Mathematical Modeling and Simulation of a Position Control Electrohydraulic Servo System

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Abstract: In this paper, the position control of a hydraulic servo system using PID control is proposed for control of an electro-hydraulicactuator in order to improve steady-state error and the adaptation scheme are applied to an experimental valve controlled cylinder. The comparisons between linear output feedback with a Proportional, Integral and Derivative control (PID). The application of a PID controller to a nonlinear output feedback is investigated by both the position and the velocity of the hydraulic servo system. The experiment was based on a microcontroller and the simulation was based on a MATLAB Simulink. The simulation and the hardware experimental results have shown that the PID controller gave the best performance as it had the smallest overshoot, oscillation, and setting time.

Keywords: PID Control, Hydraulic System, Optimal Tuning, Closed Loop Control, Microcontroller.

1.0 Introduction

Hydraulic system are widely used in many application such as press machine and metal forming, because of their vast driving ability and high dynamic response characteristic. In many applications of hydraulic servo systems, when a linear motion is required, a piston control is the first choice. In a simple application that only requires the piston to produce back and forth motions between the two end points, the valve that controls the piston simply operates in an on/off mode. With the development of advanced technology, a combination between a hydraulic servo system and electronic devices was established to take full advantages of the science. At the same time, the most common way is to provide the controller with a way to describe the PID control [11]. Most of the performances of a PID tuning for processes use frequency response. Therefore, a wide variety of PID tuning controllers have been investigated in using the Ziegler-Nichols rule [12]. An optimal tuning of PID is satisfied even in the case where the system is dynamic and the system operating points change [1]. Moreover, a variety of methods have been developed for the modelling and for the simulation such as modelling and the simulation of physical systems and the design anintegrated system [2], [4].In [6], the velocity control of the hydraulic servo system achieves fast response. An alternative approach for a complex control with a Fuzzy PID control was developed by [8]. The application of a hydraulic mini press machine has a number of characteristics that complicate the design of fault detection systems. The purpose of this paper is to discuss how to achieve the best control performance of a hydraulic mini press machine. In section 2, the discussion focuses on a dynamic model and on the method used to position the control in hydraulic servo systems. Section 3 presents the conceptual design of a PID controller. In section 4, simulations and experimental results are separated by comparing a PID control. Finally, the paper is concluded in Section 5.

2.0 System Model

The system model for a typical inertial load drive by a hydraulic servo system is shown in Figure 1.and Figure 2. The hydraulicsystem was developed by [1]. The system can be thought of as a double acting cylinder driving an inertial load at the end. The dynamic of the inertial load can be written as:

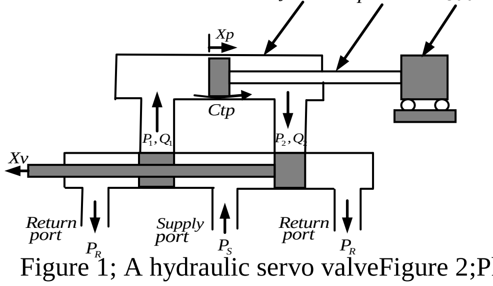

Figure 1; A hydraulic servo valveFigure 2;Photo of high performance hydraulic servo valve

$$\frac{V_t}{4\beta_e}\dot{P}_L = -A\dot{x} - C_{tp}P_L + Q_L \tag{1}$$

$$Q_L = C_d w x_v \sqrt{\frac{P_s - \text{sgn}(x_v)P_L}{\rho}} \tag{2}$$

where V_t is the total actuator volume (m^3), β_e is the effective bulk modulus (N/m^2), P_L is the pressure load (Pa), A is the actuator ram area (m^2), x is the actuator piston position C_{tp} is the total coefficient of leakage $(m^3/s.Pa)$, Q_L is the load flow (m^3/sec), C_d is the discharge coefficient, w is the spool valve area gradient (m), x_v is the spool valve position (m), P_s is the supply pressure (Pa), and ρ is the fluid density (kg/m^3). The valve of friction, a double acting hydraulic cylinder, drives a mass connected to its rod end in a horizontal direction. Thus, the load dynamics, we obtain are:

$$P_L A = m\ddot{x} + kx + F_f \tag{3}$$

where m is the total mass of the actuator, F_f is the friction force, A is the actuator ram area, and P_L is the load pressure, so $P_L = P_1 - P_2$.
Assuming Equation (1), (2), and (3), the mathematical model in the system state equations are

$$\begin{aligned} \dot{x} &= x_2 \\ \dot{x}_2 &= \frac{1}{m}(Ax_3 - F_f) \\ \dot{x}_3 &= -\alpha x_2 - \beta x_3 + \left(\sqrt{P_s - \text{sgn}(x_4)x_3)x_4}\right) \\ \dot{x}_4 &= x_5 \\ \dot{x}_5 &= -\omega_e^2 x_4 - 2\zeta\omega_e x_5 + \omega_e^2 \mu \end{aligned} \tag{4}$$

where $(\dot{x})$ is the actuator piston position, $(\dot{x}_2)$ is the actuator piston velocity, $(\dot{x}_3)$ is the load pressure, $(\dot{x}_4)$ is the valve position, $(\dot{x}_5)$ is the input current to the servo system, $\alpha:\left(4\beta_e A/V_t\right)$, $\beta:\left(4\beta_e C_{tp}/V_t\right)$, and $\gamma:\left(4\beta_e C_d w/V_t\sqrt{1/\rho}\right)$.

3.0 PID Controller

PID controllers are so far the most commonly used devices in industrial processes [11]. They have several important functions. They provide feedback and they have the ability to eliminate steady state offsets through an integral action and they can anticipate the future through a derivative action [13]. In practice, PID controllers are used at the lowest level and multivariable controllers provide the set points. The behaviour of a PID algorithm can be described as:

$$u(t) = K_p\left(e(t) + \frac{1}{T_i}\int_0^1 e(\tau)d\tau + T_d\frac{de(t)}{dt}\right) \tag{5}$$

Where K_p is the proportional action, T_i is the integral time and T_d is the derivative time. The PID controllers are parameterised by the following transfer function:

$$\frac{X(s)}{R(s)} = \frac{C(s)G(s)}{1 + C(s)G(s)} \tag{6}$$

The closed loop third order is given as:

$$G(s) = \frac{5432.58}{s^3 + 179.65s^2 + 8069.07s} \tag{7}$$

The transfer function of Eq. (6) using Eq. (7) yields:

$$\frac{k_p(1+T_{ds}+\frac{1}{T_{is}})\frac{k}{s(1+T_{p1s})(1+T_{p2s})}}{1+k_p(1+T_{ds}+\frac{1}{T_{is}})\frac{k}{s(1+T_{p1s})(1+T_{p2s})}} \tag{8}$$

The results acquired under the conditions of the PID controller are given as $k_p = 0.694,\ k_i = 1.84$ and $k_d = 1.64$. The PID controller was designed by following the standard procedure of a PID controller design.

4.0 Microcontroller and Software Development

A microcontroller based control system has been developed and has been used to provide the satisfactory control of a hydraulic servo system. The evaluations of the characteristics for the position control were obtained by using a microcontroller PIC 18F458 to control the hydraulic servo system and to reference the position information with a 256 (8 bit). The control methods were based on the position of the feedback control algorithm. They were used for the closed loop control of the hydraulic system. The PC delivered the signals to the microcontrollerPIC18F458via its parallel port. This simulator was created in order to test the performance of the controller and mechanical system being integrated. It is a C Language code that sends data to control position valve by real time control algorithms. Figure 3. shows the schematic diagram of the microcontroller system design and Figure 4. shows the microcontroller system design used in conducting the control systems.

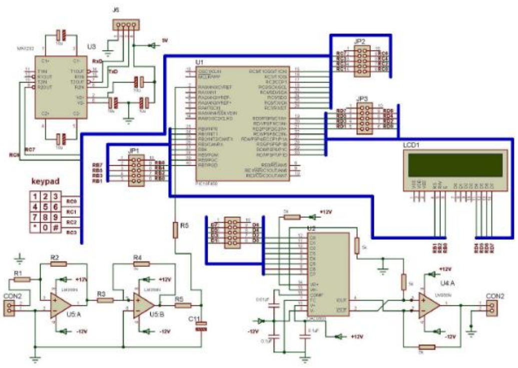

Figure 3; A schematic diagram of the microcontroller system design

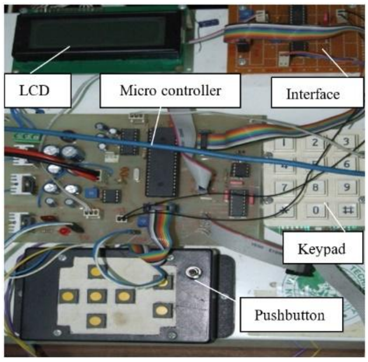

Figure 4; A photo of the microcontroller system design

5.0 Simulation and Experiment Results

The objective of Simulink was as a hydraulic actuator, which was a representative of a hydraulic servo system. A simulation of the Matlab/Simulink based study was used to evaluate the performances of the controllers on a nonlinear model of an HSS when compared to the performance of the PID control. This is shown in Figure 7. The parameters of the model had to be identified for designing the control algorithm. The steps for identifying dynamic models of the hydraulic system involve the design an experiment, selecting the model structure, choosing a criterion to fit, and devising a procedure to validate the chosen model. The system is first characterised by the position control (x_1). The piston stroke is $\pm 0.2m.$, the mass of the actuator and the load $m = 9.19Kg.$ The friction relation is approximated as linear $F_f : 0.3063N.$ Secondly, the system was characterised by the velocity control (x_2). Let $\int \dot{x}_1 dt = \int x_2 dt$

$x_1 = \frac{1}{s}(x_2)$ be a differentiable function, given by $\pm 0.16m/s.$ i. e, we obtain the result in terms of

The response is a set to the velocity and its top most displacement range was Referring the equation $[\dot{x}_2 = \frac{1}{m}(Ax_3 - F_f)]$ .The line block was required for each channel, where the velocity $\dot{x}_2$ was calculated. For that, the area and the force are added in to the block while the friction is subtracted from the block. The third method was to load the pressure system (x_3). This was the most effective method. The pressure of the system was set considering the pressure of the load. The pressure limit was the supply pressure $P_L : 5MPa.$ The value of $\dot{P}_L$ was determinedby substituting $\dot{P}_L = x_3$. The models of the components are composed and connected to the blocks. The load flow line parameters are the sum of the blocks from the function after passing through the function C_o block. Overall, the main idea of the modified hydraulic subsystems was to use valve position (x_v), the valve velocity and the input current (u). The upper and lower limits in servo valve were equal to the stroke of the servo valve , the maximum flow limit through the valve Figure 5 show the block diagram of a closed loop system.

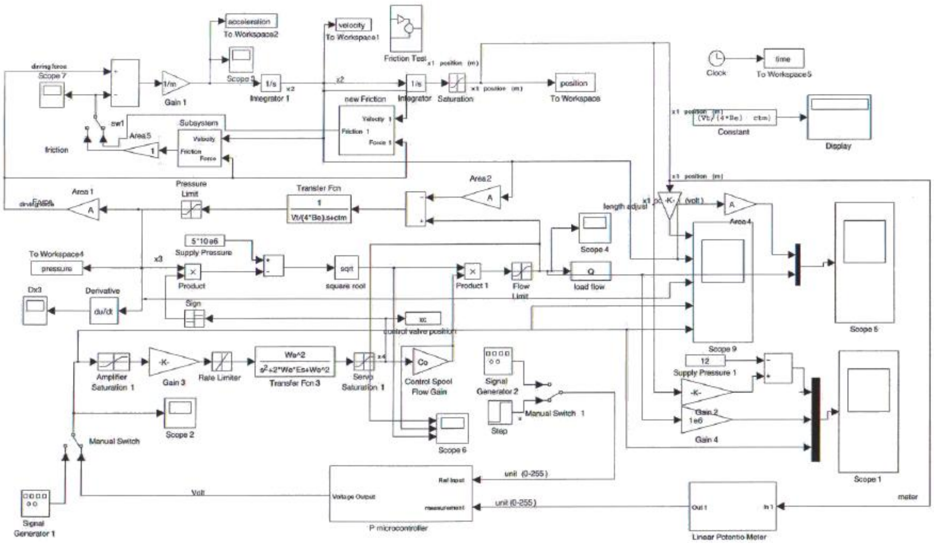

Figure 5; Matlab Simulink Model of closed loop system

6.0 Validation

The main directions to model the dynamic behaviour of a hydraulic servo system are the various controls of the model's complexity in order to achieve the different simulation and experiments. The controller was modelled using the Simulink approach in the Matlab programme, comparing the position and the velocity of the PID, which were based on an operating supply pressure of 6 Mpa as shown in Figure 6 – Figure 10.

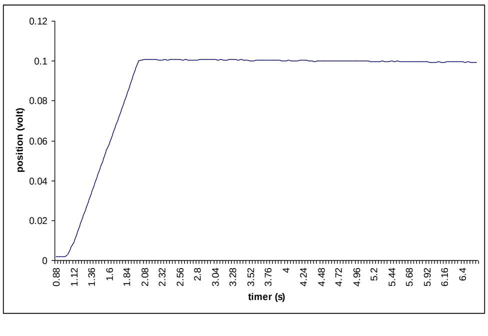

Figure 6; Unit position responses of the experiment

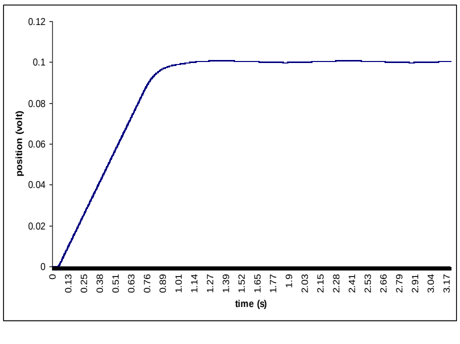

Figure 7; Unit position responses of the simulation

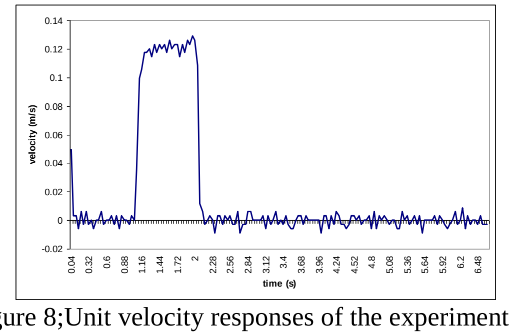

Figure 8;Unit velocity responses of the experiment

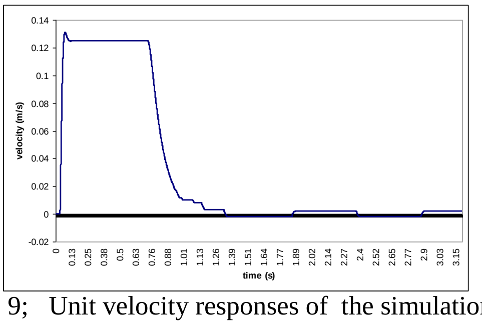

Figure 9; Unit velocity responses of the simulation

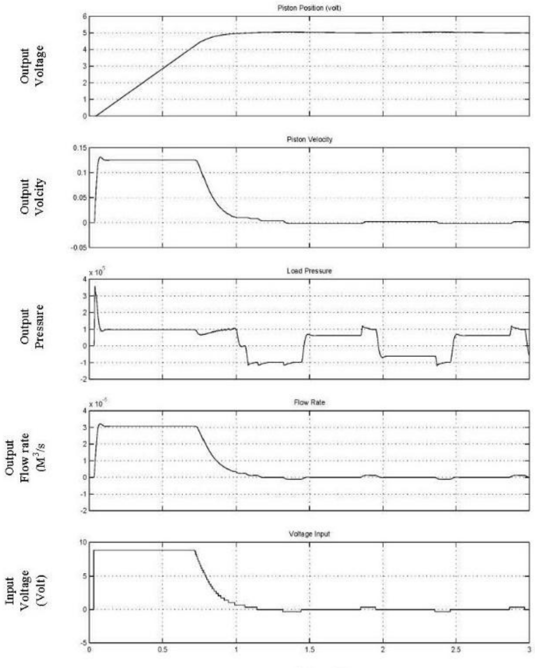

Figure 10; Observer performances of the step responses

7.0 Conclusions

This paper has documented the design, experimental evaluation, and the simulation of the position and the velocity servo control of a hydraulic mini press machine. The highly nonlinear behaviour of the system has limited the performance of the classical linear controllers that were used for this purpose. It was found that the proposed controller was better when compared with the traditional PID controller, both in the simulations and the experiments. The scheme has been tested on various processes in simulations and experiments where an accurate speed control with a fast response time was 0.123 m/s and the position was 42 mm. The simulations and the experiments found a good effectiveness of the PID method for a hydraulic servo system. The pressure control resulted in a performance error which was lower than any performance error that was controlled when keeping the system stable. Better performances of a higher control precision have been obtained in the position servo control system when compared with conventional PID controller.

References

[1]. Alleyne, A. and Liu, R.(2000),Systematic control of a class of nonlinear systems with application to Electro – hydraulic cylinder pressure control.IEEE transactions on control systems technology. Vol.8, pp. 623 – 634.

[2]. Amerongen, J. V. and Breedveld. (2003), Modelling of physical systems for the design and control of mechatronicsystems. Annual Reviews in Control. pp. 87-117.

[3]. Corke, Li, and Gurgenci, H. Z. (1997), Modelling andSimulation of an Electro - HydraulicMining Manipulator.Proceeding the 1997 IEEEInternationalConference on Robotics and Automation, Albuquerque, New Mexico.Vol.2, pp.1663 - 1668.

[4]. Ferreira, A. J. Sun P. and Gracio, J. (2006), Close loop control of a hydraulic press for springback analysis. Journal of Materials Processing Technology.177, pp.377-381.

[5]. Fung, F. R. Wang, C. Y. Yang, T. R. and Huang, H. H. (1996), Variable structure control with proportional andintegral compensations for electro- hydraulic position servo control system. Mechatronics, Vol.7, pp.67-81.

[6]. Jehan, Z. (2003), Mathematical Modeling of a Position Control Electrohydraulic Servo System. International Bhurban Conference on Applied Sciences and Technology, Bhurban, Pakistan. pp. 464-574.

[7]. Get, S. S. Lee, T.H. and Ren, S. X. (2001), Adaptive friction compensation of servo mechanisms. International Journal of Systems Science.Vol. 32, pp.523-532.

[8]. Huang, Y.J. Kuo, T.C and Lee, H.K. (2007), Fuzzy-PDController Design with Stability Equations for

Electro- HydraulicServo Systems. International Conference on Control, Automation and Systems, Seoul, Korea. pp.2407-2410.

[9]. Khan, H. Abou, C. S. and Sepehri, N. (2005), Nonlinear observer – based fault detection technique for electro-hydraulic servo-positioning systems. Mechatronics.pp.1037-1059.

[10]. Lin, C. andRen, L.(2005), The Integrated Simulation Analysis on 6 DOF Electro - hydraulic Servo – Control Parallel Platform. JSME International Journal.Vol. 48, pp.234-239.

[11]. Liu, G. P. and Daley, S. (2001),Optimal - tuning PID control for industrial systems. Control Engineering Practice 9. Vol. 2, pp.1185-1194.

[12]. Liu, G. P. and Daley, S. (1999), Optimal - tuning PID controller design in the frequency domain with application to a rotary hydraulic system Control Engineering Practice.7, pp.821-830.

[13]. Li, H. X.and Tso, S K. (1999), A Fuzzy PID with gain – scheduling control resolution for a thermal process-case study. Control Engineering Practice.Vol.7, pp.523-529.