



## Experimental Analysis of Square Position on Variable Displacement Electrohydraulic Actuation System by Open Loop Control

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### ABSTRACT

Electrohydraulic actuation systems offer definitive position control and an energy-efficient solution. Such systems are widely used in mobile machinery, robotics, and various stationary systems. Achieving good control of actuator position of the variable displacement electrohydraulic actuation system by an open loop control is the objective of this study. For square position (reference position) control, amplitude is taken as 0.1 m, at 0.05, 0.15 and 0.25 Hertz of frequency. Square position control is accomplished with LabVIEW algorithm through the application of compact RIO controller having input and output module. Appropriate control of voltage supply is obtained, when response position and reference position show appropriate accuracy. A higher Pearson's correlation coefficient near to 1 and lower the Mean absolute error, Mean deviation of error and standard deviation of error represent the best response position. It is observed that highest value of correlation coefficient achieved at 0.05 Hertz of frequency for response R3. At a lower frequency, square position control is better with higher correlation coefficient and lowest values of errors.

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## 1. INTRODUCTION

Electro-hydraulic actuator systems are essential in the engineering sector due to their compactness, ability to produce extremely high forces with excellent control accuracy, and high power-to-weight ratio [1]. Because of these features, there is a lot of curiosity among engineers and researchers about how to regulate force and position in an electro-hydraulic (EHA) system. Nowadays, electrohydraulic systems have become common in a wide range of manufacturing machines, Aircraft, mining equipment, fatigue testing, and various forms of automation, particularly in the automotive industry. Many applications requirements are quick responses, linear motions, and precise placement of heavy objects and these requirements are fulfilled with application of appropriate controller in the electrohydraulic actuation systems. For vehicle suspensions, Sam et al. [2] and Alleyne and Hedrick [3] have used hydraulic systems to compensate the vibrations. Physical modelling of hydraulic system was used by Alleyne and Liu [4], Liu

and Alleyne [5] successfully predicted the dynamics, including the friction model. This modelling has been done for the purpose of electro-hydraulic system modeling. In certain situations, obtaining a model might be extremely difficult or even impossible due to its high level of complexity and nonlinearity. The modelling and controller design are more challenging because of the electro-hydraulic control system's inherent nature includes a variety of uncertainties, significant nonlinearities, and time-varying features. Some causes of nonlinearity in the electro-hydraulic control system are nonlinear flow and pressure behaviors, friction inside the actuator, trapped oils in actuator chambers, and compressibility of oil. These factors make system modelling and control even more challenging.

Closed-loop and open-loop control are the two different types of control techniques that are utilized for the electrohydraulic actuation system. Several closed-loop control strategies, including back-stepping control [6-9], advanced adaptive control [10-12], sliding mode control [13], PID control [14], intelligent control [15],

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and unknown dynamics estimator control [16, 17], have been developed for use in control application of electrohydraulic systems. By combining feedback and dynamics estimation, these approaches can achieve extremely high control accuracy. However, closed-loop control for hydraulic systems is enormously complex and costly because a closed-loop electrohydraulic actuation system must include servo valves, a controller, and a variety of sensors. As a result of its complexity with expensiveness, closed-loop control is not appropriate for extreme conditions such as mining sites, so open-loop control is preferred in these cases. The open-loop control for hydraulic systems is primarily comprised of throttling valves [18], sequential valves [19, 20], and synchronous motors [21]. As a result of its ease of use and high level of dependability, this type of control is well-suited for extremely challenging working conditions. Position control of the actuator in an open loop for sinusoidal demand was discussed by Kumar and Mandal [22], and according to the results, up to 0.25 Hertz can be controlled in an open loop; after that, advanced control design is required for higher frequencies. In the research conducted by Sazonov et al. [23-25], an additive method of control for turbomachinery systems was discussed in details for the purpose of controlling the jet mesh system. This method of control is applicable in robotics and unmanned aerial vehicle systems. Manring and Fales [26] discuss the comparison of a pump control scheme to a valve control scheme. Multiple actuation demands can be fulfilled by valve control schemes, while pump-controlled schemes are safer, more robust, and more energy efficient (no throttling loss), so pump-controlled schemes are more demandable. In the present work, the VDEHA system is used, which is also a pump-controlled scheme. Sadeghi et al. [27] developed micro-position control system for shape memory alloy actuator using particle swarm optimisation. Developed control system gives excellent accuracy, which is suitable for application in micro-positioning. Fateh and Sadeghijaleh [28] developed voltage control strategy for tracking control of electric robot. Further, concluded that voltage control approach is better than the torque control approach.

In this experimental work, square position signals are used as the reference position signals for controlling the position of the hydraulic actuator of the VDEHA system in an open loop control. Square-position control is performed at 0.05 Hertz, 0.15 Hertz and 0.25 Hertz, which has an amplitude of 0.1 m. Control algorithms are prepared in LabVIEW and processed with a compact RIO controller. Control accuracy and error in position control are evaluated. Remaining part of this paper is organized as: in section 2, the VDEHA system is described in detail; in section 3, mathematical modelling is discussed; in section 4, results of position control and position error are presented; associated terms such as correlation coefficient and error calculations are described; and in

section 5, conclusions of using square position signals are discussed.

## 2. VARIABLE DISPLACEMENT ELECTRO HYDRO-STATIC ACTUATION (VDEHA) SYSTEM

Figure 1 presents a simplified schematic diagram of the VDEHA system. It is comprised of a symmetric hydraulic actuator, and the solenoid valves ('a' and 'b') are responsible for controlling the displacement of the actuator. The actuator moves a total length of 0.2 m in its entirety. To determine where the actuator is positioned, a linear variable differential transducer, also known as an LVDT, is utilized as a position sensor.

Figure 2 presents the experimental configuration of the VDEHA system that was used in the study. Table 1 has an overview of the components that make up the experimental rig in further detail. To interface between the personal computer (11) (LabVIEW programme) and VDEHA system, a real-time controller (12) with an input and output module is employed. NI 9215 and NI 9263 are voltage input and voltage output module, respectively.

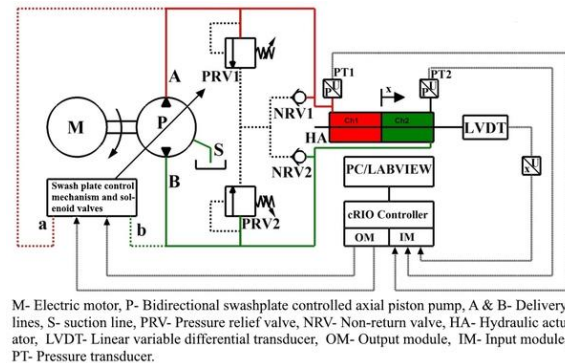
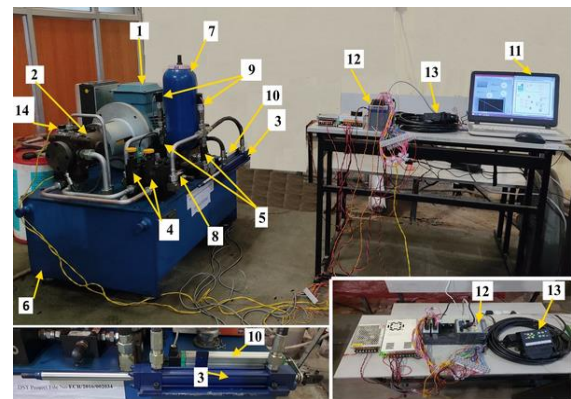


Figure 1. Schematic diagram of VDEHA system



(1) Electric motor, (2) Variable displacement axial piston pump, (3) Hydraulic actuator, (4) Pressure relief valves, (5) Non-return valves, (6) Oil tank, (7) Hydraulic accumulator, (8) Accumulator safety block, (9) Pressure transducers, (10) LVDT, (11) Personal computer, (12) Real-time controller, (13) Analogue amplifier, (14) Solenoid valves.

Figure 2. Experimental setup of VDEHA system

**TABLE 1.** Experimental components details

Label	Component	Remarks
1	Electric motor	3-phase Fixed rotational speed.
2	Variable displacement axial piston pump	Rexroth Flow = 119 L/min Displacement volume = 28CC/Rev Max pressure = 45 MPa
3	Hydraulic actuator	Symmetric Total displacement = 0.2 m
4	Pressure-relief valves	Max operating pressure = 25MPa Max flow = 100 L/min
5	Non-return valves	
6	Oil Tank	
7	Hydraulic accumulator	Volume = 6L Pre-charge pressure = 6MPa
8	Accumulator safety block	
9	Pressure transducers	Measuring pressure = 0-25MPa
10	LVDT	Measuring position = 0 - 0.2 m
11	Personal computer	
12	Real time controller	Controller – NI9030 with (IM and OM)
13	Analogue amplifier	Input = 10 to 32 V Output = 200 to 600 mA
14	Solenoid valves	Nominal resistance = 23 $\Omega$ (at 20 <sup>o</sup> C)

Both of module have voltage signal range of  $\pm 10V$  having sampling rate of 100kS/s/ch with resolution of 16-Bit. These modules are used with compact reconfigurable input output (c-RIO) system.

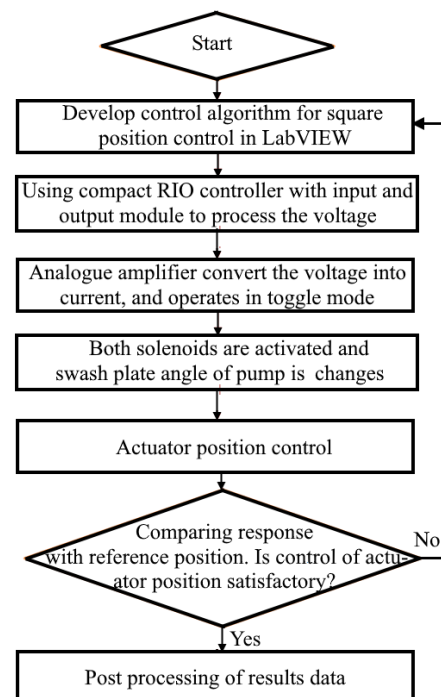
The response voltage signal from the LVDT (10), is received by the input module (NI 9215).

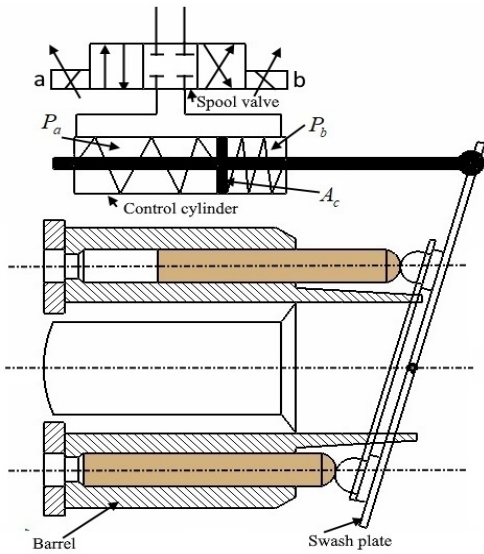
The output module (NI9263) sends the voltage signal to an analogue amplifier (13); it works in toggle mode of operation, and then electric current is supplied to solenoid valves (14). A variable-displacement axial piston pump (2) delivers the pressurized flow with the help of a fixed-speed electric motor (1). Pressure transducers (9) are used to measure the pressure in the actuator's (3) chambers. The hydraulic accumulator (7) is treated as a detached component, so it is not shown in the schematic diagram. The total displacement of the hydraulic actuator is 0.2 m, and its position is measured by an LVDT. Because of the LVDT is physically connected to the hydraulic actuator, the positions of the actuator and the LVDT are equivalent.

Controlling the flow of the pump is mostly dependent on the current that the amplifier generates. The flow rate and direction of actuator movement are based on the position of the spool valve (internal part) of the pump. An electric connection is established between a personal computer, an LVDT, a real-time controller, an analogue amplifier, and solenoid valves. To create a real-time control programme, a panel of control and simulation loops in LabVIEW software is used. LabVIEW is utilized in the manner of interfacing the hardware components of the controller, like the NI compact RIO controller, with software in the system to perform measurement and control actions. It has the ability to integrate hardware in a way that is unrivalled, and it comes with a large number of libraries that are already built in. These features make it ideal for performing sophisticated data processing during experimentation and simulation. The controlling action is performed in open loop control, which is based on the visualization of the response position with respect to the reference position. Several experiments are carried out, with the response and reference positions close to each other being considered. A flow chart of control methodology is shown in Figure 3.

### 3. MATHEMATICAL MODELING

Figure 4 shows the swash plate control mechanism and solenoid valves section of the schematic diagram (Figure 1) of the VDEHA system. The solenoid valves (14) link

**Figure 3.** Flow chart of control methodology



**Figure 4.** Swashplate control mechanism with solenoid spool valve

with the swash plate and control the pump's flow. Deflection in the swashplate either increases or decreases the displacement volume of the pump. The flow direction of the pump is determined by the direction of the deflection of the swashplate.

The control cylinder is the internal component of the pump of the VDEHA system.

In simplified form, equation of motion of control cylinder can be written as [29]:

$$m_c \ddot{x} = A_c (P_a - P_b) - F_f - F_s \tag{1}$$

where  $m_c$  is the mass of piston,  $A_c$  is the area of piston of control cylinder,  $P_a$  and  $P_b$  are chamber pressure of control cylinder,  $F_f$  is frictional force, and  $F_s$  is the self-adjusting force of controlling cylinder.

Pressure inside the chambers of control cylinder is expressed as [29]:

$$V_a \dot{P}_a = K(Q_a - A_c \dot{x} - k_{la} P_a - k_{li} (P_a - P_b)) \tag{2}$$

$$V_b \dot{P}_b = K(Q_b - A_c \dot{x} - k_{lb} P_b - k_{li} (P_b - P_a)) \tag{3}$$

where  $V_a$  and  $V_b$  are volume of chamber 'a' and 'b'.

$K$  is the bulk modulus of the oil and are the leakage through chamber 'a' and 'b', and  $k_{li}$  is the internal leakage.

Flow through the spool valve by assuming  $Q_a = Q_b$  is written as [29]:

$$Q_{sv} = Q_a = Q_b = B \cdot y \sqrt{(|P_s - \Delta P|) / 2 \text{sign}(P_s - \Delta P \text{sgn}(y))} \tag{4}$$

where  $\Delta P = P_a - P_b$ ,  $B$  is the spool valve constant,  $y$  is the spool position of the valve, and  $P_s$  is the supply pressure.

Solenoid valves allow the percentage of flow corresponding to current supply, it is expressed as:

$$Q_{sv\%} = i / 4 - 50, \text{ when } 200\text{mA} \leq i \leq 600\text{mA} \tag{5}$$

where,  $Q_{sv\%}$  is the flow through spool valve in percentage, and  $i$  is the current in mA.

An analogue amplifier is used to generate current from low voltage. In this work, the analogue amplifier operates in a toggle mode, so both solenoids operate independently. Only one solenoid is operational at any given time. Voltage and current relation are expressed as:

$$i = \begin{cases} 89.47 + 26.32 \cdot V_s, & \text{when } 4.2 \leq V_s \leq 8 \\ 0, & \text{when } 3.8 \leq V_s \leq 4.2 \\ 300 - 26.32 \cdot V_s, & \text{when } 0 \leq V_s \leq 3.8 \end{cases} \tag{6}$$

where  $V_s$  is the voltage supply to the solenoid valves.

When voltage belongs to 4.2 to 8 volts then solenoid 'a' is working, similarly for 0 to 3.8 volts, solenoid 'b' is working.

Equations (5) and (6) are the calibrated equations for the solenoid valves and analogue amplifier respectively. The behavior of actuator piston with respect to actuator position is expressed as:

$$\ddot{x}_{ap} = \frac{\pi}{4m_a} (d_{piston}^2 - d_{rod}^2) (P_A - P_B) \tag{7}$$

where  $x_{ap}$  is the actuator position,  $m_a$  is the mass of actuator,  $P_A$  and  $P_B$  are the chambers pressure of the actuator, and it is the same as the delivery pressure of the pump. To measure the pressure of chambers of actuator, pressure transducers are used.

Position of actuator is measured by LVDT, the calibration equation of the LVDT is written as:

$$x_{ap} = 0.04V_o + 0.004 \tag{8}$$

where  $V_o$  is the output voltage of LVDT.

Basically, in this work, voltage supply to the amplifier is controlled, in that voltage control actuating action of solenoid valves are performed, as a result actuator is controlled.

## 4. RESULT AND DISCUSSION

### 4. 1. Correlation Coefficient

In this work, Pearson's correlation coefficient [30] is used to determine the closeness of the response position of the actuator with the reference position of the actuator. There are three possibilities on which Pearson's correlation coefficient ( $r$ ) depends: (1) both the variables (reference

and response) are either simultaneously increasing or decreasing; (2) one variable is increasing, and another is decreasing; and (3) there is no relation between these two variables. For case 1, the correlation coefficient ( $r$ ) approaches a value of 1. For case 2, ' $r$ ' approaches -1, and for case 3, ' $r$ ' equals 0. In the current work, both variables either increase or decrease, so the ' $r$ ' value approaches 1. According to Jothivenkatachalam et al. [31], ' $r$ ' between 0.8 and 1.0 is highly significant, and ' $r$ ' between 0.6 and 0.8 is moderately significant.

The equation of correlation coefficient in term of reference and response position is expressed as:

$$r = \frac{\sum (x_{ap} - \bar{x}_{ap})(x_{apr} - \bar{x}_{apr})}{\sqrt{\sum (x_{ap} - \bar{x}_{ap})^2 \sum (x_{apr} - \bar{x}_{apr})^2}} \quad (9)$$

where  $r$  is correlation coefficient between  $x_{ap}$  and  $x_{apr}$ .

Within a sample,  $x_{ap}$  is the response position and  $x_{apr}$  is the reference position of actuator,  $\bar{x}_{ap}$  is the mean of the  $x_{ap}$ , and  $\bar{x}_{apr}$  is the mean of the  $x_{apr}$ .

#### 4. 2. Responses of Actuator Position

Closed-loop control for hydraulic systems is enormously complex and costly because it must include servo valves, a controller, and a variety of sensors. As a result of its complexity with expensiveness, closed-loop control is not appropriate for extreme conditions such as mining sites, so open-loop control is preferred in these cases. Further, when force is more essential than precision, open loop control is the preferred option. Because it is not always essential to precisely regulate the position of the actuator, open-loop control is employed in place of closed-loop control.

Numerous industrial applications have regular needs for different types of tracking requirements, such as square, sinusoidal, triangular, step, or their combination. So, application-oriented tracking requirements have been experimentally tested. In previous study of the VDEHA system by Kumar and Mandal [22], the position of the hydraulic actuator was controlled with sinusoidal demand signals in the range of 0.1 to 0.25 Hertz, and no other work was performed on this novel VDEHA system. In this experimental work, square position signals are used as the reference position signals for position control of the hydraulic actuator of the VDEHA system in an open loop control. Square-position control is performed under suggested range of frequency of 0.05, 0.15 and 0.25 Hertz, with amplitude of 0.1 m. Figures 5-7 show the results of the square position control. The equation for square position can be expressed as:

$$x_{apr} = \begin{cases} 0, & 0 \leq t < \frac{T}{4} \text{ and } \frac{3T}{4} < t \leq T \\ 1, & \frac{T}{4} \leq t \leq \frac{3T}{4} \end{cases} \quad (10)$$

where,  $T$  is the cycle time,  $T = 1/f$  and  $t$  is time in second, and  $f$  is frequency in Hertz.

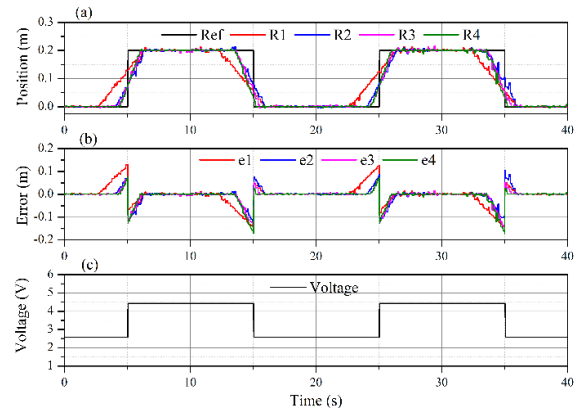
Equation (11) is used to determine the positional inaccuracy that is error ( $e$ ).

$$e = x_{apr} - x_{ap} \quad (11)$$

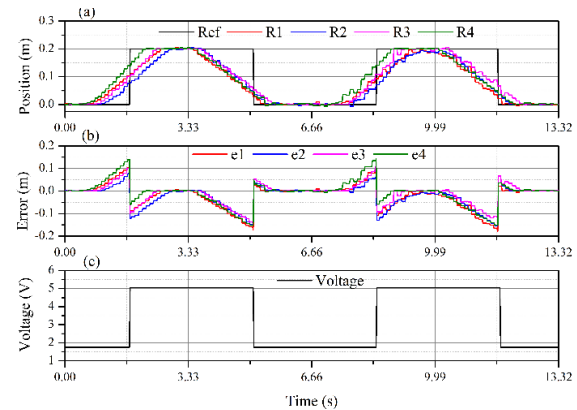
Figure 5 shows the responses of actuator (Figure 5(a)), and error in position (Figure 5(b)) at 0.05 Hertz of the frequency of the square position control. In this case, response R3 gives the highest value of the correlation coefficient ( $r = 0.957$ ), followed by other responses at the same frequency of 0.05 Hertz. The voltage supply pattern shown in Figure 5c corresponds to the better position response 'R3'.

All four variations in voltage corresponding to all four response positions are not shown because minor difference in voltage causes overlap in curve.

At 0.15 Hertz (Figure 6(a)), R3 response of the actuator position has the highest value of the correlation coefficient ( $r = 0.90$ ). Magnitude of error (Figure 6(b)) with respect to 0.05 Hertz of frequency increases in this case.



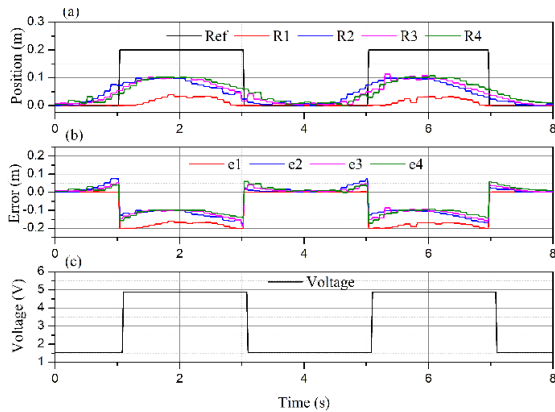
**Figure 5.** (a) Actuator position responses, (b) Error curve, and (c) Voltage supply, at 0.05 Hertz of frequency



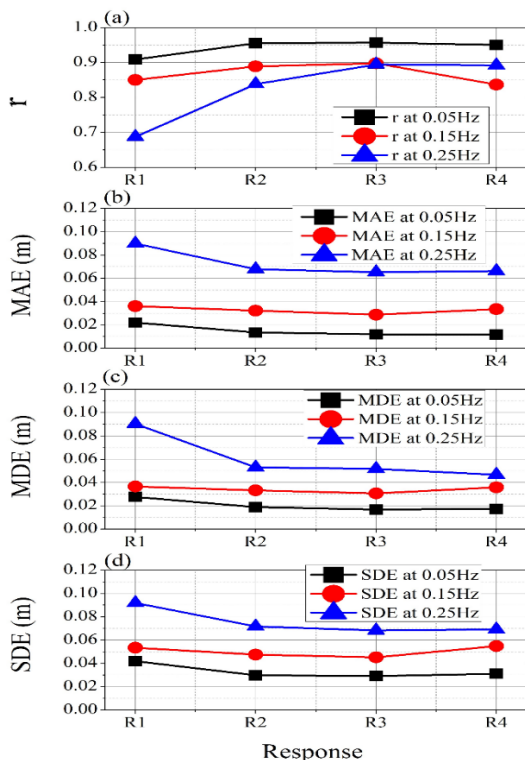
**Figure 6.** (a) Actuator position responses, (b) Error curve, and (c) Voltage supply, at 0.15 Hertz of frequency

At 0.25 Hertz (Figure 7(a)) of frequency in the square position, the actuator is unable to complete the total displacement of 0.2 m, and the magnitude of error (Figure 7(b)) is also higher than at 0.05 Hertz and 0.15 Hertz of frequency. At 0.25 Hertz, R4's response has a higher value of the correlation coefficient ( $r = 0.849$ ).

Figure 8(a) shows the correlation coefficient ( $r$ ) variation with respect to the responses (R1, R2, R3, and



**Figure 7.** (a) Actuator position responses, (b) Error curve, and (c) Voltage supply, at 0.25 Hertz of frequency



**Figure 8.** (a) Correlation coefficient ( $r$ ), (b) Mean absolute error (MAE), (c) Mean deviation of error (MDE), and (d) Standard deviation of error (SDE)

R4) at frequencies of 0.05 Hertz, 0.15 Hertz and 0.25 Hertz, and the result shows that at 0.05 Hertz of frequency, response R3 shows a higher value of " $r$ ." It means that the position response is very close to the reference position. In Figure 8(b), mean absolute error curves are shown; their minimum value is 0.011 m; similarly, MED (Figure 8(c)) and SDE (Figure 8(d)) also have values of 0.0195 m and 0.03 m, respectively, at 0.05 Hertz of frequency.

The variation in " $r$ " is minimum for 0.05 Hertz and maximum for 0.25 Hertz. It indicates that the accuracy of position control is better at 0.05 Hertz. MAE is maximum at 0.25 Hertz; variation in MAE between R1 and R2 is higher compared to variation of R2 with R3 and R4. In comparison of best-to-best " $r$ " value of responses, " $r$ " at 0.05 Hertz is 6.33% and 12.72% better than at 0.15 Hertz and 0.25 Hertz respectively. The variation in mean deviation of error (MDE) is at its minimum at 0.15 Hertz, while the minimum MDE is at 0.05 Hertz. MDE indicates how far the response signal is from the mean response signal. The minimum value of SDE indicates that the maximum number of response data is very close to its mean data, and the minimum SDE is observed at 0.05 Hertz of frequency.

### 5. CONCLUSIONS

The experimental results show that open loop control can perform satisfactorily under several square position controls with frequencies of 0.05, 0.15, and 0.25 Hertz and amplitudes of 0.1 m. In this study, the correlation coefficient assesses control accuracy between response and reference position curves. The best value (higher) of the correlation coefficient is achieved for lower frequencies, and as frequency increases, the value of the correlation coefficient gets reduced. In comparison of best-to-best response, the correlation coefficient at 0.05 Hertz is 6.33% and 12.72% better than that at 0.15 Hertz and 0.25 Hertz respectively. Mean absolute error (MAE), mean deviation of error (MDE), and standard deviation of error (SDE) are minimum for lower frequencies, and as frequency increases, their value also increases. From an application point of view, square position control can be used for low frequencies if used in open loop control. For higher frequencies, advanced control techniques like feedforward, fuzzy, and sliding mode control are suggested.

### 6. ACKNOWLEDGEMENT

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### Persian Abstract

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#### چکیده

سیستم‌های محرک الکتروهیدرولیک، کنترل موقعیت قطعی و یک راه‌حل کارآمد انرژی را ارائه می‌دهند. چنین سیستم‌هایی به طور گسترده در ماشین آلات متحرک، رباتیک و سیستم‌های ثابت مختلف استفاده می‌شود. دستیابی به کنترل خوب موقعیت محرک سیستم تحریک الکتروهیدرولیک با جابجایی متغیر توسط یک کنترل حلقه باز هدف این مطالعه است. برای کنترل موقعیت مربع (موقعیت مرجع)، دامنه ۰.۱ متر، در فرکانس ۰.۰۵، ۰.۱۵ و ۰.۲۵ هرتز در نظر گرفته می‌شود. کنترل موقعیت مربع با الگوریتم LabVIEW از طریق استفاده از کنترلر فشرده RIO با مازول ورودی و خروجی انجام می‌شود. هنگامی که موقعیت پاسخ و موقعیت مرجع دقت مناسب را نشان می‌دهند، کنترل مناسب منبع ولتاژ به دست می‌آید. ضریب همبستگی پیرسون نزدیک به ۱ بیشتر و میانگین خطای مطلق کمتر، میانگین انحراف خطا و انحراف استاندارد خطا نشان دهنده بهترین موقعیت پاسخ هستند. مشاهده شد که بیشترین مقدار ضریب همبستگی در فرکانس ۰.۰۵ هرتز برای پاسخ R3 به دست آمد. در فرکانس پایین تر، کنترل موقعیت مربع با ضریب همبستگی بالاتر و کمترین مقادیر خطا بهتر است.

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