MODELING, SIMULATION AND CONTROL OF HIGH SPEED NONLINEAR HYDRAULIC SERVO SYSTEM

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Abstract:

The purpose of this paper is to present for modeling and simulation a hydraulic servo system (HSS). This work describes the design and implementation of a control system for the operation of a hydraulic mini press machine. First, the system develops mathematical models for obtaining the system responses. While, the closed loop system is based on linearized model of feedback regulator of PD controller for high-speed control. The simulation experiment is performed using MATLAB and SIMULINK. An experimental set-up is constructed, which consists of microcontroller PIC 18F458 as controller. The simulation results, showing the effectiveness of the proposed approach.

Keywords: *modeling and simulation, hydraulic servo system, high speed hydraulic machine control.*

1. Introduction

HSS has been used in a wide range of modern industrial applications by virtue of their small size to power ratios and its ability to apply very large force and torque. However, the dynamics of hydraulic systems are highly nonlinear [6]. Stamping stroke of a typical hydraulic mini press machine is traveling 200 mm stamping distance with high speed. Closed loop feedback control for HSS provides the ability to apply very large forces and torques. The problems for HSS drives are their nonlinearities and low damping, and hence for some applications these systems are difficult for accurate control. The system conditions and parameters changes, depending on the operating conditions such as the mode of operation, which product is going to be made or what part of the process the machine is doing. Design of suitable controller to handle such type of situations is very important. The designed controller must operate properly to deal with nonlinear phenomenon and dynamics of the system parameters. The control signal errors are generally compared with velocity, position, force, pressure, and other system parameters. A HSS is a system consisting of motor, servo, controller, power supply, and other system accessories. In HSS essentially, the system controls the cylinder position to track the velocity and acceleration trajectory values enforced by the operator. The cylinder movement must precisely follow position, speed, and acceleration profiles. In [9] have proposed a controller (not adaptive), which is compared with standard PD controller but applicable to industrial practice. In [2] proposed a hydraulic press control using spring back analysis.

In this paper, which have presented a new PD controller for high-speed nonlinear hydraulic servo system. The controller is demonstrated through its application in a hydraulic servo system and its performance is evaluated by comparison with other high-speed controller developed by [3], [4], [5]. For industrial applications, which need algorithms that can adapt to nonlinear behavior tracking position and velocity, which are fully matched by the processing speed of microcontroller. At the same time, PD controller is designed to verify the performance of the model, which is developed by simulation and compared to experimental results.

The paper is organized as follows, In Section 2, a linearized mathematical model for the system is presented. Section 3 presents the development and stability analysis of PD controller. Section 4 describes hardware system architectures. Simulations and experimental results are presented in Section 5 and finally the paper is concluded.

2. Mathematical Model

The system describes detailed mathematical modeling of HSS for hydraulic mini press machine. The system consists of high-speed, electronic drives, hydraulic actuators, and position transducers. The mathematical model behavior of servo valves can be developed from the relationship between the displacement (x_p) and input voltage (μ) for the proportional valve. A third order model is sufficient for HSS, which is described by the equations given in following sections [11].

2.1. System Modeling

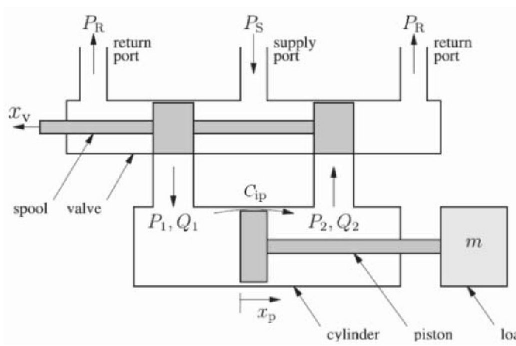

Fig. 1. A hydraulic actuator with four-way valve configuration.

In pressing machine, the hydraulic actuator is typically a double-acting hydraulic cylinder. The cylinder ports are connected to a proportional valve, and piston motion is obtained by modulating the oil flow into and out of the cylinder chambers. A servo valve provides this modulation as shown in Fig 1. The actuator can be precisely controlled by regulating theflow rates Q_1 and Q_2. However, the relationship between the piston position, x_p, and the flow rates depends on the dynamic properties of the load acting on the piston [1].

The mathematical model used in simulation is represented by the following system of equations:

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

where
V_t : total actuator volume
β_e : effective bulk modulus
P_L : load pressure
A : actuator ram area
x_p : actuator piston position
C_{tp} : total coefficient of leakage
Q_L : load flow

2.2. Hydraulic Actuator System Dynamics

The objective for developing the actuator system dynamics is to construct a strict - feedback control with fixed boundary layer to obtain precise position control of a nonlinear electro-hydraulic servo system. The relation between the servo valve spool position x_v and the input voltage u can be considered as a second order system.

$$T(s) = \frac{x_v(s)}{u(s)} = \frac{\omega_e^2}{s^2 + 2\zeta\omega_e s + \omega_e^2}$$

$$x_v s^2 = -\omega_e^2 x_v - 2\zeta\omega_e x_v s + \omega_e^2 \mu$$

$$x_v s^2 + 2\zeta\omega_e x_v s + \omega_e^2 x_v = \omega_e^2 \mu$$

$$\ddot{x}_v = -\omega_e^2 x_v - 2\zeta\omega_e^2 \dot{x}_v + \omega_e^2 \mu \quad (2)$$

Substituting $\ddot{x}_v = \dot{x}_5$ we obtain the valve velocity $\dot{x}_5$ as

$$\dot{x}_5 = -\omega_e^2 x_4 - 2\zeta\omega_e^2 \dot{x}_5 + \omega_e^2 \mu \quad (3)$$

Where, $T(s)$ is the transfer function, ω_e is the undamped natural frequency and ζ is the damping ratio of the system. Defining the load pressure P_L as $P_L = P_1 - P_2$ and the load flow Q_L as $Q_L = (Q_1 + Q_2)/2$, the relationship between the load pressure P_L and the load flow Q_L for an ideal critical servo valve with a matched and symmetric orifice can be expressed as follows:

$$Q_L = C_d w x_v \sqrt{\frac{P_s - \mathrm{sgn}(x_v)P_L}{\rho}} \quad (4)$$

The piston force equation is given by

$$P_L A = m\ddot{x} + kx + F_f \quad (5)$$

The equation of motion of the load mass is $m\dot{x}_2 = F_{hyd}$, where m is the total mass of the load, and $F_{hyd} = AP_L$ is the hydraulic force due to pressure differential across area, we obtain

$$m\dot{x}_2 = F_{hyd} = AP_L \quad (6)$$

Defining the load pressure from equations (1) and (4).

$$\dot{P}_L = -\left(\frac{4\beta_e A}{V_t}\right)\dot{x} - \left(\frac{4\beta_e C_{tp}}{V_t}\right)P_L + \left(\frac{4\beta_e}{V_t}\right)Q_L \quad (7)$$

$$\dot{P}_L = -\alpha\dot{x} - \beta P_L + \left(\frac{4\beta_e}{V_t}\frac{C_d w}{\sqrt{\rho}}\right)x_v\sqrt{P_s - \mathrm{sgn}(x_v)P_L} \quad (8)$$

where $\alpha : \left(4\beta_e A / V_t\right)$

$\beta : \left(4\beta_e C_{tp} / V_t\right)$

$\gamma : \left(4\beta_e C_d w / V_t \sqrt{1/\rho}\right)$

Combining Eq. (1)-(5) with other system parameter, results in the system state equations given below:

$$\dot{x}_1 = x_2$$

$$\dot{x}_2 = \frac{1}{m}\left(Ax_3 - F_f\right)$$

$$\dot{x}_3 = -\alpha x_2 - \beta x_3 + \left(\gamma\sqrt{P_s - \mathrm{sgn}\left(x_4\right)x_3)}x_4\right)$$

$$\dot{x}_4 = x_5$$

$$\dot{x}_5 = -\omega_e^2 x_4 - 2\zeta\omega_e x_5 + \omega_e^2 \mu \quad (9)$$

2.3. Hydraulic Servo System Parameter Identification

The data is separated into estimation data, which is used for identifying unknown system parameters and measurement data. Both factors are used for modeling in the experiments. In order to excite all the relevant frequencies of the systems and to construct a good model, the frequencies are set for sinusoidal inputs with range of 1 to 6 Hz and pressure of 5 Mpa. In conventional design of a hydraulic servo system, third order transfer function is generally used, as given below

$$G(s) = \frac{K_q \dfrac{\omega_n^2}{A_1}}{s\left(s^2 + 2\delta\omega_n s + \omega_n^2\right)} \quad (10)$$

In frequency response analysis, which measure the amplitude of oscillations at the signal frequency. At first, which carried out a set of experiments using the open loop system to determine the amplitudes of oscillations, which occur at the signal frequency. In order to observe the signal frequency component of the response only, experiments are carried out using signal frequencies of 1, 1.5, 2, 2.5, 3, 3.5, 4, 4.5, 5, 5.5 and 6 Hz at 5 Mpa pressure. Two different values of signal frequency are used in the experimental determination of sine wave and step response. Overall, the system identification is done by fitting a third order polynomial. The system transfer function is found to be:

$$G(s) = \frac{9272}{s^3 + 205.5s^2 + 1.056e4s} \quad (11)$$

The correspondence levels between predicted model and the experimental data for second order and the third order models are about 74.16% and 80.77 %, respectively. Fig. 2 shows a comparison between measured frequency responses.

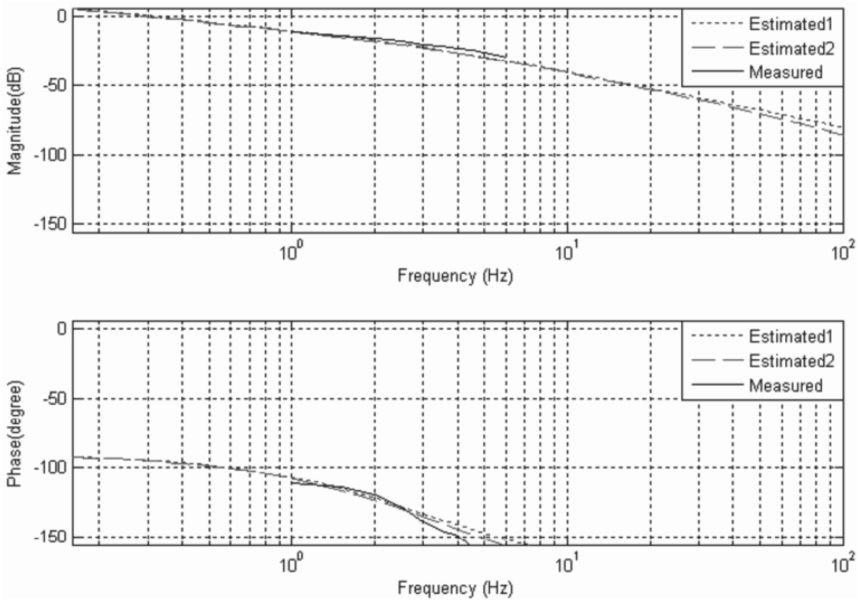

Fig. 2. Bode plot of the open loop transfer function: (esti-mated 1) second order and (estimated 2) third order.

2.4. Simulink Model

The parameter variations used in simulations for input, output, velocity, pressure, and flow rate. The system parameters were varied so as to keep the damping coefficient (ζ) constant. The values of the system and observer parameters are given in Table 1.

Table 1. Physical parameters for the HSS.

Name	System	Nominal valve	Unit
Supply pressure	P_s	$5*10^6$	Pa
Total actuator volume	V_t	$2.44*10^{-5}$	m^3
Effective modulus	β_e	$7.60*10^8$	N/m^2
Actuator ram area	A	$2.40*10^{-4}$	m^2
Actuator piston position	χ_v	0.2	m
Total leakage coefficient	C_{tp}	$5*10^{-13}$	$m^3/(s.Pa)$
Discharge coefficient	C_d	0.61	-
Spool valve area gradient	w	$2.39*10^{-3}$	m
Spool valve position (stroke)	χ	$0.238*10^{-3}$	m
Load flow	Q_L	$2.5*10^{-5}$	m^3/sec
Fluid mass density	ρ	870	kg/m^3
Mass of actuator and load	m	9.19	kg

The main idea of a new structure of HSS is the simu-link model of an open loop system. Equation (10) is obtained in system state equation of HSS.

At first, the system is characterized by position control (x_1). The piston stroke is ± 0.2m, mass of actuator and load $m = 9.19$Kg. The friction relation is measured as linear $E_f : 0.3063$N. Secondly, the system is characterized by velocity control (x_2). Let $\dot{x}_1 = x_2$ be a differentiable function, given by $\int \dot{x}_1 dt = \int x_2 dt$: ie. we obtain the result in terms of $x_1 = \frac{1}{s}(x_2)$. The response is a set of the velocity and its top most displacement range is ± 0.16 m/s. Referring the equation (9) $[\dot{x}_2 = \frac{1}{m}(Ax_3 - F_f)]$ where is $\frac{1}{m}$ gain. The line block is required for each channel, where velocity $\dot{x}_2$ is calculated. For that, area & force is added in to the block & friction is subtracted from the block.

The third method is load pressure system (x_3). This is the most effective method. The pressure of the system is set considering the pressure of the load. Pressure limit is the supply pressure P_L: 5MPa. The value of $\dot{P}_L$ is determined by substituting $\dot{P}_L = x_3$. To solve this equation, we may assume a solution x_3 of the form

$$x_3 = -Ax_2 + Q_L\left(\frac{1}{\frac{V_t}{4\beta_e} + C_{tm}}\right) \tag{12}$$

Overall, the main idea of the modified hydraulic subsystems is to use valve position (x_v), valve velocity ($\dot{x}_v$) and input current (μ). The upper and lower limits in servo valve areequal to the stroke of the servo valve x_v : $\pm$ $2.38*10^{-4}$ m and the maximum flow limit through the valve $Q_L : 2.5*10^{-5}$ m^3/s.

2.5. Simulation Result

The results of an open loop test and corresponding simulation model is presented for 1 Hz sin signal. From the result the correlation between the control signal frequency and error signal can be compared as follows:

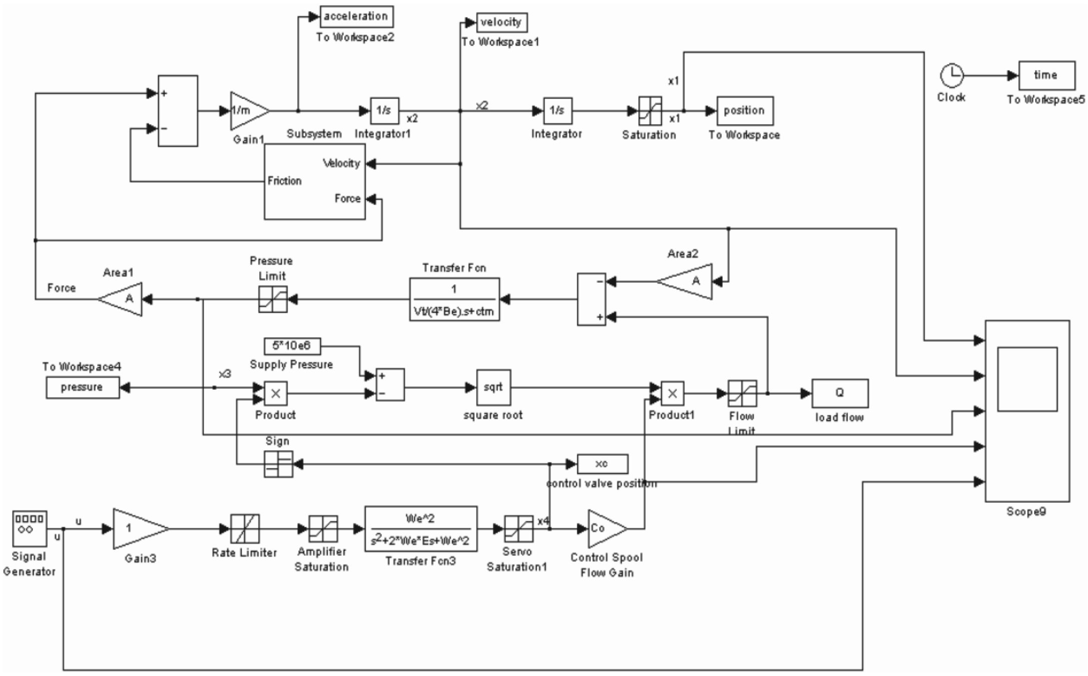

Fig. 3. The simulink model of the open loop system.

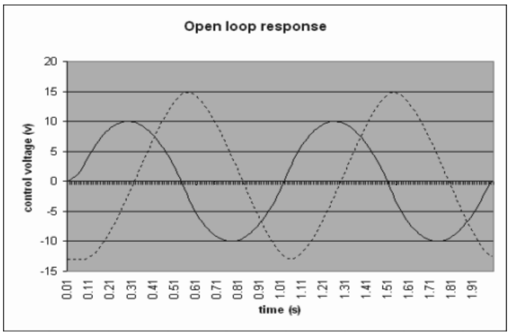

Fig. 4. Experimental of the open loop system for 5 Mpa and 1 Hz sine wave.

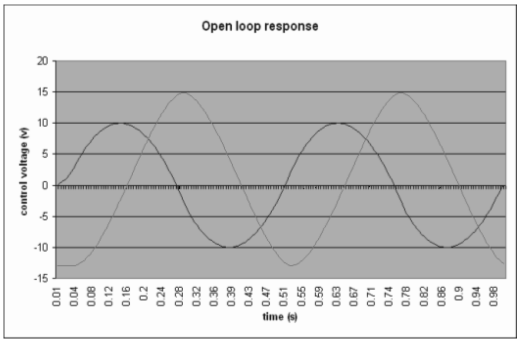

Fig. 5. Simulink model of the open loop system for 5 Mpa and 1 Hz sine wave.

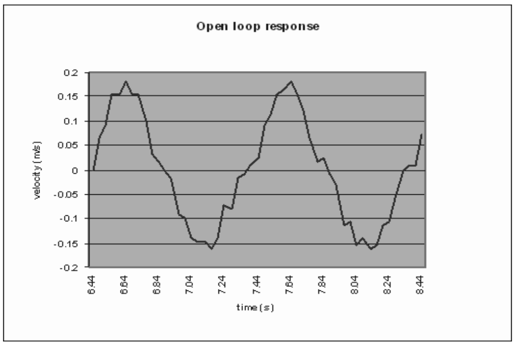

Fig. 6. Experimantal: velocity of the open loop system for 5 Mpa and 1 Hz sine wave.

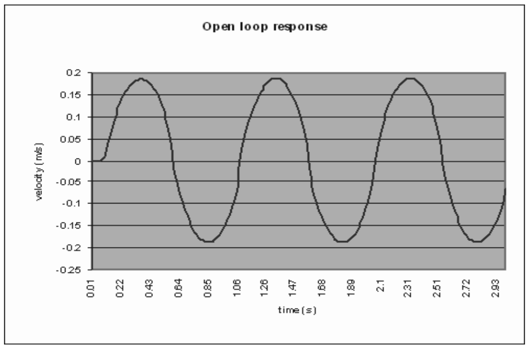

Fig. 7. Simulink model: velocity of the open loop system for 5 Mpa and 1 Hz sine wave.

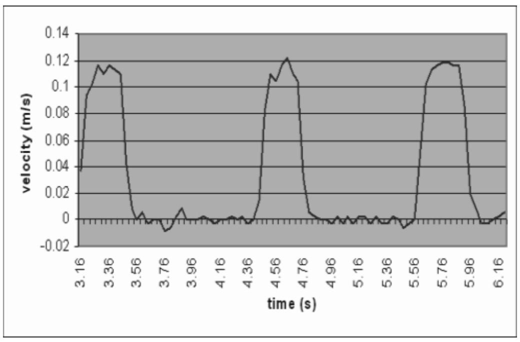

Fig. 8. Experimental: velocity of the open loop.

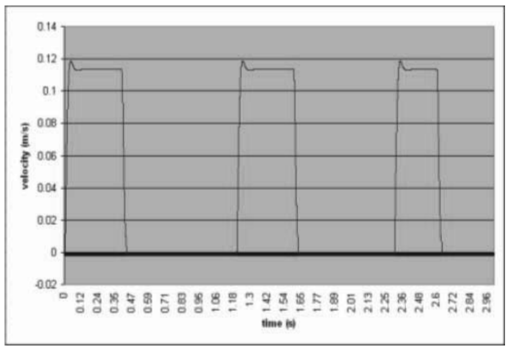

Fig. 9. Simulink model: velocity of the open loop system for 5 Mpa and step response.

3. PD Control Design

The PD regulators for process control are the basis for many hydraulic control systems. Many variations in the basic PD algorithm substantially improve its performance and operability. A PD controller is parameterized by the following transfer function.

$$\frac{X(s)}{R(s)} = \frac{C(s)G(s)}{1 + C(s)G(s)} \tag{13}$$

This can be compared wit the following, general third-order characteristic equation.

$$G(s) = \frac{9272}{s^3 + 205.5s^2 + 10560s} \tag{14}$$

Thus from equation (14), we obtain

$$= \frac{9272K_d(s + \frac{K_p}{K_d})}{s^3 + 205.5s^2 + (10560 + 9272K_d)s + 9272K_p} \tag{15}$$

The parameters of the PD controller are obtained as $K_d = 10.46$ and $K_d = 0.074$. This controller is applied to both the plant and model.

4. Hardware System Architecture

The HSS must follow the control theory guidelines, which is the main purpose to improve the velocity of piston in HSS. The hardware system of HSS is divided into two parts, which are explained below.

4.1. Hardware Design

Firstly, which construct the mechanical model of an electro-hydraulic system. The simulated response of the model provides insight into the behavior of electro-hydraulic system.

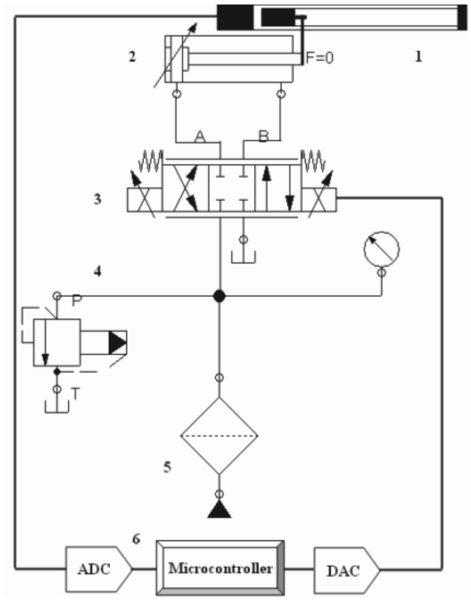

Fig. 10. Schematic diagram of the electro-hydraulic position control system.

As shown in Fig 10, (1) is the linear potentiometer; (2) is double cylinder; (3) is servo valve; (4) pressure relief valve represent fluid flows in out of the valve; (5) pressure unit is the input and output line pressures and (6) is the microcontroller to control system.

The HSS consists of a hydraulic pump, servo valve, actuator, transducer, power supply, and microcontroller. Hydraulic system model is shown in Fig.11.

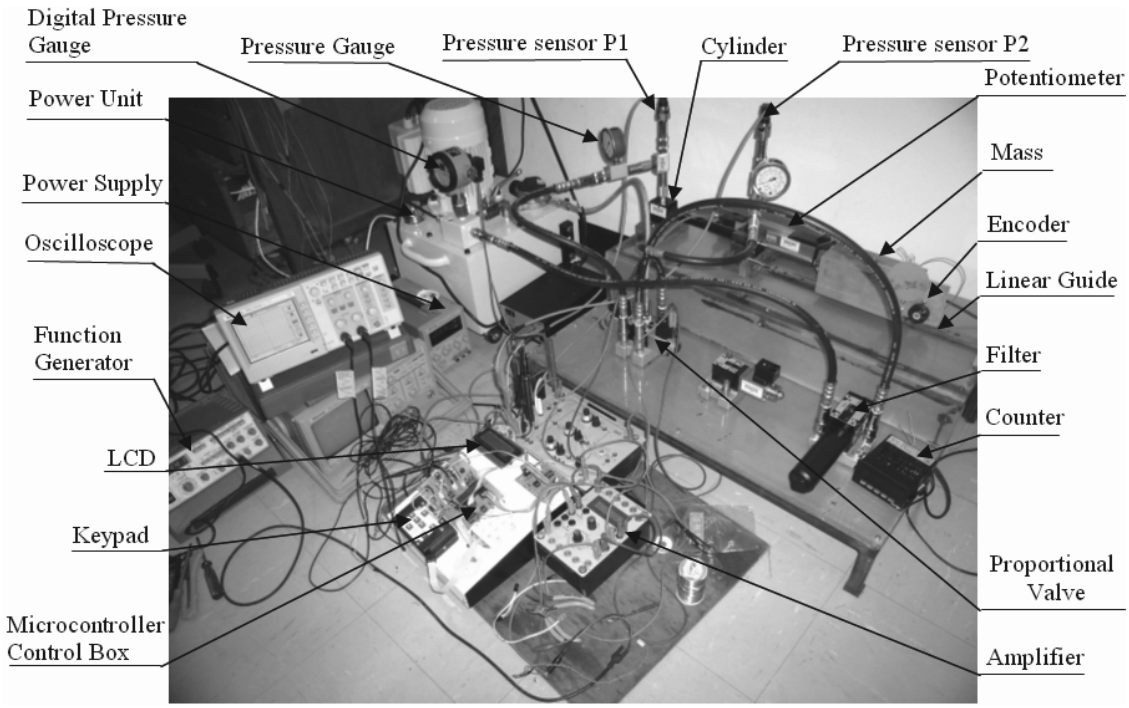

Fig. 11. Electro-hydraulic servo system model.

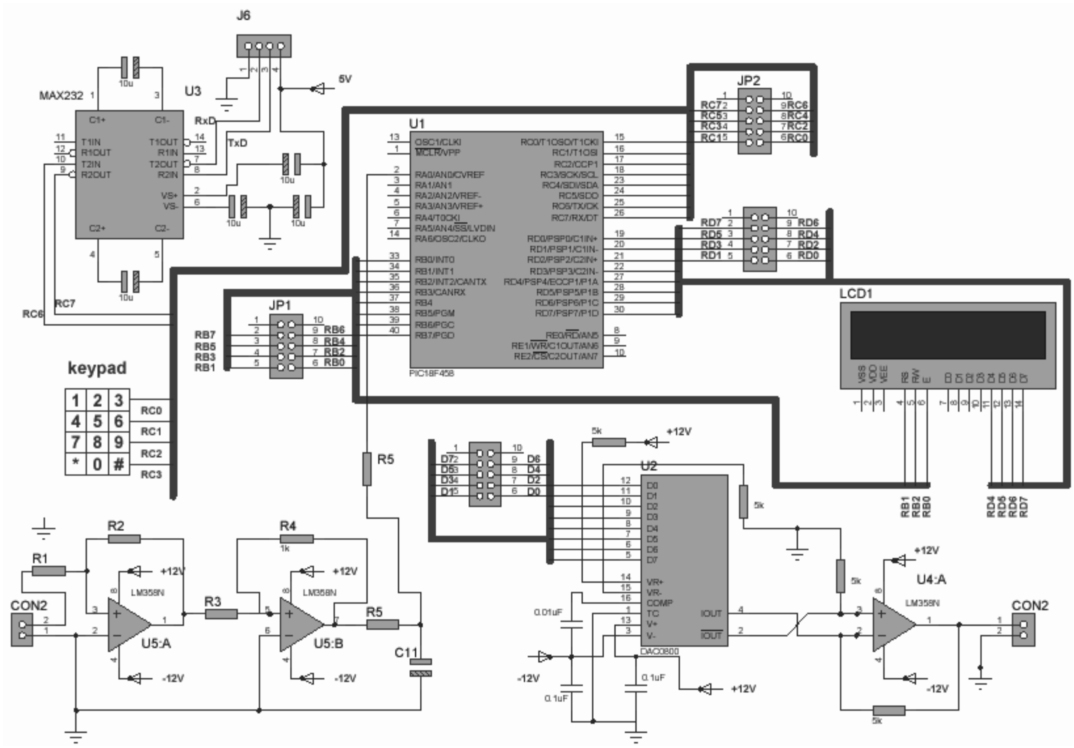

Fig. 12. Schematic of microcontroller system design.

A microcontroller based control system has been developed and used to control the hydraulic servo system. We used microcontroller PIC 18F458 to control the hydraulic servo system, in conjunction with the data acquisition processor. The schematic of microcontroller system design is shown in Fig. 12. The mass flow rate across the five-port valve is controlled by manipulating the spool offset, by controlling the current supplied to the solenoid.

5. Simulation and Experiment Results

The model is based on electro-hydraulic system control. A simulation of Matlab/Simulink based study is used to evaluate the performance of the PD controllers on the nonlinear model of HSS and compared with the performance of the controllers on linearized approximation of the system for displacement and velocity control.

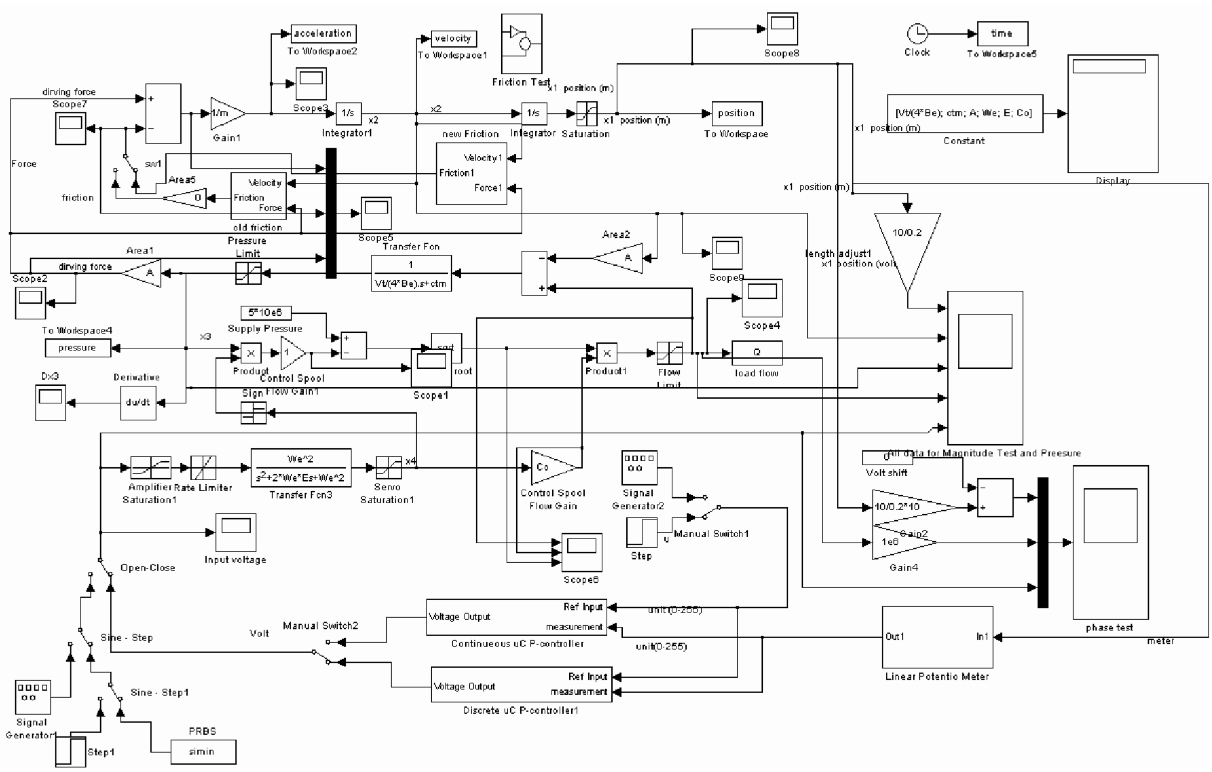

Fig. 13. Matlab Simulink Model of closed loop system.

5.1. Simulink Model for Closed Loop Control

The core of simulink is the HSS for hydraulic mini press machine. The parameters of a model must be identified for designing control algorithm and the steps of the identifying dynamic models of the hydraulic system involved designing an experiment, selecting a model structure, choosing a criterion to fit, and devising a procedure to validate the chosen model. Figure 13 shows the simulink model with Matlab program.

In simulink model, recall that for open loop system we used simulation for input and output with frequency response. Matlab simulink model is built such as, continuous microcontroller controls block, discrete microcontroller controls block, velocity, force and friction block and linear potentiometer block. The proposed model system is designed based on the use of a single-chip microcontroller PIC 18F458 with the EPROM emulator for programming the computer software. Discrete microcontroller block needs digital delay line block and two input (addition & subtraction) block for each channel. The addition point is fed with the direct input signal, while this signal was fed to the subtracted point after passing through the measurement line. The digital delay line parameter must be chosen to be equal with the number of the quantized duration samples. Figure below shows the details of the controller subsystem.

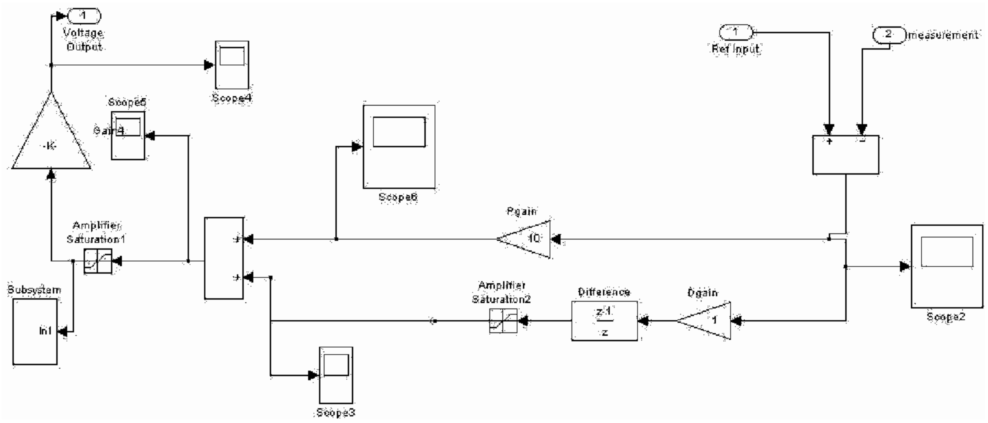

Fig.14. Block diagram of continuous microcontroller control.

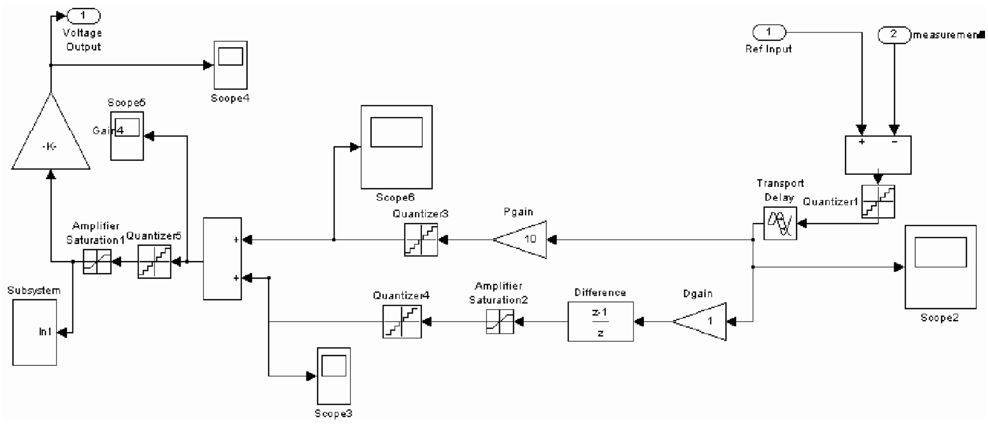

Fig. 15. Block diagram of discrete microcontroller control.

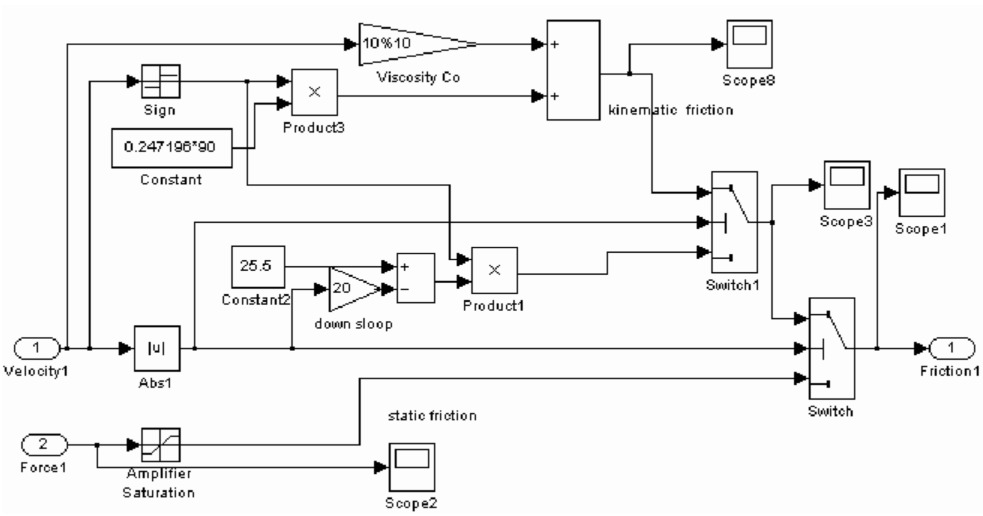

Fig. 16. Block diagram of velocity, force, and friction control.

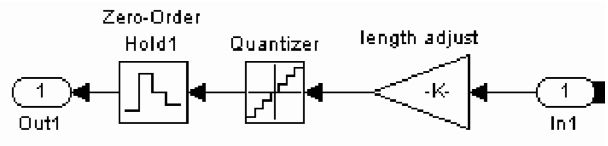

Fig. 17. Block diagram of linear potentiometer.

5.2. Simulation Results

The simulation is used to examine the effect of errors in the system parameters. These parameters include the total leakage coefficient, effective bulk modulus system mass, friction terms and so on. The results of simulations are presented in this section. Following are the system parameters used for simulation: m = 9.19 kg, A = 2.4e–4, P_s = 5e6 Pa, V_t = 2.44e–5 m^3, F_f = 0.247 N, β_e = 3.6e8 N/m^2, w = 0.00239 m, $alpha$ = 7.2e9, $beta$ = 29.5, $gamma$ = 2 .917 e9, C_0 = 4.9427e–5, C_d = 0.61, ζ = 0.7, and C_{tm} = 7.52 –12 m^3/(s.Pa).

5.3. Experimental Set Up

A set of experiments is performed to study the effect of variation in supply pressure, hydraulic parameters, and environmental stiffness and position control on velocity piston in HSS. In order to test the performance of developed system, which compared the system output with simulink model. A typical closed loop control hydraulic system consists of power supply with 4 lit/min flow rate and 5 Mpa supply pressure. Load piston position is measured by FESTO Model TP 501 with 200 mm length of stroke and four way 3 state proportional directional control valve, which is measured by VICKERS Model KBSDG4V-3. Linear potentiometer from FESTO Model FP1120 is used to record the position of the cylinder and DUPOMATIC Model PTH-100/20E0 pressure sensors are used to measure P1 and P2 (Vickers).

5.3.1. Validation

To verify the model, both static and dynamics tests are performed in simulation and compared with the experimental results. In case of the model validation and PD controller experiments, the controller is modeled using simulink approach in Matlab program. Experimental results for PD controller which are based on the control of position and velocity of position at an operating supply pressure of 5 Mpa are given below.

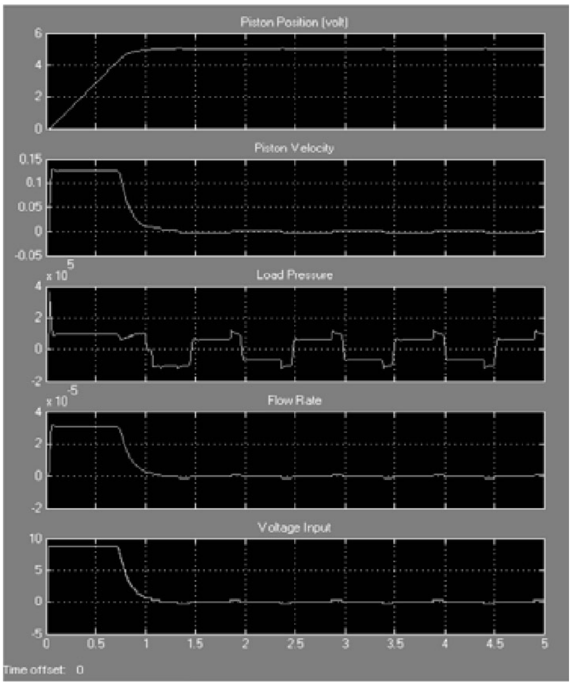

Fig. 18. Observer performance of step responses with Kp = 10 and Kd = 0.074.

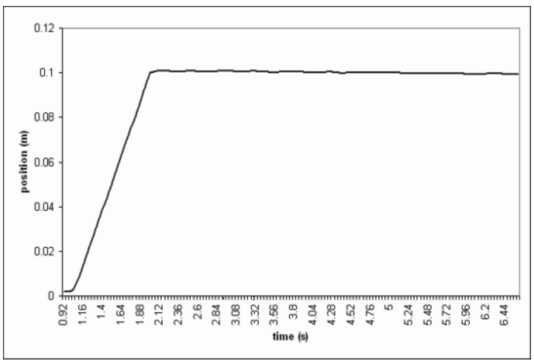

Fig. 19. Unit step responses of experiment with Kp = 10 and Kd = 0.074.

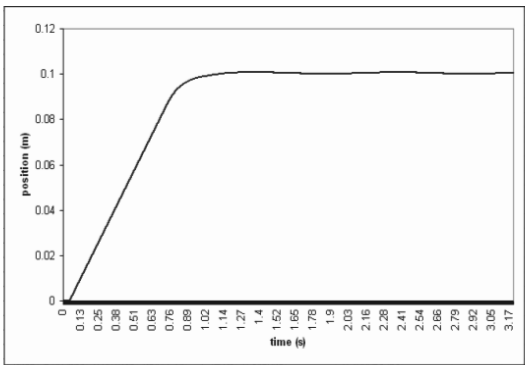

Fig. 20. Unit step responses of simulation with Kp = 10 and Kd = 0.074.

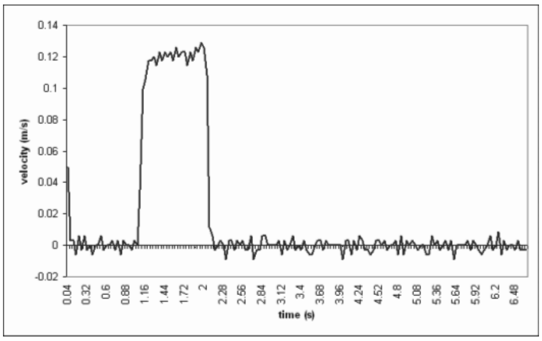

Fig. 21. Unit velocity responses experiment with Kp = 10 and Kd = 0.074.

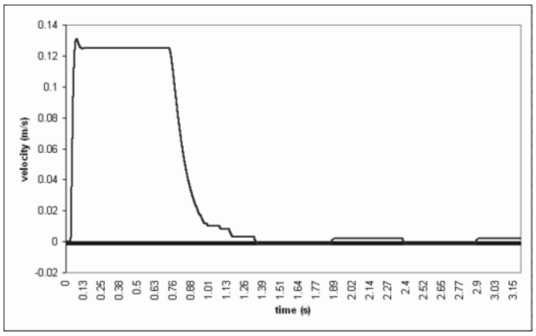

Fig. 22. Unit velocity responses simulation with Kp = 10 and Kd = 0.074.

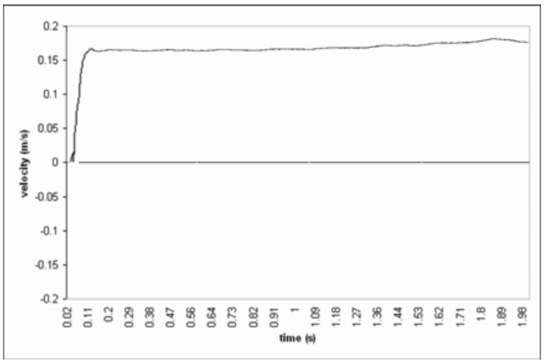

Fig. 23. Unit velocity responses simulation [4].

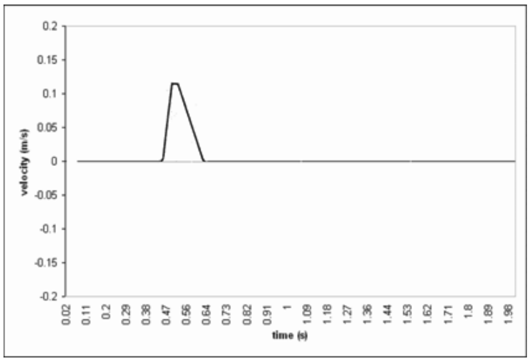

Fig. 24. Unit velocity responses simulation [8].

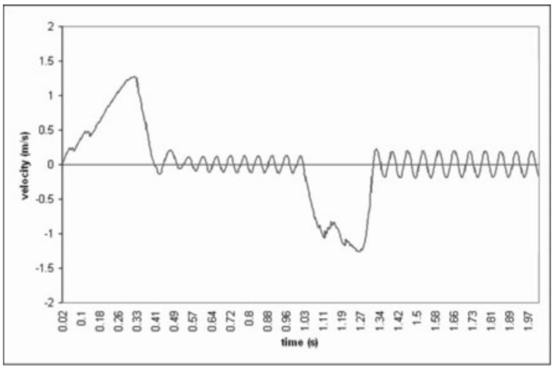

Fig. 25. Unit velocity responses simulation [5].

During the experimental test, only linear position of the movable cylinder is measured for feedback information. The velocity is obtained *via* differentiating the position with respectto time at high speeds and pressure. To test the effectiveness of the proposed controllers, we first compared it with a high-speed control system. The proposed control velocity is shown in Figure 22 with the gains $Kp = 10$, $Kd = 0.074$, $P_s = 5$ MPa, $mass = 9.19$ Kg, $\beta_e = 3.60*10^8$ Pa, $velocity = 0.125$ m/s, $rise\ time = 0.28$ s, and microcontroller control system. In [4], the control result of the proposed controller is shown in Figure 23, with corresponding feedback gains $P_s = 5$ MPa, $\beta_e = 5*10^9$ Pa, $mass = 20$ Kg, $Q = 28$ l/min, $velocity = 0.165$ m/s, and computer control system. In [8], the control result of the proposed controller is shown in Figure 24, with corresponding feedback gains $P_s = 20$ MPa, $\beta_e = 8.5*10^8$ Pa, $mass = 5$ Kg, $distace = 200$ mm, $velocity = 0.13$ m/s and computer control system. In [5], the control result of the proposed controller is shown in Figure 25, with corresponding feedback result $velocity = 1.25$ m/s, $rise\ time = 0.35$ s, overshooting is less than 5% and computer control system. The control result of the proposed controller [3], with corresponding feedback gains $P_s = 13.78$ Pa, $\beta_e = 3*10^8$ Pa, $V_t = 2.0$ m^3, $velocity = 0.8$ m/s and computer control system.

The difference in the results shown in Figure 22 demonstrates that the proposed controller is much better than Figures 23, 24 and 25 in terms of high speed, transient response and settling time. For the hydraulic mini press machine, the smaller the settling time is, the higher the production rate will be. Therefore, it is significant for practical applications. In addition, overshoot is much smaller. This is essential for hydraulic mini press machine, because a large overshoot will generate a large impact force, which would damage the sheet metal forming.

6. Conclusion

The system has presented deviation, simulation, and implementation of the nonlinear control law for HSS. The proposed controller provides performance of the PD controller for high-speed control. PD controller theory is introduced as the control technique to accomplish this goal in this study, and the controllers designed using this method are validated using experimental tests. From these tests, it can be seen that, for hydraulic systems, which have nonlinear characteristics, control theory provides a powerful control strategy that clearly improves on PD control in terms of high speed and transient response. The result shows the time constant of simulation and experiment, proportional-derivative controller with $Kp = 10$, $Kd = 0.074$.

The comparison of the results between simulink and hardware system for optimal PD controller indicates the improvement of the simulation research, where the reference velocity is 0.125 m/s and position is 39 mm. Thus, the response of the model gives a good control performance prediction of the PD controller.

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