Speed Control of Hydraulic Elevator by Using Electro-Hydraulic Servo Mechanism

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Abstract

An electro-hydraulic elevator is a new type of enhanced elevators used in low-rise buildings, no more than eight floors. In this paper, an electro-hydraulic servo system for controlling the speed of a hydraulic elevator prototype by using a proportional valve and PI controller has been investigated theoretically and experimentally. A three floors elevator prototype model with 76cm height has been built, including hydraulic components and electrical components. The elevator system is automated using an Arduino UNO board based Data Acquisition (DAQ) system. LabVIEW software is programmed to control the hydraulic elevator system through L298 DC drive via the DAQ board. The best PI gains have been calculated experimentally using Ziegler–Nichols, trial and error methods. The results showed the effectiveness of the use of Electro-hydraulic servomechanism in enhancing the performance of the hydraulic elevator in terms of comfort and smoothness when people travelled through the elevator floors.

Keywords: Speed control; Electro-hydraulic elevator; Servo mechanism; Proportional valve; PI controller

Nomenclature

A_1, A_2	*Orifice area gradient, m²*	X_p	*Piston displacement, mm*
A_{p1},A_{p2}	*Area of the piston, m²*	X_v	*Valve displacement, mm*
A_s	*Spool end area, m²*	ΔP	*Pressure difference, bar*
C_d	*Discharge coefficient of the valve, m³/sec*	ΔQ	*Flow difference, m³/sec*
C_{ip},C_{ep}	*Internal and external leakage coefficient of the hydraulic cylinder, m³/sec/Pa*	$\dot{x}_p$	*Velocity of piston, m/s*
C_{tp}	*Total leakage coefficient of cylinder, m³/sec/Pa*	$\ddot{x}_p$	*Acceleration of piston, m/s²*
D	*Derivative gain*		
d/dt	*Derivative*		*Greek Symbols*
F_{hyd}	*Total force on piston, N*	β	*Bulk modules, N/m²*
F_1	*Spring pre-load force, N*	ζ	*Damping coefficient*
K_a	*Valve flow gain, m²/sec*	w_m	*Mechanical natural frequency, Rad/sec*
K_c	*Flow pressure coefficient, m²/sec.Pa*	w_n	*Hydraulic natural frequency, Rad/sec*
K_o	*Spring rate of pipe, KN/m*	w_1	*Break frequency of sensing chamber, Rad/sec*
M_t	*Total mass, Kg*	w_3	*Break frequency of main volume, Rad/sec*
N	*Rotational speed of pump, rpm*	δ_o	*Damping ratio of pipe*
P	*Proportional gain*		
P_{So},P_{Ro}	*Supply and return pressure at initial Condition, bar*		*Abbreviations*

P_l	*Load pressure, bar*	*DAQ*	*Data acquisition*
P_s	*Source pressure from power unit, bar*	*EHSM*	*Electro-hydraulic servo mechanism*
P_1, P_2	*Pressure of port A and B, bar*	*HSV*	*High-speed valve*
Q_l	*Flow through the load, m^3/sec*	*PID*	*Proportional integral derivative*
Q_s	*Supply flow rate from power unit, m^3/sec*	*P-Q*	*Pressure-flow*
Q_1, Q_2	*Flow through proportional valve*	*PWM*	*Pulse width modulation*
V_{o1},V_{o2}	*Initial volume of chambers, m^3*	*PV*	*Proportional valve*
V_t	*Total hydraulic oil volume in cylinder,m^3*	*TF*	*Transfer function*
V_1, V_2	*Forward and return chamber volume, m^3*	*VVVF*	*Variable voltage variable frequency*

1. Introduction

Elevators are a means of transporting persons and goods vertically within a column, customized for connecting the building floors. There are three main types of commonly used elevators: traction machine room-less, traction with machine room, and hydraulic. A Hydraulic elevator is powered by a piston installed at its bottom and drives the elevator upward as an electric motor strongly pushes the oil or any other hydraulic fluid into the piston. The elevator goes down when a valve releases the fluid from the piston. The hydraulic elevators are employed for low-rise applications of about 2-8 floors and move at a maximum speed up to 200 ft/min [1].

In today's modern world, hydraulics play a very important role in the day-to-day lives of people. Any device operated by a hydraulic fluid may be called a hydraulic device [2].

Electro-hydraulic servo mechanism (EHSM) is extensively employed in mobile systems and numerous industrial applications due to its high power to weight ratio, good positioning capabilities, fast response, high stiffness, etc. There are two methods may be utilized for controlling the hydraulic cylinder speed in the systems relying on the EHSH. The first method is a variable displacement pump used to control the flow to the cylinder, while the second method is based on a servo or proportional valves. It is a closed-loop system for controlling speed, depends on the actuated error signal represented by the difference between the feedback and input signals supplied to the controller in order to decrease the error signal and make the output of the system closer to the required value [3].

Many of previous research studies had worked in the field of investigation of the performance of hydraulic cylinders in elevators through the hydraulic speed control with different sets of pumps and motors. Huayong et al. [4] have discussed and analyzed the speed control of a hydraulic elevator using a variable-voltage, variable-frequency (VVVF) technique. A PID control law was experimentally adopted. Also, testing for ascending and descending elevator at operating speed was carried out. The simulations and test results showed the existence of flutter exists when starting the elevator, but this phenomenon gradually decreases when increasing the speed. Forental et al. [5] have studied the linear motion drive depended on a hydraulic cylinder with electrical position feedback. The dynamic characteristics of the hydraulic drive based on proportional control had been investigated experimentally and by modeling methods. Mathematical models were developed for the hydraulic drive. Bode diagram for the cylinder displacement could be obtained by experiments as well as by modeling at frequencies ranged with (0.05 - 5 Hz). It is noticed that the proposed model gives an accuracy of 3% at frequencies up to 3 Hz. Also, the error amplitude is less than 10% at a frequency range of 3–5 Hz. Liu et al. [6] have presented a scheme of a hydraulic elevator via applying an electro-hydraulic proportional pressure-flow (P-Q) compound valve to a hydraulic driving control system, in order to improve the operation stability, reduce the installed power and energy consumption, and other aspects to make a beneficial attempt. The findings were as follows: the hydraulic driving control system based on P-Q valve of the hydraulic elevator has some advantages such as high control precision, smooth starting, and little effect in the case of an emergency stop, and this system can realize smooth starting of the hydraulic elevator with preventing sudden stop shock. Sun et al. [7] have investigated the development of a hydraulic elevator by reducing the requirements of the power installation and the energy consumption in the hydraulic elevator comparing with the directly driven electric elevator. In the conventional valve-controlled hydraulic elevator, the bypass throttle is usually employed for controlling the speed of the elevator cabin. They deduced that the modern generation of the hydraulic elevators could deliver a great performance of energy-saving when comparing with the conventional elevators. Also, the energy consumption of the hydraulic elevator has been significantly decreased. Xu and Wang [8] have presented traditional and self-tuning Fuzzy PID controllers for controlling the speed of a hydraulic elevator. The elevator performances were compared via analyzing their step response curves. Simulation results showed that the fuzzy PID controller has a greater effect than conventional PID controller and could meet the requirement of complicated procedures and high performance due to the short duration of response, good stability, and high precision. Wang et al. [9] have studied the performance of high-speed switching valves utilized as pilot valves. The key idea was that a 3-way main stage valve could be controlled, in each chamber, by using two high speeds of switching valves. The speed is controlled via tuning the duty ratio of the pulse width modulation (PWM) control signal. The theoretical analysis as well as dynamic simulation showed the performance of control of proportional pilot valves and high-speed switching pilot valves. The results showed that the primary stage could operate well if the continued pilot components are replaced by high-speed switching valves. Liu and Gao [10] had experimentally developed the position control of a hydraulic cylinder controlled by a high-speed on-off valve in order to achieve accurate control on position. The high-speed on-off valve HSV could be utilized to realize the exact position control through adjusting the duty of the PWM signal and the compound algorithm of PI and speed feed forward-displacement feedback is provided to efficiently reduce the position error through analyzing the flow characteristic.

The main objective of this paper is the possibility of the application of the electro-hydraulic servo mechanism EHSM on hydraulic elevators, in which control system is employed to control the proportional solenoids valve and thus control the displacement of the spool valve which leads to determine the flow rate; which leads to control the hydraulic cylinder speed and thus controlling the elevator speed up and down. The control process of the solenoids valve may be implemented through using PWM technology to control the input power provided to the solenoids, and this process of controlling depends on the value of error. This paper, experimentally investigates the control of the hydraulic elevator speed using a fixed displacement pump and proportional valve with a conventional PI controller depending on controlling the proportional solenoids valve and this control leads to determine the flow rate and thus control the hydraulic cylinder speed, in other words, control the speed of the elevator up and down.

2. Theoretical Analysis

The theoretical modeling has included the analysis of the mathematical model of the main components of the experimental rig such as a proportional spool valve, relief valve, hydraulic cylinder, hydraulic pump, and flexible hose.

2.1 Proportional valve control

The proportional valves are one type of the spool valves shown in Fig. 1. The general relations and performance characteristics of the proportional valve have been derived and developed to meet the following assumptions:

- Stabilizing the pressure source
- Neglecting the fluid inertia
- No flow reversal or cavitation's
- Neglecting the return line pressure (P_o)
- Symmetrical and matched and Orifices
- Considering the power matches to the hydraulic cylinder and the relative valve

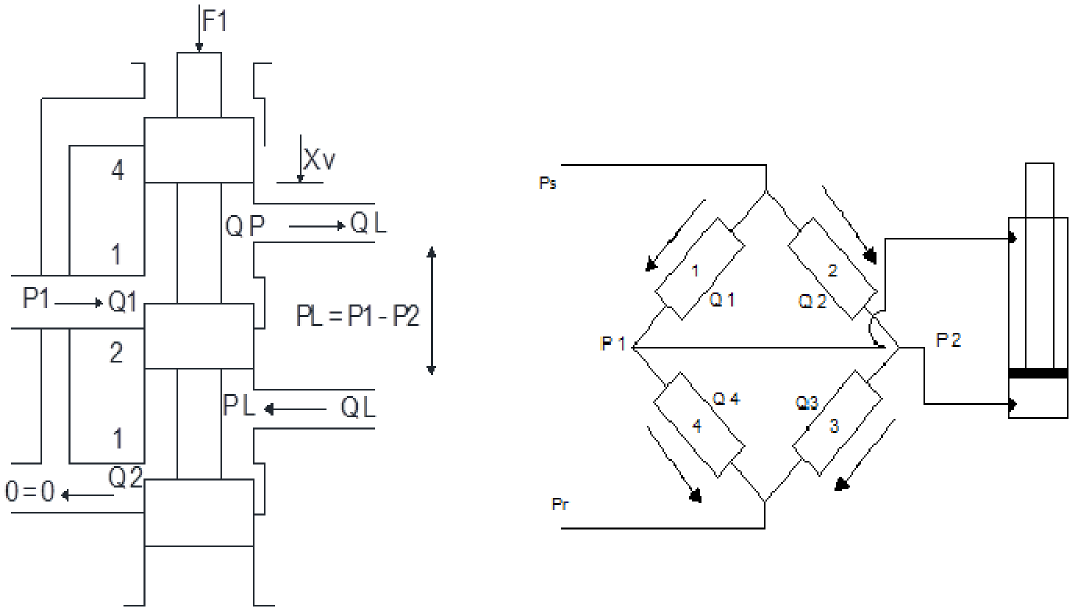

Fig. 1: Schematics of 4-way, 3-position spool valve [11]

If the area of the orifice of the slide valve is matching and symmetrical then the flow pressure equation in the valve would be [12] (see Appendix):

$$Q_l = Kqx_v - KcP_l \qquad \text{......... (1)}$$

Eq. (1) represents the Linearized equation of the pressure-flow to the control valve [12].

2.2 Double-acting single rod cylinder

Considering the hydraulic cylinder of Fig. 2, the application of the continuity equation to the two sides of cylinder yields [13]:

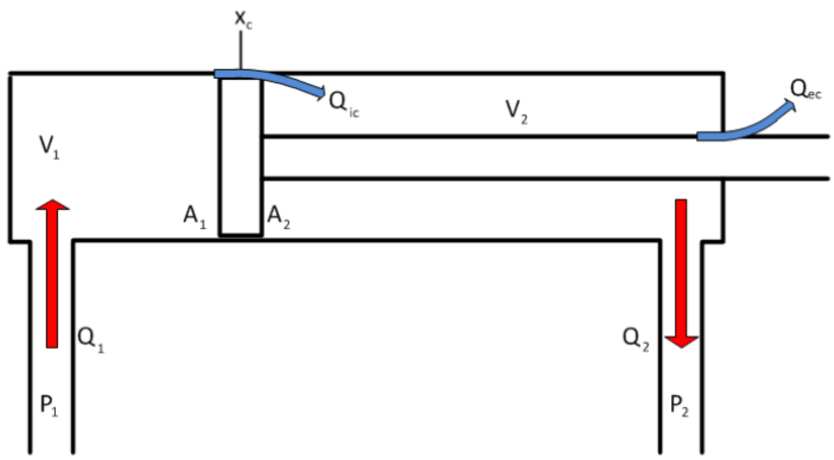

Fig. 2: Schematic diagram of a double-acting cylinder [14]

The flow through the load will be (see Appendix):

$$Q_l = Aav.\dot{x_p} + C_{ip}(P_l) + \frac{V_t}{4\beta}\dot{P}_l \qquad \text{.......... (2)}$$

2.3 Dynamic equations

Now, by applying Newton's second law to the forces on the piston [12] (see Appendix):

1. The hydraulic natural frequency (w_n) is:

$$w_n = 2Aav.\sqrt{\frac{\beta}{V_t\,M_t}} \qquad \text{……… (3)}$$

Where $Aav. = \frac{Ap_1+Ap_2}{2}$

The cylinder data label has the following specifications: Stroke = 76.2cm, Bore = 3.81cm, Rod diameter = 1.9cm.

This will give the following results:

Ap_1= 11.4cm^2, Ap_1= 8.56cm^2, $Aav.$ = 9.98cm^2 (calculated)

Spool diameter = 0.0125 m (manufacturing data)

M_t =14.1 Kg×9.81 = 138.32 N and V_t =8.686×10^{-4} m^3 (calculated)

Ps = 10 bar (regulated)

β = 108×10^8 N/m^2 and C_{ip} = 1.8×10^{-11} m^4.sec/Kg (manufacturing data)

Kc = 1.9×10^{-12} m^4.sec/Kg and Kq = 0.832 m^2/sec (laboratory section)

$$\omega_n = 2 \times 0.998 \times 10^{-3} \times \sqrt{\frac{108 * 10^8}{8.686 * 10^{-4} * 14.1 * 9.81}} = 598.43\ {rad}/{sec}$$

2. The damping ratio (ξ) is:

$$\xi = \frac{M_t(C_{ip}+Kc)}{Aav.\sqrt{\frac{M_tV_t}{\beta}}} = \frac{C_{ip}+Kc}{Aav.}\sqrt{\frac{M_t\beta}{V_t}} \qquad \text{……… (4)}$$

$$\xi = \frac{1.8*10^{-11}+1.9*10^{-12}}{0.998*10^{-3}} * \sqrt{\frac{138.32*108*10^8}{8.686*10^{-4}}} = 0.827$$

Now, substituting Eqs. (A11) and (A12) into (A10) in the Appendix yields the transfer function of the valve-controlled cylinder:

$$G_s = \frac{V_p}{x_v} = \frac{\frac{Kq}{Aav.}}{S(\frac{1}{w_n^2}S^2+\frac{2\xi}{w_n}S+1)} \qquad \text{……… (5)}$$

2.4 Pressure relief valve modeling

The schematic of a single-stage pressure control valve (relief valve) is shown in fig. (3). The equations which describe the spool motion are [12]:

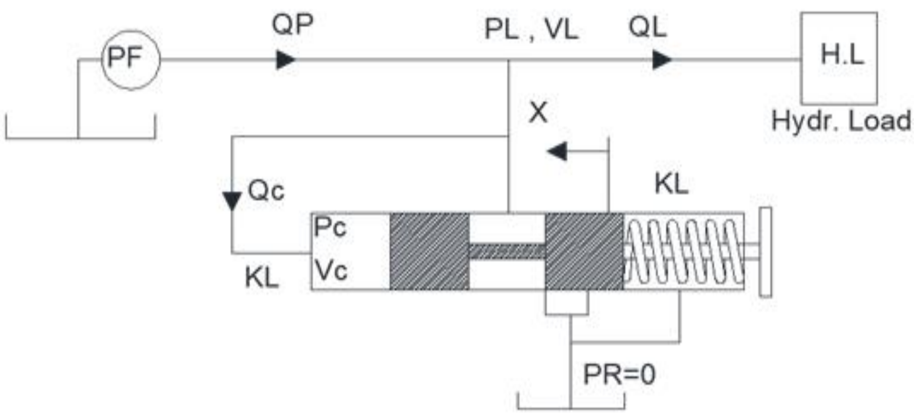

Fig. 3: Single-stage pressure control relief valve

$$G_r(s) = \frac{x}{F_1} = \frac{\frac{1}{K_e}\left(1+\frac{s}{w_1}\right)}{\frac{s^3}{w_m^2 w_1}+\frac{s^2}{w_m^2}+\left(\frac{1}{w_1}+\frac{1}{w_2}\right)s+1} \quad \text{..........} (6)$$

Where

$w_2 = \frac{K_1 K_e}{A^2}$ break frequency due to restrictor

$w_m = \sqrt{\frac{K_e}{M_v}}$ mechanical natural frequency

2.5 The long pipeline modeling

Considering fluid physical properties and motion feature in the pipeline such as mass, pressure, and damping, so that the damping dynamic model of the simple mass-spring shown in fig. 4 could be used for simulating the liquid in the pipeline. In this model, M denotes the liquid mass, B the damping coefficient, K the spring rate, $F_{(t)}$ the external force, and $X_{(t)}$ the displacement. The transfer function model may be derived as follows [15]:

$$G_p(s) = \frac{\frac{1}{K_o}}{\frac{s^2}{w_o}+\frac{2\delta_o}{w_o}s+1} \quad \text{..........} (7)$$

Where$w_o = \sqrt{\frac{K_o}{M_o}}$, and$\delta_o = \sqrt{\left(\frac{B_o^2}{4\,M_o K_o}\right)}$

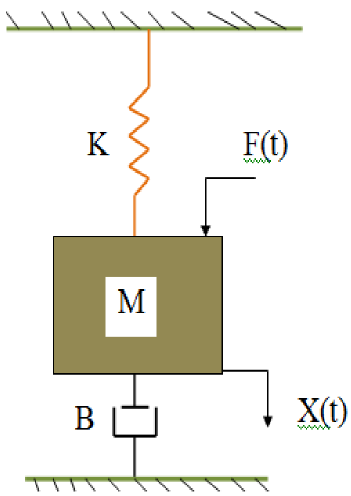

Fig. 4: Simulated model for the pipeline liquid [2]

2.6 The pump modeling

Assuming there is no slip between the electric motor and the hydraulic pump so that the rotational speed of the electric motor is equal to the rotational speed of the hydraulic pump. The ideal flow rate of the pump is given by [16]:

$$Q_p = D_p N \quad \text{..........} (8)$$

The actual flow rate of the pump (Qp. actual) is less than the ideal flow rate due to the fluid leakage and fluid compression. The continuity equation of the hydraulic pump may be given as [2] (see Appendix):

$$Q_p - Q_l = C_l\left[1+\left(\frac{s}{w_p}\right)\right]P_s \quad \text{..........} (9)$$

Where $w_p = \frac{C_l \beta_e}{V_t}$ and $Q_p - Q_l = losses\ in\ flow$

3. Methodology

The experimental side consists of two main parts; the mechanical part involves the hydraulic system components and the electrical part is represented by the electrical control system and other accessories.

3.1 Hydraulic system

A hydraulic system apparatus is shown in figs. 5 and 6 which involve atmosphere reservoir manufactured by Bratt Hydraulics, made of cast iron with one glass eye, with a capacity of 245 liters of hydraulic oil with HL32 type. The pump used in the system is a gear type, group 1P – P3000 series, from (DOWTY Company Production). The pump has a flow of 16.4 L/min at a speed of 1500 rpm and can provide a hydraulic pressure of 207 bar. The displacement or the geometric volume of this pump is 10.93cm3/rev. It is driven by a 3-phase induction motor, 50 HZ type ASEA to provide a 3 kW power to the pump with a speed of 1500 rpm. In order to control the flow direction in the system, 4/3 proportional control valve (Rexroth Company Production) model 4 WRE size 6 solenoid 24V as direct operated spool valves were used in the system. Fluid pressure was adjusted at the required pressure by a relief type Pratt Hydraulics Company Production, valve's DBD size 6. Manual flow control valve size 10 type (DV(10−i)/OP350) was also used. This valve works on the oil that has a viscosity limit between (2.8-380) cst and oil temperatures ranging between (-15 to 200oC) with a maximum pressure of 350 bar.

To protect the pump from reverse pressure, a check valve type (S10A1) size 6 was used. Double acting with a single rod cylinder was used as an actuator. This cylinder has a stroke length of 762mm, a bore of 38.1mm and a rode diameter of 19mm.

3.2 Electrical control system

The general structure of the electro-hydraulic elevator system is shown in fig. (7), which is divided into three main parts, electro-mechanical part, data acquisition (DAQ) board, and LabVIEW software within the computer. The electro-mechanical part consists of a hydraulic cylinder which drives the elevator cabin up/down, the proportional valve used as an actuator that controls the hydraulic cylinder, the position sensor type transducers potentiometric series TLH transducers with total length (750 mm) and pressures sensors and the elevator door with its electrical driver. Arduino UNO DAQ board is used as an interface card between the computer and the electromechanical system. The LabVIEW software is programmed to control the hydraulic elevator system via the DAQ board.

Fig. 5: Electro-hydraulic elevator system prototype; 1. Tank, 2. Electrical motor, 3. Constant displacement pump, 4. Hydraulic cylinder, 5&6. Pressure transducers, 7. Gauge pressure, 8. Proportional valve, 9. Pressure valve, 10. Flexible hose, 11. Elevator frame, 12. Distance sensor, 13. Cabin frame, 14. Cabin door

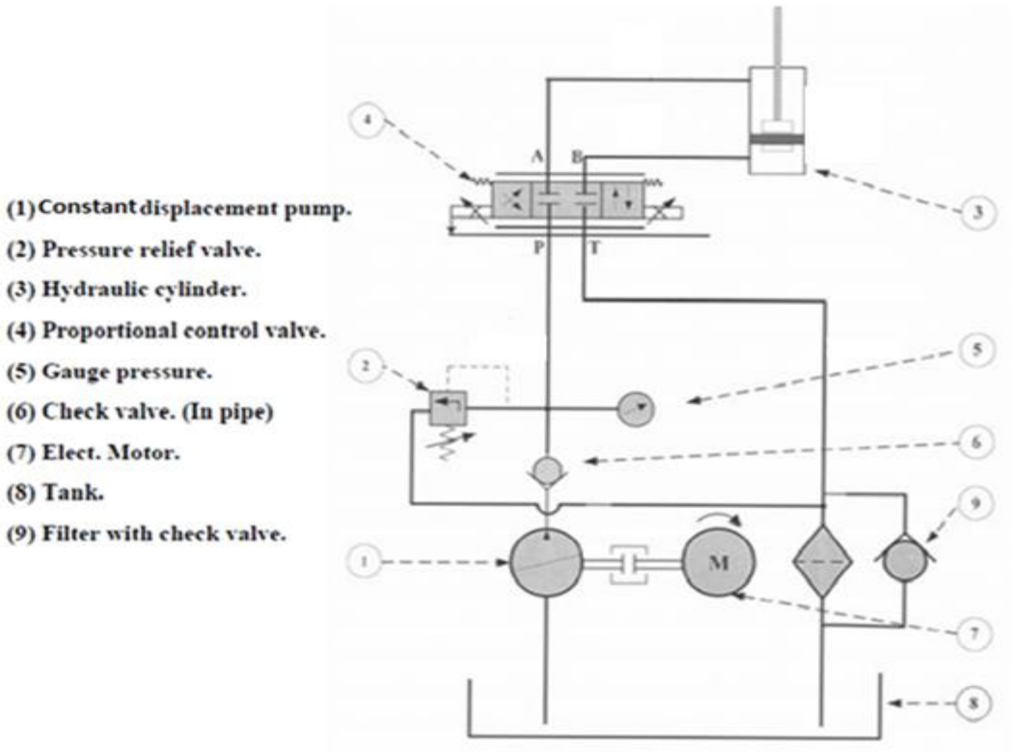

Fig. 6: Schematic diagram of the hydraulic system of the elevator

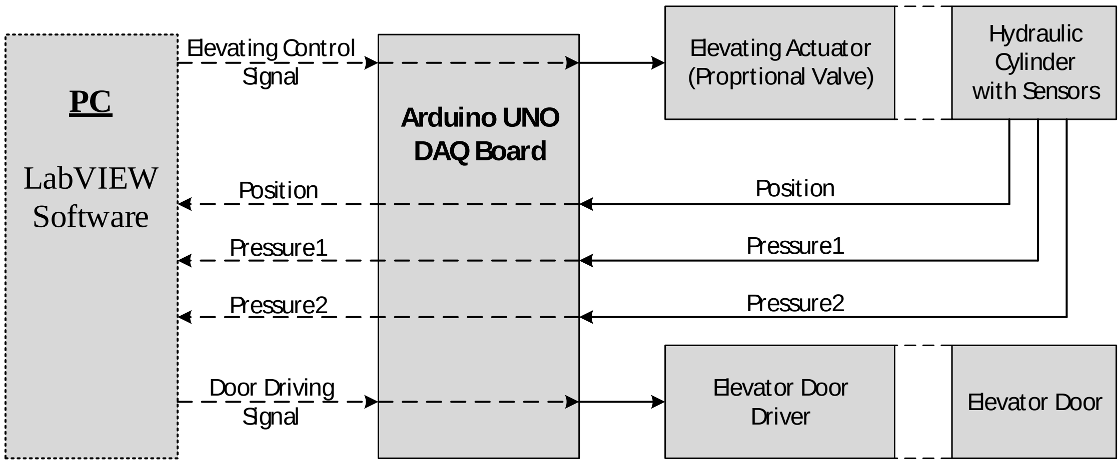

Fig. 7: The general structure of the electro-hydraulic elevator system

The electronic circuit of the hydraulic elevator is demonstrated in fig. (8), which controls the proportional solenoid valve by using solenoid driver (L298) based on PWM technique, and reading the signals of sensors statuses and sending the control signals to the electromechanical system via the DAQ board.

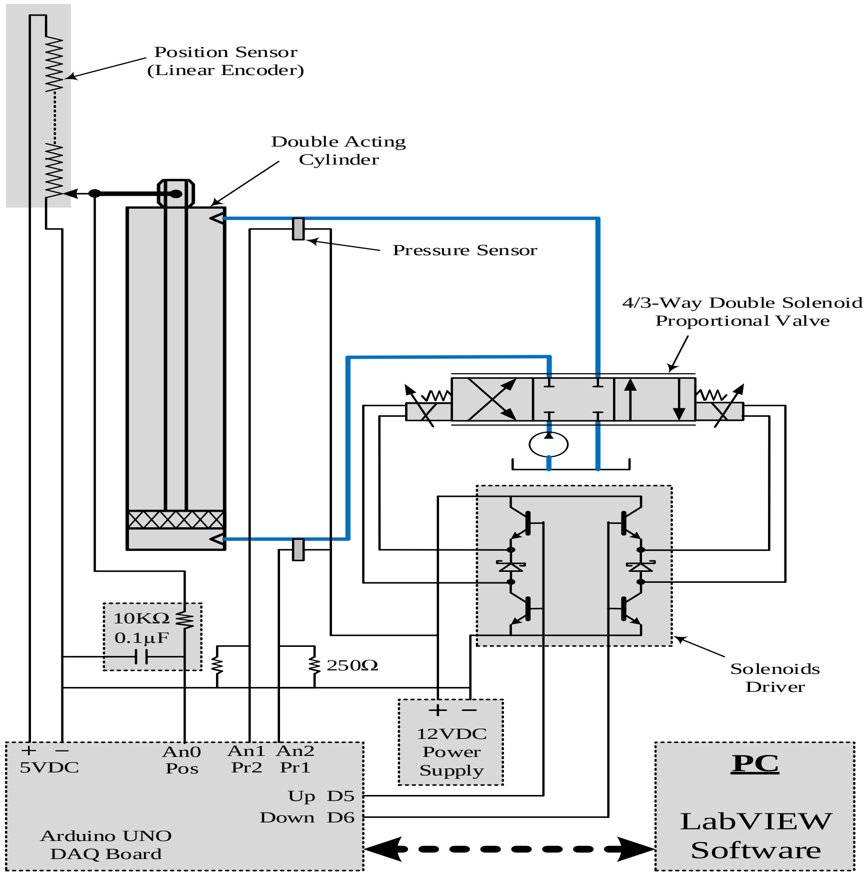

Fig. 8: Hydraulic elevator control Circuit

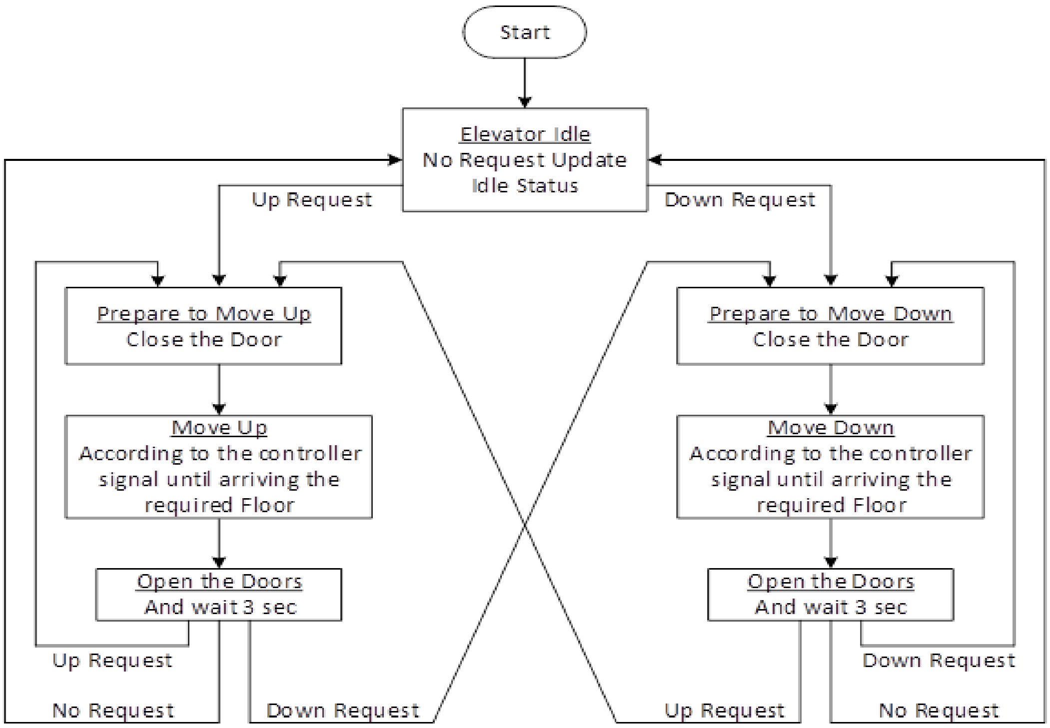

Fig. (4.23) Hierarchical state chart for the hydraulic elevator control

4. Results and Discussion

4.1 Theoretical and simulation tests

The results of the stability analysis of the hydraulic elevator system with the PI controller depend on the transfer functions of the system which were previously estimated in Section Two.

The elevator system simulation using MATLAB Simulink software is accomplished using the transfer function which simulates the open-loop elevator hydraulic cylinder with its actuator (proportional valve). The linear encoder (displacement sensor) returns the position signal as feedback, where it is compared with the setpoint to get the error which is supplied to the system via PI controller having the parameters (Kp = 0.003, Ki = 0.001).

Fig. 9 shows the variation of stroke displacement (H), setpoint for a proposed travel pattern of six steps between the three floors as (1→2→3→2→1) and then (1→3→1) with time, where the elevator starts motion from a distance (5cm to 35cm and from 35cm to 65cm) upward, and the downward distance from (65cm to 35cm, 35cm to 5cm), respectively.

From floor 1 to 2, it is noticed that the elevator starts moving with acceleration to reach the desired speed. After 1.5 sec, when a position elevator reaches to 30cm, the motion being to decelerate gradually and smoothly to reach a rise time (tr) of about (95 – 100) % of the set point, when the rise time from floor 1 to floor 2 requires 4 sec. This procedure is repeated when traveling from floor 2 to floor 3, 3 to 2, and 2 to 1.

The same figure shows the elevator move from 1 to 3 directly without a stop in floor 2. The elevator accelerates rapidly, while begin to decelerate at 60 cm height and then stops smoothly.

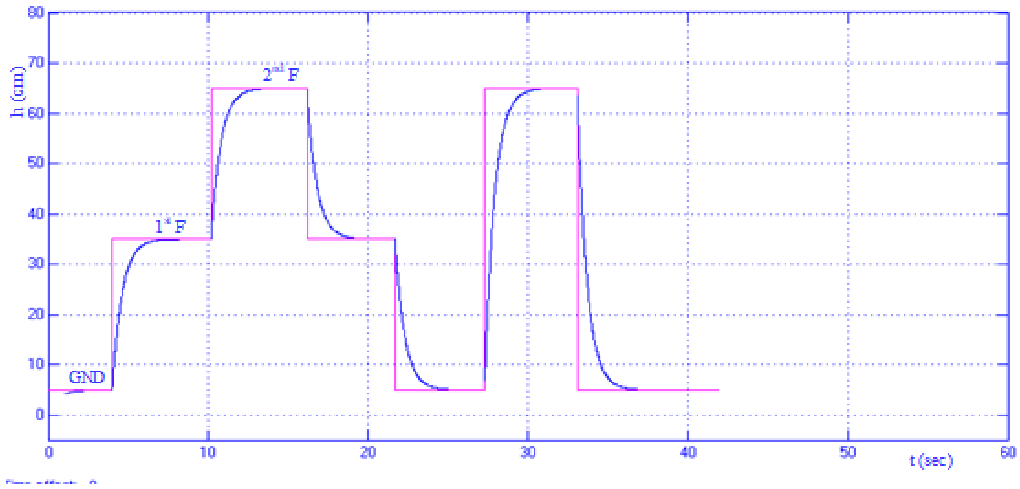

Fig. 9: the variation of stroke displacement with time

4.2 Experimental results

To evaluate the performance of the designed hydraulic elevator under study, the pressure supply was applied with values of 7, 8, and 10 bar for the two cases without load and then with a load of 15 Kg.

The elevator was constructed from three floors, firstly, it transfers from 1 to 2 with stroke of 30cm from position (5cm to 35cm) and secondly from 2 to 3 with a stroke of 30cm from position (35cm to 65cm). To enhance the elevator performance, a PI controller has been used and aided by Lab VIEW software.

4.2.1 Accurate position with time

The aim of developing the position control is to guarantee accurate motion of a standalone hydraulic servo-mechanism by relying on the experimental validation of the servomechanism itself. The figures below were derived through laboratory testing.

Fig. 10 shows the variation of stroke displacement (H) with time for traveling up and down across floors (1→2→3) and vice versa.

Supply pressure of (7 bar) was used with controller coefficients of (Kp = 5.3 and Ki = 0.17) when the elevator is loaded with a weight of 15 Kg.

From floor 1 to 2, it’s noticed that the elevator starts moving with acceleration to reach the desired speed. After 4.15 sec, when a position elevator reaches to 27 cm, the motion will begin to decelerate gradually and uniformly. After 3 sec of deceleration, the elevator will reach to a rising point (95-100) % of the setpoint. Therefore, it requires 7.15 sec rise time to transfer from floor 1 to floor 2.

This procedure is repeated from floor 2 to floor 3 with a difference in time required to start deceleration with 4.1 sec. The time needed for this stroke to reach rise point in floor 3 is 7.1 sec.

It's seen that a period time required to reach rise time to move from floor 1 to floor 2 is less than moving from floor 2 to 3 because the inactive volume in active chamber firstly needs some interval time to accumulate pressure to reach the supply pressure value.

For a downward motion from floor 3 to 2, due to reduce the area of the piston, weight gain and inertia force, the elevator drops with rapid acceleration for a short interval time of 4.25 sec then begin to decelerate in a period time of 6 sec. Where, it needs 10.25 sec to reach an accurate position to achieve the drop time.

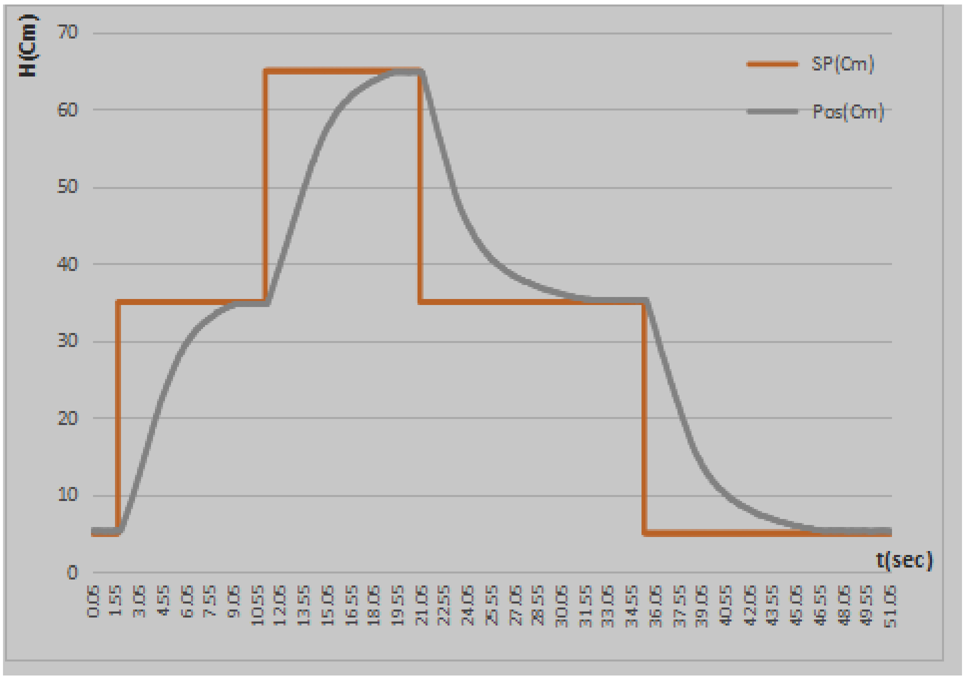

Fig. 10: variation of position with time (PI controller, 15Kg load, and 7 bar supply pressure); floors pattern (1→2→3) and (3→2→1).

The variation of position with time at 7 bar pressure supply, no-load and PI controller is represented in fig. (11). Firstly, the elevator begins to move down from floor 3 to 2. It is noticed the starting with rapid acceleration and then a gradual deceleration begins when it approaches the rising time (tr). Also, the period time required to reach rise time at no-load is less than with a load of 15Kg due to the inertia effect of the load, which makes the elevator, in this case, takes more time than without load, the same procedure is repeated when moving from floor 2 to 1.

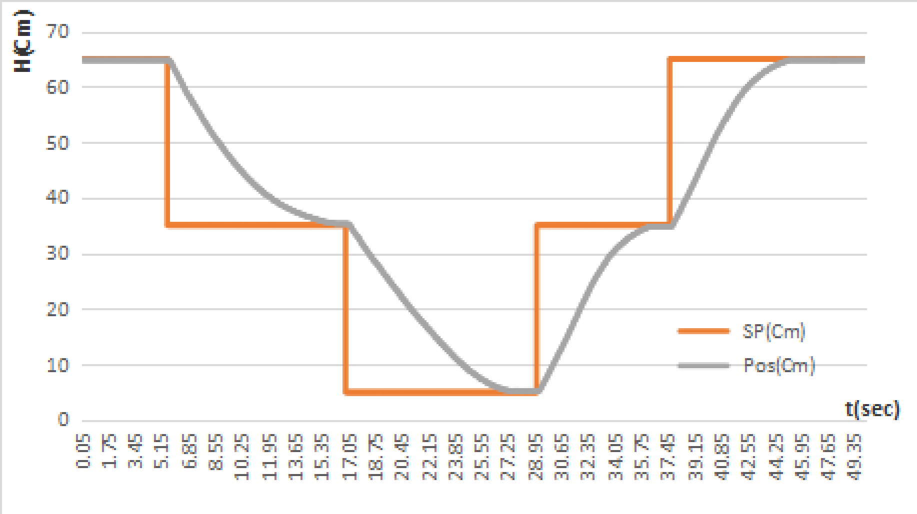

Fig. 11: Variation of position, with time (PI controller and 7 bar supply pressure at no-load); floors pattern (3→2→1→2→3).

The variation of both the stroke position and pressure of chamber (active chamber and passive chamber), with load of 15 Kg is shown in fig. (12). In order to make the elevator motion more comfortable because of pressure difference increases, the speed of stroke starts from 5cm with acceleration and begins to decelerate at position 30.5 cm, at which the pressure drops to less than 8 bar, due to the stroke speed increasing. The interval time required to reach the rise time is less than that of 7 bar when the elevator is moving upward from floor 1 to floor 2 and from floor 2 to floor 3, and moving downward from floor 3 to floor 2 and floor 2 to floor 1. This is because of the inertia force of the fluid and the load. To reach the accurate position floor 3 to floor 2 and from floor 2 to floor 1, the system requires 9.5 seconds.

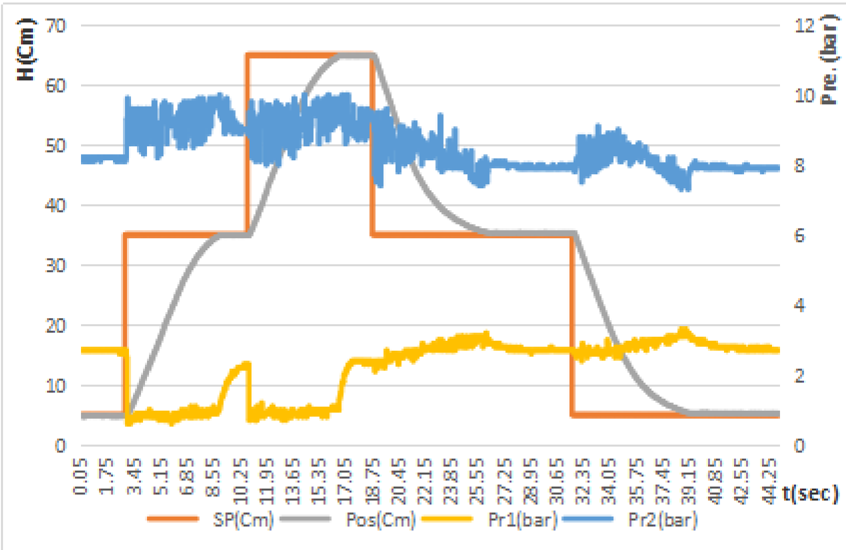

Fig. (12): Variation of position, pressure Pr_1, and pressure Pr_2 with time (PI controller, 15Kg load, and 8 bar supply pressure).

The elevator moving from floor 1 to 3 directly without a stop at floor 2 as shown in fig. (13). The elevator accelerates rapidly, while starts slowing down in 55 cm more smoothly to make the elevator getting ready to stop. Thereby reaching the rise time as less than moving from floor 1 to 2 or 2 to 3, because the long-stroke takes more time for motion to decelerate in which it begins at distance longer than 2 to 3.

The performance of the elevator without using any controller is illustrated in fig. (14). The main differences in the performance between the uncontrolled elevator and PI controlled elevator are as follows:

1. The stroke starts and ends for up motion without any acceleration and deceleration, respectively.
2. For down motion, the stroke motion is slightly decelerated for a small interval time.
3. There is a difference in time period to reach final position from floor 1 to 2 as compared with 2 to 3.
4. There is no sufficient time to reach the set point for all strokes (up and down), which leads to uncomfortable motion and generates increment in the speed and acceleration and vibration.
5. Pressures Pr_1 and Pr_2 without controller are more difficult to analysis compared with using the PI controller. In order to make a comparison, another value of pressure supply of (10 bar) may be used.

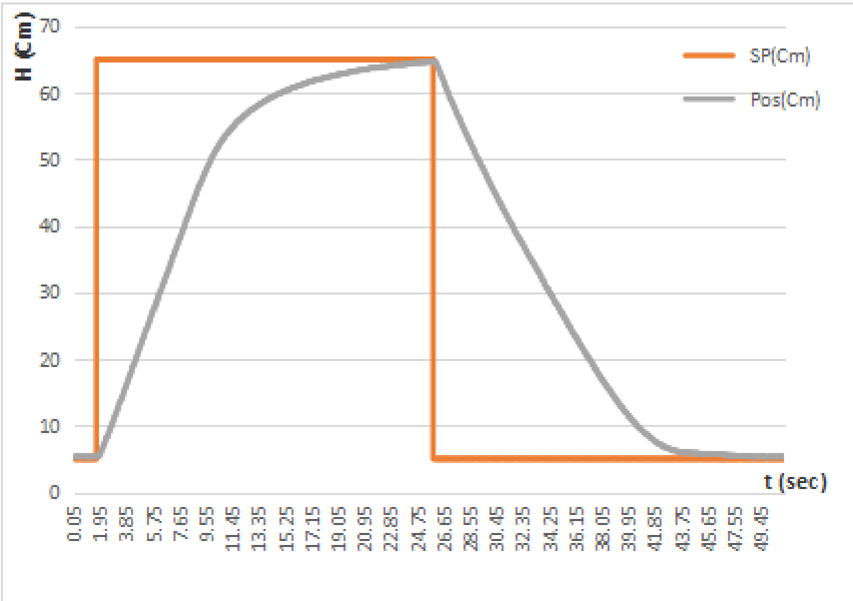

Fig. 13: Variation of position with time (PI controller and 7 bar supply pressure); floors pattern 1→3.

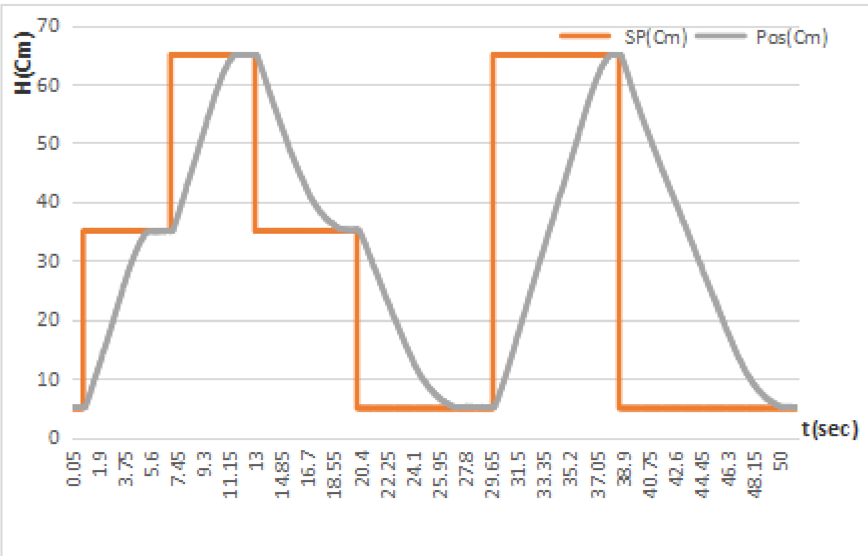

Fig. 14: Variation of position, pressure Pr_1, and pressure Pr_2 with time (without controller, no-load, and 8 bar supply pressure).

4. Conclusions

To enhance the performance of a conventional hydraulic elevator prototype, a servo mechanism based on a PI controller was used with LabVIEW software. Most important conclusions may be drawn:

1. It is preferable to use a PI controller to reduce both the fluctuating in pressure supply to the actuator and reach an accurate position as less as possible time.
2. Using a PI controller with proportional gain Kp = 5.3, integral gain Ki = 0.17 offers higher response and stability than without PI controller.
3. No deceleration is happening in the system without a controller, while a good deceleration is gotten in the system with PI controller, which reduces the vibration and jerk at a stop.
4. The results proved that the combination of the servo system and conventional hydraulic elevator improved its performance.

CONFLICT OF INTERESTS.

- There are no conflicts of interest.

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Appendix

Theoretical Analysis of the Proportional valve control

$$Q_l - Q_{l1} = \Delta Q_l = \frac{\partial Q_l}{\partial X_v}|_t \Delta x_v + \frac{\partial Q_l}{\partial P_l}|_t \Delta P_l \quad \text{...... (A1)}$$

$$Kq = \frac{\partial Q_l}{\partial X_v} \quad \text{...... (A2)}$$

$$Kc = -\frac{\partial Q_l}{\partial P_l} \quad \text{...... (A3)}$$

Where Kq is the flow gain and Kc is the flow pressure coefficient. Now, substituting Eqs. (A2) and (A3) in (A1) will give:

$$\Delta Q_l = Kq\Delta x_v - Kc\Delta P_l \quad \text{...... (A4)}$$

$$Q_l = Kqx_v - KcP_l \quad \text{...... (A5)}$$

Double acting single rod cylinder:

$$Q_l = Aav\frac{dx_p}{dt} + C_{ip}(P_l) + \frac{V_t}{4\beta}\frac{d}{dt}(P_l) \quad \text{...... (A6)}$$

$$\frac{dx_p}{dt} = \dot{x_p}\ \frac{d}{dt}(P_l) = \dot{P_l} \quad \text{...... (A7)}$$

$$Q_l = Aav.\dot{x_p} + C_{ip}(P_l) + \frac{V_t}{4\beta}\dot{P_l} \quad \text{...... (A8)}$$

Dynamic equations:

$$\frac{V_p}{x_v} = \frac{Kq}{[\frac{V_t\ M_t}{4\beta Aav.}\dot{V_p} + (\frac{M_t}{Aav.}(C_{ip} + Kc))V_p + Aav.]} \quad \text{...... (A9)}$$

Therefore, the second order transfer function equation will be:

$$\frac{x_o}{x_i} = \frac{K}{\frac{1}{w_n^2}D^2 + \frac{2\xi}{w_n}D + 1} \quad \text{...... (A10)}$$

$$\frac{x_o}{x_i} = \frac{b_o}{a_2D^2 + a_1D + a_0}$$

Where $w_n = \sqrt{\frac{a_0}{a_2}}$ and $\xi = \frac{a_1}{2\sqrt{a_o\ .\ a_2}}$

Now, by comparing Eq. (A9) with the second order Eq. (A10), the hydraulic natural frequency and damping ratio could be found as follows:

1- The hydraulic natural frequency (w_n) is:

$$w_n = \sqrt{\frac{Aav.}{\frac{v_t}{4\beta}.\frac{M_t}{Aav.}}} = \sqrt{\frac{Aav.^2\ 4\beta}{v_tM_t}}$$

$$w_n = 2Aav.\sqrt{\frac{\beta}{v_{tM_t}}} \quad \text{...... (A11)}$$

2- The damping ratio (ξ) is:

$$\xi = \frac{M_t(C_{ip} + Kc)}{Aav.\sqrt{\frac{M_tV_t}{\beta}}} = \frac{C_{ip} + Kc}{Aav.}\sqrt{\frac{M_t\beta}{V_t}} \quad \text{...... (A12)}$$

$$\xi = \frac{1.8 * 10^{-11} + 1.9 * 10^{-12}}{0.998 * 10^{-3}} * \sqrt{\frac{138.32 * 108 * 10^{8}}{8.686 * 10^{-4}}} = 0.827$$

Now, substituting Eqs. (A11) and (A12) into (A10) yields the transfer function of the valve-controlled cylinder:

$$G_s = \frac{V_p}{x_v} = \frac{\frac{Kq}{Aav.}}{S(\frac{1}{w_n^2}S^2 + \frac{2\xi}{wn}S + 1)} \qquad \text{...... (A13)}$$

The pump modeling

$$Q_p = D_p N \qquad \text{....... (A14)}$$

$$Q_p - C_l P_s - Q_l = \frac{V_t}{\beta_e} s P_s \qquad \text{....... (A15)}$$

$$Q_p - Q_l = C_l \left[1 + \left(\frac{V_t}{C_l \beta_e}\right) S\right] P_s \qquad \text{....... (A16)}$$

$$Q_p - Q_l = C_l \left[1 + \left(\frac{S}{w_p}\right)\right] P_s \qquad \text{....... (A17)}$$

Where$w_p = \frac{C_l \beta_e}{V_t}$, and$Q_p - Q_l = losses\ in\ flow$.

Programming of LabVIEW Software

```
int  x1 ;
 if(x==0)
        x1 = 65;
 else if (x==1)
        x1 = 35;
else   x1 = 5 ;
if (abs(SP0-Pos)<=0.2)
                OpenD = 1 ;  // open the door
if (abs(x1-Pos)>0.5)
                OpenD = 0 ;  // close the door
if (Dor==1)                        // door is opened
                SP=SP0;
if ((Dor==0) &&(abs(x1-Pos)>0.5)) // door is closed
               SP =x1;  // change set point
```

```
int Flash =0;
if( abs(e) <0.2)
       {
          UpPin = 0 ;
          DnPin = 0 ;
          Dither = 1 ;
        }
else if(e >= 0.2)
       { Flash =0.6* DutyC*120+85  ;
         DnPin = 0;
         UpPin = Flash;
         Dither = 0 ;
        }
else
        { Flash = 0.6*DutyC*(-140)+95 ;
           UpPin = 0 ;
           DnPin = Flash;
           Dither = 0 ;
        }
```

السيطرة على سرعة المصعد الهيدروليكي باستخدام نظام المؤازرة الكهرو هيدروليكي

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الخلاصة

المصعد الهيدروليكي هو أحد أنواع المصاعد المستخدمة في المباني ذات الارتفاع المنخفض والذي لا يزيد عن الثلاث طوابق. في هذا البحث تم نصب وتنفيذ وتشغيل نموذج مصغر لمصعد هيدروليكي والتحقق النظري والتجريبي باستخدام صمام تناسبي، ومسيطر نوع PI. تم تنفيذ المصعد مع ثلاث طوابق بارتفاع كلي يبلغ 76 سم مع جميع المكونات والملحقات الهيدروليكية والكهربائية، حيث ان اتمتة المصعد كانت باستخدام مايكرو كونترولر اردوينو Arduinoنوع أونو UNO.

برنامج الحاسوب اللابفيو LABVEIWتم استخدامه للسيطرة على المصعد من خلال مسوق التيار المستمر L298 DC.

أفضل قيم لباراميترات المسيطر PIتم الحصول عليها تجريبيا.

النتائج أظهرت دعم وتحسين الأداء للمصعد الهيدروليكي من خلال استخدام نظام المؤازرة الكهروهيدروليكي لتحقيق التوقف السلس وراحة المستخدمين للمصعد في التنقل بحركة انسيابية بين الطوابق.

الكلمات الدالة: سيطرة السرعة، مصعد هيدروليكي، نظام المؤازرة، صمام تناسبي، مسيطر نوع PI