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MODELING AND CONTROLLING OF AN ELECTRO-HYDRAULIC ACTUATOR NOVEL BY USING INDEPENDENT METERING VALVE

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Abstract*-In recent year, hydraulic system research mainly focus on way of saving energy. Because in hydraulic systems the efficiency is quite low. In mobile hydraulic equipment, conventional hydraulic spool valves with pressure have already been replaced by valve assemblies with four-valve independent metering valve (IMV) with electronically controlled pressure compensation. The independent metering concept and microprocessor control have potential to save energy rather than conventional proportional valve control. This paper presents the structure and dynamic of a new hydraulic cylinder circuit using independent metering valves combine with control variable displacement pump to saving energy. This hydraulic circuit is simulated and compared with conventional by AMESim and Matlab Software. In IMV circuit, five Modes are compared together for verify the saving energy. Moreover, the mathematical model presented to model the dynamic of cylinder and all parameter of circuit get from the real system*.

Keywords: Independent metering valve, Load feedback, Energy saving, Hydraulic cylinder

1. INTRODUCTION

In present situation, considering the development of the industry, a trend towards automation of working cycles and supporting the users has increased strongly. Hydraulic systems have been considered as the potential choices for modern industries ranging from heavy-duty manipulators to precision machine tools because of their advantages such as durability, high productivity, efficiency, power weight ratios, controllability, accuracy, reliability, cost, etc. Operating power plants reliably, at full power and maximum efficiency, is desirable target because of its great economic significance. However, conventional electro-hydraulics typically increase machine costs, have difficulty controlling systems which contain control valves. Proportional directional four-way control valves are the most frequently used valves in motion control in hydraulic applications. Nowadays, in industrial applications of hydraulic systems, electronic control is becoming widespread [1]. A typical four way directional valve is shown by Bu and Yao [2]. Due to dependency of metering in and metering out orifices which are opened with respect to one driving input, it has only one degree of freedom (DOF) in controlling the valves to control the spool position. Only one chamber's pressure can be controlled. Hence it cannot provide precise motion control while controlling pressure in both actuator chambers. This further means that its potential for energy savings is low. Besides, different applications require different four way spool valve designs to accommodate the different needs of the various systems. Hence, the current trend in industry is to design a valve that can be adapted to the various systems in practice. Moreover, almost traditional hydraulic systems that work this kind of technology only sense the pressure in the inlet chamber of the actuator, while being unknown to the pressure in the other chamber and it uses pressure compensators to keep a constant pressure margin across the spool valve inlet and thus achieve a linear relationship between the valve opening and flow. The technical solution is that the Pressure Compensated Load Sense (PCLS) systems are blind to the pressure in the return chamber and it will drive the supply pressure high even when not needed, which means more energy consumption. This is an important reason greatly affects to efficiency of system. Therefore, if the drawback of typical four way directional control valves is broken then the flexibility of the valve could be drastically increased, making the way for significant improvements in hydraulic efficiency [3]. The technique of breaking the mechanical linkage between the meter-in and meter-out orifice is well known and has been used in heavy industrial applications for several years. The technology is called the independent metering hydraulic circuit.

The idea has been used in industry for several years. Aardema suggest using two conventional four way directional valves: one to control meter-in and the other to control meter-out [4]. This is a luxurious solution because it replaces one valve by two four-way directional valves, and it does not achieve complete independence of meter-in and meter-out. Liu and Yao investigated the use of a fifth valve to allow for cross port flow [5]. Their control scheme was composed of a task level controller and a valve level controller. The task level controller calculates the load that the cylinder has to move and determines which valves to be used. The valve level controller achieves pressure control and motion control of the hydraulic cylinder. His research used a robot arm setup that has three degrees of freedom and concentrated his work on boom motion control. Ruth Book used a similar valve concept that called the “Smart Valve” [6]. She used a five-valve configuration which can be adapted to any other hydraulic systems through making software changes to the valves. She included anti-cavitation algorithms to control the pressure in the cylinder inlet chamber. Whenever cavitation occurred, the piston halted its movement until the void could be filled with oil. The idea was to draw oil from the return port when the system senses an impending cavitation to fill the void in the inlet chamber. Due to the low flow capacity of the valves, the scheme was not successful all the times and cavitation occurred.

Moreover, the control of hydraulic systems is far from trivial due to the highly nonlinearities in the system, in addition to changes in fluid properties with temperature changes and some parameters of system unknown such as external disturbances, leakages, viscous friction. That is why researchers who worked on independent metering valves tried to apply adaptive control schemes to solve these problems inherent in hydraulic systems [7, 8]. The final goal of independent metering concepts is not only a more controllable system but a more energy efficient system as well. Because of fuel economy and the increasingly stringent environmental requirements, energy efficiency of hydraulic systems has been under research for a long time even before the concepts of programmable valves, independent metering, and regeneration flow appeared.

In this paper, an independent metering hydraulic circuit is proposed. Here, four proportional valves are used in this configuration to reduce the system cost and energy consumption. Moreover, regenerative flow and high pressure flow from the pump does not connect directly so it is possible to save more energy by flow regeneration. The force feedback characteristic is used to develop the mathematical modelling to make this system capable of reacting faster and more accurately even facing with disturbances. The control equations for all modes of independent metering are developed in such a way that these can be expressed as a simple orifice equation based on equivalent valve conductance and equivalent pressure concepts. Analysis and control of all modes is compared with respect to conventional hydraulic circuit to ensure the overall performance.

2. INDEPENDENT METERING VALVE (IMV) WITH FOUR VALVE

In this research, velocity control of hydraulic cylinder with energy saving is considered as the target. The IVM circuit mainly includes four electro-hydraulic proportional valves (EHP) and one check valve. It allows energy regeneration and reduces the required pump energy subsequently, saves valuable energy. In this concept, two valves are used to connect between two ports of the cylinder to the supply line, the other ones are to connect to the return line as depicted in Fig. 1.The five working modes of this system can be listed down as

(i) Power Extension Mode (PE)
(ii) Power Retraction Mode (PR)
(iii) High Side Regeneration Extension Mode (HSRE)
(iv) Low Side Regeneration Extension Mode (LSRE)
(v) Low Side Regeneration Retraction Mode (LSRR)

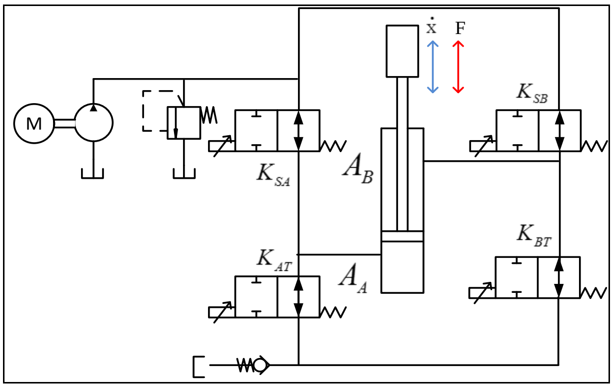

Fig.1: Schematic Diagram of IMV system

2.1 Power Extension Mode

The operation of this mode is not conceptually different from the operation of a cylinder controlled by a conventional proportional direction control valve. A schematic that describes this mode is shown in Fig. 2. Flow is supplied at high pressure from the pump through valve Ksa into head chamber of the cylinder casing the piston to extend. This piston motion forces the flow out of road chamber through valve Kbt to the tank. The difference, however, is that in this four-valve configuration Ksa and Kbt can be controlled separately, while in a conventional proportional valve they cannot be controlled indecently and thus flow to tank need not always be restricted.

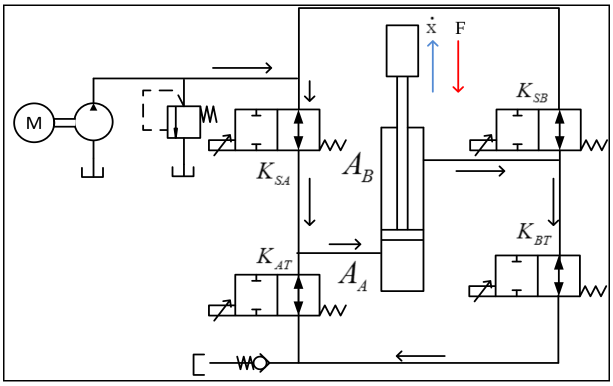

Fig.2: Power Extension Mode

2.2 Power Retraction Mode

This mode is shown in Fig. 3 with arrows showing the essential directions of forces and velocities. Flow is supplied at high pressure from the pump through valve Ksb into rod chamber of the cylinder causing the piston to retract. This piston motion forces the flow out of the head chamber through valve Kat to the tank. Again, in the four-valve configuration Ksb and Kat can be controlled separately, while in a conventional proportional valve they cannot be controlled indecently.

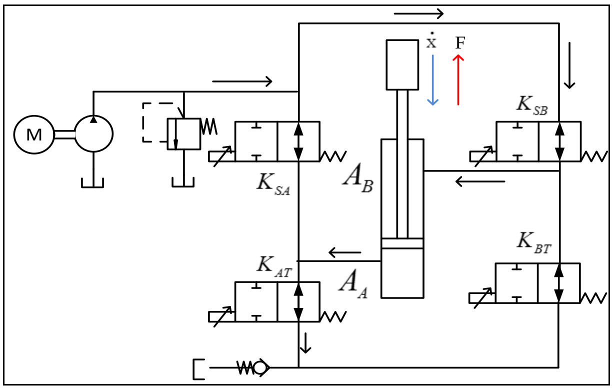

Fig.3: Power Retraction Mode

2.3 High Side Regeneration Extraction Mode

This mode could not be achieved by a conventional proportional direction control valve because the working principle of HSRE mode is described as Fig. 4. Flow coming out of rod chamber at high pressure does not go to tank through valve Kbt, but it circulates through valve Ksb and Ksa into head chamber causing the piston to extend. However, the flow coming out of rod chamber is less than the flow needed in head chamber to achieve a certain speed because of the different in areas ($A_A > A_B$). Flow out of either chamber for a given velocity and with the different in area ($Q_A > Q_B$). The remaining flow is supplied from the pump, but it is just enough to make up of the difference in flow while in powered extension mode all the flow is supplied from the pump. Thus, the high regeneration extension mode has the potential to save energy. It is called high side because the flow goes through a path involving Ksa and Ksb that is on the pump side (high pressure side).

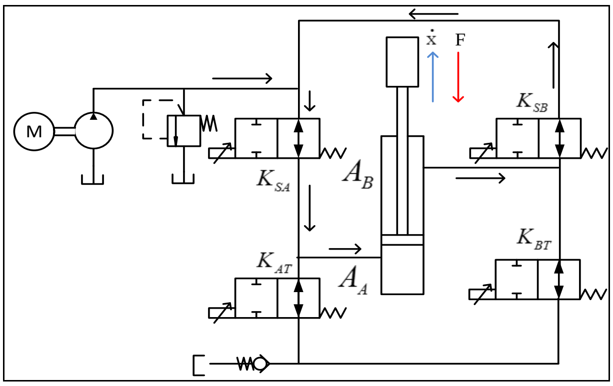

Fig.4: High Side Regeneration Extension Mode

2.4 Low Side Regeneration Extraction Mode

This mode is also best suited for overrunning loads, which, for example, happens when lowing a load with gravity assistance. A schematic that described this mode is shown in Fig. 5. Flow coming out the rod chamber does not go to tank but it circulates through valve Kat and Kbt into head chamber causing the piston to extend. However, the flow coming out the rod chamber is less than the flow needed in head chamber to achieve a certain speed as explained above. The remaining flow can be supplied from the pump, but it is just enough to make up of the difference in flow and it is supplied at a low pressure.

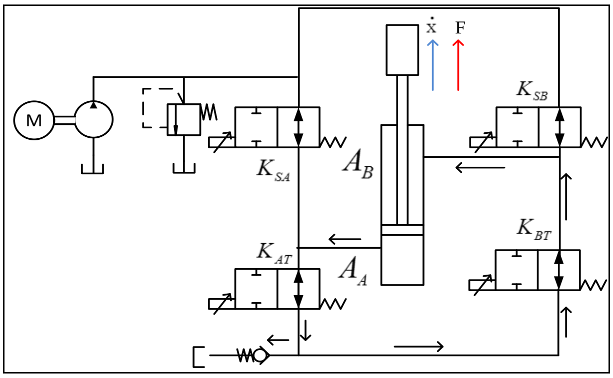

Fig.5: Low Side Regeneration Extension Mode

2.5 Low Side Regeneration Retraction Mode

This mode is also best suited for overrunning loads, which, for example, happens when lowing a load with gravity assistance. A schematic that described this mode is shown in Fig. 6. Flow coming out the head chamber does not go to tank but it circulates through valve Kat and Kbt into rod chamber causing the piston to retract. The flow coming out the rod chamber is less than the flow needed in head chamber to achieve a certain speed as explained above. The extra flow goes to tank through the check valve. Thus, no pump flow is needed at for this mode, which means a high potential for saving energy whenever this mode is used. It is called low side because the flow goes through a path involving Kbt and Kat that is the tank side (low pressure side).

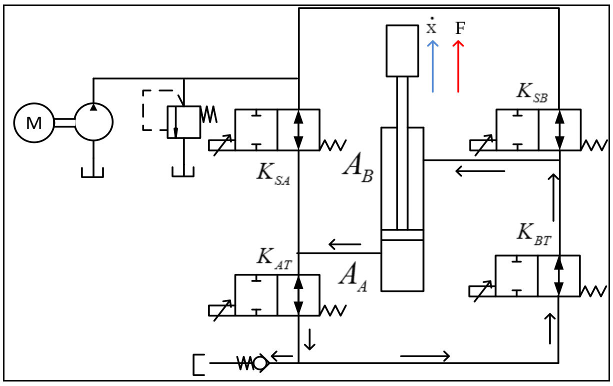

Fig.6: Low Side Regeneration Retraction Mode

3. QUASI-STATIC SYSTEM MODELING OF IMV

The dynamics model of the system shown in Fig. 1 include the dynamics of the oil path from a variable displacement pump through a proportional valve to a piston that has some differential pressure across it that depends on the load, and subsequent return of oil to a tank through another proportional throttle valve. This path, consequently, contains oil lines, proportional throttle valves, and expanding and compressing chambers (volumes). The dynamic model of valve can be further simplified as:

$$\ddot{K}_v + 2\xi\omega_n\dot{K}_v + \omega_n^2 K_v = k_n\omega_n^2 i_n \quad (1)$$

where, K_v is the conductance of valve; ω_n is nature frequency, ξ is damping coefficient, k_n is proportional coefficient of valve.

The orifice flows can be expressed as following:

$$Q_{sa} = K_{sa}\sqrt{|P_s - P_a|}\,\mathrm{sgn}(P_s - P_a) \quad (2)$$

$$Q_{sb} = K_{sb}\sqrt{|P_s - P_b|}\,\mathrm{sgn}(P_s - P_b) \quad (3)$$

$$Q_{at} = K_{at}\sqrt{|P_a - P_r|}\,\mathrm{sgn}(P_a - P_r) \quad (4)$$

$$Q_{bt} = K_{bt}\sqrt{|P_b - P_r|}\,\mathrm{sgn}(P_b - P_r) \quad (5)$$

where, K_{sa} , K_{sb} , K_{at} , K_{bt} is the conductance of valve SA, SB, AT and BT; P_s , P_a , P_b , P_r is supplied , head chamber, rod chamber and return pressure.

The compressibility is computed as:

$$Q_a = \frac{V_a}{B_e}\dot{P}_a \quad (6)$$

$$Q_b = \frac{V_b}{B_e}\dot{P}_b \quad (7)$$

where, $V_a = V_{a0} + A_a x$ and $V_b = V_{b0} - A_b x$, V_{a0} is the initial volume of head chamber, V_{b0} is the initial volume of rod chamber.

The mass conservation as:

$$Q_{sa} - Q_{at} - Q_L = Q_a + A_a\dot{x} \quad (8)$$

$$Q_{sb} - Q_{bt} + Q_L = Q_b - A_b\dot{x} \quad (9)$$

where, Q_L is the leakage flow form one chamber to the other around this piston, which is too small and it will be neglected.

Conservation of momentum is:

$$P_a A_a - P_b A_b = M\ddot{x} + F + F_f \quad (9)$$

where, F is the external force applied on the position and can be dependent on position. F_f is the cylinder friction force, which is declared by parameter in the AMESim model, also viscous coefficient and temperate.

4. NUMERICAL SIMULATIONS

In this section, numerical simulations with speed control of boom actuator in an excavator were carried out to investigate the applicability of the IMV circuit. For the simulations, the models using the IMV was built in a co-simulation in which the hydraulic circuits are constructed using AMESim software while the control logics derived for these IMV circuits are performed in Simulink environment of MATLAB software.

4.1 Comparison between IMV and Conventional Hydraulic Circuit

The structure of conventional hydraulic circuit and IMV circuit-Power Extension Mode apply for boom cylinder (CHC) systems are described as Fig. 7 and Fig. 8. In the IMV systems, proportional valves were used to control the actuator speed. The conductances of these valves were defined based on the reference speed and feedback pressure signals Pa, Pb, Ps and Pr of the hydraulic circuits.

For comparison between conventional hydraulic circuit and IMV system have to select only PE and PR mode because conventional hydraulic circuit can performed only those mode and other three modes of the IMV system describe the performance effectiveness and energy saving.

Table 1: System parameters

Parameters	Specification	Value
Cylinder	Piston diameter× Rod diameter × Length of stroke	40 mm× 22 mm× 0.50m
Fluid properties	Density	850 kg/m^3
	Bulk modulus	17000 bar
	Temperature	30°C
	Absolute viscosity	51 cp
Hydraulic pump	Pump displacement	8 cc/rev
	Pump Speed	1000 rev/min
Relief valve	Cracking pressure	60 bar
Proportional valve	Cross sectional area at maximum opening	6 mm^2
	Damping ratio	1.8
	Natural frequency	50 Hz
Direction control valve	Damping ratio	0.8
	Natural frequency	80 Hz

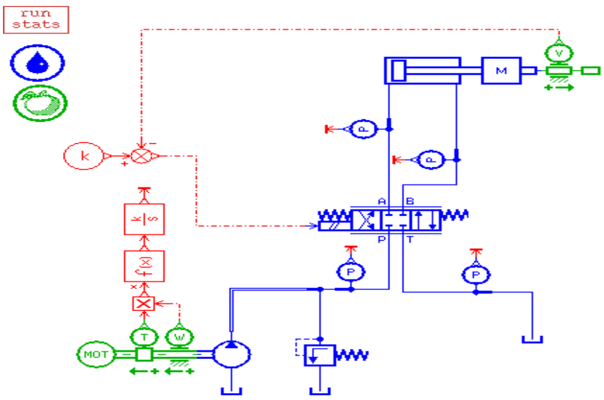

Fig.7: AMESim model of Conventional Hydraulic Circuit

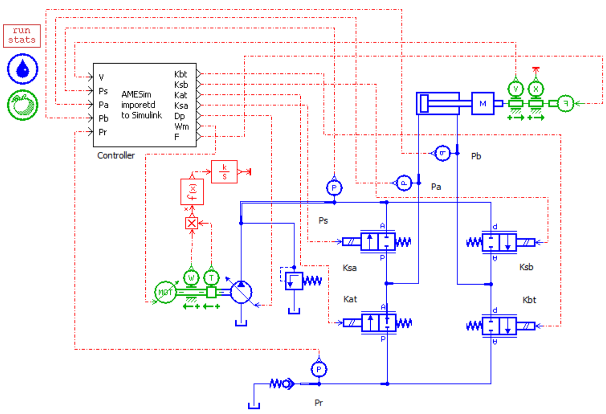

Fig.8: AMESim model of IMV system

Firstly, the simulations were done with the PE and PR modes. The system response as well as energy consumption comparisons are then shown in Fig. 9 and 10. As seen in Fig. 9 and Fig. 10, both the CHC and IMV configurations showed almost similar performances for the PE and PR modes. However, the IMV performances were better than those of the CHC in terms of actuator speed and energy consumption.

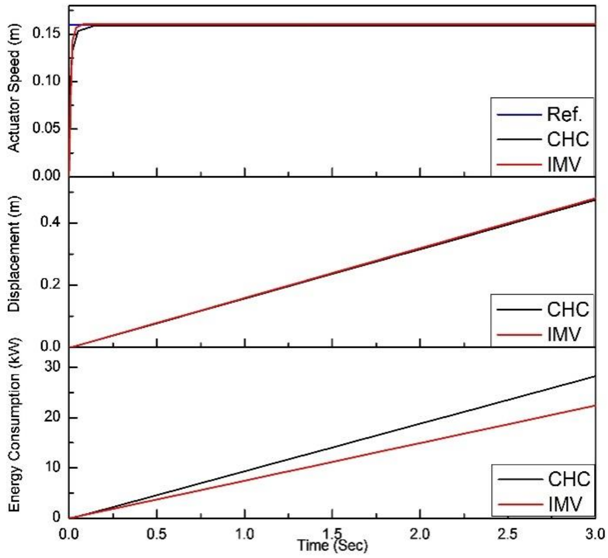

Fig.9: Power Extension Mode

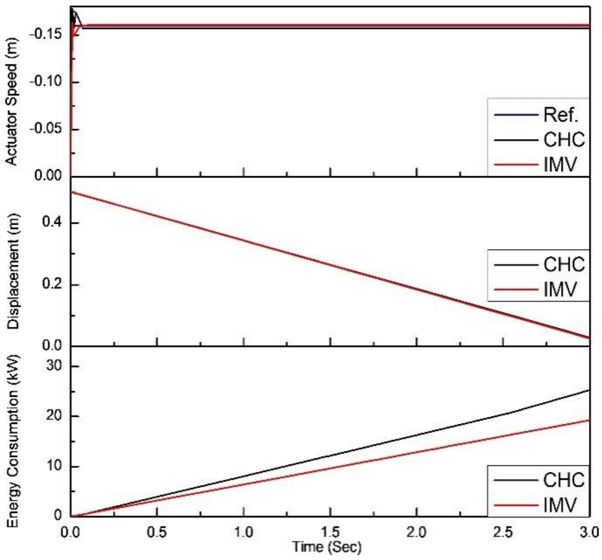

Fig.10: Power Retraction Mode

In the case of HSRE, LSRE and LSRR modes, IMV configuration can perform. CHC system cannot perform this modes. HSRE, LSRE and LSRR modes were shown in Fig. 11, Fig. 12 and Fig. 13, respectively. In HSRE mode, it was consumed low energy rather than PE and PR mode because in oil line, oil did not went to tank. Both line and pump pressure were utilized for cylinder piston extract. Both LSRE and LSRR modes, it required little energy to perform because this modes were worked utilizing the gravitational force. However, the tracking performance were satisfactory level in both mode.

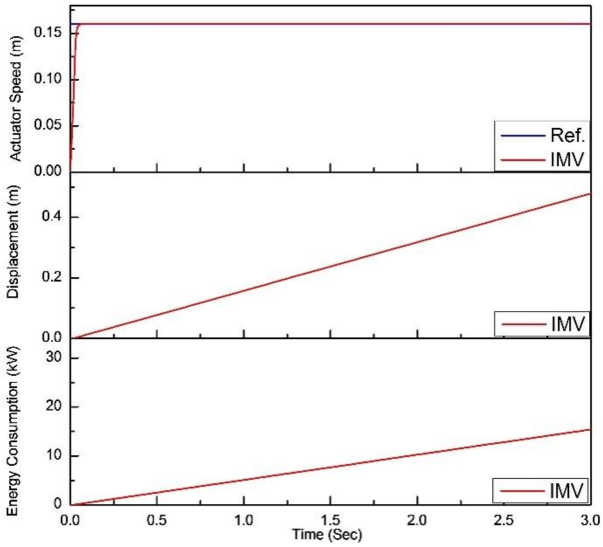

Fig.11: High Side Regeneration Extension Mode

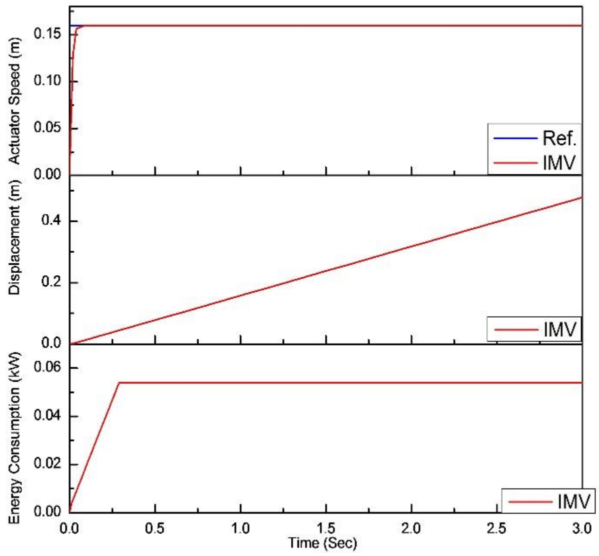

Fig.12: Low Side Regeneration Extension Mode

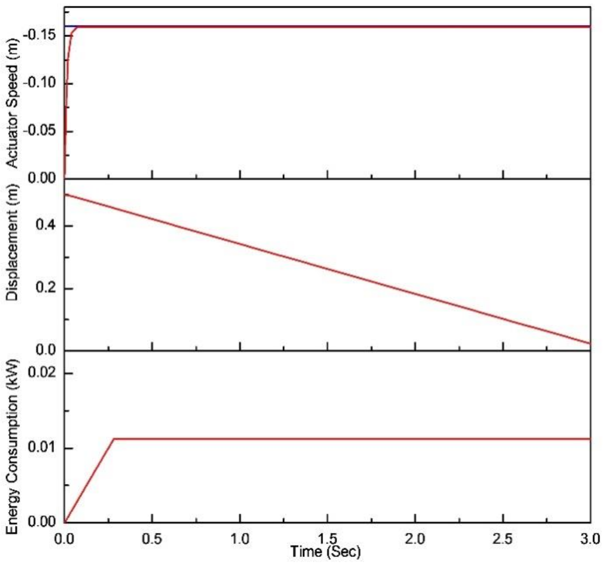

Fig.12: Low Side Regeneration Retraction Mode

Then the energy consumptions were applied to check the energy saving in each case by using the following Eq. (10).

$$\text{Energy saving}(\%) = \frac{E_{CHC} - E_{IMV}}{E_{CHC}} \times 100\% \tag{10}$$

where, E_{CHC} and E_{IMV} are energy consumption of the CHC and IMV configuration respectively.

Finally, The energy consumption comparison in between the CHC and IMV configuration were listed out in Table 2.

Table 2: Energy consumption comparison

Mode	Energy Consumed (kW)		Energy Saved (%)
	E_{CHC}	E_{IMV}	
PE	28.18	22.35	20.69
PR	23.90	18.05	24.48

5. CONCLUSION

In this paper, the four valve independent metering configuration has been investigated including its potential for saving energy. Independent metering combine with control variable displacement pump promised intuitively to be more energy efficient than conventional metering. The four valve independent metering configuration has shown better tracking performance rather than conventional hydraulic circuit. In addition, the paper is proposed using the proportional throttle valve to apply for independent metering valve circuit.

6. REFERENCES

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7. NOMENCLATURE

Symbol	Meaning	Unit
A_A	Head side piston area	(mm^2)
A_B	Rod side piston area	(mm^2)
F	Force	(N)
K_{SA}	Valve conductance of SA valve	(lph/bar$^{1/2}$)
K_{SB}	Valve conductance of SB valve	(lph/bar$^{1/2}$)
K_{AT}	Valve conductance of AT valve	(lph/bar$^{1/2}$)
K_{BT}	Valve conductance of BT valve	(lph/bar$^{1/2}$)
P_A	Pressure at head chamber	(bar)
P_B	Pressure at rod chamber	(bar)
P_S	Pressure at return line	(bar)
P_R	Pressure at Supply line	(bar)
$\dot{x}$	Velocity of piston	(m/s)