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Comparative Analysis of a 4/3 Manual Lever and a 4/2 Electric Solenoid Hydraulic Directional Control Valve with a Double-Acting Actuator Using Regression Modeling

OSHUNDAIRO MUSHAFAU IDOWU[1], ANAIDHUNO UFUOMA PETER[2], IBRAHIM ADEBAYO ALAMUTU[3]
[1]*Department of Marine Engineering,*[2]*Department of Mechanical Engineering, Federal University of Petroleum Resources, Effurun, Delta State Nigeria.*
[3]*Department of Mechanical Engineering, Petroleum Training Institute, Effurun, Delta State, Nigeria.*

Abstract - This study investigates and compares the operational performance of a 4/3 Manual Lever Directional Control Valve (MLDCV) and a 4/2 Electric Solenoid Directional Control Valve (ESDCV) in a hydraulic system utilizing a double-acting actuator. The primary objective was to analyze the piston traveling speed, force characteristics and time under varying valve openings and to develop predictive regression models for each hydraulic directional control valve system. Experimental data revealed that while the MLDCV exhibited predictable piston extension speeds, its retraction speeds were erratic and inconsistent, largely attributed to fluid dynamic phenomena - turbulence, cavitation. The responses were fast to predict on the experiment. The regression model for MLDCV extension speed was highly significant ($R^2 = 0.9999$), but the model for retraction speed demonstrated a lower fit ($R^2 = 0.086$), making its coefficients unreliable. The ESDCV provided stable and predictable piston speeds for both extension and retraction, with strong and statistically significant regression models ($R^2 > 0.99$ for both) that reliably explained piston dynamics and exhibited lower prediction errors. This research underscores the superior control and predictability offered by the ESDCV in applications demanding consistent and precise performance. The developed regression models serve as valuable tools for predicting hydraulic system design and selecting the most appropriate directional control valve based on application requirements.

Key Words: Solenoid, Control, Valve, Actuator, Regression, Modeling

I. INTRODUCTION

Hydraulic systems are integral to numerous industrial applications for assurance of effective operation of modern industries and mobile machinery which critically dependent on fluid power systems mainly governed by adequate management of force and motion [1]. Central to this management are hydraulic control valves, components that regulate the fluid's path, pressure, and rate of flow to govern actuator performance [2].

At its most fundamental level, a hydraulic system functions by transmitting energy via a contained, non-compressible fluid, a concept derived from Pascal's law and aligned with Newton's laws of motion [3][4][5]. These systems are generally classified into two primary domains: hydrostatic and hydrodynamic. Hydrostatic applications, such as braking systems and hydraulic presses, leverage the pressure of a relatively stationary fluid to generate force [6]. Conversely, hydrodynamic systems, exemplified by turbines, derive power from the kinetic energy and flow of the fluid itself.

The application of hydraulic systems in modern automation technology is widespread. A basic distinction is made between stationary hydraulic and mobile hydraulic systems. Mobile hydraulic systems, such as those used on wheeled or tracked vehicles, often feature manually operated valves. In contrast, stationary hydraulics, common in manufacturing and industrial settings, predominantly utilizes solenoid valve [7]. Other specialized areas include marine, mining, and aircraft hydraulics, with aircraft systems demanding a special position due to the critical importance of safety measures.

The ubiquity of hydraulic technology stems from its significant benefits, including a high power-to-weight ratio, exceptional force generation, and versatile speed regulation. Its applications are extensive, ranging from heavy construction equipment like excavators to precision manufacturing tools and vital aerospace systems. Despite these strengths, hydraulic systems present certain challenges. Potential fluid leaks pose environmental and safety risks, the viscosity and performance of hydraulic fluids can be negatively affected by temperature fluctuations, and the initial investment in high-quality components can be substantial.

This investigation will closely examine the functional differences between the hydraulic 4/3 manual lever directional valve (MLDCV) and 4/2 electric solenoid directional control valve (ESDCV). [8][9]. While their purpose is identical, their method of actuation one relying on physical or pilot-pressure signals and the other on an electrical current creates distinct operational profiles. By systematically gathering data across various working conditions, a predictive regression model will be developed. This model will enable a detailed analysis aimed at predicting key performance metrics [10][11][12], ultimately providing clear, data-driven recommendations for selecting and implementing the most efficient control valve for specific engineering applications.

II. METHODOLOGY

This section provides a comprehensive description of the empirical methods used to compare the performance of a ¾ manual lever directional control valve (MLDCV) against a 4/2 electric solenoid directional control valve (ESDCV). The methodology covers the experimental apparatus, and procedural operation of data acquisition.

2.1 Experimental Apparatus and Components

Laboratory hydraulic test equipment was set up as shown in schematic diagrams in Figure 2 and Figure 3 to simulate a fundamental hydraulic linear actuation circuit.

Hydraulic Power Unit (D): The system is a stationary unit consists of a hydraulic Pump driven by a 2 HP, single-phase AC electric motor. This pump was rated for a constant flow rate of 10 liters per minute at a maximum pressure of 120 bar. Gear pump are chosen for their reliability and consistent fluid flow, which is required for repeatable experiments. The units include a 5-liter base reservoir to hold the hydraulic fluid, a suction strainer, and a return line filter (10-micron) to maintain fluid cleanliness. A system pressure relief valve was installed to prevent over-pressurization and ensure safe operation.

Actuator (A): An actuator labeled (A) in the schematic diagrams (Figure 1and Figure 2) is a standard industrial-grade, double-acting hydraulic cylinder was used as the load actuator. It had a bore diameter of 31mm, a piston rod diameter of 14 mm, and a full stroke length of 194 mm. A double-acting cylinder was chosen because it allows for powered motion in both extension and retraction.

Directional Control Valves (B): the directional control valves for both 4/3 Manual/Hydraulic-Valve and 4/2 Electric solenoid directional control valves are labeled (B) in the Figure (2)and Figure (3) .

i. Manual Lever Valve: This is a 4-way, 3-position (4/3) directional control valve with a tandem-center spool configuration. Actuation was achieved via a manual lever. In the center position, the tandem-center design unloads the pump by connecting the pressure port to the tank port, reducing heat generation and energy loss.
ii. Electric Solenoid Valve: This is a 4-way, 2-position (4/2) directional control valve with specifications identical to the manual valve. Actuation was controlled by two 24V DC solenoids. Energizing one solenoid shifted the spool to direct flow for piston extension, while energizing the other directed flow for retraction with the aid of a switch mounted on the electric panel labeled (E) in Figure 3.

Flow Control Valve (C): A 1-way flow valve with calibrated flow control, labeled C in Figure 2 and Figure 3, was mounted to vary the fluid flow to the actuator and is micrometrically calibrated.
Hose and Fittings: 10mm hoses are used to connect and link the components to create the hydraulic circuit conveying the hydraulic fluid through the system..

Instrumentation: Pressure gauges labeled PP (pump pressure) and PS (pressure supplied) in Figure 2 and Figure 3 were installed. Analog pressure gauges were used to measure the tank pressure and output pressure across the full scale, allowing for precise measurement of the pressure required to move the piston in and out of the actuator.

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Time and Distance Measurement: A digital stopwatch (±0.01s accuracy) was used for manual time recording, and a meter rule was used for measuring the piston displacement traveling distance.

2.2 Principle of Operation and Experimental Procedure

The hydraulic system operates based on Pascal's Law, which states that pressure applied to an enclosed, incompressible fluid is transmitted undiminished to every portion of the fluid and the walls of the containing vessel [3].The electric motor drives the hydraulic pump (D), which draws hydraulic fluid from the reservoir and forces it into the system, generating flow through the one-way flow valve (C).The flow is directed to the directional control valve DCV (B). When the valve is actuated (either by the manual lever or an energized solenoid), it shifts its internal spool, opening a path for the pressurized fluid to travel to the actuator, the hydraulic cylinder (A).The pressurized fluid enters the cylinder on one side of the piston, creating a force equal to the pressure multiplied by the piston's surface area. This force overcomes any resistance (load) and causes the piston to move forward. As the piston extends, fluid on the opposite side of the piston is pushed out and directed back to the reservoir through the return path in the directional control valve. When the DCV is return by the manual lever or re-energized by the solenoid control the fluid filled second side of the actuator cylinder to retract the piston inwardly as shown in Figure 2 and 3. The fluid flow was varied by the micrometer calibration turning of the one-way flow valve (C) from 3mm to 9mm. The corresponding out put pressure was recorded, while the piston traveling distance and time taken were recorded respectively. The experimental set up for the electrically operated require panel board (E) with banana cables for electrical switch control connection to energize the electrically controlled solenoid DCV.

2.3 Variables calculation

Piston Output Force F_{po} : This is the force required to overcome the load resistance to push out the piston from the cylinder as a result of the applied pressure, mathematically represented using Equation (1)

$$\mathrm{F_{po}} = \mathrm{P_{po}} \times \mathrm{A_b} \quad (1)$$

Where $\mathrm{F_{po}}, \mathrm{P_{po}}$ and $\mathrm{A_b}$ are piston output forces, piston out put pressure, and bore area respectively.
Piston Retraction Force $\mathrm{P_{fr}}$: This Is the force required to overcome the load resistance to retract the piston in the cylinder as a result pressure applied, mathematically represent using Equation 2

$$\mathrm{F_{pr}} = \mathrm{P_{pr}} \times \mathrm{A_a} \quad (2)$$

Where $\mathrm{F_{pr}}, \mathrm{P_{pr}}$ and $\mathrm{A_a}$ are piston retraction force, piston retraction pressure, and annular area respectively.
Annulus Area $\mathrm{A_a}$: This is the differential area between the cylinder bore and the area of the piston rod, calculated using Equation 3

$$\mathrm{A_a} = \mathrm{A_b} - \mathrm{A_p} \quad (3)$$

where $\mathrm{A_a}, \mathrm{A_b}$ and $\mathrm{A_p}$ are annulus area, area of cylinder bore and piston area respectively. The areas $\mathrm{A_{a,b,p}}$, are determine using $\frac{\pi d^2}{4}$ and schematically describe in Figure 1

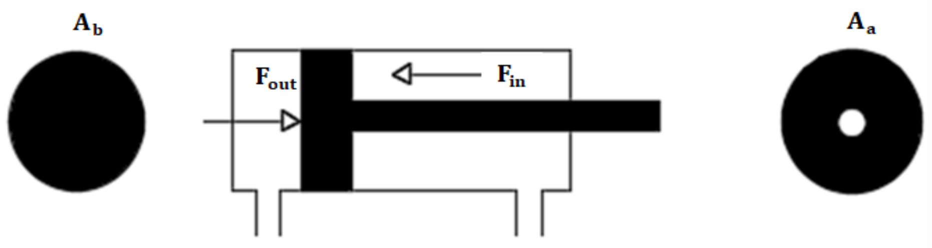

Figure 1: schematic diagram of area and applied force

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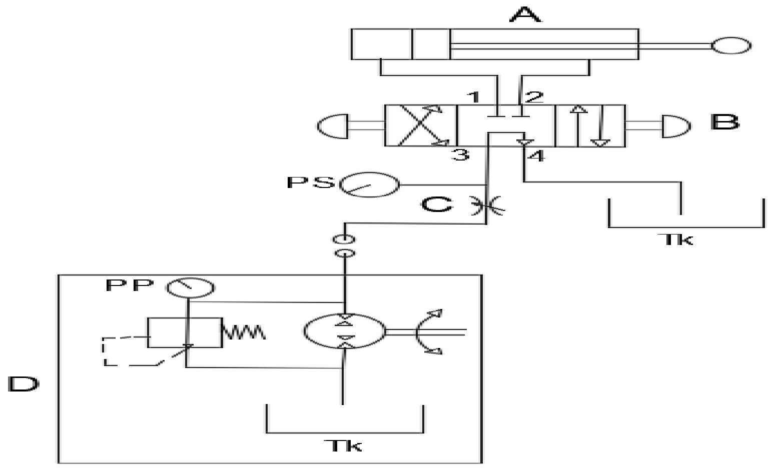

Figure 2: Schematic hydraulic set up with the use of ¾ Manual lever Directional Control valve (MLDCV).

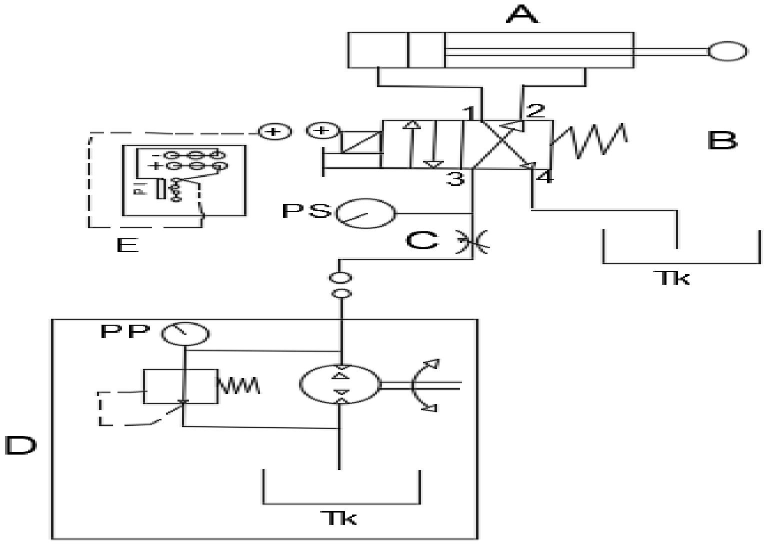

Figure 3: Schematic hydraulic set up with the use of ¾ Electric Solenoid Directional Control valve (ESDCV).

III. RESULTS AND DISCUSSION

This section presents experimental results, recorded data, and analysis for MLDCV and ESDCV hydraulic setup. The analysis of the results follows the flow chart shown in Figure 3.

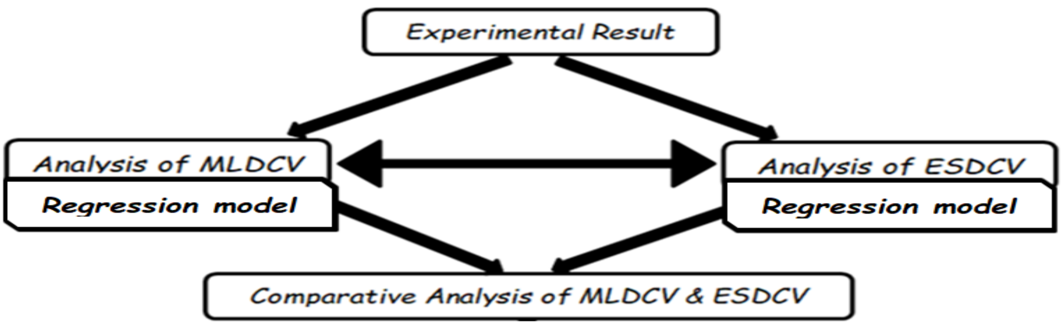

Figure 3: Result analysis flow chart

3.1 Experimental Results for MLDCV

The experimental data and calculated variables are recorded in Table 1 for the MLDCV. The responses, such as piston traveling speed and the required force for the inward and outward movement of the piston, are analyzed against the variation of the micrometric valve opening.

Table 1: Data presentation of 4/3 Manual lever Directional Control valve (MLDCV)

Valve Opening (mm)	Piston out pressure (N/mm^2)	Piston in pressure (N/mm^2)	Piston Traveling Time (sec)		Piston Area (mm^2)	Bore Area (mm^2)	Annulus Area (mm^2)	Piston Force-out (N)	Piston Force2-in (N)	Piston Distance (mm)	Piston Travelling speed-out (mm/s)	Piston Traveling speed-in (mm/s)
			Out	in								
3	0.5	0.9	3.15	3.2	38.50	188.78	150.27	94.4	135.2	194	61.6	60.6
4	0.5	1.0	3.14	3.73	38.50	188.78	150.27	94.4	150.3	194	61.8	52.0
5	0.6	1.0	3.03	3.22	38.50	188.78	150.27	113.3	150.3	194	64.0	60.2
6	0.6	1.0	3.04	3.15	38.50	188.78	150.27	113.3	150.3	194	63.8	61.6
7	0.6	1.0	3	3.4	38.50	188.78	150.27	113.3	150.3	194	64.7	57.1
8	0.55	1.0	2.8	3.2	38.50	188.78	150.27	103.8	150.3	194	69.3	60.6
9	0.5	0.9	2.8	3.34	38.50	188.78	150.27	94.4	135.2	194	69.3	58.1

3.1.1 Relationship between the Valve Opening and Piston Traveling Speed using MLDCV.

The first graph, labeled Graph A in Figure 4, shows the speed of the piston as it extends (moves out). The piston's extension speed generally increases as the gate valve opening increases from 3 mm to 8 mm. The speed starts at approximately 61.5 mm/s at a 3 mm valve opening and rises to 69 mm/s at an 8 mm valve opening, leveling off until the 9 mm opening. This relationship is expected, as a wider valve opening allows for a higher flow rate of hydraulic fluid, causing faster movement of the piston [13]. This implies a direct relationship between the gate valve opening and the piston's outward traveling speed.

The second graph (Graph B) shows that the piston's retraction speed is highly erratic and unpredictable when varying the valve opening. The speed fluctuates significantly, dropping from over 60 mm/s to 52 mm/s at a 4 mm opening and peaking at 61 mm/s at a 6 mm opening. This indicates that there is no apparent and consistent relationship between the valve opening and the retraction piston speed in this study. This inconsistency is likely a result of complex fluid dynamic phenomena such as turbulence, cavitation, friction, and load characteristics [14] [15], which cause the piston to move inward in a jerky manner.

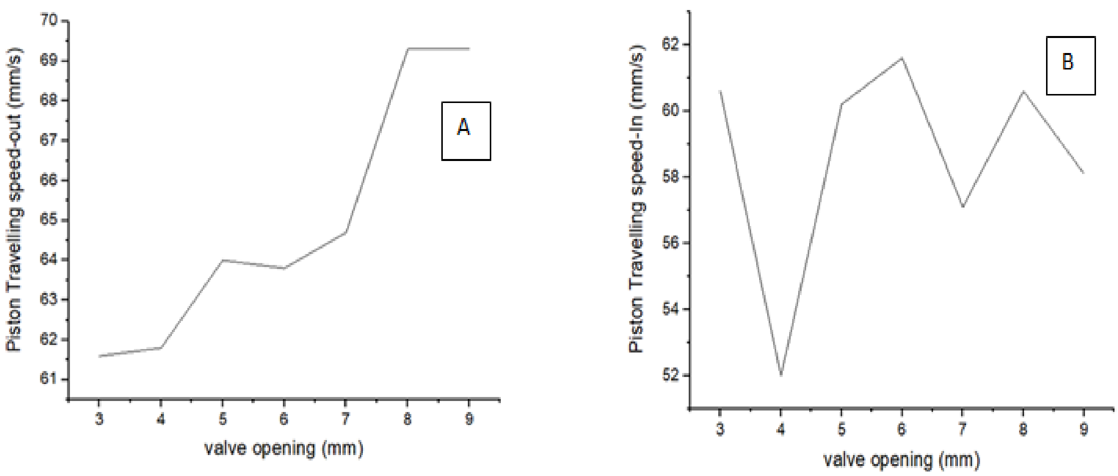

Figure 4: Piston Traveling speed against Valve Opening using MLDCV

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3.1.2 Relationship between Valve Opening and Piston Traveling Force using MLDCV.

Graph C in Figure 5 shows the force exerted by the piston during the extension stage. The trends of the extension and retraction force graphs are similar, where the force remains relatively constant across a range of valve openings. The force rapidly increases to about 113 N at a 5 mm opening and remains constant until the valve opening reaches 8 mm. This implies that the piston's outward traveling force is primarily determined by the load [17], which is lower in the outward movement (113 N) than in the inward movement (150 N) shown in Graph D.

Graph D in Figure 5 displays the retraction force exerted by the piston during its inward movement. The force remains constant across a wide range of valve openings. The force shows a quick rise to approximately 150 N as the valve opens to 4 mm and drops below 136 N at a 9 mm valve opening. This indicates that the force required for the piston's inward movement depends on overcoming the load working against its direction, rather than being solely dependent on the fluid flow rate regulated by the valve opening [18]. At this operating range (4–6 mm), the force is independent of the piston's speed.

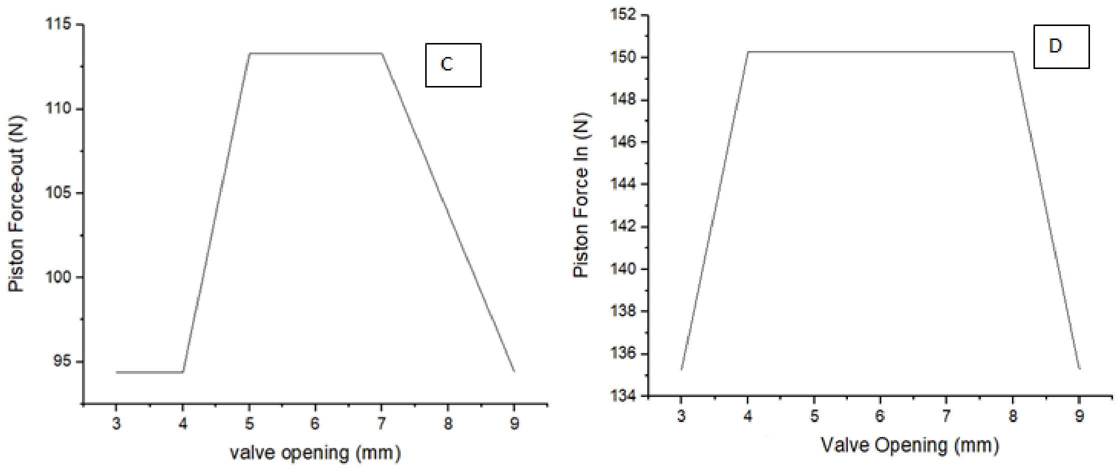

Figure 5: Piston Traveling speed against Valve Opening using MLDCV

3.2 Experimental Results for ESDCV

The experimental data and calculated variables are recorded in Table 2 for the ESDCV. The responses, such as piston traveling speed and the required force for the inward and outward movement of the piston, are analyzed against the variation of the micrometric valve opening.

Table 2: Data presentation of 4/2 Electric Solenoid Directional Control valve (ESDCV)

Valve Opening (mm)	Piston out Pressure (N/mm²)	Piston in pressure (N/mm²)	Piston Traveling Time (sec)		Piston Traveling Area mm²	Bore Area (mm²)	Annular Area (mm²)	piston Force1-out(N)	piston Force2 in- (N)	Piston Distance (mm)	Piston Travelling Speed-out (mm/s)	Piston Traveling Speed- in (mm/s)
			out	in								
3	0.00	0.4	144.2	38.50	38.50	188.78	150.27	0.0	60.1	194	0.9	1.3
4	0.05	0.4	142.6	38.50	38.50	188.78	150.27	9.4	60.1	194	1.0	1.4
5	0.10	0.5	5.7	38.50	38.50	188.78	150.27	18.9	75.1	194	25.9	34.0
6	0.25	0.7	3.77	38.50	38.50	188.78	150.27	47.2	105.2	194	43.1	51.5
7	0.50	0.8	3.2	38.50	38.50	188.78	150.27	94.4	120.2	194	56.7	60.6
8	0.50	0.8	3.1	38.50	38.50	188.78	150.27	94.4	120.2	194	62.6	62.6
9	0.50	0.8	3.1	38.50	38.50	188.78	150.27	94.4	120.2	194	62.6	62.6

3.2.1 Relationship between the Valve Opening and Piston Traveling Speed using ESDCV.

Graph E shows a steady position of the piston's extension speed near zero until the valve opens to 4 mm. It then rises smoothly to a maximum speed of 65 mm/s at a 7.5 mm valve opening. At this stage, a further opening of the valve does not significantly affect the speed, as shown in Figure 6.

Graph F displays the retraction speed of the piston, which follows a very similar pattern to Graph E. The speed is negligible until the valve opens to 4 mm, then increases smoothly and predictably to a maximum of 68 mm/s at an 8 mm valve opening, as shown in Figure 6. This research generally notes that the extension and retraction stroke speeds are well-controlled by the gate valve. The use of an electric solenoid directional control valve appears to be more stable and predictable.

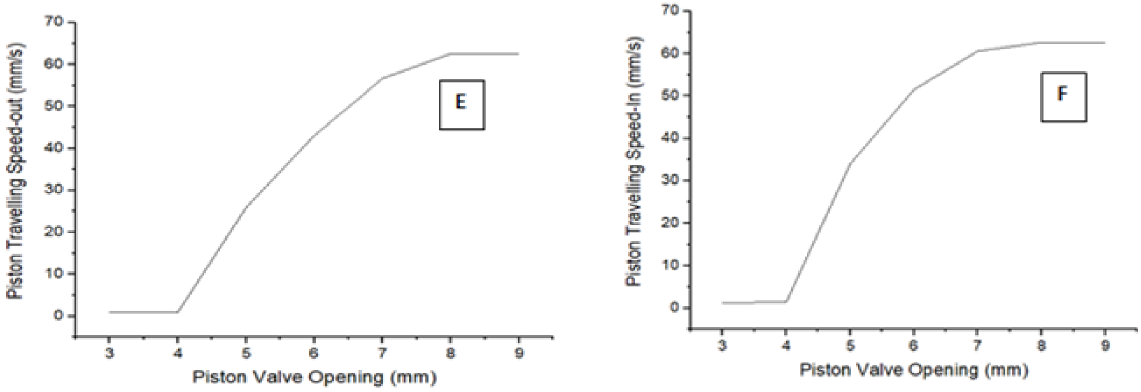

Figure 6: Piston Traveling Against the Valve Opening Using ESDCV

3.2.2 Relationship between Valve Opening and Piston Traveling Force using ESDCV.

Graph G reveals that the force generated by the piston increases as the valve opening increases, eventually leveling off. The force starts near zero and rises in a curved pattern as the valve opens, reaching the maximum piston force of 98 N at a 7 mm valve opening, as shown in Figure 7. This indicates a direct relationship between the valve opening (i.e., the flow rate/pressure) and the output force, up to the system's maximum capability for the load.

Graph H shows the retraction force increasing as the valve opening increases, rising to a maximum of approximately 120 N at a 7 mm valve opening. The retraction force also depends on the valve opening [19]. The maximum retraction force (120 N) is higher than the maximum extension force (98 N), as shown in Figure 7.

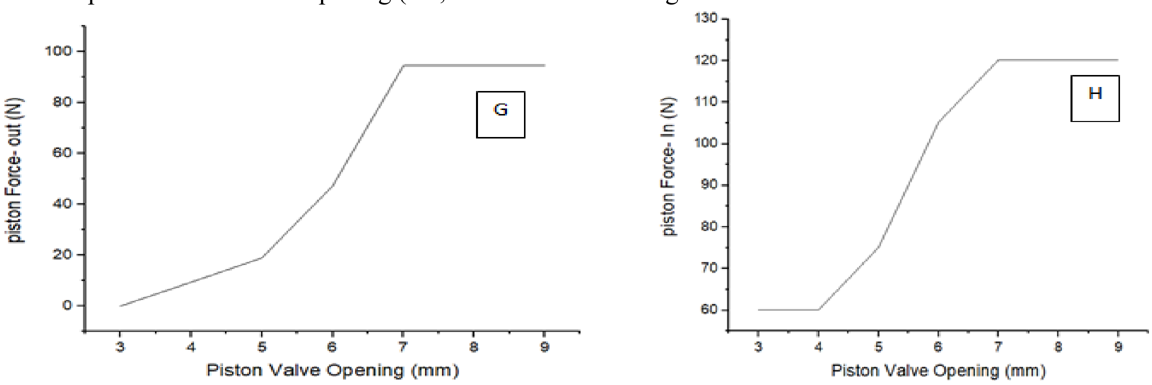

Figure 7: Piston Traveling speed against Valve Opening using ESDCV

3.3 Actuator Piston Traveling Speed Analysis with 4/3 Manual Lever Direction Control Valve Using Regression Model

3.3.1 Statistical Analysis of the Hydraulic System with MLDCV

Piston Traveling Speed (Outward Movement)

The statistical analysis for the piston's outward movement (extension) shows a highly effective regression model. The coefficient of determination (R^2) is 0.9999, indicating a very strong positive linear relationship, which implies that the model fits the data almost perfectly in terms of correlation. The ANOVA results reveal that the P-value ($1.07x10^{-8}$) is significantly less than the F-value, strongly implying that the predictors have a meaningful and reliable relationship with the outcome as shown in Table 3.

Piston Traveling Speed (Inward Movement)

In contrast, the statistical result for the piston's inward movement (retraction) demonstrates a poor fit and no statistical significance, as shown in Table 3. The R^2 value is extremely low at 0.0860, and the P-value is very high at 0.8354. This confirms that the model poorly fits the data, which is consistent with the observed erratic behavior of the retraction speed (Figure 4b) due to fluid turbulence, cavitation, friction, and load characteristics due to fast response to piston movement in that direction [14] [15].

Table 3 Regression Statistic Result with MLDCV

Piston Traveling Speed	Coefficient of Determinant	Adjusted R Square	Multiple R	Standard Error	P-Value	F-value	Sample
out	0.9999	0.9999	0.9999	0.0398	1.07x10-8	19347.6	7
In	0.0860	0.3710	0.2933	3.8806	0.8354	0.1882	7

3.3.2 Regression Model Equation of 4/3 Manual Lever Direction Control Valve

The models are mathematical structure as linear regression equations, revealing the piston traveling speed as the depended variable and the independent variable as force due to fluid pressure and piston traveling time for both inward and outward movement of the piston as expressed in Eq 4 and 5 .

$$\text{Piston Traveling speed (outward)} = 132.4098 - 0.01044\text{F}_{out} - 22.1771\text{t}_{out} \qquad (4)$$

The model Eq 4 shows a high level of statistical significance, and its excellent fit R^2 (0.9999) allows the coefficients to be interpreted with high confidence. The positive intercept term (132.4) represents the theoretical baseline piston traveling speed (outward) under ideal and unconstrained conditions. The negative coefficient for piston force (-0.01044) indicates an inverse relationship between the piston force parameter and the outward traveling speed. The negative coefficient for time (-22.1771) has a large magnitude, indicating that the time term is the dominant factor in determining the piston's outward traveling speed.

$$\text{Piston Traveling speed (Inward)} = 84.53766 - 0.059406\text{F}_{in} - 5.764286\text{t}_{in} \qquad (5)$$

The model eq 4 ,is critical to note that this model was not statistically significant as P- value of 0.835 and shows a very poor fit with R2 of 0.086 therefore the mathematical coefficients cannot be reliably interpreted as real physical relationships as revealed in Table 3.

3.4 Actuator Piston Traveling Speed Analysis with 4/2 Electric Solenoid Direction Control Valve Using Regression Model

3.4.1 Statistical Analysis of the Hydraulic System with ESDCV

Piston traveling speed inward and outward with ESDCV

The statistical analysis presents the linear regression model to predict the traveling speed of a hydraulic piston under ESDCV at two different condition ‘out and in’ movement of the piston from the actuating cylinder as shown in Table 4. The coefficient of determinant R2 value for both the piston traveling speed ‘out and in’ are 0.9903 and 0.9992 respectively, which means that over 90% of the variation of the piston speed can be explained by the variables force and time, also shows a very strong and reliable predictive relationship [19]. The P-values from ANOVA for both models out ($9.42\text{x}10^{-5}$) and in ($6.48\text{x}10^{-7}$) is far lower than 0.05, this indicate that the results are highly statistically significant [20], as the F-values of 204.116 for out-speed and 2482.16 for in-speed further confirm the overall significance of the models. A high F-value means that the variables in the models are reliably predict the piston speed. The adjusted R2 values of the speed-out and in (0.9855 and 0.9988) are very close to their coefficient of determinant values, this implies a good sign and the model are not over fitted . The standard error determines the typical distance between the observed values and regression line. The speed-in model has a lower error (0.09616) than the speed-out model (3.2866), suggesting that the predictions for the retraction (Piston traveling speed-in) are closer to average, more precise than for extension (Piston traveling speed-out).

Table 4 Regression Statistic Result with ESDCV

Piston Traveling speed	Coefficient of Determinant	Adjusted R^2	Multiple R	Standard Error	P-Value	F-value	Sample
out	0.9903	0.9855	0.9951	3.2866	9.42x10-5	204.1166	7
In	0.9992	0.9988	0.9996	0.9616	6.48 x10-7	2482.16	7

3.4.2 Regression Model Equation of 4/2 Electric Solenoid Direction Control Valve

The mathematical model of linear regression equations, revealing the piston traveling speed as the depended variable and the independent variables as force due to fluid pressure and piston traveling time for both inward and outward movement of the piston as expressed in Equation 5 and 6.

$$\text{Piston Traveling speed} = 20.49958 + 0.43140\text{F}_{out} - 0.10564\text{t}_{out} \quad (5)$$

The model Eq 6 suggests that the speed of the piston as it moves out is great influenced two factors according to this research work which force-out and the negative time-out. The equation further shows that the speed increases as the force increases while the time decreases with a positive coefficient of the intercept [21].

$$\text{Piston Traveling speed} = -11.2808 + 0.611578\text{F}_{in} - 0.16825\text{t}_{in} \quad (6)$$

The retraction speed (piston traveling speed-in) similarly predicted by the terms Force-in and time-in, the positive term force if increases the piston traveling speed increases, while the time reduces.

This research successfully conducted a comprehensive comparative analysis of the operational characteristics and developed predictive regression models for 4/3 Manual Lever Directional Control Valves (MLDCV) and 4/2 Electric Solenoid Directional Control Valves (ESDCV) within a double-acting hydraulic actuation system. The findings reveal critical distinctions in performance and predictability between the two Hydraulic Directional Control valve types, which are vital for informed hydraulic system design and component selection.

The MLDCV exhibited consistent piston extension speeds (piston traveling speed-out), allowing for a highly robust regression model to predict this motion effectively. However, its piston retraction speeds (piston traveling speed-In), were found to be highly erratic and unpredictable, a characteristic attributed to complex fluid dynamics including turbulence, cavitation, and friction. This inherent variability rendered the regression model for MLDCV retraction statistically insignificant and unreliable for practical application.

Conversely, the ESDCV demonstrated significantly more stable and predictable piston traveling speeds for both extension and retraction stages. Its controlled electric actuation mechanism enabled a more consistent response across varying valve openings. The regression models developed for the ESDCV were highly robust and statistically significant for both piston movements, with high coefficients of determinant (R^2 values exceeding 0.99) and lower prediction errors, confirming their superior predictive power and reliability in characterizing piston speed dynamics. This indicates that electric solenoid control offers a more precise and controllable hydraulic system.

IV. CONCLUSION

This study provides a valuable data-driven and quantitative comparison of two fundamental directional control valve types, offering empirical evidence of their distinct operational performance. It quantitatively establishes that while MLDCVs may be suitable for less precision-demanding applications, ESDCVs offer superior stability, predictability, and control, which are essential for modern automation and precision engineering. The development of robust and statistically significant regression models for the ESDCV, alongside the clear identification and explanation of the MLDCV's limitations in specific operating regimes, represents a significant contribution. These findings provide engineers and researchers with critical insights and analytical tools necessary for making informed decisions regarding prediction of hydraulic component selection, system design, and performance to develop more efficient, reliable, and precise hydraulic systems.

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