THE VELOCITY CONTROL OF THE ELECTRO-HYDRAULIC SERVO SYSTEM

Magdy M. Abdelhameed[1], M. A. Abdelaziz[2], A. Bakarman[3]

[1]*Head of department, Professor, Mechatronics Engineering Department, Ain Shams University, Cairo*
[2]*Assistant Professor, Mechatronics Engineering Department, Ain Shams University, Cairo*
[3]*Engineering, Mechatronics Engineering Department, Ain Shams University, Cairo*

Abstract

In general two basic methods are used for controlling the velocity of a hydraulic cylinder. First by an axial variable-displacement pump for controls flow to the cylinder. This configuration is commonly known as a hydrostatic transmission. Second by proportional valve powered by a constant-pressure source, such as a pressure compensated pump, drives the hydraulic cylinder. In this study, the electro-hydraulic servo system (EHSS) for velocity control of hydraulic cylinder is investigated experimentally and its analysis theoretically. Where the controlled hydraulic cylinder is altered by a swashplate axial piston pump or by proportional valve to achieve velocity control. The theoretical part includes the derivation of the mathematical model equations of combination system. Velocity control system for hydraulic cylinder using simple (PID) controller to get constant velocity range of hydraulic cylinder under applied external variable loads . An experimental set-up is constructed, which consists of the hydraulic test pump unit, the electro-hydraulic proportional valve unit, the hydraulic actuator unit , the external load control unit and interfacing electronic unit. The experimental results show that PID controller can be achieve good velocity control by variable displacement axial piston pump and also by proportional valve under external loads variations.

Keywords: *Velocity control, Swashplate, Proportional valve, Hydraulic cylinder, PID controller, Axial piston pump*

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1. INTRODUCTION

EHSS is widely used in many industrial applications and mobile systems because of their high power to weight ratio, high stiffness, fast response, self cooling, good positioning capabilities, etc. EHSS systems can be classified into two different systems, hydraulic valve-controlled systems and hydraulic displacement-controlled systems. The hydraulic valve-controlled systems that have been applied widely in industrial servo control have higher response but lower energy efficiency. In contrast, hydraulic displacement-controlled systems, where displacement is adjusted by a variable displacement pump, have higher energy efficiency but lower control response owing to their complexity.

Velocity control of EHSS problems are treated by (Anderson, 1988; Bake, 1992; Jen and Lee,1992; Chern and Wu, 1992; Lierschaft, 1993; Boes, 1995; Huang and Wang, 1995; Bernzen, 1999)[9]. This study will focus on the velocity control of hydraulic cylinder by two methods. First method by swashplate axial piston pump where swashplat controlled by stepper motor to change flowrate of pump and then control velocity of hydraulic cylinder. Second method by Proportional valve to control flow from pump to cylinder and hence control velocity of hydraulic cylinder.

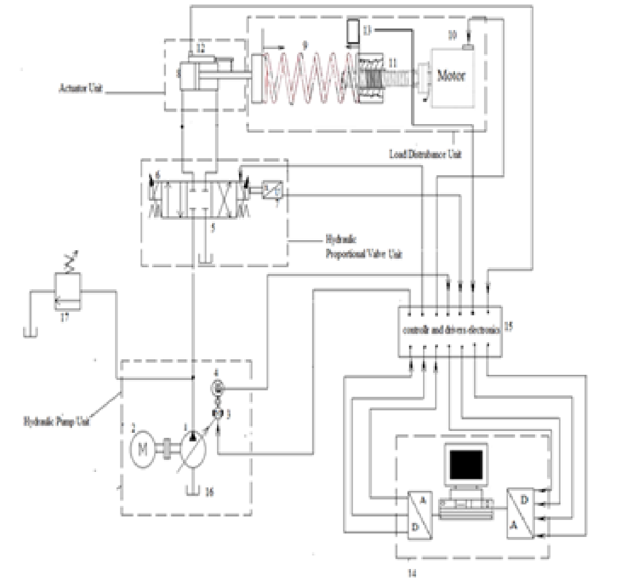
The rest of the paper is organized as follows. In Section 2, a short description of the experimental set-up. Then, in Section 3 mathematical model for the system is presented. In section 4, the implementation of PID controller is presented. Experimental results are presented in Section 5 and finally the paper is concluded.

2. EXPERIMENTAL SETUP DESCRIPTION

Figure 1 illustrates schematically the proposed system, it consists of five main units; the hydraulic test pump unit, the electro-hydraulic proportional valve unit, the hydraulic actuator unit, the load disturbance unit and interfacing unit.

As shown in Figure 1, the unit of the hydraulic pump consists mainly of the test pump (1), which is a piston pump type (Model PVQ 06L ASY, OilGear) that has flowrate about 6 L/min at 1000 rpm. The test pump is coupled with an electric motor (2) that has 1.5 kW of output power and uses a flexible coupling to absorb any axial and/or lateral vibration between the connected shafts. The swashplate is rigidly connected to a stepper motor (3), which drives the swashplate to change its inclination angle.

As shown in Figure 1, the main part of electro-hydraulic proportional valve unit is directional control valve (5),which is 4/3 proportional control valve (Model TGL 55074, ORSTA hydraulic). Two solenoids (6) actuate the spool of valve in two sides using a coil. When the coil is energized, the spool is moved to the desired position.



1. Variable displacement piston pump, 2.Electric motor, 3.stepper motor swash plate, 4. Incremental encoder, 5.Main proportional valve, 6.solenoid valve, 7. Linear variable differential transformer (LVDT), 8.Hydraulic cylinder, 9.Mechanical spring, 10.Stpper motor, 11. Ball screw lead, 12. potentiometer transducer, 13. Pressure gauge transducer, 14.Computer with DAQ, 15.Controller and drivers electronics and electronic interface, 16.Tank and 17.Pressure relief valve.

Fig-1: Schematic diagram of the proposed system setup.

The actuator used is double acting hydraulic cylinder (8) as shown in figure 1 . Hydraulic cylinder is equipped with potentiometer transducer (12) (Gefran type) which has a useful stroke of 250 mm. The velocity of hydraulic cylinder detected with potentiometer, when potentiometer measured physical position and convert it to analog signals. The analog signals are sent to acquisition system and with derivative function can be estimated the velocity of hydraulic cylinder.

The load disturbance unit is built to simulate different modes of change in external load. When return to figure 1 ,seen that mechanical spring (9) represents to virtual different load patterns. Mechanical spring connected to hydraulic cylinder (8) in one side, while in other side connected to ball screw lead (11) and dc motor (10). Through properties of mechanical spring, find that the changes force of spring proportional to its compression distance. So that, the motor with ball screw lead can control the changes force of spring based on other acting side force through hydraulic cylinder. Light distance transducer (13) used to estimated external different loads patterns by knowing variable length of spring and fed back into electronic control and data acquisition.

In interfacing and electronic unit, the PC sends an input voltage signal through an analog and digital output ports on the data acquisition card (DAQ) board (NI USB-6009) to the drivers and amplifiers (such as swashplate stepper motor driver and solenoid valve amplifier). The output voltage from each sensor (potentiometer and light distance sensor) is sent to the PC through analog input ports on the DAQ. Simulink and Matlab are used as the user interface to generate and analyze all data to and from the PC. A photograph of the constructed experimental test set-up is shown in Figure 2.

Fig-2: Photograph of the experimental setup.

3. MATHEMATICAL ANALYSIS AND MODELING OF EHSS

3.1 Pump Unit Modeling

3.1.1 Mathematical Model of the Stepper Motor of Swashplate

If a sinusoidal characteristic of the magnetic field in the air gap is assumed, the contribution of each phase *j* on the motor torque T_{Mj} can be written as, [1][2][3]

$$T_{Mj} = k_m \sin\left(\Phi_j + N\theta_p(t)\right) I_j(t) \tag{1}$$

Where, k_m is the motor constant , θ_p is the actual rotor position, Ij is the current in the coil as function of time, Φ_j is the location of coil j in the stator, N is the number of rotor pole and P is the number of stator phases.

However the current $I_j(t)$ in the coil is a function of the supplied voltage Vj (t) and the coil properties. A general equation between $I_j(t)$and V_j (t) is given by,

$$V_j(t) = emf_j + RI_j(t) + L\frac{dI_j(t)}{dt} \tag{2}$$

Where emf_j is the electromotive force induced in the phase j, R is the resistance of the coils and L is the inductance of the coils However, the emf in each coil can be expressed as

$$emf_j = k_m \sin\left(\Phi_j + N\theta_p(t)\right) \omega_d \tag{3}$$

Where ω_d is the rotational velocity of the rotor.

The total torque produced by the stepper is given as

$$T_M = \sum_{j=1}^{P} T_{Mj} \tag{4}$$

Using Equation (4), and considering the equation of motion of a stepper motor,

$$T_M = J_d \frac{d\omega_d}{dt} + B_d\omega_d + T_d \quad (5)$$

Where, J_d is the inertia of the rotor, B_d is the viscous damping constant and T_d is the Load torque acting on the stepper motor shaft.

The angular velocity is given by

$$\omega_d = \frac{d\theta_p}{dt} \quad (6)$$

Hence for a 2 phase PM motor with Nr rotor teeth and the two phases Φ_j at 0 and (π/2) the following state space equations can be derived.

$$\frac{d\omega_d}{dt} = \frac{-k_m \sin(N\theta_p)I_a + k_m \cos(N\theta_p)I_b - B_d\omega_d - T_l}{J_d} \quad (7)$$

$$\frac{dI_a}{dt} = \frac{V_a - RI_a + k_m\omega_d \sin(N\theta_p)}{L} \quad (8)$$

$$\frac{dI_b}{dt} = \frac{V_b - RI_b + k_m\omega_d \sin(N\theta_p)}{L} \quad (9)$$

3.1.2 Mathematical Model of the Pump

3.1.2.1 Torque Model

Figure 3 illustrates the components and forces that have an effect on the total torque.

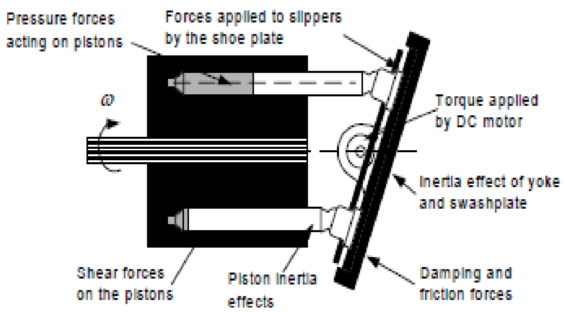


Fig-3: Forces that give rise to torques acting on the swashplate and yoke assembly

The load exerted by the swashplate can be divided into the following torques: [4][5-7]

- Torque produced by friction forces, T_f,
- Torque relating to the pressure effect, T_p,
- Torque relating to the rotation of the barrel, T_r and
- Torque required to overcome the swash plate inertia, $J_p\ddot{\theta}_p$

$$T_d = J_p\ddot{\theta}_p + T_f - T_p - T_r \quad (10)$$

where J_p is the average moment of inertia of swashplate yoke assembly and T_d is the torque applied to the yoke by the stepper motor.

The frictional torque includes coulomb friction, viscous damping friction and stiction. As mentioned, the stiction friction is assumed to be negligible. Hence the frictional torque can be represented by

$$T_f = \mathrm{sign}(\dot{\theta}_p)T_{fc} + B_p\dot{\theta}_p \quad (11)$$

where T_{fc} is the torque produced by the coulomb friction force, B_p is the damping coefficient of the swashplate yoke assembly.

However, the torque applied to the swashplate due to the pressure effect is significant. This torque is a function of both the pump pressure and swashplate angle and can be written as

$$T_p = K_{p1}P_p - K_{p2}P_p\theta_p \quad (12)$$

where K_{p1} is the pressure torque constant, K_{p2} is the pressure torque constant and P_p is the pump pressure.

When the pump is in operation, there is a torque applied to the swashplate by the piston slippers. This force is a result of the inertia of pistons and the shoe plate and is known to be a function of the swashplate angle. Therefore, this torque is related to the rotation of the barrel and can be represented as

$$T_r = -S_1 - S_2\theta_p \quad (13)$$

where S_1 is the simplified pump model constant and S_2 is the simplified pump model constant.

Substituting Eqs. (11)-(13) into Eq.(10),

$$T_d = J_p\ddot{\theta}_p + S_1 + S_2\theta_p + \mathrm{sgn}(\dot{\theta}_p)T_{fc} + B_p\dot{\theta}_p + K_{p2}p_p\theta_p - K_{p1}P_P \quad (14)$$

3.1.2.2 Flow Model of the Pump

The displacement of the pump is defined as follows: [4][8][5-7]

$$D_p = NA_pR_p\tan\theta_p/\pi \quad (15)$$

where D_p is the displacement of the pump, R_p is the radius of the piston pitch, N is the Number of pistons and A_p is the Area of the piston.

Assuming that the rotational speed of the prime mover is ω_p, the ideal flow rate of the pump is as follows

$$Q_{pidea} = \omega_pD_p = \omega_pNA_pR_p\tan\theta_p/\pi \quad (16)$$

From the continuity equation, the flow equation for the pump can be written as:

$$Q_{pidea} - Q_{ip} - Q_{ep} - Q_p = \frac{V_p}{B}\frac{dp_p}{dt} \quad (17)$$

where Q_p is the output flow of the pump, Q_{ip} is the internal leakage flow of the pump, Q_{ep} is the external leakage flow of the pump, V_p is the volume of the pump forward chamber and B is the bulk modulus of the fluid.

Since the suction pressure is assumed to be zero, the leakage flow of the pump (including the internal leakage and the external leakage flow) can be approximated by:

$$Q_{lp} = Q_{ip} + Q_{ep} = C_{tp} p_p \tag{18}$$

where C_{tp} is the Total leakage flow coefficient.

Substituting Equations (16) and (18) into Equation (17), yields

$$\omega_p N A_p R_p \tan\theta_p / \pi - C_{tp} p_p - Q_p = \frac{V_p}{B}\frac{dP_p}{dt} \tag{19}$$

Equation (19) is thus the flow model of the pump.

3.2 Proportional Valve Model

3.2.1 Dynamics of the Spool Valve

The dynamic of a spool valve follows approximately a linear differential equation of second order described in equation,[1][35]

$$m_v \ddot{x}_v + c\dot{x}_v + kx_v = F \tag{20}$$

where m_v is the mass of the spool, c the damping factor ,k is the spring stiffness and x_v is the spool valve position. The force F can be seen as an input to the system, e.g. the input voltage to the valves.
By substituting $k = m_v\omega_n^2$ and $c = 2\boldsymbol{\zeta}\sqrt{\mathbf{km_v}}$, equation (20) becomes

$$\ddot{x}_v + 2\omega_n \boldsymbol{\zeta}\dot{x}_v + \omega_n^2 x_v = \omega_n^2 k_v u \tag{21}$$

where u is the voltage control input, k_v is the proportional valve gain, ω_n is the natural frequency and ζ is the damping ratio.

Written equation (21)as a relation between input and output finally yields in the transfer function of the form,

$$\frac{x_{v(s)}}{u(s)} = \frac{\omega_n^2 k_v}{s^2 + 2\omega_n \boldsymbol{\zeta s} + \omega_n^2} \tag{22}$$

3.2.2 Flowrate Through the Proportional Valve Restriction Areas

The proportional valve spool displacement throttles oil flow through the four control gaps of the valve since the valve open center type as in figure 4., [11]

$$Q_a = C_d w(x_{vmax} - x_v)\sqrt{\frac{2(p_A - p_t)}{\rho}} \tag{23}$$

$$Q_b = C_d w x_v \sqrt{\frac{2(p_p - p_A)}{\rho}} \tag{24}$$

$$Q_c = C_d w(x_{vmax} - x_v)\sqrt{\frac{2(p_p - p_B)}{\rho}} \tag{25}$$

$$Q_d = C_d w x_v \sqrt{\frac{2(p_B - p_t)}{\rho}} \tag{26}$$

Where p_p is the Supply pressure, p_t is the Return pressure, p_A and p_B are the pressure acting on the two sides of the piston of the cylinder, ρ is the Oil density, C_d Discharge coefficient and w is the width of the valve port.

$$Q_A = Q_b - Q_a \tag{27}$$

$$Q_B = Q_c - Q_d \tag{28}$$

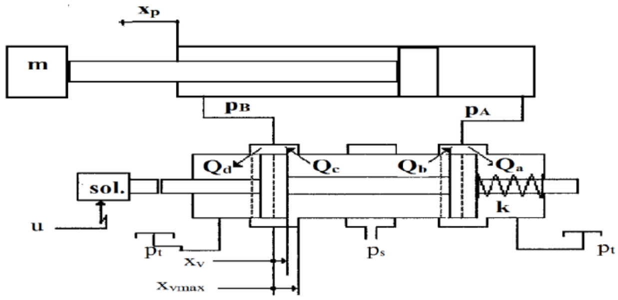


Fig-4: Hydraulic actuator with four-way valve configuration

3.3 Hydraulic Cylinder Modeling

As in figure 4 the application of the continuity equation to the cylinder chambers yields the following equations,[9][11]

$$p_A = \frac{B}{V_o + A_p x_p}\int (Q_A - A_p\dot{x}_p)\, dt \tag{29}$$

$$p_B = \frac{B}{V_o - A_p x_p}\int (Q_B - A_p\dot{x}_p)\, dt \tag{30}$$

where Ap is the Piston area, Vo is the half of volume of oil filling the cylinder, B is the fluid bulk modulus and x_p is the piston position.

The piston is driven by the pressure difference in the hydraulic cylinder chambers. The equation of motion of the piston becomes as follows,[9][2]

$$(p_A - p_B)A_p = m\ddot{x}_p + f_v\dot{x}_p + F_L \tag{31}$$

Where m is the total mass of the piston and f_v is the viscous friction cofficient and F_L is the external load variables.

3.4 Modeling of External loads Unit

Such as in Figure 1, the external load can be represented by mechanical spring which attached by ball screw to control the force. Th equation of external force formulated as follows,

$$F_L = k_{mech}(x_p + x_{screw}) \tag{32}$$

Where k_{mech} is the stiffness of spring and x_{screw} is the displacement of spring due to ball screw motion.

4. PROPORTION INTGRAL DERIVATIVE (PID) CONTROLLER

Classical PID controllers are very popular in industries because they can improve both the transient response and steady state error of the system at the same time. The performance specifications of the systems such as rise time, overshoot, settling time and error steady state can be improved by tuning value of parameters K_P, K_I and K_D of the PID controller, because each component has its own special purposes. K_P or proportional controller is used to assure the output reach the reference input. However, the output of the system with this controller will never reach zero steady state error. In order to obtain zero or very small steady state error, KI or integral controller is given to the system. Derivative controller or KD will improve the speed performance of the system. Mathematically it is represented as

$$u(k) = K_P e(k) + K_I \sum_{i=0}^{k} e(i) + K_D[e(k) - e(k-1)] \tag{33}$$

where e(k) is the error signal.

The PID control method has been widely used in industry during last several decades because of its simplicity. To tune these parameters, the model is linearized around different equilibrium points. PID controller could be determined by many methods which lead to regulate the (PID) controller for example trial and error method or Ziegler and Nichols method.

5. THE EXPERIMENTAL RESULTS

The aim of this section is to test the velocity control strategy on the experimental set-up that shown in Figure 2. PID controller applied to the variable pump controlled and valve controlled by simulink and DAQ as the development platform and shown in figure 6 and 7. A testing of response of the system was performed using a step input to ensure constant velocity of hydraulic cylinder .

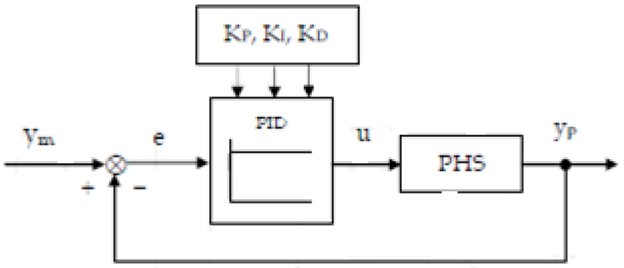


Fig-5: Block diagram of a PID controller.

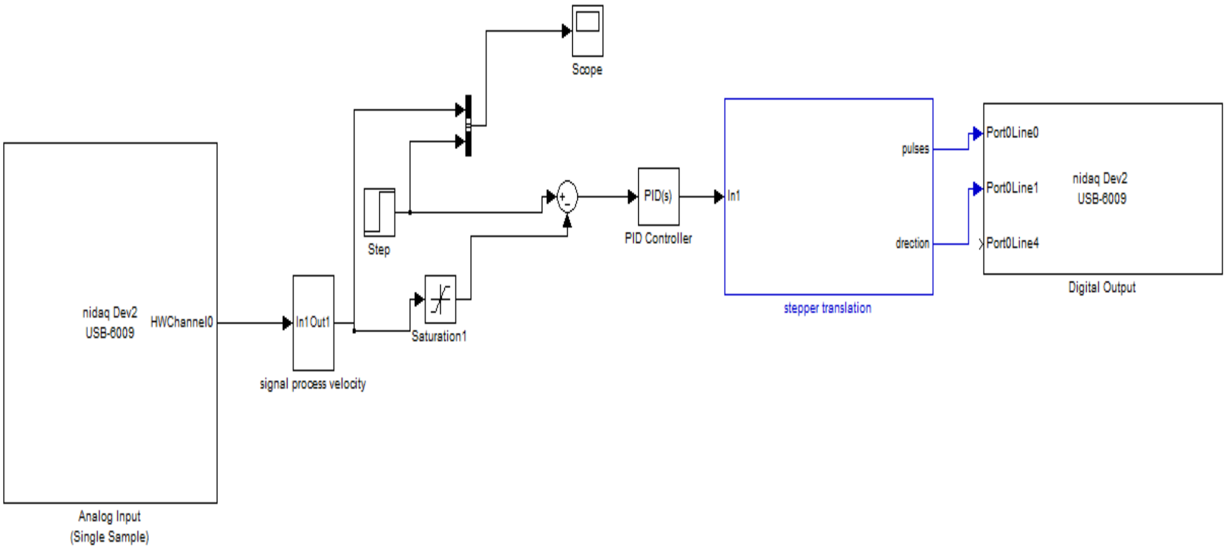


Fig-6: The control velocities by variable pump and developed by simulink

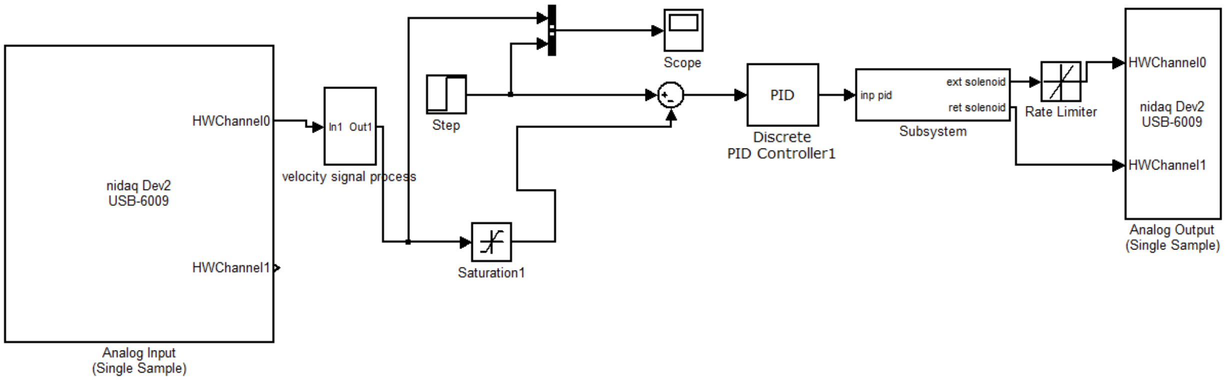
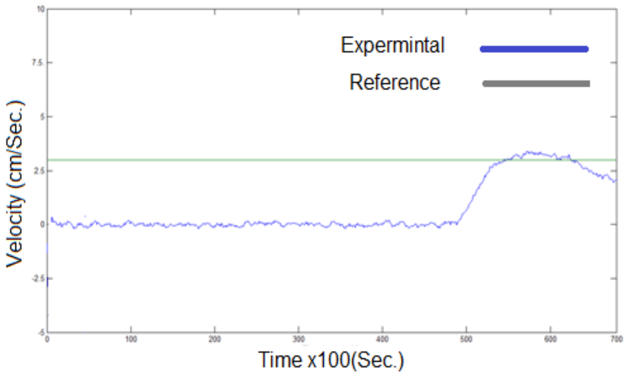


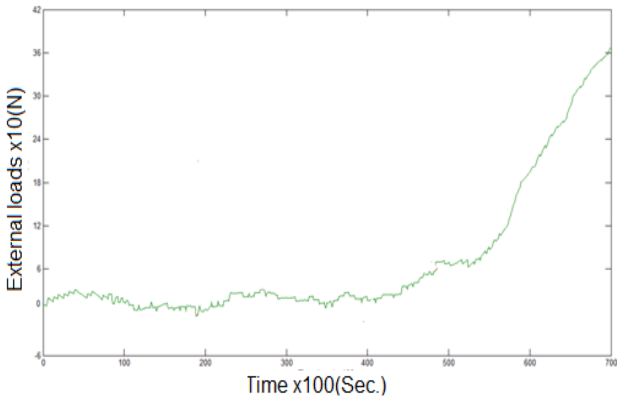
Fig-7: The control velocity by proportional valve and developed by simulink

5.1. Velocity Control of Hydraulic Cylinder by Proportional Valve

Figure 8 shows the velocity form controlled under variable linear external load. The parameter values tuned of the PID controller are P=1.7, I=0.9, and D=0.1. Where external load form changed, velocity response can be presented in figure 9 by same PID parameters. In figure 10 shows the tested velocity response without any external load by PID parameters as P= 1.2, I= 0.8 and D = 0.1.

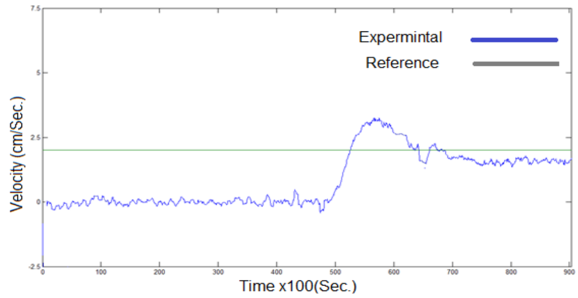


(a)

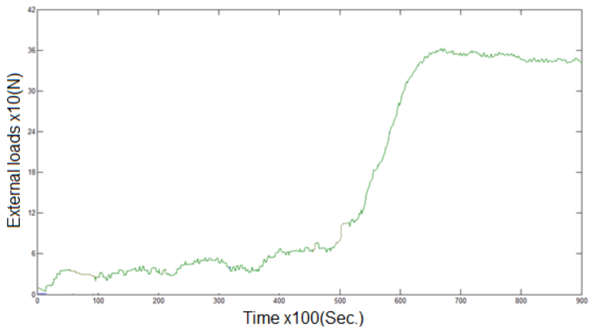


(b)

Fig-8: Experimental results of constant velocity by controlled valve under form of external loads.



(a)



(b)

Fig-9: Experimental results of constant velocity by controlled valve under form of external loads

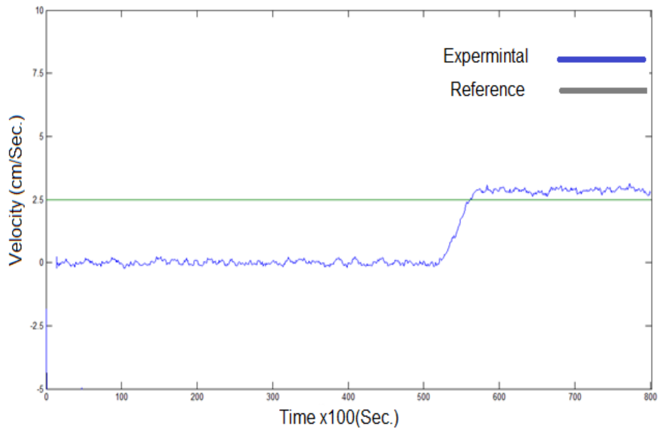


Fig-10: Experimental results of constant velocity by controlled valve without any external loads

5.2 Velocity Control of Hydraulic Cylinder by Swashplate Axial Piston Pump

Fig. 11 shows the velocity form controlled under variable external loads. The parameters values tuned of the PID controller are P=70, I=70 and D=5.

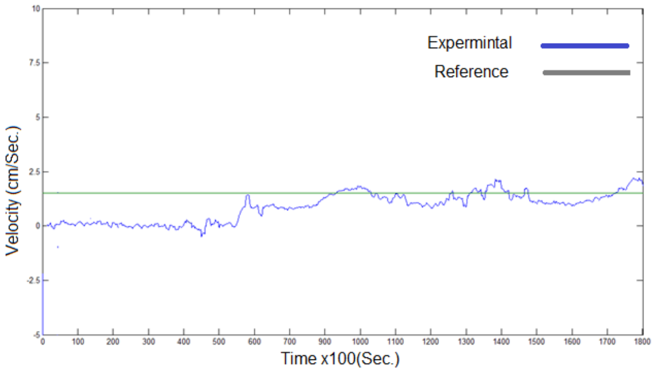


Fig-11: Experimental results of constant velocity by variable pump under external loads

6. CONCLUSION

A model equations for EHSS has been derived. PID 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. The comparison of the results between test responses indicates the rising time of the various velocity valve controlled responses are at 5 sec., while the rising time of the velocity variable pump controlled response is at 6 sec. Thus, the control velocity by controlled valve has higher responses than variable pump controlled.

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BIOGRAPHIES

Magdy Abdelhameed: received his B.S.c., M.S.c. and PhD degrees from AinShams University, faculty of Engineering, Mechanical Engineering Department in 1983, 1989 and 1994 respectively. Currently he is enrolled as a head of the mechatronics Department at AinShams University, faculty of Engineering. His research interests lie in the field of electro-hydraulic system.

Mohamed A. Abdelaziz: received his B.S.c., degrees from Ain Shams University, faculty of Engineering, Mechanical Engineering department in 2001. He then received his M.S.c and PhD. degrees from University of Illinois, Chicago, USA, faculty of Engineering, Mechanical Engineering Department in 2007. Currently He is enrolled as an associate professor in the Automotive Engineering Department at Ain Shams University, faculty of Engineering .His research interests lie in the field of automotive systems.

Amgad A. Bakarman : received his B.Sc. degree in Mechanical engineering department from Aden Univerity, Aden, Yemen, in 2008 and the M.Sc. student in Mechatronics at Ain Shams University, Cairo, Egypt, Currently.