Electro-hydraulic proportional system real time tracking control development based on pulse width modulation method

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ABSTRACT

The present work is directed to develop the dynamic performance of an electro-hydraulic proportional system (EHPS). A mathematical model of (EHPS) is presented using electrohydraulic proportional valve (EHPV) by the aim of Matlab-simulink which facilitate the simulation of the hydraulic behavior inside the main control unit. Experimental work is done and closed loop system is designed using linear variable displacement transducer sensor (LVDT). The controller of the system is an Arduino uno which is considered as the processor of the system. The model is validated by the experimental system. The study also presents a real time tracking control method based on pulse width modulation by controlling the speed of the actuator to achieve position tracking with minimum error.

KEYWORDS hydraulic control; proportional valve; electro hydraulic proportional system; PWM control method; position tracking control
INTRODUCTION

The proportional valve is a valve which produces a proportional output to an electronic control input or a valve operates by proportional solenoids instead of on-off solenoids. It can be classified into three types: pressure control valves, flow control valves and directional control valves.

The pressure control valves are designed mainly to control the pressure while the proportional flow control valves are designed mainly to control the flowrate but Proportional directional valve is used to control both the flow direction and the flow rate.

Many studies dealt with the proportional valve investigations, Vaughan-ND et al.[1] presented a nonlinear dynamic model of a high-speed direct acting solenoid valve the model accurately predicted both the dynamic and steady state response of the valve to voltage inputs. Simulated voltage, current, and displacement results were presented which agreed well with experimental results.

Lai-JiingYih et al.[2] proposed an adaptive self-tuning controller to enable a hydraulic proportional valve to achieve accurate set-point flowrate control. the performance of the closed-loop system was very robust as the system response remained the same under various operating conditions.

M. F. Rahman, et al.[3] began the first step of converting a conventional on-off solenoid into a proportional solenoid. They studied the dynamic behavior of a
conventional solenoid by letting the simulation depending on linear magnetic principle, by using the simulation package SIMNON. A comparison was done between the results from simulation model and experimental work.

Niksefat-Navid, et al.[4] study the development and experimental evaluation of a hydraulic force controller, using nonlinear Quantitative Feedback Theory (QFT) design method. The designed controller was implemented on an industrial hydraulic actuator equipped with a low-cost proportional valve.

Elmer-KF, et al. [5] presented a generic non-linear dynamic model of a direct-acting electro-hydraulic proportional solenoid valve. The model accurately and reliably predicted both the dynamic and steady state responses of the valve to voltage inputs. Simulated results were presented, which agreed well with experimental results.

Dasgupta-K, et al.[6] studied the dynamics of a proportional controlled piloted relief valve through bond-graph simulation technique. The simulation results were also verified with experimental results.

Chu-MingHui, et al.[7] studied the nonlinear model of a variable displacement axial piston pump (VDAPP) with a three-way electro-hydraulic proportional valve (EHPV) which controlled the swash plate actuators. The time response for the swash plate angle was analyzed theoretically by the simulation model and experimentally, and a favorable model-following characteristic was achieved. The proposed neural controller which conducts nonlinear control in VDAPP, enhanced adaptability and robustness, and improved the performance of the control system.
Wei Liu, et al.[8] studied an optimization of a throttle poppet valve based on hydraulic feedback principle to use it in machines. The study can be used as a guideline to design various sizes of proportional valves.

Riccardo Amirante, et al.[9] designed a new methodology of spool surfaces of 4/3 proportional directional valve. Based on redesign of both the compensation profile and spool lateral surfaces to reduce the acting flow forces on it. The proposed methodology achieved lower actuation forces compared to the commercial configuration.

Wilber Acuña-Bravoa, et al.[10] presented the application of a model-based control structure called Embedded Model Control (EMC), the position tracking of the spool presented a better result by using the EMC. Compared with those from using an industrial manufacturer.

**MODEL DISCRIPTION**

The system shown in fig (1) consists of hydraulic pump (1) which feed the electro hydraulic proportional valve (EHPV) (3), through pressure relieve valve (2), and one way valve (4), the (EHPV) controls the output hydraulic feed delivered to the hydraulic actuator (5) to allow the hydraulic actuator tracking the required input position by the aim of PID controller (6), which receive a feedback signal from position sensor (7).
MATHEMATICAL MODEL

The proposed system is mathematically done by the aim of Matlab-simulink and the system components configuration are selected as same as the experimental components.

hydraulic pump

The type of the hydraulic pump used in the electro-hydraulic system is a fixed displacement gear pump, operates with nominal speed of 600 rpm and maximum output flow rate of 6.05 liter/min. The pump is driven by an AC electric motor operates with 220 Volts, 50 Hz, and 1.5 kW.

Figure 1. EHPS hydraulic circuit
The relief valve is used to control the pressure in the hydraulic systems to protect its individual elements, pipes and hoses from over pressure problems, it adjusted at a pressure of 60 bar.

**Proportional directional valve**

The 4/3 electro-hydraulic proportional directional valve fig (2) is a Hydraulic Ring manufacture of type NG6. It has maximum operating pressure of 315 bar, its valve spool has zero overlap, and its motion is controlled by two electrical proportional solenoids.

**Hydraulic actuator**

The hydraulic actuator used in this study is a double acting steel with cam for operating the limit switch with diameter of 32/22 mm area ratio 1.6:1 and max pressure of 160 bar.

![Figure 2. typical electrohydraulic proportional valve](image)

**Model equations**
the $Q$-$P$ mathematical relation of the pump which has been found as following

$$Q_p = Q_{th} - (7 \times 10^{-3} P_p)$$

Where

$Q_{th}$ : maximum theoretical flowrate of the gear pump = 6.05 liter/min.

$P_p$ : pump output pressure.

The equation of poppet motion of the relief valve:

$$m_p \frac{d^2 z}{dt^2} + f_z \frac{dz}{dt} + k_z (z + z_o) = p_c A_p + F_{val}$$

$$F_{val} = K_r z_o$$

The flow rate equation:

$$Q_{rv} = C_d A \sqrt{\frac{2(P - P_t)}{\rho}}$$

Equation of motion of the valve spool

$$F_s = m_s \frac{d^2 x}{dt^2} + f_s \frac{dx}{dt} + kx$$

Where:

$m_s$ = mass of spool + 0.5(mass of return spring) = 0.021 Kg

mass of spool = 19.58 gm [measured]

mass of spring = 1.972 gm [measured]

$f_s$ = 50 : 100 N.sec/m.

$k$ = 24500 N/m [measured experimentally]

Flow rate equations through the proportional valve:
Fig (3) illustrates the flow through the proportional valve, where (P) is the valve pressure port, (T) is the valve return to the tank, (A) and (B) are the valve outputs to the actuator, also a,b,c and d are the areas where the flow is subjected through the valve.

Figure 3. proportional directional valve Internal orifices

\[ Q_a = C_d A_a (x) \sqrt{\frac{2(P_b - P_t)}{\rho}} \]  \hspace{1cm} (6)

\[ Q_b = C_d A_b (x) \sqrt{\frac{2(P - P_b)}{\rho}} \] \hspace{1cm} (7)

\[ Q_c = C_d A_c (x) \sqrt{\frac{2(P - P_a)}{\rho}} \] \hspace{1cm} (8)

\[ Q_d = C_d A_d (x) \sqrt{\frac{2(P_a - P_t)}{\rho}} \] \hspace{1cm} (9)

Continuity equations through the hydraulic actuator:

\[ Q_{AR} - \frac{d}{dt} \left( Q_i - Q_v - \frac{(V_y + ay)}{B} \times \frac{dp_R}{dt} \right) = 0 \] \hspace{1cm} (10)

\[ A \frac{dy}{dt} + Q_i - Q_{mR} - \frac{(V_y - Ay)}{B} \times \frac{dp_R}{dt} = 0 \] \hspace{1cm} (11)

Actuator Equation of motion:

\[ a_p A_p - A.p_a = m_c \frac{d^2 y}{dt^2} + f_f \frac{dy}{dt} + F \] \hspace{1cm} (12)
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These equations are modelled in Matlab-simulink to facilitate the study of fluid behavior through the hydraulic circuit and also a controller is designed by the aim of Simulink tool.

Fig (4) shows the simulation code of the system on Matlab-simulink, all system equations are represented in the code with different input signals, the system is also represented by a transfer function by the aim of system identification tool of the program and it gives the following transfer function:

\[ \frac{8.468s + 11.05}{s^2 + 8.522s + 11.11} \]  

(13)

Figure 4. EHPS mathematical model code

By the transfer function the controller of the model can be selected which is (PID) controller which is used to adjust and resolve the error of the system, it controls
the system by evaluate the feedback of the system and compensate the system error, the mathematical equation of the PID controller is:

\[ u(t) = K_p e(t) + K_i \int_0^t e(t) dt + K_d \frac{de(t)}{dt} \] (14)

Where

\[ u(t) \] is the controller output, \[ e(t) \] is the system error, \[ K_p, K_i, \text{and } K_d \] are the controller constants where: \[ K_p \) (0.428), \[ K_i \) (5.58) and \[ K_d \) (0).

EXPERIMENTAL WORK

The experimental work is done by using the same components configuration as mentioned in section (3) which is the same as the mathematical model configurations. Fig (5) shows the test rig used in this study.

Figure 5. Electrohydraulic test rig

Fig (6) shows the electrical circuit used in controlling and monitoring the position tracking of the actuator, the input signal transfers from Matlab-simulink to the control
A new method of controlling the system is proposed in this study, by experimentally testing the system by means of oscilloscope connecting to the two terminals of the proportional valve, it has been found that the actuator response depending on changing the pulse width modulation (PWM).

Fig (7) shows (PWM) input signal which is detected by the oscilloscope with pulse width of 180 as an example of the input (PWM).

![Electric circuit for controlling the system](image)

**Figure 6.** Electric circuit for controlling the system
Fig (8) shows a four (PWM) input to the system and its response on the actuator position with time, as the pulse width increases the speed of the actuator increases as when the (PWM) is 160 the speed of the actuator reaches 50 mm/sec and when the (PWM) reaches 180 the speed of the actuator reaches 74 mm/sec, the maximum actuator speed is reached when the (PWM) reaches 240 as the speed reaches 105 mm/sec and also the inertia of the actuator increases which effects badly on the system control, this experiment help to produce a new method of electro-hydraulic proportional system control, by controlling the speed of the actuator to avoid the inertia of the actuator, so the following controller depend on adjusting the speed, when the actuator position is far from the target position the input (PWM) signal should be high to increase the speed of the actuator and when it approaches its target the (PWM) signal should be reduced to avoid the actuator inertia.
Figure 8. Effect of pulse width modulation on actuator speed

Fig (9) shows the Matlab-simulink code of this controller which consists of Arduino pin (2) block that receive the displacement of the actuator from Arduino which receives it from the displacement sensor (LVDT), Arduino pin (10) and (11) blocks which send the required (PWM) signal to the Arduino that delivered this signal to the two coils of the proportional valve, the two function blocks control the speed by calculate the difference between the input signal and the actuator position.

Figure 9. EHPS controlling code
RESULT AND DISCUSSION

The mathematical model is validated by the experimental work by varying the input signal with constant input 50 mm, step input 70 mm, square signal 70 mm and sinusoidal signal with amplitude 30 mm, bias 30 mm and frequency (0.5 rad/sec) 0.08 HZ, these results are shown in fig (10,11,12, and 13) as it shown the model has a good agreement with the experimental result in the constant, step and square signal inputs.

**Figure 10.** Hydraulic actuator response by applying constant input of 50 mm (experimental vs simulation)

**Figure 11.** Hydraulic actuator response by applying step input of 70 mm (experimental vs simulation)
when applying a sinusoidal wave, the experimental result shows a relatively delay compared to the mathematical model, this delay is due to the controller which reduce the actuator speed to avoid the actuator inertia that lead to high overshooting, however the mathematical model result is still give a good agreement with the experimental result.

The mathematical model helps in studying the fluid behavior inside the hydraulic circuit which is calculated.
Fig (14) shows the pressure behavior in the hydraulic circuit by applying constant input while fig (15) shows the pressure behavior by applying sinusoidal wave, these results are serving in testing the system failure when applying a high load or object to any component malfunction.

**Figure 14.** Hydraulic circuit pressure behavior by applying constant input of 50 mm.

**Figure 15.** Hydraulic circuit pressure behavior by applying sinusoidal wave input of 30 mm amplitude and 30 mm bias with frequency 0.08 HZ.

**CONCLUSION**

The study presented a new method for controlling the electro hydraulic proportional actuator by controlling the speed of the actuator this is done by controlling
the band width modulation signal that operates the two coils of the proportional valve,
also this study presented a mathematical model of the system and its transfer function,
also a controller of the model is done by (PID) controller.
This study also facilitates to study the pressure behavior inside the hydraulic circuit.

**NOMENCLATURE**

| Symbol | Description |
|--------|-------------|
| a      | Area of cylinder rod side, m². |
| A      | Area of cylinder piston side, m². |
| A_a    | Throttle area a in directional proportional valve, m². |
| A_b    | Throttle area b in directional proportional valve, m². |
| A_c    | Throttle area c in directional proportional valve, m². |
| A_d    | Throttle area d in directional proportional valve, m². |
| A_p    | Poppet area of the relief valve, m². |
| B      | Bulk modulus of elasticity of hydraulic oil, N/m². |
| C_d    | Discharge coefficient. |
| C      | Radial clearance, m. |
| d      | Diameter of cylinder rod, m. |
| D      | Diameter of cylinder piston, m. |
| f_r    | Damping coefficient of relief valve poppet, N.sec/m. |
| f_s    | Damping coefficient of directional proportional valve spool, N.sec/m. |
| F_s    | Proportional solenoid force, N. |
| F_seat | Seat reaction of the relief valve, N. |
| k      | Stiffness of return spring in directional proportional valve, N/m. |
| m_p    | Mass of poppet in the relief valve, kg. |
| m_s    | Mass of moving parts in the directional proportional valve, kg. |
| p      | Pressure, Pa. |
| P_A    | Pressure at port A of the direction proportional valve, Pa. |
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$P_B$  Pressure at port B of the direction proportional valve, Pa.

$P_P$  Pressure at port P of the direction proportional valve, Pa.

$P_t$  Tank pressure, Pa.

$Q$  Flow rate, m$^3$/sec.

$Q_a$  Flow rate through opening area a at directional proportional valve block, m$^3$/sec.

$Q_{AR}$  Flow rate from port A to rod side chamber in the hydraulic cylinder, m$^3$/sec.

$Q_b$  Flow rate through opening area b at directional proportional valve block, m$^3$/sec.

$Q_c$  Flow rate through opening area c at directional proportional valve block, m$^3$/sec.

$Q_d$  Flow rate through opening area d at directional proportional valve block, m$^3$/sec.

$x$  Displacement of directional valve spool, m.

$y$  Displacement of cylinder rod, m.

$z$  Displacement of relief valve poppet, m.

Declarations

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Figure Captions List

Fig. 1  EHPS hydraulic circuit
Fig. 2  typical electrohydraulic proportional valve
Fig. 3  proportional directional valve Internal orifces
Fig. 4  EHPS mathematical model code
Fig. 5  Electrohydraulic test rig
Fig. 6  Electric circuit for controlling the system
Fig. 7  Pulse width modulation input signal 180
Fig. 8  Effect of pulse width modulation on actuator speed
Fig. 9  EHPS controlling code
Fig. 10  Hydraulic actuator response by applying constant  input of 50 mm
Fig. 11  Hydraulic actuator response by applying step  input of 70 mm
Chinese journal of mechanical engineering

Fig. 12    Hydraulic actuator response by applying square input of 70 mm

Fig. 13    Hydraulic actuator response by applying sinusoidal wave input of 30 mm
            amplitude and 30 mm bias with frequency) 0.08 HZ

Fig. 14    Hydraulic circuit pressure behavior by applying constant input of 50 mm.

Fig. 15    Hydraulic circuit pressure behavior by applying sinusoidal wave input of
            30 mm amplitude and 30 mm bias with frequency) 0.08 HZ