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Performance Improvement of a Conventional Hydraulic Elevator by Using Electro-Hydraulic Servo Mechanism

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K E Y W O R D S

Electro-hydraulic; servo mechanism; elevator; proportional valve; PID controller

A B S T R A C T

An electro-hydraulic elevator is a new type of enhanced elevators that are used in low-rise buildings that do not exceed more than three floors. In this paper, an electro-hydraulic servo system for controlling the speed of a hydraulic elevator prototype by using a proportional valve and PID controller was investigated theoretically and experimentally. A three floors elevator prototype model with 76cm height was built including hydraulics components and electrical components. The elevator system is fully automated using the Arduino UNO board based Data Acquisition (DAQ) system. LabVIEW software is used to control the hydraulic elevator system through L298 DC drive via the DAQ board. The best PID gains was obtained experimentally. The elevator system prototype could be emplimented for educational purposes; such as learning the undergraduate students in the Electromechanical Engineering Department in the University of Technology how to structuring the electro-hydraulic elevator as well as the appropriate control strategy.

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1. Introduction

Elevators are 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. 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 stories and move at a maximum speed up to 200 ft/min. In the modern world of today, hydraulics plays 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 [1].

Electro-hydraulic servo mechanism (EHSM) is extensively used in numerous industrial applications and mobile systems due to its high power to weight ratio, fast response, good positioning capabilities, high stiffness, etc. Two methods may be used to control the speed of the hydraulic cylinder in the systems that rely on the EHSH. The first method is a variable displacement pump that controls the flow to the cylinder, while the second method based on a servo or proportional valves. It is a closed-loop system for speed control, depends on the actuated error signal represented by the difference between the input signal and the feedback signal supplied to the controller to decrease the error signal and make the system output approaching to the desired value. The search [2] had focused on studying the fixed displacement pump-fixed displacement motor through the use of proportional valve PV and PID controller.

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. [3] 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 elevators 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. [4] 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 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 of up to 3 Hz. Also, the error amplitude is less than 10% at a frequency of 3–5 Hz. Liu et al. [5] 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, 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 case of an emergency stop, and this system can realize smooth starting of the hydraulic elevator with preventing sudden stop shock. Sun et al. [6] 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 used for controlling the cabin's speed. They deduced that the modern generation of the hydraulic elevator could deliver a great energy-saving performance as compared with the conventional elevators. And the energy consumption of the hydraulic elevator has been reduced remarkably. Xu and Wang [7] presented a traditional PID controller and self-tuning Fuzzy PID controller 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 better effect than a 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. [8] studied the performance of high-speed switching valves used as pilot valves. The key idea was that a three-way main stage valve could be controlled, in each chamber, by two high speeds of switching valves. The speed is controlled via adjusting the duty ratio of the control PWM signal. The theoretical analysis and dynamic simulation showed the control performance of proportional pilot valves and high-speed switching pilot valves. The results showed that the primary stage could work well if the continued pilot components are replaced by high-speed switching valves. Liu and Gao [9] had developed experimentally the position control of a hydraulic cylinder controlled by a high-speed on-off valve to realize the accurate position control. The high-speed on-off valve HSV could be utilized to realize the exact position control through adjusting the PWM signal and the compound algorithm of PI and speed feed forward- displacement feedback is provided to effectively reduce the position error through analyzing the flow characteristic.

The main objective of the current research is the possibility of applying 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 is leading to determine the flow rate and thus control the hydraulic cylinder speed, this represents control of elevator speed up and down. The process of control of the solenoids valve may be implemented through using pulse width modulation (PWM) technology, through which it can control the provided voltage to the solenoids, and this process of controlling depends on the value of error. This paper, experimentally investigates on controlling the velocity of a hydraulic elevator using a fixed displacement pump and proportional valve with a conventional PID controller depending on controlling the proportional solenoids valve and thus controlling leads to determine the flow rate and thus control the speed of the hydraulic cylinder, in other words, control the speed of the elevator up and down.

2. Theoretical Analysis

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

Ⅰ. Proportional valve control

The proportional valves are one type of spool valves, Figure 1; the general relations and performance characteristics for the proportional valve have been derived and developed to meet assumptions:

- Pressure source is constant
- Neglecting the fluid inertia
- No flow reversal or cavitation's
- Neglecting the return line pressure (P_o)
- Matched and symmetrical orifices
- Considering the power match to the hydraulic cylinder and the relative valve

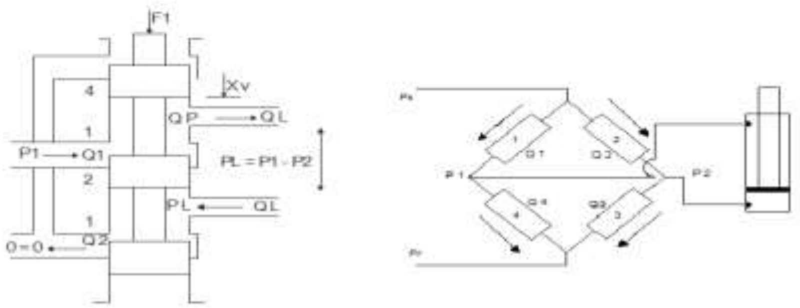

Figure 1: Schematics of 4-way, 3-position spool valve

If the area of the orifice of the slide valve is matching and symmetrical then the flow pressure equation in the valve would be [10] (see Appendix-A):

$$Q_l = Kqx_v - KcP_l \tag{1}$$

Eq. (5) represents the Linearized equation of the pressure-flow to the control valve [10].

Ⅱ. Double acting single rod cylinder

Considering the hydraulic cylinder of Figure 2, the application of the continuity equation to the two sides of cylinder yields [12]:

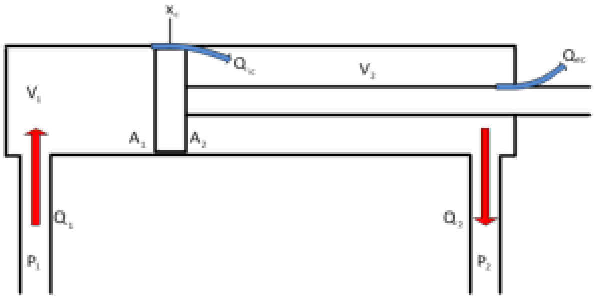

Figure 2: Schematic diagram of a double-acting cylinder [13]

Eq. (2) will be (see Appendix-A):

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

Ⅲ. Dynamic equations

Now, by applying Newton's second law to the forces on the piston [10] (see Appendix-A):

1- The hydraulic natural frequency (w_n) is:

$$w_n = 2Aav.\sqrt{\frac{\beta}{v_{tM_t}}} \qquad (3)$$

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

Note that the cylinder data label has the following specifications:

Stroke = 76.2 cm, Bore = 3.81 cm, Rod diameter = 1.9 cm.

This will get the following:

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

Spool diameter = 0.0125 m (measured)

Mt =14.1 Kg×9.81 = 138.32 N, *Vt* =8.686×10^{-4} m^3 (calculated)

Ps = 10 bar (regulated)

β = 108×10^8 N/m^2, Cip = 1.8×10^{-11} m^4.sec/Kg (manufactures data)

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

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

2- The damping ratio (ξ) is:

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

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

Now, substituting Eqs. (11) and (12) into (10) yields the transfer function of the valve-controlled cylinder:

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

Ⅳ. Pressure relief valve modeling

Figure 3 shows a schematic of a single-stage pressure control valve (relief valve). The equations describing spool motion are [10]:

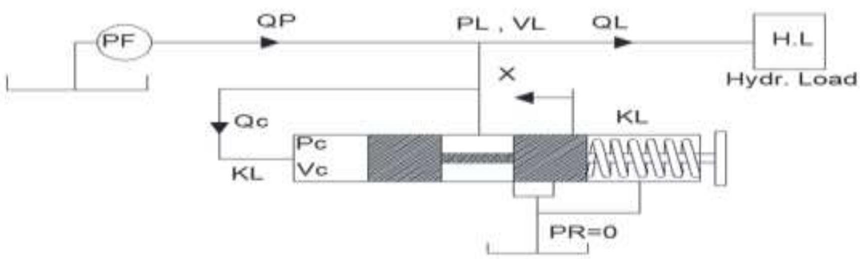

Figure 3: Single-stage pressure control relief valve

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

Where

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

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

V. The long pipeline modeling

Considering fluid physical properties and motion features in the pipeline such as mass, pressure, and damping, so that the damping dynamic model of the simple mass-spring shown in Figure 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 [1]:

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

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

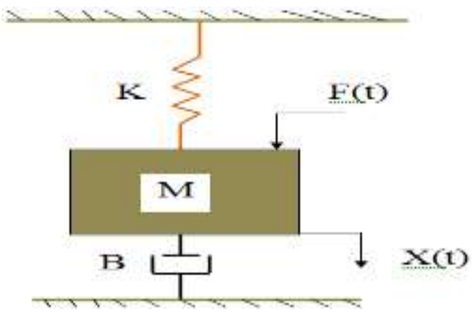

Figure 4: Simulated model for the pipeline liquid [1]

VI. 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:

$$Q_p = D_p N \quad (8)$$

The actual flow rate of the pump (Qp. actual) is less than the ideal flow rate due to fluid leakage and fluid compression. The continuity equation of the hydraulic pump may be given as [1] (see Appendix-A):

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

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 represented by the electrical control system and other accessories.

I. Hydraulic system

A hydraulic system apparatus is shown in Figures 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 10.93 cm^3/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. 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 adjustable 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 200°C) 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.

Ⅱ. Electrical control system

Figure 7 shows the general structure of the electro-hydraulic elevator system 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 that drives the elevator cabin up/down, the proportional valve used as an actuator that controls the hydraulic cylinder, the position 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.

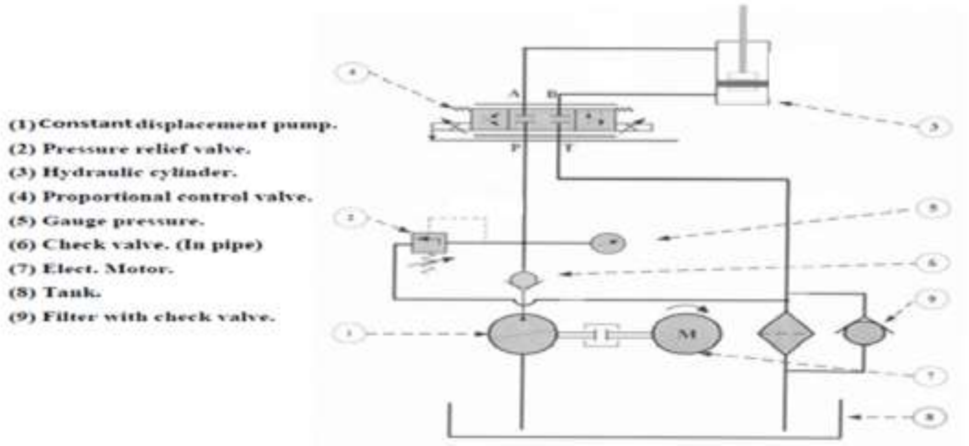

Figure 6: Schematic diagram of the hydraulic system of the elevator[11]

Figure 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

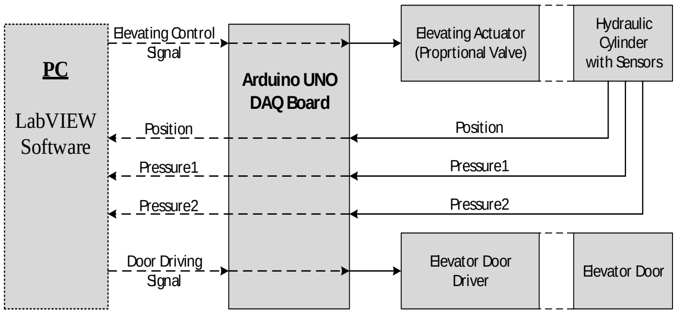

Figure 7: The general structure of the electro-hydraulic elevator system

Figure 8 demonstrates the electronic circuit of the hydraulic elevator which controls the proportional solenoid valve by using a 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.

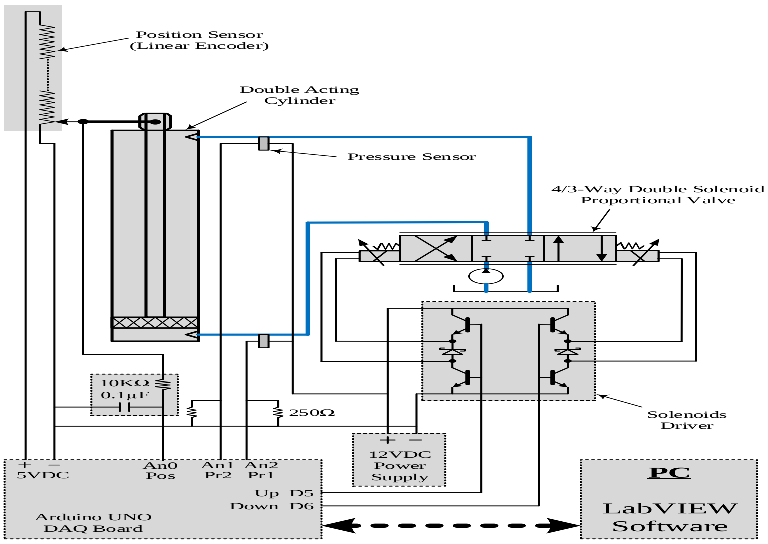

Figure 8: Hydraulic elevator control Circuit

4. Results and Discussion

To reduce oscillation in the system and to obtain a comfortable motion with reach an accurate position as less as possible time in the designed elevator under steady, PI controller was used by LabVIEW software with Arduino circuit.

Figure 9 shows the variation of stroke displacement (H) with time, Figure 10 shows the variation in pressure oscillation at the active chamber (Pr_1) and inactive chamber (Pr_2) with time for up and down floors motion 1 to 2, 2 to 3 and vice versa. Supply pressure was used (7 bar) with the controller coefficient ($Kp = 5.3$, $Ki = 0.17$) with 15 Kg load. From floor 1 to 2 its noticed that the elevator starts moving with acceleration to reach the desired speed. After 4.55 seconds when a position elevator reaches 29 cm the motion being to decelerate gradually and uniformity. At 3 seconds after deceleration, the elevator moment being to reach rise time that represents (95-100) % of setpoint in which a period time required from floor 1 to floor 2 to reach a (rise time) is 7.55 second. This procedure repeated from floor 2 to floor 3 with a difference in a period required to start deceleration with 2.2 seconds, the time for this stroke to reach rise time in floor 3 is 8.1 second. Its seen that a time required to reach rise time from floor 1 to floor 2 is less than floor 2 to 3. Because inactive volume in the active chamber firstly needs some interval time to accumulate pressure to reach supply pressure value. For a down motion from floor 3 to 2 the elevator drop with rapid acceleration for a short interval time (4.4) second then begins to deceleration in a period time (5.95) second. Where it needs 10.35 seconds to reach an accurate position to achieved rise time.

Also, interval time needed to reach rise is longer than up position, because of inertia force result load to be again with pressure supply. On another hand figure (10) demonstrate the variation in both pressure oscillation chamber (Pr_1) and chamber (Pr_2) between (6 – 8) bar and (0 – 2) bar respectively.

The pressure transients result from pump ripple, opening/closing of valves, actuators bottoming out and so on. Also, it's noted that when stroke ended at 29.5 seconds the pressure chamber Pr_1 drops to minimum value while pressure chamber Pr_2 increases to maximum value due to cut off pressure supplying and have increased backpressure. Figure 11 shows the variation of stroke displacement (H) with time, Figure 12 shows the variation in pressure oscillation at the active chamber (Pr1) and passive chamber (Pr2) with time for up and down floors and both figures without the controller, It found that ended motion when reach to rise time occurs without deceleration in which load to effect on the cylinder and increases the vibration in the system, the process repeated from 2 to 3 and for down motion. Also, it found a large degree in fluctuating pressure Pr1 in which starts from 8.6 bar and drop nonlinearity to 7 bar as well as a high range of oscillation in which enclosed between ± 1 bar.

For comparison purposes to perform the elevator with and without the controller. Figure 11 and 12 illustrator performance the elevator was used without any controller uses in which can be listed below the main difference between uncontrolled and PI controller used.

1. The stroke starts and ended for up motion without any acceleration and deceleration respectively.
2. For down motion stroke motion is slightly decelerate for a small interval time.
3. There is a difference in the time period to reach a final position from 1 to 2 as a compare with 2 to 3.
4. There is no sufficient time to reach setpoint for all stroke (up and down) in which load to uncomfortable motion which increases the speed and acceleration and vibration generated.
5. Pressure Pr_1 and pressure Pr_2 are more difficult to analyze compared with PI controller.

When the system was built, installed and operated practically. The oscillation system was not obtained only when adding the proportional gain value (Kp=160).

Figure 13 show the variation in the displacement of the stroke elevator with time for the floor pattern (1 →2 → 3) and vice versa with 160 proportional gain. From floor 1 to 2 the elevator starts moving without acceleration to reach a setpoint (35 cm). After the setpoint the elevator starts oscillating up and down as an unstable system, the system reaches to the rise time at 2.25 second, the peak time at which it arrives the first overshoot at 5.35 second, and the settling time the time required to reach and stay within a range about 2% from the final value at 5.45 second. Where the maximum peak value of the response curve measured from the setpoint (Mp) equal to 0.9 cm. This procedure repeated from floor 2 to floor 3 with the difference in a period time required.

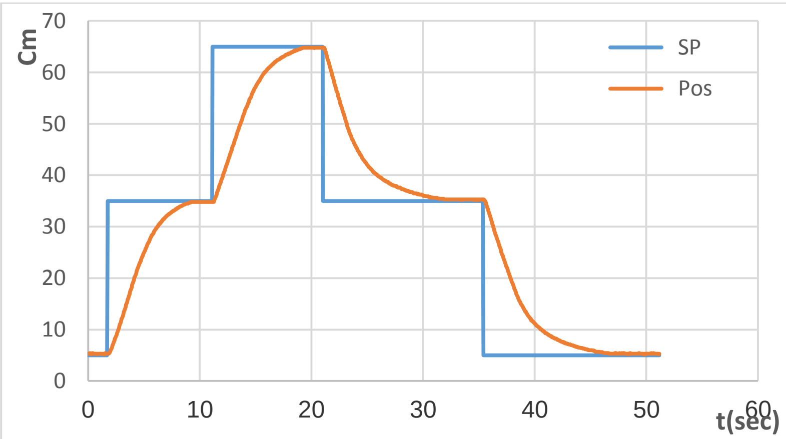

Figure:9 variation of position, with the time (PI controller and load 15Kg, 7 bar supply pressure)

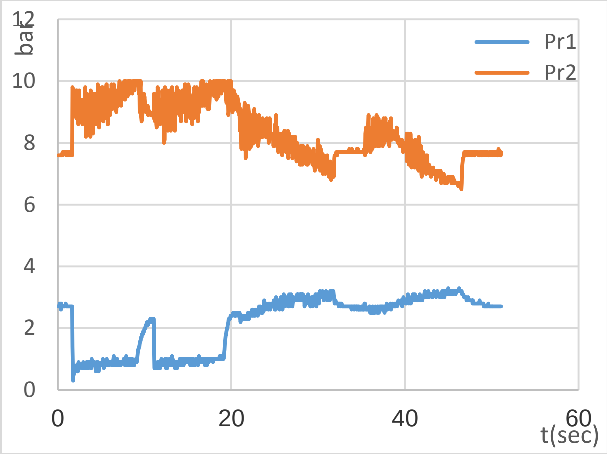

Figure: 10 variations of pressure Pr1 and pressure Pr2 with the time (PI controller and load 15Kg, 7 bar supply pressure)

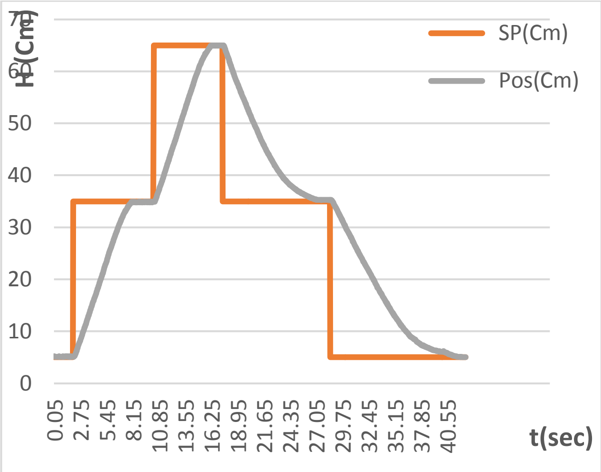

Figure: 11 variations of position, with the time (without PI controller, and load 15Kg, 7 bar supply pressure)

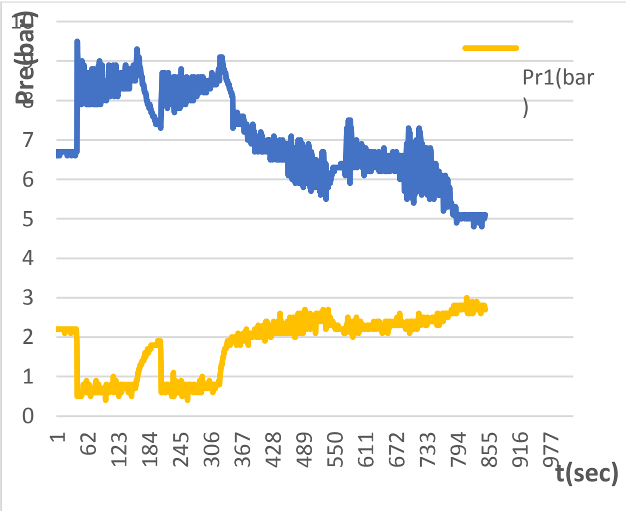

Figure: 12 variations of pressure Pr1 and pressure Pr2 with the time (without PI controller and load 15Kg, 7 bar supply pressure)

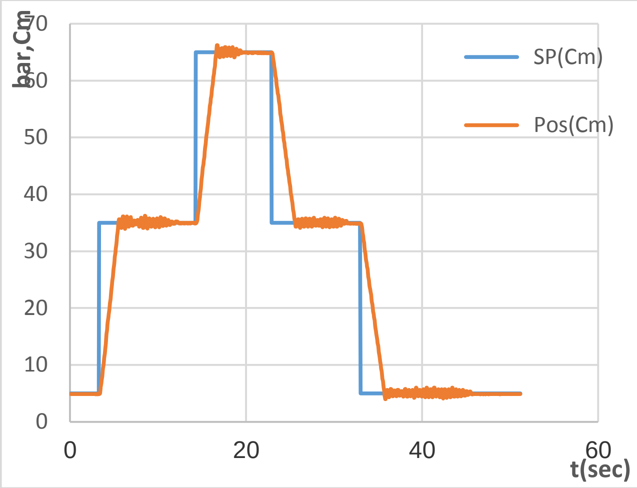

Figure: 13 variations of position, with the time (proportional gain =160, and load 15Kg, 7 bar supply pressure)

5. 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 controller to reduce both the fluctuating 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 = 0.1 offers higher response and stability than without PI controller.
3. There is no deceleration getting in the system without the controller, while a good deceleration getting in the system with PI controller which reduces the vibration and jerk at the stop.

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Nomenclature

A_1, A_2	*Orifice area gradient, m²*
A_{p1}, A_{p2}	*Area of piston, m²*
A_s	*Spool end area, m²*
C_d	*Discharge coefficient of valve, m³/sec*
C_{ip},C_{ep}	*Internal and external leakage coefficient of the hydraulic cylinder, m³/sec/Pa*
C_{tp}	*Total leakage coefficient of cylinder, m³/sec/Pa*
D	*Derivative gain*
d/dt	*Derivative*
F_{hyd}	*Total force on piston, N*
X_p	*Piston displacement, mm*
X_v	*Valve displacement, mm*
ΔP	*Pressure difference, bar*
ΔQ	*Flow difference, m³/sec*
$\dot{x}_p$	*Velocity of piston, m/s*
$\ddot{x}_p$	*Acceleration of piston, m/s²*

Greek Symbols

β	*Bulk modules, N/m²*
F_1	*Spring pre-load force, N*
K_a	*Valve flow gain, m²/sec*
K_c	*Flow pressure coefficient, m²/sec.P_a*
K_o	*Spring rate of pipe, KN/m*
M_t	*Total mass, Kg*
N	*Rotational speed of pump, rpm*
P	*Proportional gain*
P_{So},P_{Ro}	*Supply and return pressure at initial Condition, bar*
P_l	*Load pressure, bar*
P_s	*Source pressure from power unit, bar*
P_1, P_2	*Pressure of port A and B, bar*
Q_l	*Flow through the load, m³/sec*
Q_s	*Supply flow rate from power unit, m³/sec*
Q_1, Q_2	*Flow through proportional valve*
V_{o1},V_{o2}	*Initial volume of chambers, m³*
V_t	*Total hydraulic oil volume in cylinder,m³*
V_1, V_2	*Forward and return chamber volume, m³*
ζ	*Damping coefficient*
w_m	*Mechanical natural frequency, Rad/sec*
w_n	*Hydraulic natural frequency, Rad/sec*
w_1	*Break frequency of sensing chamber, Rad/sec*
w_3	*Break frequency of main volume, Rad/sec*
δ_o	*Damping ratio of pipe*

Abbreviations

DAQ	*Data acquisition*
EHSM	*Electro-hydraulic servo mechanism*
HSV	*High-speed valve*
PID	*Proportional integral derivative*
P-Q	*Pressure-flow*
PWM	*Pulse width modulation*
PV	*Proportional valve*

TF *Transfer function*
VVVF *Variable voltage variable frequency*

APPENDIX – A

Theoretical Analysis

Proportional valve control

$$Q_l - Q_{l1} = \Delta Q_l = \frac{\partial Q_l}{\partial X_v}|_t \Delta x_v + \frac{\partial Q_l}{\partial P_l}|_t \Delta P_l \quad \ldots (1)$$

$$Kq = \frac{\partial Q_l}{\partial X_v} \quad \ldots\ldots (2)$$

$$Kc = -\frac{\partial Q_l}{\partial P_l} \quad \ldots\ldots (3)$$

Where Kq is the flow gain and Kc is the flow pressure coefficient. Now, substituting Eqs. (2) and (3) in (1) will give:

$$\Delta Q_l = Kq\Delta x_v - Kc\Delta P_l \quad \ldots.. (4)$$

$$Q_l = Kqx_v - KcP_l \quad \ldots.. (5)$$

Double acting single rod cylinder

$$Q_l = Aav\frac{dx_p}{dt} + C_{ip}(P_l) + \frac{V_t}{4\beta}\frac{d}{dt}(P_l) \quad \ldots\ldots\ldots (6)$$

$$\frac{dx_p}{dt} = \dot{x_p}\frac{d}{dt}(P_l) = \dot{P_l} \quad \ldots\ldots\ldots. (7)$$

$$Q_l = Aav.\dot{x_p} + C_{ip}(P_l) + \frac{V_t}{4\beta}\dot{P_l} \quad \ldots\ldots\ldots. (8)$$

Dynamic equations

$$\frac{V_p}{x_v} = \frac{Kq}{[\frac{V_t}{4\beta}\frac{M_t}{Aav.}\dot{V_p} + (\frac{M_t}{Aav.}(C_{ip} + Kc))V_p + Aav.]} \quad \ldots. (9)$$

Therefore, the second order transfer function equation will be:

$$\frac{x_o}{x_i} = \frac{K}{\frac{1}{w_n^2}D^2 + \frac{2\xi}{w_n}D + 1} \quad \ldots\ldots\ldots\ldots..(10)$$

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

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

Now, by comparing Eq. (9) with the second order Eq. (10), the hydraulic natural frequency and damping ratio could be found as follows:

3- The hydraulic natural frequency (w_n) is:

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

$$w_n = 2Aav.\sqrt{\frac{\beta}{v_{tM_t}}} \quad \ldots\ldots\ldots..(11)$$

4- The damping ratio (ξ) is:

$$\xi = \frac{M_t(C_{ip} + Kc)}{Aav.\sqrt{\frac{M_tV_t}{\beta}}} = \frac{C_{ip} + Kc}{Aav.}\sqrt{\frac{M_t\beta}{V_t}} \quad \ldots\ldots(12)$$

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

Now, substituting Eqs. (11) and (12) into (10) yields the transfer function of valve-controlled cylinder:

$$G_s = \frac{V_p}{x_v} = \frac{\frac{Kq}{Aav.}}{S(\frac{1}{w_n^2}S^2 + \frac{2\xi}{w_n}S + 1)} \quad \dots\dots..(13)$$

The pump modeling

$$Q_p = D_pN \quad \dots\dots. (16)$$

$$Q_p - C_lP_s - Q_l = \frac{V_t}{\beta_e}sP_s \quad \dots\dots.. (17)$$

$$Q_p - Q_l = C_l\left[1 + \left(\frac{V_t}{C_l\beta_e}\right)S\right]P_s \quad \dots\dots. (18) \qquad Q_p - Q_l = C_l\left[1 + \left(\frac{S}{w_p}\right)\right]P_s$$

……. (19)

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