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Experimental Study of Leakage Compensation for Actuator Speed Control in Electro-Hydraulic Systems

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***Abstract*— Hydraulic actuators are widely used on mobile equipment and robots, due to their good capability in positioning especially in servo system, fast and smooth response, high power density, environment tolerance, and compact size. Using actuators for constant speed operation hydraulic systems is highly required in many applications such as steel sheet rolling, pressing operations, production, oil and gas subsea applications assembly lines, robotics, aircrafts equipment and submarine systems. However, accepted positioning in the applications requires an accurate electro-hydraulic actuator facing several problems such as (internal and external leakage, backpressure, and load).**

A variable-pump and/or valve-controlled may be used to control the actuator velocity in a constant speed operation. The valve control may be done by using a proportional directional valve while the variable-pump may be used either by a variable displacement or variable speed pump. They may be used to satisfy the actuator flow-demand due to any system disturbance or system leakage.

An experimental test rig has been carried out, equipped with variable displacement vane pump and 4/3 Electro-hydraulic proportional direction control valve EHPDV with its proportional amplifier, to investigate the hydraulic system performance and to measure the operating parameters to evaluate the system dynamic and steady state characteristics of the studied hydraulic system.

The measured parameters have been directly connected to the data acquisition system to store the measured parameters. A LABVIEW program has been constructed to read the measured value from the sensors; displacement and pressure.
Investigations have been carried out using both methods; pump control and valve control to decide which one is more effective in a certain operation conditions and pressure ranges in the experimental work.

Results for different directional proportional valve setting and the variable vane pump speed have been illustrated to show their effect on the dynamic and steady state response of hydraulic cylinder. Intentionally, different system external leakages have been introduced to show its effect on the system performance. The leakage compensation has been shown to be achieved by either controlling the proportional electro-hydraulic valve input volt or controlling the pump speed through frequency modulator.

Keywords*— Leakage Compensation; Constant Speed Actuator.

I. INTRODUCTION

Hydraulic actuators are widely used in mobile equipment, industry and robots, due to their good capability in positioning and affording the required speed especially with servo systems which are characterized with fast and smooth responses, high power density, environment tolerance, and compact size. System leakage problems include internal and external leakage. Internal leakage is essentially affected by the flow geometry such as clearance, working pressure in addition to the fluid properties such as oil viscosity while the external leakage is affected by the seal type and the hose connections.

The actuator constant velocity is considered very critical in some applications like aviation, robotics, subsea systems, etc. The change in actuator velocity in hydraulics systems occurs due to a lot of un-certainties like internal leakage, external leakage, back pressure, pressure drop inside the hydraulic circuit, and change in oil viscosity due to temperature.

Leakage compensation in constant speed hydraulic systems is highly required to have a precise control of the job target. To achieve this, two options are available; either pump control or flow control. For flow control, one of the most methods used in achieving the leakage compensation is to develop an effective actuator velocity control using a proportional electro-hydraulic control valves with either open loop or closed loop. Another method is to use pump control through motor speed control.

In the present work, experimental investigation of a constant speed system, comprising proportional electro-hydraulic directional control valve (EHDCV) and variable pump rotation speed control, is studied. The system performance has been studied for different working conditions for dynamic and steady state response, the effect of external leakage of the system performance has been highlighted for different working conditions for the valve input, pump speed.

The external leakage compensation has been achieved by controlling the proportional electro-hydraulic valve input volt or by controlling the pump rotation speed through frequency modulator. The main objective is to compensate the minor expected external leakage inside the hydraulic system that leads to sudden change in the actuator speed and to maintain constant operation parameters.

The variable speed hydraulic system, as shown in Figure 1, is an energy-saving system, which usually utilizes a variable speed pump, a fixed displacement pump driven by a variable speed electric motor, and varies the output of actuators by regulating the rotary speed of the electric motor. Compared with traditional variable displacement hydraulic systems, utilizing a constant speed electric motor to drive a variable displacement pump, the variable speed hydraulic systems have many advantages, such as higher reliability, wider range of speed regulation, better energy-saving, and lower cost.

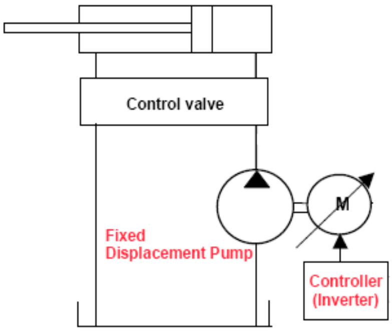

Fig. 1: Inverter based hydraulic pump

The inverter based hydraulic pump adds brains and causes the system to pump only as much as it needs to maintain constant pressure. The improved inverter drive system used in hydraulic applications along with the combination control tandem pump, were used to create a unit. This unit automatically controls the pump depending on the load pressure during operation, standby, and pressure maintenance. For example, during pressure maintenance, the motor runs at minimum power to operate only the small pump.

The inverter based pump has the following advantages; minimal heat generated by the motor, contributing to consistent accuracy in process, much less oil needed for heat reduction, less oil required to operate these units over conventional systems, service life increase, operate at a very low noise level, units operate just above room temperature with continuous operation, because of less oil heating the oil in tank actually lasts up to 3 times longer, less oil changed and less oil used producing less cost in waste removal, lower CO2 emissions than conventional units, less cooling equipment needed, reduced air-conditioning load, reduces the system losses, relief valve is closed, reduce heat exchanger size, reduce thermal effect and avoid properties change in oil . But it has the following disadvantages, sensitive to inertia, sensitive to speed variation, and sensitive to working pressure due to internal leakage.

Imamura and Sawada (2008) developed a novel energy-saving hydraulic pump system that comprises a high-efficiency motor, low-inertia pump and an inverter controller. Compared with conventional hydraulic pumps each driven by a constant speed induction motor, our novel hydraulic pump system driven by a high-efficiency variable-speed IPMSM (Interior Permanent Magnet Synchronous Motor) features energy-saving of 40% or more when used for an injection molding machine which is a typical application of this new pump system. By means of a pressure feedback control function and run speed control software on the controller, the pressure and flow rate of the hydraulic fluid are accurately controlled according to a command from the controller of the molding machine.

Park and Hwan (2008) studied electro hydraulic actuator (EHA) system in cascade system. The inner loop consists of an electric motor, a gear pump, and an angular velocity controller, and the outer loop consists of a hydro-actuator and a position controller. Especially, dead-band nonlinearity that exists between the electric motor and the gear pump and friction that occurs between the cylinder and the piston are considered. The tracking performance of EHA position control systems becomes unsatisfactory due to the dead-band and friction effects. Thus, in order to improve the position tracking performance of EHA systems with disturbance, back stepping control scheme for the desired position tracking is proposed, which was compared with the conventional PID scheme.

Lovrec et al. (2009) study the applicability of a low-priced drive concept using a speed-controlled induction motor in combination with a constant-displacement pump applied in a load-sensing control strategy. The suggested approach of the drive concept has been experimentally verified on a prototype of the drive. A hydraulic press-brake used for the machining of casting products in the automotive industry was taken into consideration. Experimental results prove that the suggested drive concept fulfils, on the whole, the goals and expectations aimed at in this work: low energy losses, low noise emission, improved control dynamics, reduced steady-state error, and convenient cost-effectiveness.

Hu and Ding (2011) discussed about the velocity tracking control of the system, in which the compound algorithm of PD & feed forward-feedback control was proposed. The simplified working principle of this kind of pump/inverter-controlled system was first introduced. The mathematic models of the system and the transfer function were established based on the hydraulic elevator system. The frequency-domain analysis of the system showed that it is a type 0 system with the feature of low frequency and poor damping. The steady-state error of velocity is still existed by unity-feedback control without compensation, and the velocity lag occurs with the integral compensation.

Pournazeri and Khajepour (2011) proposed a new lift control strategy based on the hydraulic supply pressure and flow control. In order to control the peak valve lift, the hydraulic pump speed is precisely controlled using a two-input gearbox mechanism. This eliminates the need for precision control of the solenoid valves opening interval within every cycle. To achieve a smooth control signal, it is worthwhile to control the maximum valve lift within few engine cycles rather than every cycle; therefore, instead of using the governing nonlinear differential equations of the mechanism, a novel average model of the system is developed based on energy conservation equations. Moreover, the new lift control technique is implemented experimentally by reconfiguration of the existing electro hydraulic valve system prototype and empirical results are then compared with those obtained from the simulations.

Xu and Jin (2013) proposed an energy regulation based variable-speed electrohydraulic drive. This novel drive principle is combined with the advantages of variable-speed drive and valve-control drive. The speed-control strategy, which is aimed at the multiple-input multiple-output (MIMO) structure of the proposed drive principle, is analyzed. The results of simulations and experiments comparing it with three other drive principle systems show that the proposed drive principle has not only good speed-control accuracy, but also a perfect energy-saving performance.

Xu et al. (2014) presented an accumulator-based power assisted unit (PAU). The PAU is an energy assisting & recycling device, which can release or absorb hydraulic energy according to system requirements. The proposed drive is expected to improve response and control precision compared with the variable-speed drive. The proposed drive principle system is a multi-input-multi-output (MIMO) complicated nonlinear system with time-varying, which increases the control difficulty. A mathematical model of proposed drive was firstly deduced. Then, a hybrid control strategy was presented. A four drives have been tested using three common variable-load disturbances. The comparisons of simulation results show that the proposed drive principle system demonstrates a good dynamic performance, which can not only achieve the expected energy saving target, but also significantly improve the response and control precision over the existing variable-speed drive system.

II. EXPERIMENTAL TEST RIG

Experimental study has been carried out to show the effect of changing the different flow control methods namely proportional directional valve control, pump speed control on the system static and dynamic performance. In addition, different flow control methods with external system leakage are used to keep actuator speed constant, neglecting the speed variation due to oil compressibility. External system leakage is intentionally introduced and the leakage compensation has been achieved by using the proportional electro-hydraulic valve and by controlling the pump speed through frequency modulator. The study aims to highlight the difference between these two control options on the dynamic and steady state response of the system.

The developed hydraulic system used for investigation consists of a double acting actuator with horizontal load and connecting hoses equipped with a position sensor linear variable differential transformer (LVDT) and pressure sensors. The load force acting on the actuator includes the mass and back pressure acting on the actuator in the return line by a throttle valve.

The experimental test rig has been shown in Figure 2 while the circuit of the studied system has been introduced in Figure 3. The experimental test rig consists of three phase AC motor, variable displacement vane pump, 4/3 Electro-hydraulic proportional direction control valve EHPDV with its proportional amplifier, double acting cylinder, pressure relief valve, check valve, oil tank and filter.

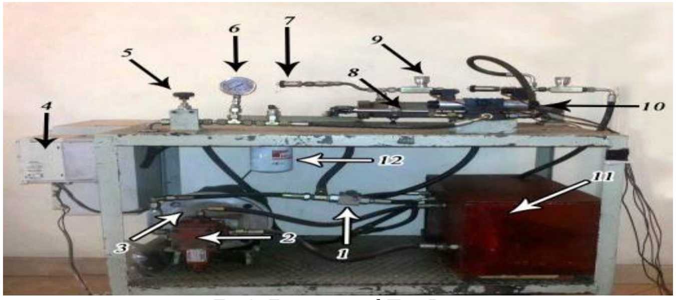

Fig 2: Experimental Test Rig

(1) ball valve, (2) variable speed vane pump, (3)AC motor, (4) variable frequency driver, (5) pressure relief valve, (6) pressure gauge, (7) Pressure sensor, (8) LVDT sensor, (9) throttle valve, (10) electro hydraulic proportional control valve, (11) tank, (12) filter.

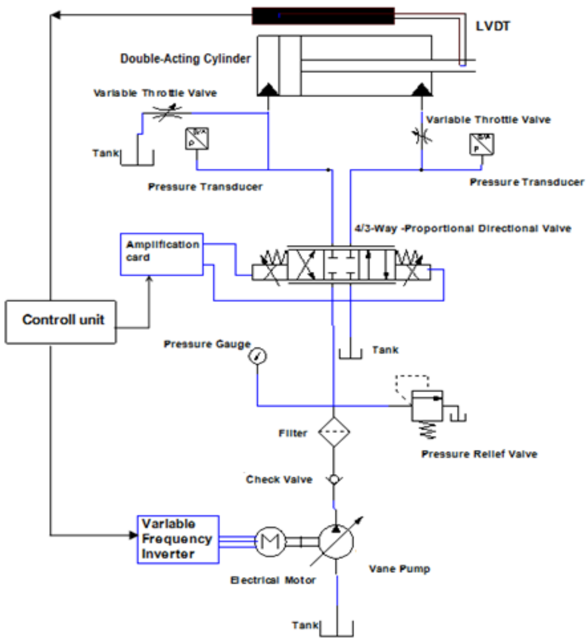

Fig 3: Schematic diagram of the hydraulic circuit.

The electro-hydraulic proportional directional control valve is used to provide a precise flow quantity at the desire time to the actuator chambers to perform the required task. The valve under investigation is manufactured by Parker-Hannifin of 4/3 electro-hydraulic open loop proportional directional control valve controlled by a proportional electrical solenoid with zero lapping to regulate the flow rate. The maximum flow is 12 L/min with solenoids (9v, 2.7 A).

The used measuring instrumentations are pressure transducer, Linear Variable Differential Transformer (LVDT). The pressure transducer is used to measure the pressure at the output ports of the proportional valve with pressure range up to 40 bar and supply voltage of 12 to 36 VDC and output signal from 4 to 20 mA, it has been calibrated with a percentage error of 0.7%. A linear variable differential transformer (LVDT) is used to measure linear displacement of the actuator with range up to 230 mm, supply voltage of 9-24 VDC and sensitivity of 200 mV/mm with output signal of 0-5 V and it has been calibrated with a percentage error of 0.4%. A variable frequency driver (VFD) is used to control the speed of the AC motor by controlling the frequency of electric power supplied to the motor.

To develop an adequate velocity control for hydraulic cylinder actuators, a hardware-in-the-loop electro hydraulic linear actuating system simulator was developed. This interactive simulator was established using LABVIEW software which is consisted of a pulse width modulation (PWM) valve control driver, an electro-hydraulic proportional directional control valve. The control flow chart of the experimental test rig is illustrated in Figure 4.

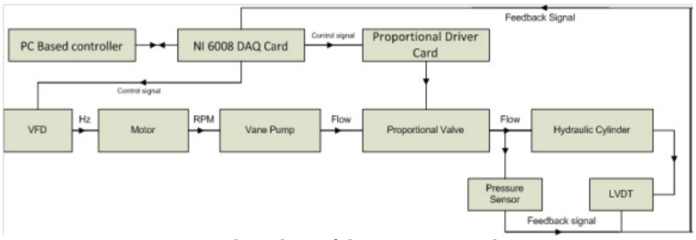

Fig 4: Flow chart of the system control

III. EXPERIMENTAL RESULTS

Many experiments have been conducted to show the effect of change the directional proportional valve setting and the vane pump speed on the dynamic and steady state response of hydraulic cylinder. Intentionally, different system external leakages have been introduced to show its effect on the system performance. Different input voltage for proportional valve and different input frequency for the pump speed have been applied. Pressure has been read using pressure gauge, piston displacement has been also measured using a LVDT sensor then actuator velocity were calculated using LABVIEW software.

A. System Performance using different flow control methods without leakage

System performance such as piston displacement, piston velocity, stroke time (time needed until piston reaches the end of stroke) and velocity rise time (time to constant velocity) has been studied, static and dynamic performance, using different flow control methods namely proportional directional control valve, pump speed control. Different experimental investigations have been conducted with different proportional valve setting at constant pump speed. In addition, different pump speeds setting at constant valve setting have been investigated.

1. System Performance for Different Pump Speed Control:

Experimental tests were conducted for fixed input value of 40% for the directional proportional control valve for different input frequency values applied to show their effect on the system performance.

System performance for different pump speed has been illustrated in Figure 5. The piston displacement stroke varied from 0 to 230 mm while varying the pump speed from 450 rpm to 900 rpm , the piston reached its target in 25s at maximum pump speed of 900 rpm and the longest time was 54s at a minimum pump speed of 450 rpm.

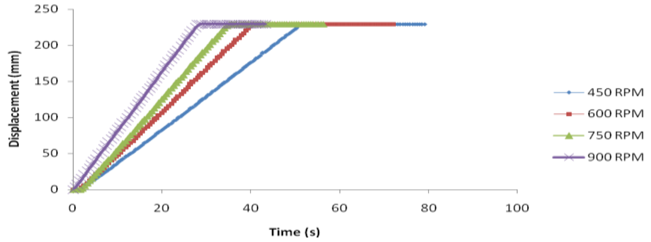

Fig 5: Piston displacement at different pump speeds

In Figure 6, the stroke time decrease with the increase of controlling the inverter frequency for controlling pump speed which reflects that the cylinder moves faster with the increase of the input pump rpm.

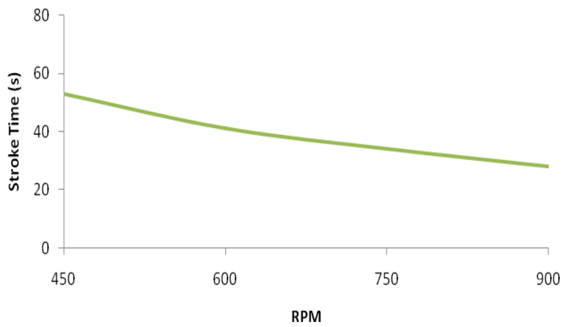

Fig 6: Piston stroke time for different pump speed.

In Figure 7, the piston velocity varied from 0 to 4.5 m/s while varying the input pump speed from 450 to 900 rpm and the piston reached its maximum velocity in 10s at maximum pump speed of 900 rpm. While the minimum velocity at 2.5 m/s at minimum pump speed of 450 rpm, and the velocity rise time in this case is kept constant in about 35s of actuation.

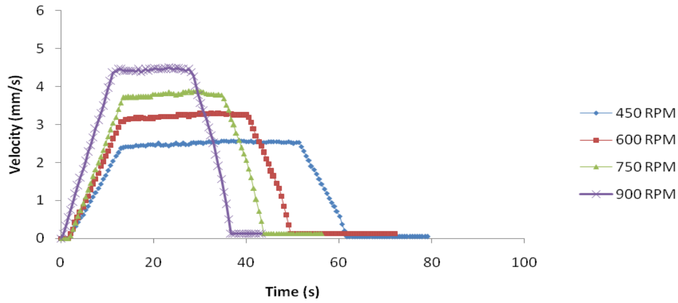

Fig 7: Piston Velocity for different pump speed.

The following experimental results conducted for different pump speed control in the range of 450 to 900 rpm and proportional DCV opening setting of 40%, 50% and 60% as shown in Figure 8. In this figure as the pump speed increased, the cylinder stroke rise time has been decreased for different proportional valve setting. It has been shown that at 40% DCV setting, the cylinder stroke time is about twice increase of that for 50% while the DCV setting of 60% is about 30% reduction of that at 50% valve setting and this difference is decreased with the increase of pump speed. This discrepancies is due to the increase of the pump flow with the increase of the pump speed while the effect of this variation with the different DCV setting show that at 50% valve setting the performance of stroke time is more effective rather than that at 40% DCV setting and at 60% does not has a great effect from that at 50%.

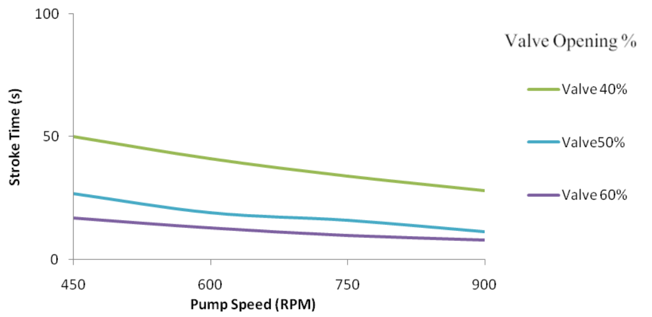

Fig 8: Cylinder stroke rise time for different pump speed and valve setting.

In Figures 9 to 10, the cylinder velocity and velocity rise time have been plotted. In these figures, as the pump speed increased, the cylinder velocity rise time have been decreased for different proportional valve setting. It has been shown that for different DCV setting, the velocity value at 50% DCV setting is about more 200% of than at 40% DCV setting and this difference increase with the pump speed increased. Also, at DCV setting of 60% is about 120% of that at 50% valve setting and this difference is decreased and reached a constant value with the increase of pump speed. This discrepancies is due to the DCV overlapping and show that at 50% valve setting the performance of stroke rise is more effective rather than that at 40% DCV setting and at 60%.

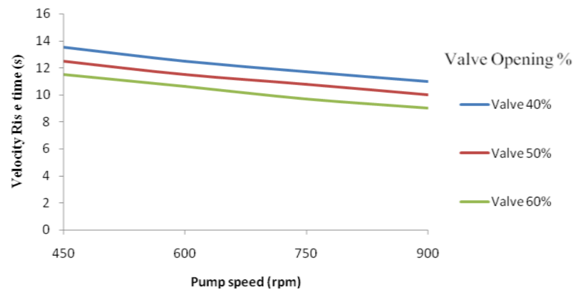

Fig 9: Cylinder velocity rise time for different pump speed.

Figure 10 shows that the variance in the valve opening at constant rpm has a large effect on the piston steady state velocity more than changing the pump speed at constant valve opening.

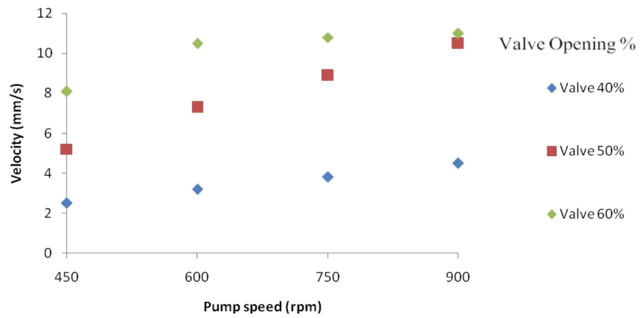

Fig 10: Steady state cylinder velocity for different pump speed.

2.Effect of proportional DCV setting on System Performance

System performance, cylinder displacement and velocity, has been recorded for different input volt for the directional proportional control valve for a pump speed of 600 rpm for the pump inverter of the controlling pump in case of no external leakage.

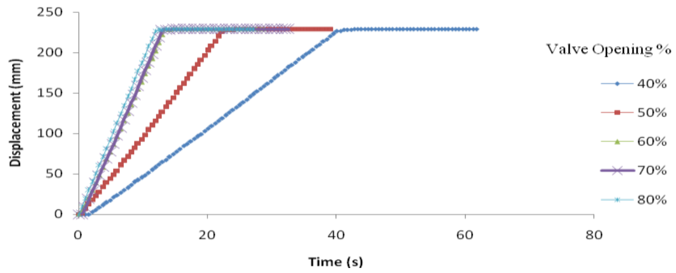

Fig 11: 1 Piston displacement for different valve opening .

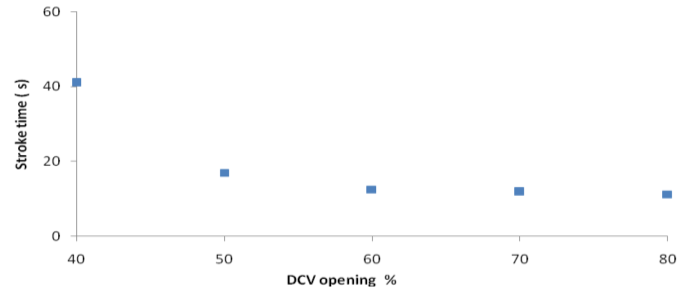

Fig 12: Piston stroke time for different valve opening at 600 rpm.

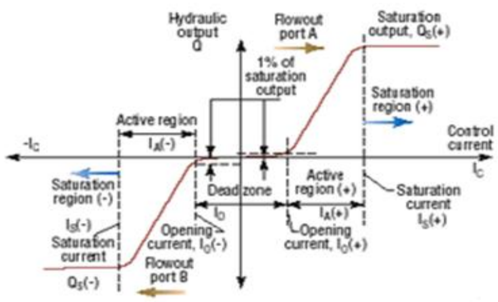

Fig 13: proportional valve overlapping effect

Piston displacement varies from 0 to 230 mm while directional proportional control valve opening varies from 40% to 80%, the piston reaches its end in 11s at maximum voltage of 80% opening and the longest time exceeded 40s at minimum voltage of 40% opening.
With the increase of the input directional proportional volt, the piston displacement reached its maximum displacement rapidly until it has no effect on the cylinder displacement for a range over 60% opening as shown in Figure 12 due to valve overlapping effect as it enter the saturation region . Figure 13 explains the effect of valve over lapping as it divide the valve performance curve into main three areas dead zone , active region and saturation region, the valve enter the saturation region after 60% opening .In addition, piston velocity for different directional proportional control valve voltage varies from 0 to 11 m/s while varying proportional valve voltage from 40% to 80% opening as shown in Figure 14.

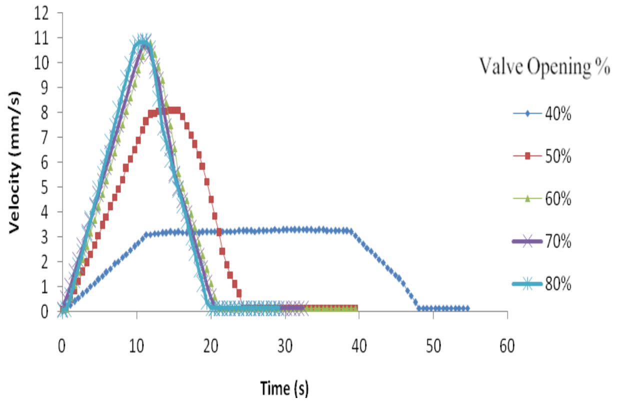

Fig 14: Piston velocity for different valve opening at 600 rpm.

The piston reaches its maximum velocity in 11s at maximum voltage of 80% opening while its minimum velocity of 2.5 mm/s was recorded at minimum voltage of 40% opening and kept constant velocity for about 30 seconds.

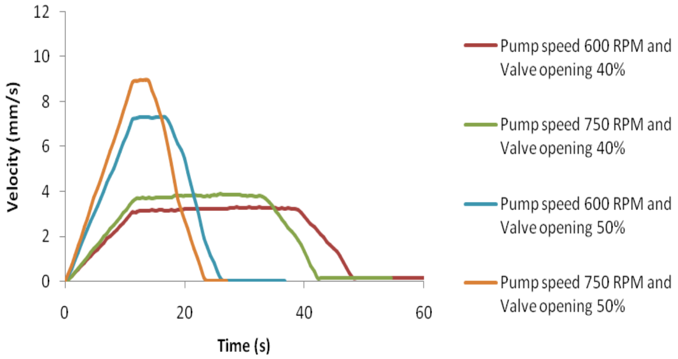

Fig 15: Performance curves at 600 and 750 rpm at different valve opening

Figure 15 shows the variance between velocity curves at pump speed 600 and 750 rpm at valve opening 40% and 50%, it shows that the change in the valve opening has a large effect on the velocity value and the velocity rise time more than the effect of changing the pump speed.

The following experimental results conducted for different proportional DCV setting in the range of (40% to 80%) opening and pump speed range of (450 to 1050) rpm. In Figures 16 to 17, the cylinder velocity has been plotted for different pump rpm and different valve opening. In these figures, as the proportional DCV opening increased, the cylinder velocity rise time has been decreased for pump speed It has been shown that for different DCV opening, the cylinder velocity rise time decreased and the reducing behaviour are not all the same trend and with the increase of the DCV valve setting the trend become more dramatic decreased.

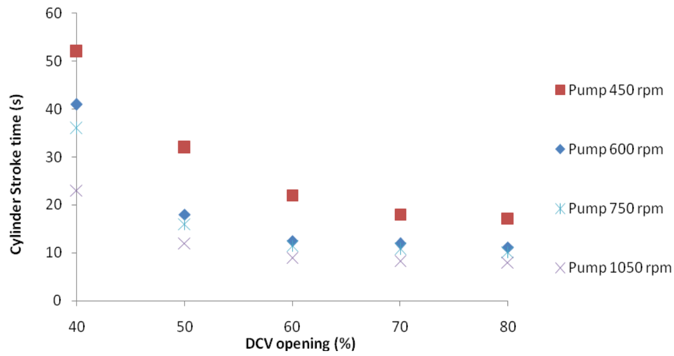

Fig 16: Cylinder stroke time for different Valve opening.

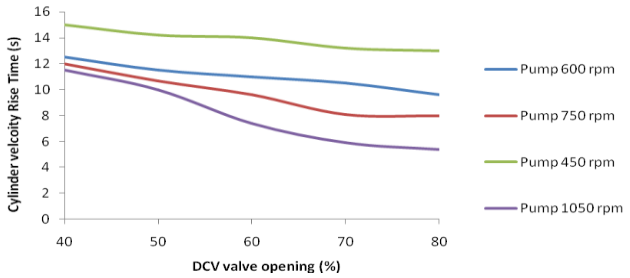

Fig 17: Cylinder velocity rise time for different DCV opening

In these figures as the proportional DCV setting increased, the cylinder stroke rise time has been decreased for different pump speed and with the increase of the pump speed, flow increase, the rise time decreased and the difference between the different proportional DCV settings become smaller with the increase of pump speed. It has been noticed that the cylinder rise time becomes saturated for about 60% due to the valve overlapping and that the valve best performance could be achieved around 50% of its opening.

B. System Performance for Different System Leakage

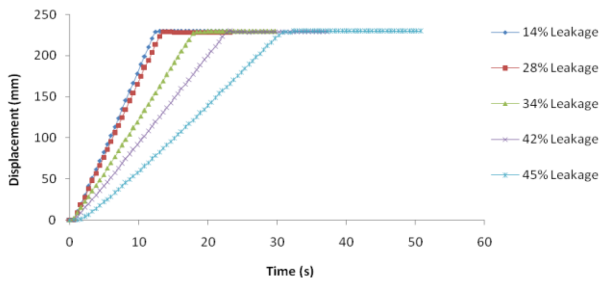

Fig 18: Piston Displacement for different external leakage percent

The tests were conducted while considering the leakage's effect on the expansion stroke of the piston, several leakage values were used to show their effect on both displacement and velocity. Piston displacement stroke varies from 0 to 230 mm with a constant pump speed of 600 rpm and valve opening of 60%. It has been shown that as the leakage value decrease as the piston displacement response time reaches maximum values faster as shown in Figure 18.

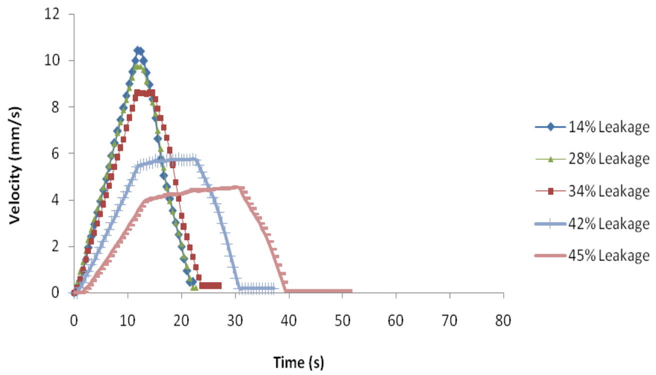

Fig 19: Piston velocity for different external leakage values

Piston velocity varies from 0 to 11 m/s, such velocity is attained when the pump speed is 600 rpm and the valve opening is 60% while the piston reaches its maximum velocity 10 mm/s in 10 s and minimum velocity at 3.2 mm/s in 40s as shown in Figure 19.

C. Performance with System leakage

1.System Performance with Proportional Directional Control Valve:

Piston displacement stroke varies from 0 to 230 mm; such displacement is attained while pump speed is kept constant at 600 rpm and valve input voltage is variable and the longest time for the piston to reach the target was with a minimum voltage of 4V as shown in Figure 20, while piston stroke time for different input valve control signal with leakage and without external leakage has been illustrated in Figure 21.

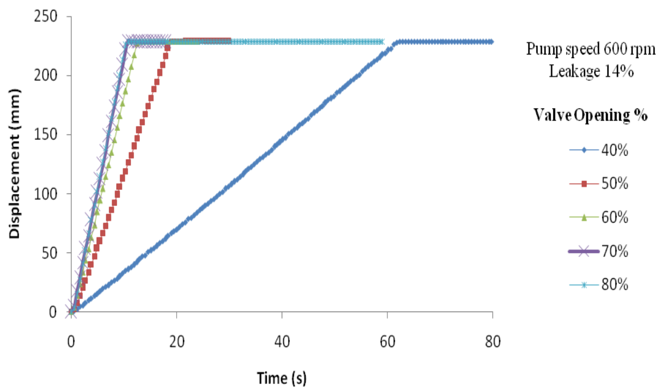

Fig 20: Piston displacement for different valve opening with system leakage

Piston velocity varies from 0 to 11 m/s, such velocity is attained while varying Valve opening from 40% to 80%. It has been shown in Figure 22 that piston reaches its maximum velocity in 10s at maximum valve opening of 80%, and minimum velocity at 2.5 m/s at minimum valve opening of 40%. The velocity in this case is kept constant in about 36s. Figure 23 shows the different between velocity rise time in case of with leakage and without leakage.

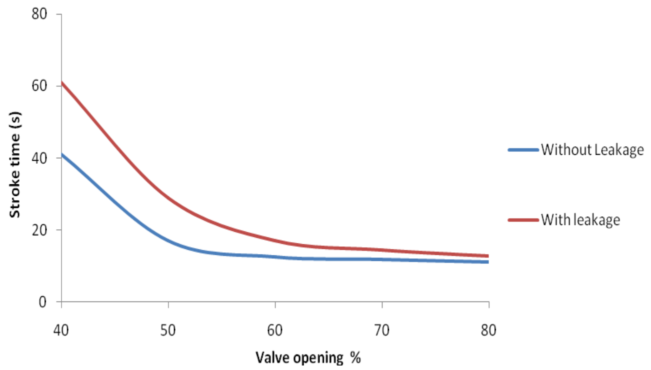

Fig 21: Piston displacement for different valve opening with and without system leakage

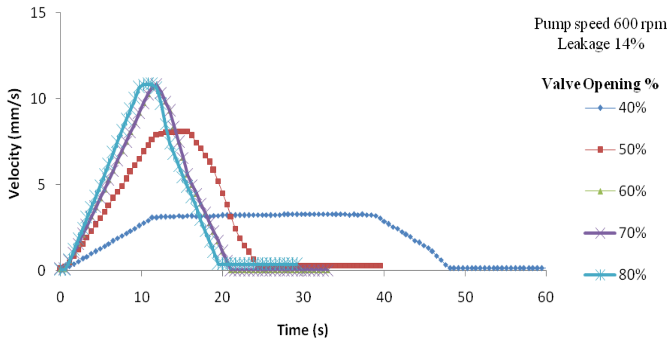

Fig 22: Piston Velocity for different valve opening with system leakage.

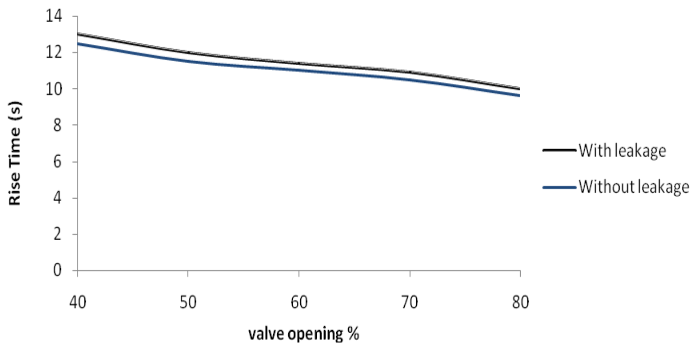

Fig 23: Velocity rise time for given valve opening with and without system leakage

2.System performance with controlling pump speed

For a fixed valve opening 40%, the piston displacement stroke varied from 0 to 230 mm while varying the pump speed from 450 to 900 rpm, the piston reached its target in 40s at maximum pump speed 900 rpm and the longest time was 95 s at a minimum pump speed of 450 rpm.

System performance for different pump speed has been illustrated in Figure 24. The variance in piston stroke time has been shown in Figure 25 for both cases with leakage and without leakage.

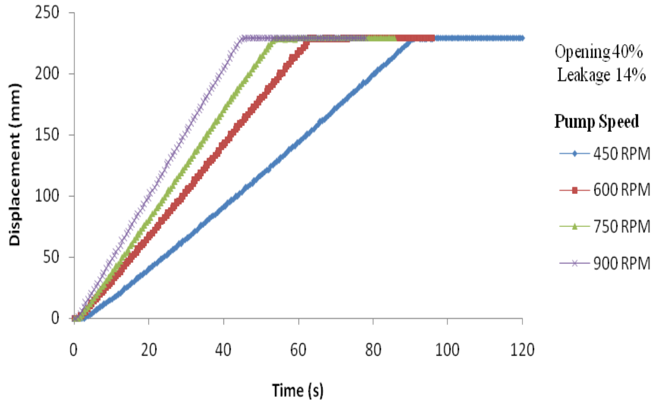

Fig 24: Piston displacement for different pump speed with system leakage

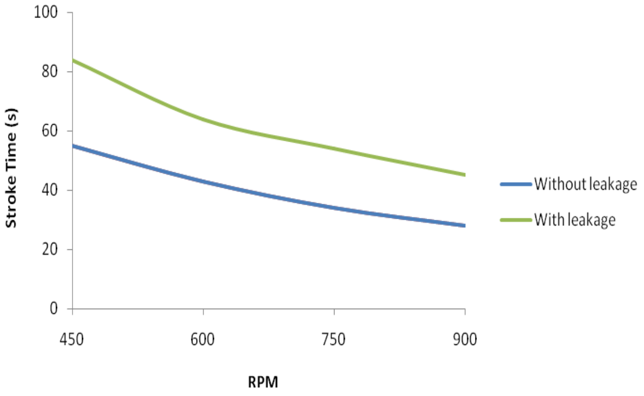

Fig 25: Piston stroke time for different pump speed with and without system leakage

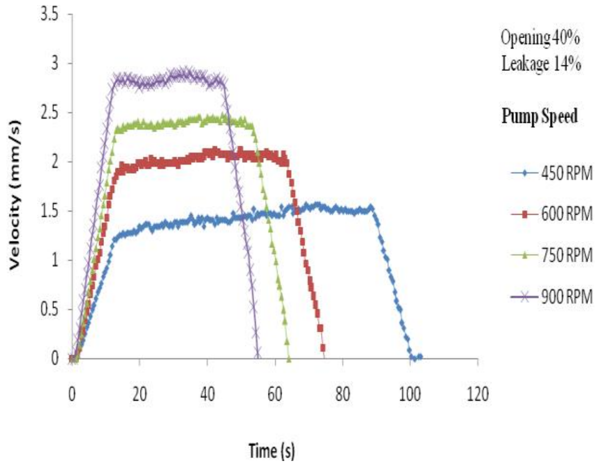

Fig 26: Piston Velocity for different pump speed.

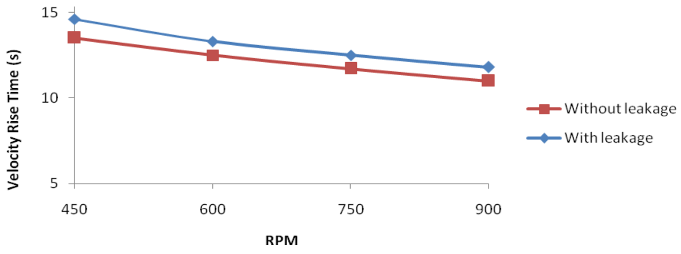

Fig 27: Velocity rise time for given pump speed with and without system disturbance

Piston velocity varied from 0 to 2.8 mm/s while varying the inverters input pump speed from 450 to 900 rpm and the piston reached its maximum velocity in 15 s at maximum pump speed of 900 rpm. The minimum velocity at 1.3 mm/s is shown at minimum pump speed 450 rpm, as shown in Figure 26. Variance in velocity rise time has been shown in Figure 27.

D. System Dynamic Characteristics:

The system dynamic characteristics affect heavily the overall performance of electro hydraulic systems. These particular characteristics include overshoot response, settling time and rise time. While conducting the different load and leakage iterations on this system, it was concluded that settling time and steady state error varied heavily between the existing of external leakage and without external leakage cases. So, more experiments were conducted to highlight this effect to adjust the required voltage levels in leakage cases to match the no external leakage case.

1.Leakage compensation in case of pump control

While controlling the system using pump inverters, leakage has a great effect on response time of cylinder displacement as shown in Figures 28, the required controlling voltage values to compensate the leakage effect have been shown in Figure 29.

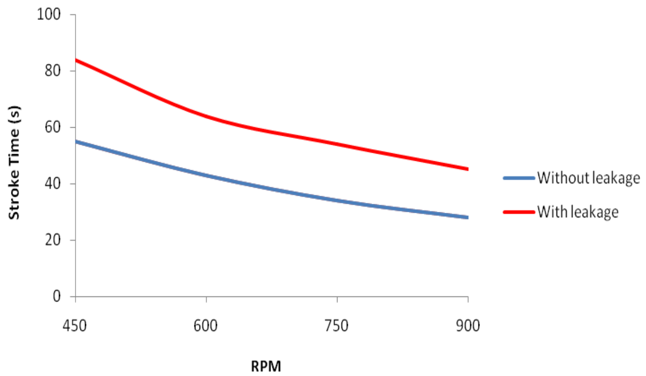

Fig 28: Piston dynamic response time for different pump speed.

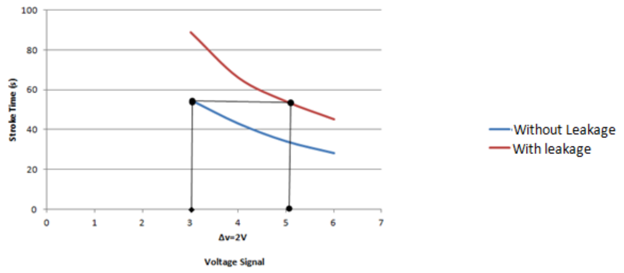

Fig 29: Voltage required overcoming system leakage.

2. Leakage compensation in case of valve control

Also, while controlling the system using valve controller, leakage has a considerable effect on response time of cylinder displacement as in Figure 30, the required control voltage values to compensate the leakage effect is shown more clearly in Figure 31.

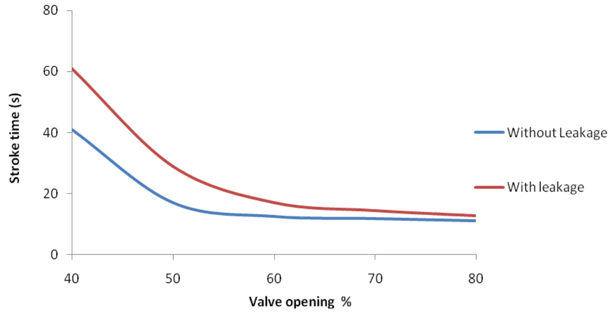

Fig 30: Piston stroke time for valve voltage

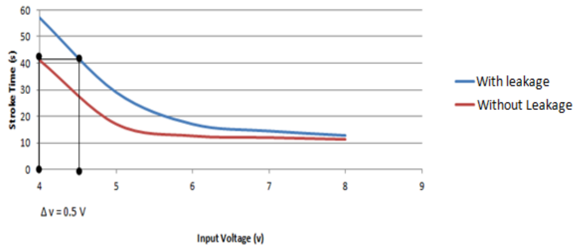

Fig 31: Voltage required overcoming system leakage

3.Steady state error adjustment in case of pump control

Steady state error is the difference between the measured velocity in case of leakage and without leakage. The cylinder velocity in case of leakage and without leakage has been illustrated to show the difference steady state error has been illustrated in Figure 32.

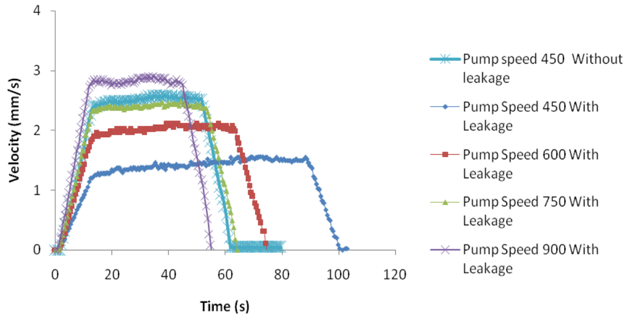

Fig 32: steady state error in piston velocity in pump control at 40% opening

4.Steady state error in case of valve control

For DCV control of 40% opening, the cylinder velocity in case of leakage and without leakage has been illustrated to show the difference steady state error has been illustrated in Figure 33.

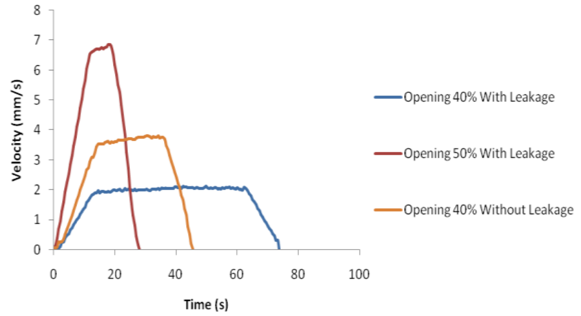

Fig 33: steady state error in piston velocity during valve control at 600 rpm

The volt control signal for proportional DCV and speed control for pump control will be used for leakage compensation in future work through controller with a precise control algorithm to compensate leakage in electro-hydraulic proportional systems.

IV. CONCLUSIONS

The dynamic and steady state system performance for the hydraulic pump and proportional valve have been investigated. It has been shown that settling time and steady state error are varied from normal while introducing external leakage in the hydraulic system and these variations have been observed and plotted. In order to overcome the external leakage, controlling the proportional valve opening and the pump speed has been illustrated.

- It has been found that controlling the proportional valve opening has a better effect on piston performance than controlling the pump speed.
- It has been shown that varying the input speed for the pump has a considerable effect on the response time of the actuator velocity as well as the actuator displacement response time.
- It has been shown that valve opening higher than 60%, there is a very small effect of varying on the response time of the actuator velocity while there is a minor effect on the actuator displacement response time due to valve over lapping.
- It has been shown that the optimum range for pump speed operation is about 600 to 750 rpm at a proportional DCV opening in the range of 40% to 60% due to the valve overlapping.

V. REFERENCES

[1] Jiing-Yih Lai And Yuan Rong " Adaptive Flow Rate Control of a Hydraulic Proportional Valve" JSME International Journal Series III,Vol. 35 , 1992, pp. 582-590

[2] Shiuh-Jer Huang And Shinn-Horng Chen "Optimal Control And Analysis Of Hydraulic Machine Tool Servo System" International Journal of Machine Tools and Manufacture ,Vol 06, 1992 , pp. 361-377

[3] Sung-Mu, Chung-Oh And Chang Pyung H."An Experimental Study On The Position Control Of An Electro Hydraulic Servo System Using Time Delay Control, Control Engineering Practice, Volume 2, Issue 1, February 1994

[4] Y.Liu And H.Handroos, "Technical Note Sliding Mode Control For A Class Of Hydraulic Position Servo," Mechatronics 9 1999.

[5] Knohl, T. & Unbehauen H."Adaptive Position Control Of Electrohydraulic Servo Systems Using ANN". Mechatroincs. 10, (2000) 127–143, ISSN 0957 4158,2000

[6] Bin Yao, Fanping Bu, J. Reedy, G. T. -C. Chiu "Adaptive Robust Motion Control Of Single-Rod Hydraulic Actuators: Theory And Experiments" IEEE/ASME Transactions On Mechatronics, Vol. 5, No. 1, March 2000

[7] Bin Yao, Fanping Bu, George T. C. Chiu, "Non-Linear Adaptive Robust Control Of Electro-Hydraulic Systems Driven By Double-Rod Actuators" Int J Control , 74(8), 761-775 2001

[8] W.Dobchuk "Control Of A Hydraulically Actuated Mechanism Using A Proportional Valve And A Linearizing Feedforward Controller", Phd, Department Of Mechanical Engineering, University Of Saskatchewan, Saskatoon, Canada, Aug. 2004.

[9] Phanindra Garimella And Bin Yao "Fault Detection Of An Electro-Hydraulic Cylinder Using Adaptive Robust Observers" Anaheim, California, USA, November 13 – 19, 2004

[10] Ke Lia, M.A. Mannan "Electro-Hydraulic Proportional Control Of Twin-Cylinder Hydraulic Elevators" Control Engineering Practice Volume 9, Issue 4, April 2004

[11] Ferreira, J. A., Gomes De Almeida, F., Quintas, M. R. And Estima De Oliveira, J. P. "Hybrid Models For Hardware-Inthe-Loop Simulation Of Hydraulic Systems. Part 1: Theory. References" Proc. Instn Mech. Engrs, Part I: J. Systems And Control Engineering, 2004

[12] J.-C. Renn And C. Tsai Development Of An Unconventional Electro-Hydraulic Proportional Valve With Fuzzy-Logic Controller For Hydraulic Presses The International Journal Of Advanced Manufacturing Technology, Volume 26, July 2005

[13] K. Dasgupta, R. Karmakar, "Dynamic Analysis Of Proportional Solenoid Controlled Piloted Relief Valve By Bondgraph" Journal Of Simulation, Modelling, Practice And Theory 13 2005

[14] C. Schwartz , V. J. De Negri "Modeling And Analysis Of An Auto-Adjustable Stroke End Cushioning Device For Hydraulic Cylinders" Journal Of The Brazilian Society Of Mechanical Sciences 2005

[15] Bora Eryilmaz, And Bruce H. Wilson, "Unified Modeling And Analysis Of A Proportional Valve". Journal Of The Franklin Institute, 2006.

[16] Sang-Min ,Woo-Yong Han "Induction Motor Servo Drive Using Robust PID-Like Neuro-Fuzzy Controller" Control Engineering Practice 14, 2006

[17] Sam And Hudha "Modelling And Force Tracking Control Of Hydraulic Actuatorfor An Active Suspension System" Industrial Electronics And Applications, 1ST IEEE Conference, 2006

[18] Ayman A. Aly "A Non Linear Optimal PID Control Of A Hydraulic Crane "Journal Of Engineering, Sciences, Assiut University, Vol. 35, 2007

[19] Branimir D. Kikovi, Nemanja S. Visnji "Modelling The Vehicle With Hydrodynamic Brakes On Downhill Routes And Defining The Brake Parameters With The Pneumatic Valve" International Journal Of Information And Systems Sciences Computing And Information Volume 4, 2008

[20] R Amirante, A Innone, LA Catalano. "Boosted PWM Open Loop Control Of Hydraulic Proportional Valves", Energy Conversion And Management. 2008

[21] H. Du & N. Zhang, "Fuzzy Control For Nonlinear Uncertain Electrohydraulic Active Suspensions With Input Constraint," Ieee Transactions On Fuzzy Systems, Vol. 17, (2) Pp. 343-356, 2009

[22] Menshawy, T., Moghazy, M. And Lotfy, A., "Investigation Of Dynamic Performance Of An Electro-Hydraulic Proportional", 13th International Conference On Aerospace Sciences And Aviation Technology, ASAT-13, May 26 – 28, M.T.C., Cairo, Egypt. 2009

[23] Shaymaa Mahmood "Controlling A Nonlinear Servo Hydraulic System Using PID Controller With A Genetic Algorithm Tool" IJCCCE, Vol. 12, No. 1, 2012

[24] Ajit Kumar, J. Das "Performance Investigation Of A Proportional Valve Controlled Hydraulic Motor Used In Hydrostatic Transmission System" International Journal Of Mechanical And Production Engineering, Issn: 2320-2092, Volume- 1, Issue- 6, Dec-2013

[25] Takayuki Imamura, Yuzo Sawada "Energy-Saving Hybrid Hydraulic System Comprising Highly Efficient Ipm Motor And Inverter, For Injection Molding And Manufacturing Machine "7th Jfps International Symposium On Fluid Power, Toyama 2008 September 15-18, 2008

[26] Min Gyu Park, Sung Hwan "Improvement Of Position Tracking Performance Of Electro Hydraulic Actuator Systems With Disturbance" 7th Jfps International Symposium On Fluid Power, Toyama 2008

[27] Darko Lovrec & Mitja Kastrevc "Electro-Hydraulic Load Sensing With A Speed-Controlled Hydraulic Supply System On Forming-Machines" Int J Adv Manuf Technol 2009

[28] Dongming Hu, Shoubao Ding "Velocity-Tracking Control Of The Variable-Speed Controlled Hydraulic System: Using Compound Algorithm Of PD & Feedforward-Feedback Control", Third International Conference On Measuring Technology And Mechatronics Automation, IEEE 2011

[29] Mohammad Pournazeri, Amir Khajepour "An Efficient Lift Control Technique In Electro-Hydraulic Camless Valvetrain Using Variable Speed Hydraulic Pump" SAE International 2011

[30] Ming Xu, Bo Jin "Speed-Control Of Energy Regulation Based Variable-Speed Electrohydraulic Drive "Journal Of Mechanical Engineering Volume 59, 2013

[31] Ming Xu- Jing Ni "Dynamic Simulation Of Variable-Speed Valve-Controlled-Motor Drive System With A Power-Assisted Device", Journal of Mechanical Engineering Volume (60) , 2014

[32] Qin Zhang "Hydraulic Linear Actuator Velocity Control Using A Feedforward-Plus-Pid Control'' , International Journal Of Flexible Automation And Integrated Manufacturing ,2001

[33] Maria Werlefors , Alexander Medvedev "Observer-Based Leakage Detection In Hydraulic Systems With Position And Velocity Feedback" 17th IEEE International Conference On Control Applications Part Of 2008 IEEE Multi-Conference On Systems And Control San Antonio, Texas, USA, 2008.