989). It is a common practice to simplify the system into the orifice equation for the servo-valve, the pressure dynamics at the cylinder, and the dynamic equation of motion.

In particular for the linearized model, the latter equations are expressed in terms of the load pressure given by $P_L = P_A - \alpha P_B$, and it is considered that the flow through the orifices is in a steady state, i.e. $Q_A = A_p\dot{y}$ and $Q_B = \alpha A_p\dot{y}$.

Taking an operating point $\boldsymbol{P_0} = [x_{v0},\ P_{A0},\ P_{B0}]$, and assuming dominance of the first order term from the Taylor series expansion, the set of linearized equations are stated as follows.

Linearized pressure equation:

$$\delta P_A = \begin{cases} \dfrac{R_{h1}}{R_{h1} + \alpha^3 R_{h4}}\delta P_L & \text{for } x_v \geq 0 \quad \text{(a)} \\ \dfrac{R_{h2}}{R_{h2} + \alpha^3 R_{h3}}\delta P_L & \text{for } x_v < 0 \quad \text{(b)} \end{cases} \tag{1}$$

$$\delta P_B = \begin{cases} -\dfrac{\alpha^2 R_{h4}}{R_{h1} + \alpha^3 R_{h4}}\delta P_L & \text{for } x_v \geq 0 \quad \text{(a)} \\ -\dfrac{\alpha^2 R_{h3}}{R_{h2} + \alpha^3 R_{h3}}\delta P_L & \text{for } x_v < 0 \quad \text{(b)} \end{cases} \tag{2}$$

where $R_{hi} = 1/(x_v c_{vi})^2$.

Linearized orifice equation for the servo-valve:

$$\delta Q_A = K_{x_A}\delta x_v + K_{P_A}\delta P_A, \tag{3}$$

$$\delta Q_B = K_{x_B}\delta x_v + K_{P_B}\delta P_B. \tag{4}$$

Linearized pressure dynamics: Adopting that fluid flows into chamber A and flows out of chamber B while the pistons's rod extends with positive velocity (see Fig. 1), then

$$\delta Q_A = A_p\delta\dot{y} + \frac{V_{A0}}{\beta}\frac{d}{dt}[\delta P_A], \tag{5}$$

$$\delta Q_B = \alpha A_p\delta\dot{y} - \frac{V_{B0}}{\beta}\frac{d}{dt}[\delta P_B]. \tag{6}$$

Linearized equation of motion: In order to simplify the model, it is considered that the friction is governed by the classic *static+viscous* friction model given by $F_f(y) = F_s + \sigma\dot{y}$, where F_s is the static friction coefficient and σ is the viscous friction coefficient. Therefore, the linearized equation of motion is given by

$$A_p(\delta P_A - \alpha\delta P_B) - \delta F_{ext} - \delta F_f = m\delta\ddot{y}. \tag{7}$$

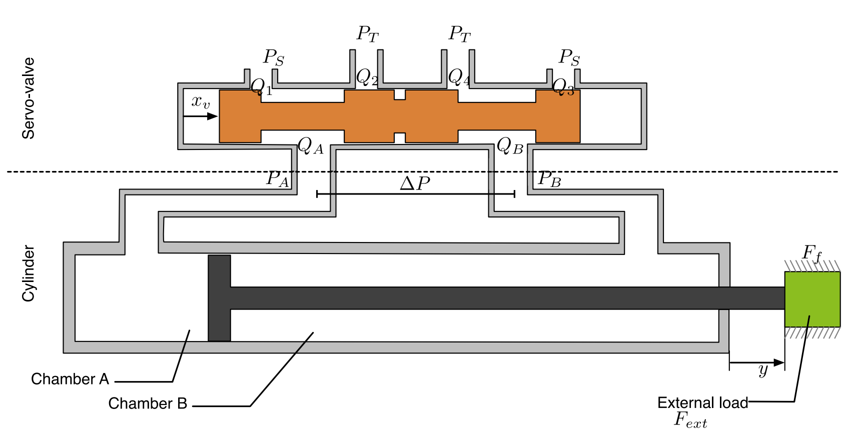

Fig. 1. Schematic diagram of the simplified SHA.