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Design and Performance Analysis of PID Controller for Position Control of Hydraulic Systems actuated by Servo Motor Drives

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ABSTRACT

The efficiency of hydraulic position control in industrial applications is essential factor which affect productivity of earth-moving equipment, pressing machine and most hydraulic devices with accurate positioning systems. However the conventional controlled hydraulic systems have problem of low energy efficiency, hence, a new technique was developed to enhance a position control of a hydraulic cylinder using servo motor drive. A servo motor drive is used to control the pump speed through a PID controller where the cylinder position feedback is used as the control reference. A hydraulic setup is simulated where a two staged cylinder is used as an actuator and load source. The proposed system is used to as an alternative to variable displacement pumps with proportional valve to control the cylinder position. The result shows a good positioning accuracy with high gain of energy efficiency.

Keywords: Servo motor drive, Variable rotation speed, Load cylinder, PID, Position control.

1. INTRODUCTION

In hydraulic systems, electro-hydraulic (EH) systems have been widely used in many industrial applications because of its wide range of advantages [1] like high accuracy, availability, controllability, easy to maintain, durability, large power, high stiffness, fast response and more other advantages [2]. The electro-hydraulic systems (EHS) have several ways to be controlled, such as controlling the system through the control valve which have several types, the first one is the classical control (ON-OFF) control which is simple enough for normal hydraulic system, the second one is the proportional control valve by using a proportional driver which can be connected to a software to enhance the efficiency. The third one is to use the servo valve which is used and controlled by continuous signal. Another method of controlling the EHS is by using a variable displacement pump (VDP) by controlling the swash plate angle which can be effective to reduce the wasted energy used in a fixed displacement pump. A new approach is discussed by controlling the speed of the motor driving the pump hence controlling the flow of a fixed displacement pump (FDP).

Wide range of hydraulic pumps are driven by internal combustion engine (ICE) or an AC motor, the speed control in (ICE) can be done though the throttle valve and fuel injection variation but AC motor need a driver which alternate frequency hence the speed of the AC motor changes and give the required speed for pump which generate accurate flow rate to control position of the cylinder.

The servo motion control fundamentals have not changed significantly in the last 50 years. The basic reasons for using servo systems in contrast to open loop systems include the need to improve transient response time, reduce steady state errors and reduce the sensitivity to load parameters. Improving the transient response time generally means increasing the system bandwidth. Faster response times mean quicker settling allowing for higher machines throughput. Reducing the steady state errors will relate to the system accuracy.

The dynamic models of the Electro-Hydraulic System (EHS) have many uncertainties, which are consequences of physical characteristics, disturbances and load variation. Literature survey shows the key principles concepts and methods of components performance analysis and describe their possible constructive solutions [3-7]. The existing literature presents the ways of realizing hydraulic system using the available components. The electro-hydraulic systems are often employed in high performance applications. These applications include the control of active suspension systems [8-9], power steering systems [10], material testing systems [11], cam less engines [12], earth moving industries [13], load simulators [14], the control of multi-axis robotic manipulators [15], accurate positioning of an industrial hydraulic manipulator [16], position control of hydraulic robots with valve dead-bands [17].

The hydraulic valve controlled system have high response but low energy efficiency, some researchers have focused on the improvement of energy efficiency of the hydraulic valve controlled systems [18-22]. However, it is still lower than that hydraulic pump controlled systems [21-22]. Recently, high response pump controlled systems driven by AC servo motor are introduced [23-26]. The investigation of higher response and higher efficiency is still in progress.

The aim of the present work is to develop a simulation of a hydraulic system model, investigate the servo performance at different loads and control them using the servo motor drive for higher energy efficiency. The other sub-objective is to develop and implement a control law that can control the position of main cylinder. The performance of the controller is also assessed effectively by performing simulation and different experiments work in real-time environment so that the motion of the main piston can be monitored and controlled at any time. The control algorithm is based on the PID philosophy.

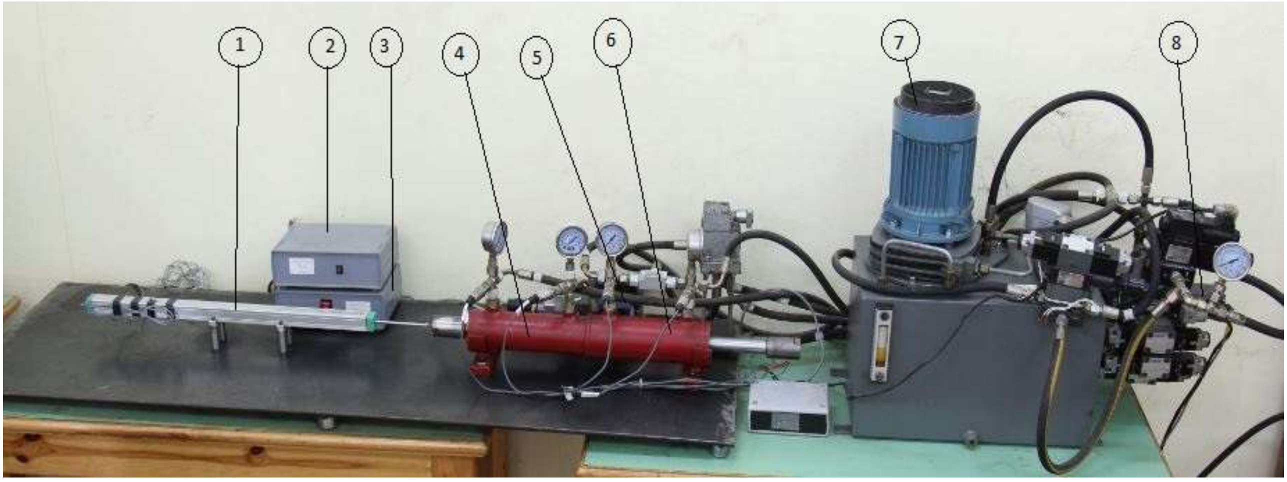
2. LAY-OUT OF THE EXPERIMENTAL TESTRIG PLATFORM

The electro-hydraulic system (EHS) can be experimentally setup which is used to simulate the position control or velocity control under certain loads and evaluate the developed control algorithm. The technical specification of the components used to build up the electro-hydraulic setup is described in this paper.

The experimental setup is developed to evaluate different control concepts. The main control system consists of a main cylinder which is controlled by FDP with different range of speed using a Servo Motor. The position of the system is defined as the position of the load cylinder and is measured by using a linear potentiometer. The measured signal is used for control algorithm. Control algorithm is executed through the controlled computer equipped with an Arduino card. A picture of the experimental setup to control the position of EHS is shown in Fig. 1. The experimental setup was developed in order to evaluate different control concept. In this experimental model both the main and the load cylinders are through a 20mm rod with a 60 and 70 mm bore respectively and 150mm stroke.

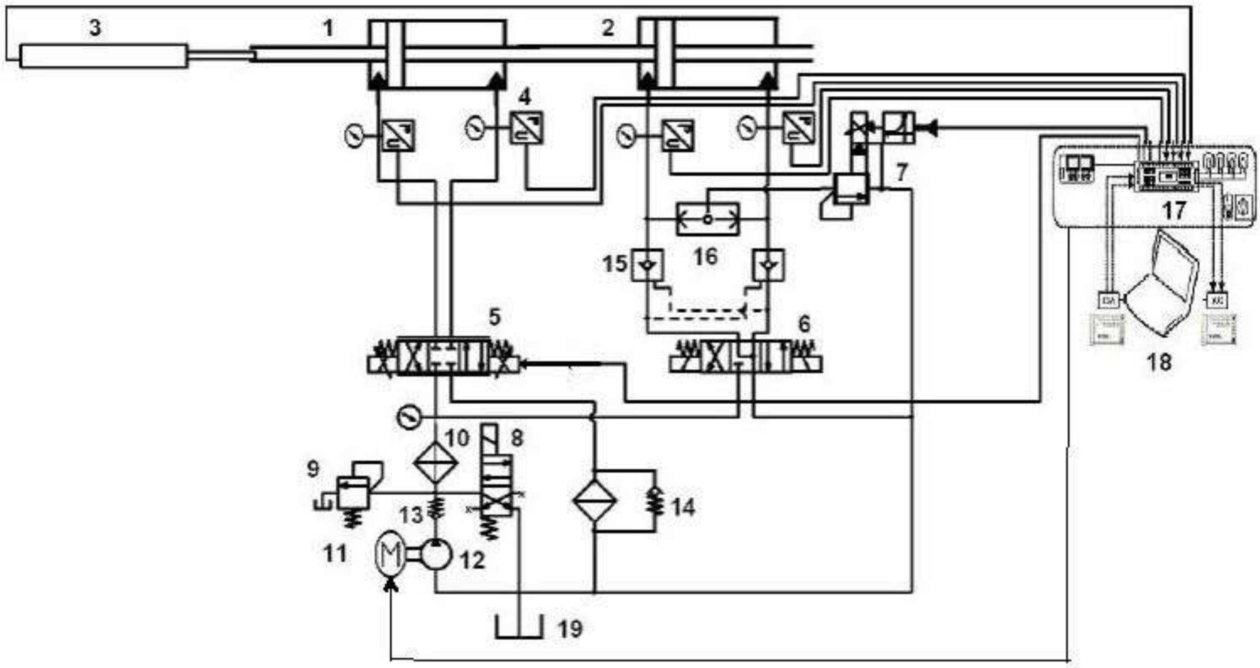
The AC motor and its drive which controls the speed are manufactured by ABB. It is directly attached with a hydraulic power unit manufactured by (Mannesmann Rexroth) which includes a vane pump which has a flow rating of 12 l/min at a nominal speed of 1450 rpm and the pressure can be regulated up to 70 bars. The power unit includes a pressure relief valve, an unloading valve, an oil filter, an oil strainer, an oil reservoir of size 40 liters, a breather, a pressure gauge, an oil level indicator, and a cooling system which is equipped by a piping system and fins mounted next to the motor shaft to apply forced convection for a purpose of heat removal is shown in Fig.1. the nominal working flow amounts of 40 l/min at 35 bar valve pressure difference. The load is achieved by using a proportional relief valve (manufactured by BOSCH REXROTH), which is on the side of the load cylinder. The pressure transducers manufactured by DWYER INSTRUMENTS are used in the present work with an output signal from 0 to 10 V. The linear Potentiometer manufactured by GEFRAN with maximum stroke 600mm is embedded in the experimental setup to measure the cylinder position which is connected to the rod of the main hydraulic cylinder. The controller receives the position input from the linear potentiometer. The linear potentiometer is connected to the main hydraulic cylinder rod and calculates a control output for the AC motor Drive hence to the AC motor. The controller in the actual system is divided into two units, the computer control system using MATLAB Simulink real time and interface with the hardware by an Arduino card as shown in Fig.2.

The cylinder rod displacement is controlled in direction and speed by The AC motor Drive and the load on the cylinder is controlled by proportional relief valve which receive its driving signal from an amplifier card connected to the proportional solenoid. The PC is used to give the reference value of the position and load pattern. The Arduino card transmits the command signal of the load from the PC to the amplifier card through a digital-to-analog converter and the position reference to the motor Drive. The position feedback needed for cylinder position is done using a resistive displacement transducer that is connected with the main cylinder rod to send the real-time measurements to the Arduino through analog-to digital converter.



1. Linear Potentiometer 2. Electronic hardware 3. ABB Motor Drive 4. Main cylinder 5. Proportional Relief valve 6. Load cylinder 7. Power Unit (Motor, Tank , breather, indicator, etc) 8. Directional control Valve

Fig. 1: Experimental Test rig



1. Main cylinder 2. Load Cylinder 3. Linear potentiometer 4. Pressure transducer 5. Directional control valve 6. solenoid valve 7. Proportional relief valve 8. Unloading valve 9. Pressure relief valve 10. Pressure filter 11. Servo Motor 12. Fixed displacement Pump 13. Ball check valve 14.return flow filter 15.pilot check valve 16.shuttle valve 17. Electronic interface (arduino) 18. controlled computer 19. Tank

Fig. 2. Schematic diagram of the experimental setup

3. CONTROLLERS DESIGN

The objective of work is to enhance the dynamic performance through decreasing the nonlinear effects found in the system. The position control of hydraulic cylinder has some uncertainties that make the controller design more challenging. It is necessary to check if the controller can keep stability and performances in all uncertainties. This objective is achieved by designing suitable types of control system which has the capability to overcome these phenomena. This section focuses on design of PID controller. The plant should be reformulated to adapt to the framework of different types control techniques. This section illustrates the design of PID controller which is used for the closed loop system to eliminate the steady state error and improve the dynamic response of the EHS. The error (difference between the generated signal and the actual) is fed into the controller. The produced command signal is then fed to the AC motor driver as shown in Fig. 4. To change the speed hence rotating the FDP shaft to give a certain flow rate for the main cylinder to eliminate the error.

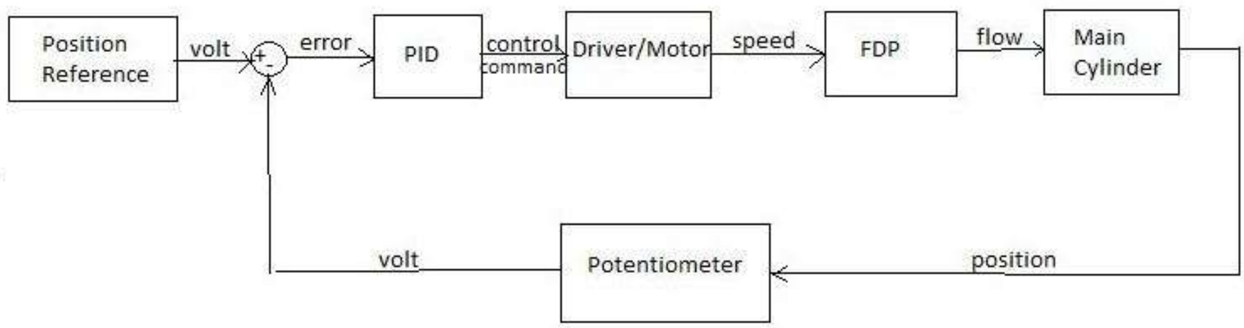


Fig. 4: Control Block Diagram

The values of the K, Ti, and Ts are calculated using the Ziegler and Nicholas rule to improve the system stability and precision. However, a final tuning must be done iteratively to obtain the desired response. For the simulation work, after tuning using the Ziegler-Nicholas rule the final gain values are KP =2.5, Ti =1.7 and Td=0.8

4. EH System model

The nonlinear dynamic model of electro-hydraulic system consists of a hydraulic cylinder controlled with a servo motor. The model will take into consideration modeling of AC Motor drive, pressure and flow rate dynamics, compressibility and piston dynamics.

A. pump speed:

The dynamics of pump speed can be described by the following [27].

$$n_P = Qp / \ (\eta * Dp) \qquad (1)$$

Where η is the volumetric efficiency, Qp is Pump Flow, n_p is the motor rotational speed and D_p is the pump displacement.

B. Pressure and flow rate dynamics :

The derivative of the pump pressure can be obtained in the form: [2-7]

$$\dot{P}_P = \frac{\beta}{V_P}(Q_P - Q_v - Q_{PA} - Q_{PB}) \qquad (2)$$

Where Q_p & Q_v are the pump and relief valve flow rate respectively, Q_{PA} and Q_{PB} are flow rates of directional valve which are described in the following equation from (4 to 13)

$$Q_P = Q_{th} - P_p / R_L \qquad (3)$$

Where Q_{th} is pump theoretical flow rate, R_L is pump resistance to internal leakage.

$$Q_v = C_d A_t \sqrt{2P_P / \rho} \tag{4}$$

Where ρ is the oil density, C_d is the coefficient of discharge, and A_t is the throttle area and is given by:

$$A_t = \pi X \sin(\alpha/2)\{D_d - X \sin(\alpha/2)\cos(\alpha/2)\ \} \tag{5}$$

Where α is Poppet cone vertex angle, D_d is damping spool diameter and X is Poppet displacement which can be calculated from the following equation:

$$P_d A_d + F_{SR} - P_P(A_d - A_p) = m\ddot{X} + f\dot{X} + K_{rv}(X_0 + X) \tag{6}$$

Where P_d is the pressure in the damping chamber, A_d is Damping spool area, P_P Valve inlet pressure (pump pressure), m is the reduced mass of the moving parts, F_{SR} is the seat reaction force, K_{rv} is the spring stiffness.

$$\dot{P}_d = \frac{\beta}{V_o + A_d X}(Q_d - A_d \dot{X}) \tag{7}$$

Where A_d damping spool area, V_0 is initial volume of the damping spool chamber, B is the bulk modulus of oil.

$$Q_d = \frac{\pi D_d C^3}{12\mu L}(P_P - P_d) \tag{8}$$

Where Q_d is flow rate in the radial clearance of the damping spool, C is the radial clearance of the damping spool, μ is the dynamic viscosity and L is damping spool length.

The Equations of flow through the directional control valve are derived from the application of flow continuity through the orifice [28].

$$Q_{PA} = C_d A_{PA} \sqrt{\frac{2}{\rho}(P_P - P_A)} \tag{9}$$

$$Q_{PB} = C_d A_{PB} \sqrt{\frac{2}{\rho}(P_P - P_B)} \tag{10}$$

$$Q_{AT} = C_d A_{AT} \sqrt{\frac{2}{\rho}(P_A - P_T)} \tag{11}$$

$$Q_{BT} = C_d A_{BT} \sqrt{\frac{2}{\rho}(P_B - P_T)} \tag{12}$$

Where P_A and P_B are the cylinder chamber pressures, P_P is the supply pressure, P_T is the tank pressure, C_d is the valve coefficient of discharge, where A_{PB}, A_{PA}, A_{AT}, A_{BT} are the areas of throttle through the valve.

C. Compressibility:

Hydraulic fluid compressibility is governed by the equation of bulk modulus, where β is the fluid bulk modulus and V is the chamber volume. Applying this to the two sides of the cylinder yields

$$\dot{P}_A = \frac{\beta}{V_{Ao} + A_P X_P}(Q_{PA} - Q_{AT} - A_P \dot{X}_P) \tag{13}$$

$$\dot{P}_B = \frac{\beta}{V_{Bo} - A_P X_P}(Q_{PB} - Q_{BT} + A_P \dot{X}_P) \qquad (14)$$

A_P is the annulus area of the piston and rod side of the cylinder, V_{A0} & V_{B0} is the cylinder half-volume.

D. Piston motion:

$$(P_A - P_B)A_P = m\ddot{X}_P + f_V \dot{X}_P + F_L \qquad (15)$$

Where m is the total mass of the piston and the load, f_V is the friction force, F_L is the external load disturbance which is given by:

$$F_L = P_L A_{PL} Sign(\dot{X}_P) \qquad (16)$$

E. Model Identification and validation:

The dynamic characteristics of the hydraulic systems are intricate due to the large number of components involved and their nonlinear behavior. Access to a dynamics model of such a system allows for understanding and designing closed-loop controllers for detecting system failures by running models. To achieve the desired level of accuracy in modeling and estimating the corresponding parameters, the system was broken into its components. Each of these is modeled individually, and the overall dynamic model is assembled from the individual models. The components modeled include Motor, pump, directional valves and cylinders. The enormous size, weight, and power of the experimental system were made available for experimentation, made identification experiments quite challenging. The system parameters were estimated or identified:
Spring stiffness: k_{rv}=210000 N/m, Bulk Modulus of oil: B=1.5.109 Pa, Discharge coefficient: C_d=0.6, Oil density: ρ=867 Kg/m^3, Dynamic viscosity: μ=0.02 Pa s, pump flow rate: Q_P= 12 L/min, proportional relief valve setting pressure: P_{relif}=7.106 N/m^2, Min. area of spool clearance: AMIN=1e-6 m2, Friction force:F_v=1000 N, Tank pressure: P_T=105 Pa.

F. Simulation results:

Using MATLAB Simulink to simulate the EHS model from equation 1 to 16 and using the model identification and verification we conduct that result in Figure 5. Showing servo motor, pressure, flow rate and piston dynamics.
This section presents the simulation results for PID controller of the EHS. The controller parameters obtained and applied in simulation with the same numerical values as another paper which controls the position using proportional directional valve only.

The cases with load forces are considered and presented in Figs 6, 7 and 8. The tested load 15, 20, 30, 50 bars (corresponding to 4.7, 6.3, 9.4, 15.9 KN respectively) for sinusoidal, step and square load. The desired position (0.038m) is achieved in different loads. PID controller shows an arbitrary small position error. Our result shows that PID controller is able to restrain the external disturbance in the form of load force.

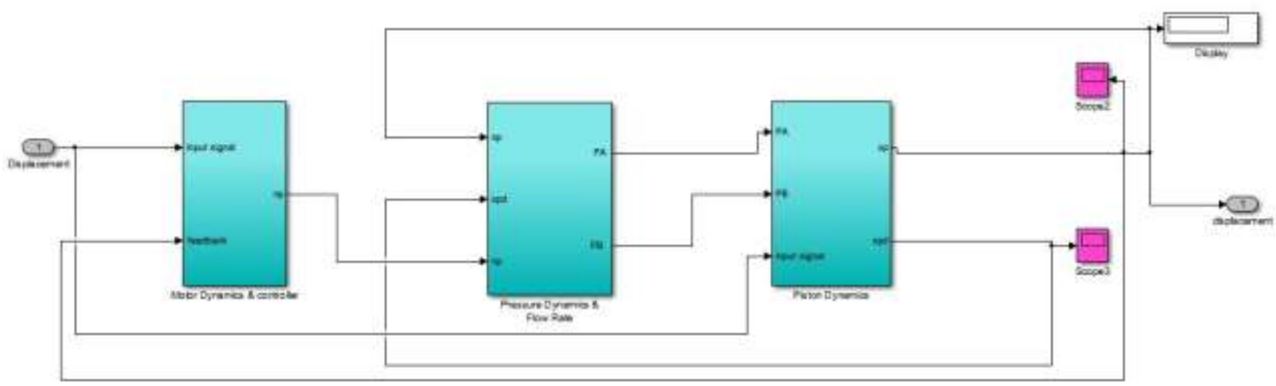


Fig. 5. MATLAB Simulink model

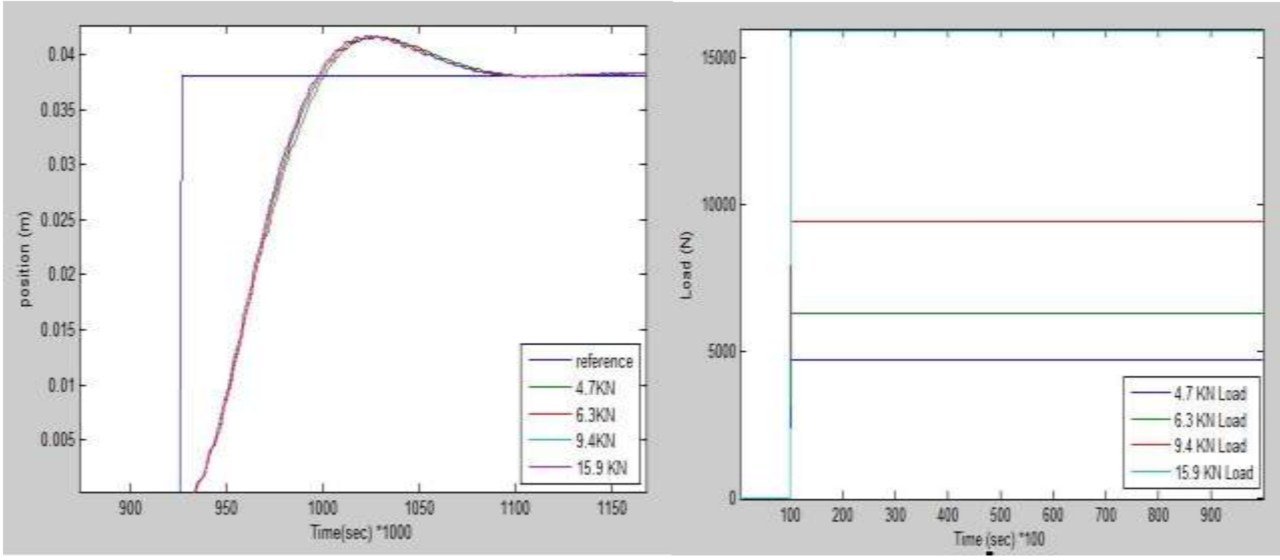


Fig. 6. Step Load response and its Load graph

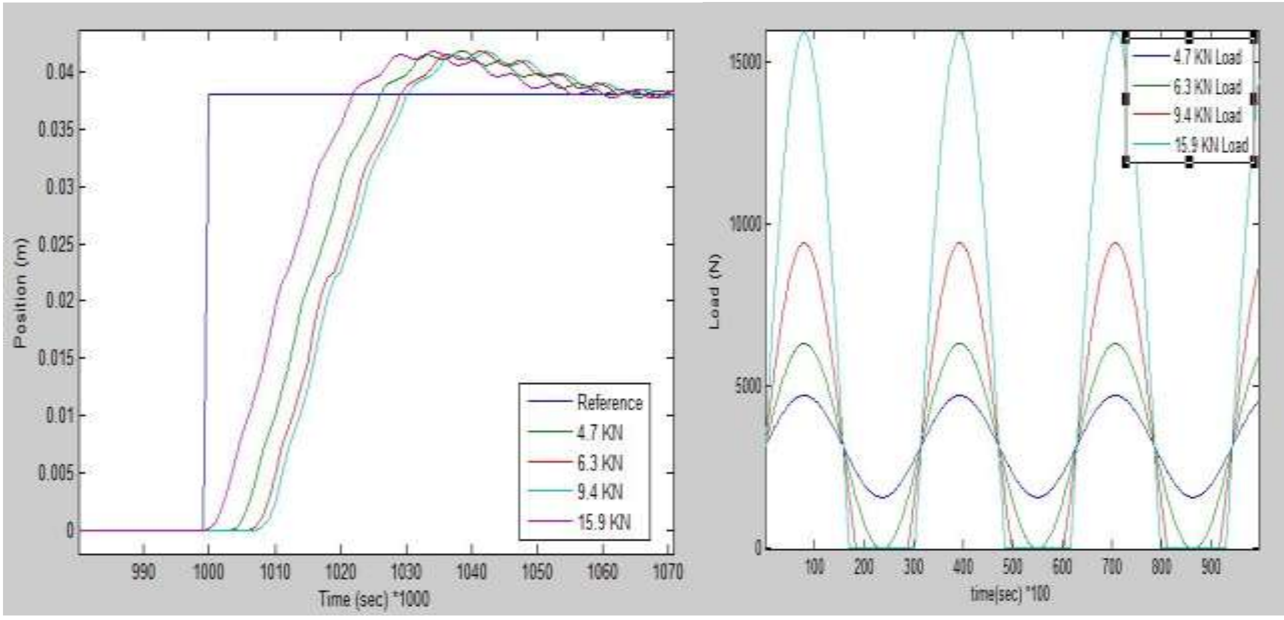


Fig. 7. Sin wave Load response and its Load graph

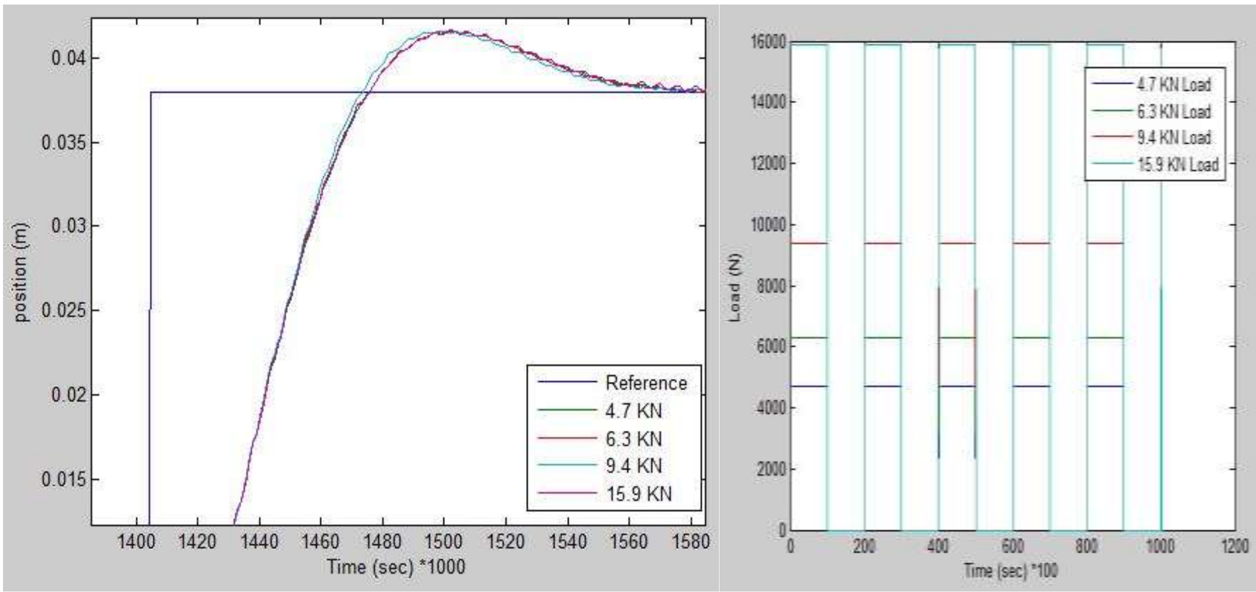


Fig.8. Square wave Load response and its Load graph

5. CONCLUSION

The position control of a hydraulic system using servo motor has been tested and verified. A sinusoidal, square and step load patterns have been applied. A PID controller used to control the position of double acting cylinder. The EH system is modeled using Simulink software. The control algorithm (PID) is developed using MATLAB Simulink.

The proposed PID control system requires position effectively at different patterns of load using a servo motor. PID controller can work well, especially considering how little information is provided for the design.

For constant load, the desired position achieved after a time 5 sec for no load and increases by increasing the load up to 8 sec at 15.9 KN load. A load pattern of industrial application was used in the experimental result to show the stability of PID controller to reach a required position after 5.5 sec. For all loads the PID controller position provides much faster settling times when compared to the no load position response and independent of the load values. PID loop with fixed tuning parameters would be severely challenged by these conditions. If it had been tuned for optimal performance with the load nearly constant, it would be much too slow about correcting position deficiencies when the variable load occurred. Conversely, it had been tuned to be aggressive enough for a constant variable load.

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