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VELOCITY CONTROL OF ELECTRO-HYDRAULIC PUMP CONTROL SYSTEM USING GEAR PUMP

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Abstract. *Two types of velocity control electro-hydraulic systems, valve and pump flow control systems, were extensively investigated in this study. Valve flow control system tested in the study was one of the conventional types with the use of proportional valve and conventional gear pump. The proposed pump flow control system was of the simplest one. An inverter type variable speed drive was used to drive the same gear pump in order to adjust the pump speed and hence the discharge flow rate according to the desired cylinder velocity. Mathematical model of the pump flow control system is presented in the article, and its numerical simulation results were obtained by solving the state space equations. Open loop and proportional-integral (PI) closed loop performances of both valve and proposed pump flow control systems were tested and compared. Velocity tracking performance of valve flow control system either under open loop or PI closed loop control was always better than the proposed pump flow control system. However, the power consumption of the proposed system was much better than the valve flow control system. Response speed in terms of bandwidth frequency of the proposed system under open loop control was less than the valve flow control system by half due to the large inertia of motor-pump rotor. Under PI closed loop control, bandwidth frequency of the proposed system was improved to be 15% less than the valve flow control system.*
Keywords: Electro-hydraulic system, Velocity control, Pump control, Variable speed drive, Gear pump

1. **Introduction.** The electro-hydraulic system (EHS) has been an industrial transmission workhorse from the past due to many advantages, such as high power-weight ratio, ease of use and great reliability. EHS technology has been constantly evolving to this day. Not only improvement in efficiency and controllability of EHS was concerned, but its environmental friendliness and energy-saving have also been key research issues [1,2].

Hydraulic oil flow rate of the EHS could be controlled at the control valve, at the pump or both. The valve control EHS commonly uses fixed displacement pump to generate a constant oil flow rate. The opening area of proportional control valve is adjusted in order to control the hydraulic oil flow rate. EHS valve control systems were studied for controlling position, velocity and force [3-7]. For the valve flow control system, not all the oil flows through the proportional valve. The rest of the oil would return to tank via relief valve. This makes the system operate at the oil relief pressure; hence, power consumption is kept at maximum at all times.

Hydraulic pump is directly controlled to discharge oil flow rate as needed in pump control EHS. This leads to a better energy consumption of pump control EHS when

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compared with valve control EHS. However, the drawbacks of pump control EHS are slower response and less precision. Therefore, the performance improvement of pump control EHS has been studied by several researchers recently [8-16]. The work in [8] presented a pump control EHS with the use of load sensing cylinder to control pump swash plate in order to control oil flow rate. Proportional valve was also used to direct the flow and further adjust the flow rate. A similar pump control EHS was proposed in [9]. It differs from [8] in that instead of load sensing cylinder, a stepping motor is used to control pump swash plate. A hybrid force control EHS force control for pressing machine with bidirectional piston pump driven by alternating current (AC) servo motor is proposed in [10,11]. In addition to AC servo bidirectional pump, an auxiliary pump was used in [12], an accumulator was used in [13], in order to compensate the oil flow.

Drawbacks of the pump control EHS with variable displacement pump or bidirectional pump [8-13] are the high costs of the equipment and the complexity of the circuits. Alternatively, pump control EHS could also be constructed with a lower costed fixed displacement pump. In order to be able to adjust the discharge flow rate from a fixed displacement pump, variable speed electric motor would be used to drive the pump [14-16]. A pump control EHS with the use of fixed displacement pump and proportional valve was presented in [14]. Auxiliary devices were proposed to add to the fixed displacement pump control EHS. [15] added an accumulator to the system in order to improve acceleration response. Proportional relief valve was added in [16] to reduce energy loss.

This paper proposes a simple low cost pump control EHS. The pump used in the proposed system is a conventional gear pump, and is driven by an inverter type variable speed drive. Instead of proportional valve, a simple 4/3 directional control valve is used to direct the oil flow. The cost of the proposed system in study is far lower than pump control EHSs explored in other studies [8-16]. Mathematical model and numerical simulation of the proposed system are also presented. Both open loop and closed loop performances of the proposed system and the valve control EHS would be experimentally tested and compared. Frequency responses of both proposed and valve control systems would be experimentally conducted in order to understand the limitation on response speeds of both systems.

Nomenclature

A_{HE}: cross sectional area of cylinder head end
A_{RE}: cross sectional area of cylinder rod end
A_{PA}: cross sectional area of P-A port
A_{BT}: cross sectional area of B-T port
A_{PT}: cross sectional area of P-T port
C_d: orifice discharge coefficient
C: damping value
L_{stroke}: total cylinder movement
m_{piston}: mass of piston
P_T: ambient pressure
Q_{PAHE}: flow from pump to cylinder head end
Q_{BTRE}: flow from cylinder rod end to tank
Q_{PT}: flow from pump to tank
V_{HE}: initial fluid volume in cylinder head end
V_{hose}: initial fluid volume in hydraulic hose
V_{RE}: initial fluid volume in cylinder rod end
$x_1 = x$: piston displacement
$x_2 = \dot{x}$: piston velocity
$x_3 = P_{HE}$: piston head end pressure
$x_4 = P_{RE}$: piston rod end pressure
$x_5 = P_P$: pump pressure
$x_6 = N$: motor speed
β: hydraulic fluid bulk modulus
ω_P: volume displacement of pump

2. **Flow Control Systems.** Two types of velocity control EHSs are studied and compared experimentally in this study: valve flow control and pump flow control systems.

2.1. **Valve flow control system.** The valve flow control system used in this study (Figure 1) is one of the conventional EHSs. The system utilizes a fixed displacement gear pump driven by a fixed speed electrical motor. The control signal from the PC controller calculated based on the measuring data is sent via D/A card to the proportional control valve. The opening area of the proportional control valve is adjusted according to the control signal, metering the oil flow rate through it. The response time of 0-100% spool displacement of proportional control valve according to the valve manufacturer is 50 ms [17].

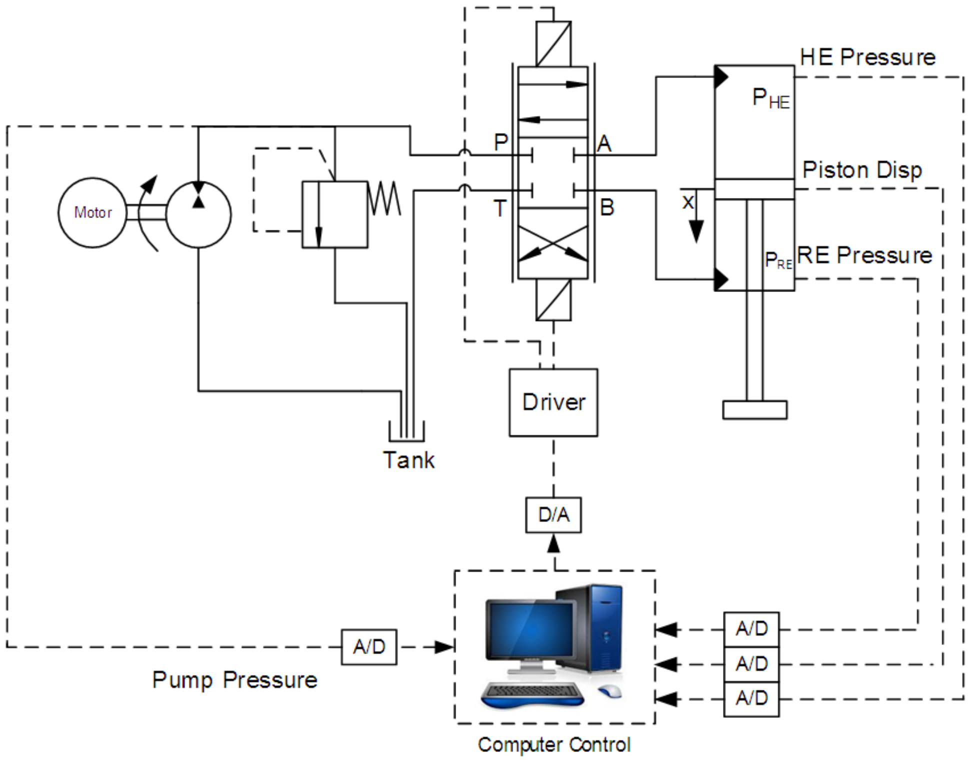


FIGURE 1. Schematic diagram of the valve flow control system

2.2. **Pump flow control system.** Figure 2 shows the schematic diagram of the proposed pump flow control system. The proposed pump flow control system is of the simplest one, with the use of the same fixed displacement gear pump used in the valve control system. The pump is driven by the same electrical motor used in the valve control system. However, the electrical motor speed is controlled by an inverter type variable speed drive (VSD). A 4/3 directional control valve is used to direct the oil flow to either the head-end (HE) side or the rod-end (RE) side of the cylinder according to the velocity command. At the cylinder sudden stop command, the downstroke of directional control valve is faster than pump due to the valve's smaller inertia, and the valve tends to completely shut off before the pump does. Open center type directional control valve is then used (Figure 2) in order that the oil could bypass into tank during the sudden shutoff; thus, oil pressure surge could be avoided. The directional control valve utilized in pump flow control system, unlike the proportional control valve in the valve flow control system, cannot meter the flow. Therefore, additional pilot operated check valves are placed at both sides of the cylinder (Figure 2) in order to meter or restrict the oil flow during the downward motion of the cylinder. This act could prevent gravitational overrunning motion. The control signals calculated based on the measuring data are sent to both the

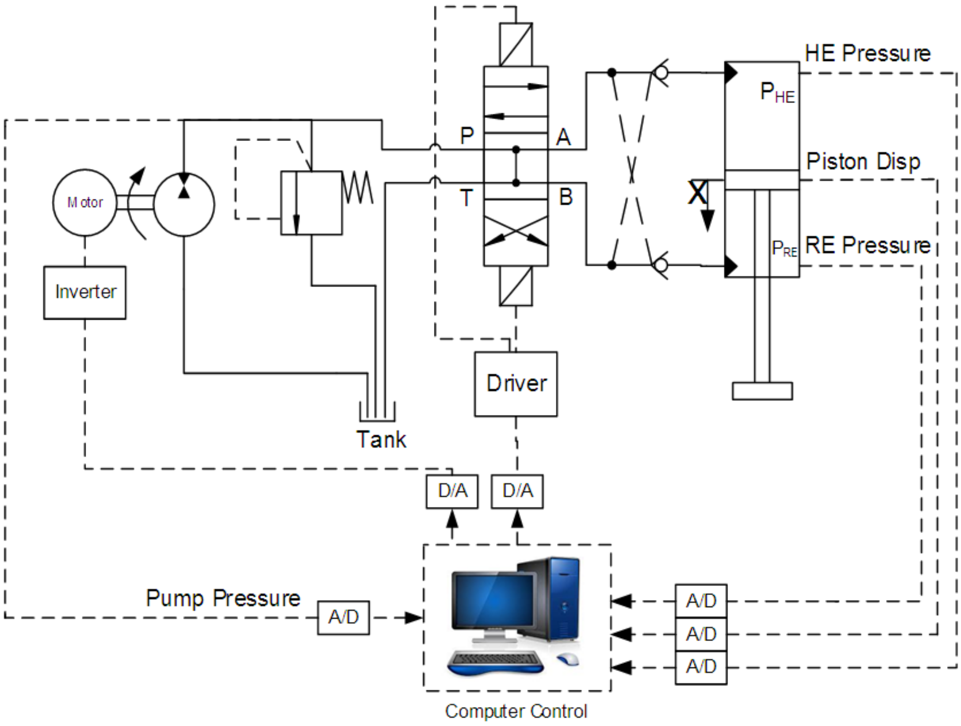


FIGURE 2. Schematic diagram of the proposed pump flow control system

electrical motor variable speed drive and the direction control valve. The response time of 0-100% spool displacement of direction control valve according to the valve manufacturer is 60 ms [18]. The combined cost of VSD, directional control valve and check valves, used in the pump flow control system (Figure 2) should always be lower than the comparable proportional valve and its driver used in the valve flow control system (Figure 1). Thus, the total cost of the proposed pump flow control system is lower than the conventional valve flow control system.

For both valve and proposed pump flow control systems, the piston position, pump pressure, and cylinder HE and RE pressures are measured and sent to the computer via the National Instrument 6221 data acquisition card. The velocity feedback signal is obtained by numerically differentiating the piston position measured by a draw wire potentiometer. The sampling rate implemented in both control systems was 100 Hz. The fixed displacement gear pump used in both control systems has a volumetric displacement of 11 cm^3/rev, and discharges a maximum flow rate of 260 cm^3/s. The specifications of EHS components used in both systems are shown in Table 1.

3. **Mathematical Modeling of the EHS.** This topic discusses the mathematical model of the proposed pump flow control system. The mathematical model comprises the modeling of piston dynamics, electrical motor dynamics, and cylinder pressure dynamics. The detail can be explained as follow.

The motion of the piston is described by the Newton's second law equation as shown in Equation (1).

$$\ddot{x} = \frac{1}{m_{piston}} \left(P_{HE}A_{HE} - P_{RE}A_{RE} - C\dot{x}\right) \tag{1}$$

For the ease of modeling, the dynamics of induction motor and pump driven by variable speed drive can be modeled as a first order system. The pump revolving speed, N, is the function of the inverter frequency input, and is obtained from Equation (2). The values

TABLE 1. The specifications of the EHS components

Component	Specification	
Variable speed drive	Manufacturer:	Toshiba: VF-S11-3PH-380V
	Power:	5.5 Kw
Electrical motor	Manufacturer:	Mitsubishi: SF-JR 2HP 4P
	Power:	1.5 Kw
	Maximum speed:	1450 rpm
Hydraulic pump	Manufacturer:	Honor: 2GG1U11R
	Type:	Gear Pump
	Volumetric displacement:	11 cc/rev
Proportional valve	Manufacturer:	Tokimec: COM-3-2C-AN-11
	Type:	4/3 closed center
	Time response:	50 ms
Direction control valve	Manufacturer:	Vickers: DG4V-3S-0C-M-U-H5-60
	Type:	4/3 open center
	Time response:	60 ms
Position sensor	Manufacturer:	Penny Giles: DLS-750-P60-CR-P
	Type:	Draw wire potentiometer
Pressure sensor	Manufacturer:	Wika: A-10
	Type:	Piezo-electric

of constant gain, K_m, and time constant, T_m, were obtained from the experiment to be 29 and 0.2, respectively.

$$N = \frac{K_m}{T_m s + 1} f_{in} \tag{2}$$

The volumetric efficiency of the pump is not considered in the modeling. Therefore, the pump discharge flow, Q_P, is assumed to be proportional to the driven speed as shown in Equation (3).

$$Q_P = \omega_P N \tag{3}$$

The flow from pump to cylinder head end, Q_{PAHE}, via valve port A and the flow from cylinder rod end to tank, Q_{BTRE}, via valve port B equations are derived from orifice flow equations as shown in Equations (4) and (5).

$$Q_{PAHE} = \left(C_d\sqrt{\frac{2}{\rho}}\right) A_{PA}\sqrt{P_P - P_{HE}} \tag{4}$$

$$Q_{BTRE} = \left(C_d\sqrt{\frac{2}{\rho}}\right) A_{BT}\sqrt{P_{RE} - P_T} \tag{5}$$

The flow return to tank through the relieve valve occurs only when the value of pump pressure, P_P, is higher than the relieve valve cracking pressure, P_{cr}, as described in Equations (6) and (7).

$$Q_{PT} = 0 \text{ if } P_P < P_{cr} \tag{6}$$

$$Q_{PT} = \left(C_d\sqrt{\frac{2}{\rho}}\right) A_{PT}\sqrt{P_P - P_T} \quad \text{if } P_P > P_{cr} \tag{7}$$

The compression of the hydraulic oil causes the changes in pressure at pump discharge and in the cylinder. The changes of pressures are described by the bulk modulus equations, Equations (8) to (10).

$$Q_P = Q_{PAHE} + Q_{PT} + \frac{V_{hose}}{\beta}\dot{P}_P \tag{8}$$

$$Q_{PAHE} = A_{HE}\dot{x} + \frac{(V_{HE} + A_{HE}x)}{\beta}\dot{P}_{HE} \tag{9}$$

$$Q_{BTRE} = A_{RE}\dot{x} - \frac{(V_{RE} + A_{RE}(L_{stroke} - x))}{\beta}\dot{P}_{RE} \tag{10}$$

Equations (1)-(10) can be rewritten in the state space form as shown in Equations (11) to (19). The state equations explaining the changes in pump revolving speed and cylinder motion are shown in Equations (11)-(13).

$$\dot{x}_6 = -5x_6 + 145f_{in} \tag{11}$$

$$\dot{x}_1 = x_2 \tag{12}$$

$$\dot{x}_2 = \frac{1}{m_{piston}}(x_3A_{HE} - x_4A_{RE} - Cx_2) \tag{13}$$

Because of the different flow directions during the extension and retraction of the cylinder, the state equations explaining the changes in oil pressures are different for both cases, and are shown in Equations (14) to (19). The values of parameters used in the model, Equations (11) to (19), are shown in Table 2.

Extension:

$$\dot{x}_3 = \frac{\beta}{(V_{HE} + A_{HE}x_1)}\left(\left(C_d\sqrt{\frac{2}{\rho}}\right)A_{PA}\sqrt{x_5 - x_3} - A_{HE}x_2\right) \tag{14}$$

$$\dot{x}_4 = \frac{\beta}{(V_{RE} + A_{RE}(L_{stroke} - x_1))}\left(-\left(C_d\sqrt{\frac{2}{\rho}}\right)A_{BT}\sqrt{x_4 - P_T} + A_{RE}x_2\right) \tag{15}$$

$$\dot{x}_5 = \frac{\beta}{V_{hose}}\left(-\left(C_d\sqrt{\frac{2}{\rho}}\right)A_{PT}\sqrt{x_5 - P_T} - \left(C_d\sqrt{\frac{2}{\rho}}\right)A_{PA}\sqrt{x_5 - x_3} + \omega_P x_6\right) \tag{16}$$

Retraction:

$$\dot{x}_3 = \frac{\beta}{(V_{HE} + A_{HE}x_1)}\left(\left(-C_d\sqrt{\frac{2}{\rho}}\right)A_{AT}\sqrt{x_3 - P_T} - A_{HE}x_2\right) \tag{17}$$

$$\dot{x}_4 = \frac{\beta}{(V_{RE} + A_{RE}(L_{stroke} - x_1))}\left(\left(C_d\sqrt{\frac{2}{\rho}}\right)A_{PB}\sqrt{x_5 - x_4} + A_{RE}x_2\right) \tag{18}$$

$$\dot{x}_5 = \frac{\beta}{V_{hose}}\left(-\left(C_d\sqrt{\frac{2}{\rho}}\right)A_{PT}\sqrt{x_5 - P_T} - \left(C_d\sqrt{\frac{2}{\rho}}\right)A_{PB}\sqrt{x_5 - x_4} + \omega_P x_6\right) \tag{19}$$

Table 2. The EHS parameters used in the mathematical model

Parameters	Value	Unit
Pump volumetric displacement	11	cc/rev
Cylinder stroke	25	cm
Piston diameter	40	mm
Rod diameter	28	mm
Piston mass	2	kg
Relief valve cracking pressure	30	bar
Effective bulk modulus of hydraulic oil	700	MPa
Density of hydraulic oil	850	kg/m^3
Damping coefficient	500	kg/s
Orifice discharge coefficient	0.62	–
Initial fluid volume in hydraulic hose	1.28×10^{-5}	m^3

Table 2 shows the values of the parameters used in the mathematical model. They were obtained from equipment data sheet and direct measurements.

4. **Controller Design.** Open loop and PI closed loop controls are tested and compared in this study. The details of both controllers are explained as follows.

4.1. **Open loop control.** Figure 3 shows the block diagram of open loop control for both valve and pump flow control systems. Open loop valve flow control system uses inverse valve modulation, whereas open loop pump flow control system uses inverse pump modulation.

The valve or pump modulation is the static relationship between the valve or pump control command to the EHS output. For velocity control EHS, the EHS output could be either oil flow rate or the cylinder velocity.

Figure 4 shows the valve modulation with oil flow rate as the EHS output. To obtain the valve modulation, various constant voltage inputs were sent to the proportional valve in the valve flow control system. The pump which operates at constant speed discharges a constant flow rate. The valve port areas vary according to the varying valve control inputs. Corresponding steady state oil flow rate to each valve control input was then recorded, and the relationship is constructed. The oil flow rate when divided by cross sectional area of cylinder head end or rod end can be interpreted as cylinder extension or retraction velocity. The inverse valve modulation which is used to convert cylinder velocity back to the valve control command could then be obtained from Figure 4.

Figure 5 shows the pump modulation with oil flow rate as the EHS output. In the same procedure of obtaining valve modulation, various constant voltage inputs were sent to the inverter variable speed drive in the pump flow control system. Corresponding oil flow rate to each pump control input was recorded, and the relationship is constructed.

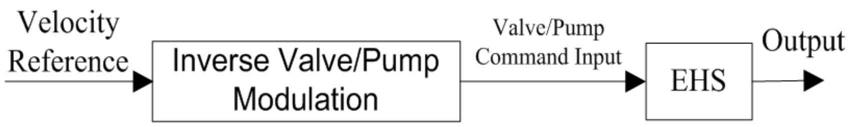


FIGURE 3. Block diagram of open loop control system

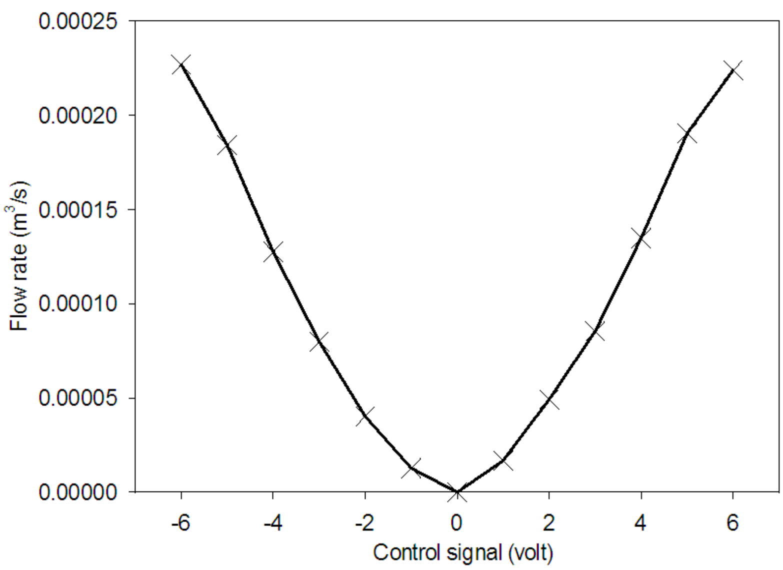


FIGURE 4. Valve modulation between valve control signal versus oil flow rate

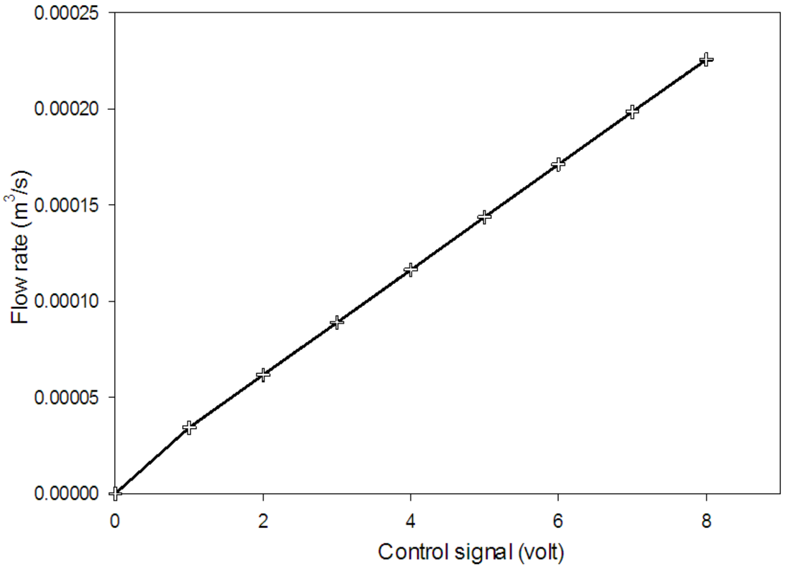


FIGURE 5. Pump modulation between pump control signal versus oil flow rate

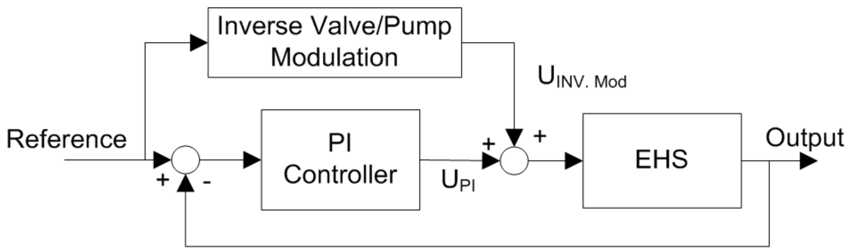


FIGURE 6. Block diagram of PI closed loop control system

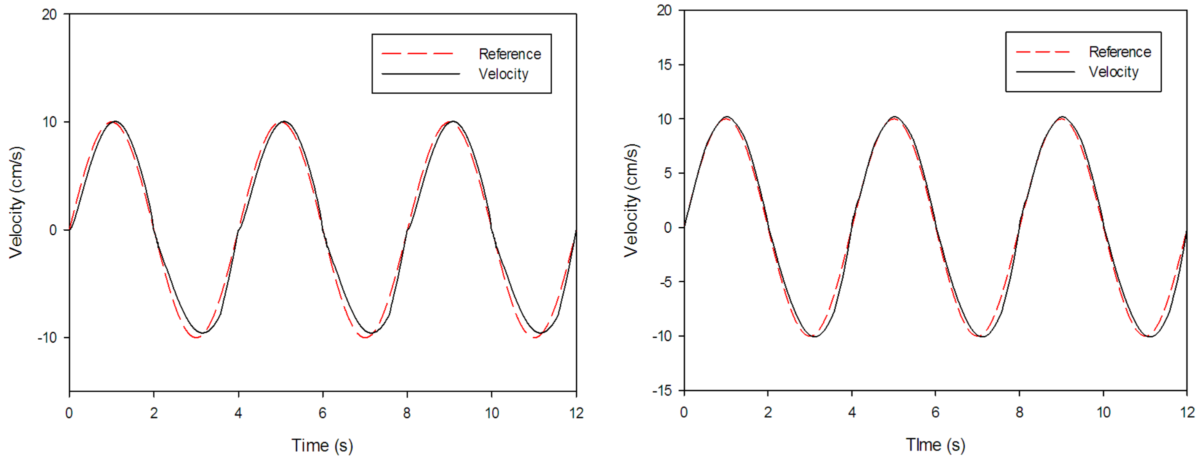
Both valve and pump inverse modulations were modeled as lookup tables in the control code.

4.2. **PI closed loop controller.** Figure 6 shows the block diagram of PI closed loop control system. The inverse valve or pump modulation previously used in the open loop control system is used in the feedforward loop of the closed loop control system. Tracking performance is improved upon using feedback PI closed loop control. The total control action is then the combination of the PI and inverse modulation outputs, as shown in Equation (20).

$$u(t) = K_P e(t) + K_I \int e(t)dt + U_{Inverse\ Modulation} \tag{20}$$

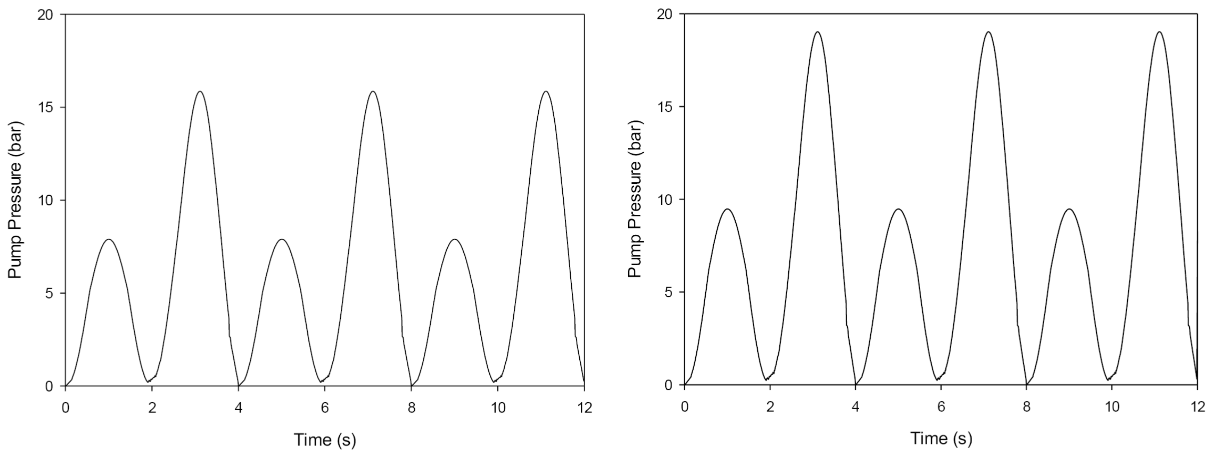
where $e(t)$ is the velocity tracking error. In this study, the PI gains, K_P, and K_I were tuned experimentally by using the Ziegler-Nichols method. For valve flow control system, the control signal is sent to the proportional control valve. While for the pump flow control system, the control signal is sent to both VSD and direction control valve.

5. **Simulation Results.** This topic explains only the simulation results of pump flow control system. The numerical simulations of the valve control system could be found in several articles such as [19]. The simulation results of the proposed pump control system were obtained by numerically solving the state space equations (Equations (13)-(21)). Fourth order Rung-Kutta integration method was used to calculate the solutions of the state space equations.



(a) (b)

FIGURE 7. Tracking response simulation of the proposed pump flow control system under (a) open loop control and (b) PI closed loop control



(a) (b)

FIGURE 8. Pump pressure simulation of the proposed pump flow control system under (a) open loop control and (b) PI closed loop control

Figure 7 shows the computer simulation of sinusoidal velocity tracking responses of the pump flow control system. The inverse pump modulation obtained from the actual experiments (Figure 5) was used in the computer simulation. Figure 7(a) shows the tracking response of open loop pump flow control system. Figure 7(b) shows the velocity response of PI closed loop control. PI controller could reduce root mean square (RMS) velocity tracking error by almost half, from a value of 1.14 cm/s in case of open loop to a value of 0.61 cm/s.

Figure 8 shows the pump pressures obtained from the same computer simulation as of Figure 7. The pump pressures vary periodically according to desired velocity. The pressure during retraction is higher than one during extension because of different piston areas of both sides. For the PI closed loop control (Figure 8(b)), the pump pressure varies in the same trend as the open loop control (Figure 8(a)) at a little higher value due to extra control effort output by PI controller.

6. **Experimental Result.** The motor speed was kept constant at 1450 rpm for the valve flow control system, and was varied between 0 to 1450 rpm according to the control scheme for the pump flow control system. The key performance indices (KPIs) to be compared between both control systems are RMS velocity tracking error and power consumption. PI gains of the PI closed loop control were tuned experimentally by the Ziegler-Nichols method. The results in this study are divided into 2 cases: open loop control and PI closed loop control.

The dynamics of pump flow control system largely depends on the dynamics of the motor-pump because of the large inertia of the rotating combined elements. Figure 9 shows the pump speed open loop step response of the pump flow control system. The pump discharge flow was diverted back to the tank in this test; thus, the hydraulic system was unloaded. Constant control signals at the values of half and full scale VSD commands were sent to the VSD, and the pump speeds were recorded. Delay times of 0.05 second could be observed in both responses. The pump speed response has a 10%-90% rise time of approximately 0.22 second which is roughly equivalent to a natural frequency and a bandwidth frequency between 1.8-2.5 Hz at damping ratio between 0.7-1 according to any classical control textbooks. The topic of system bandwidth will be lately discussed.

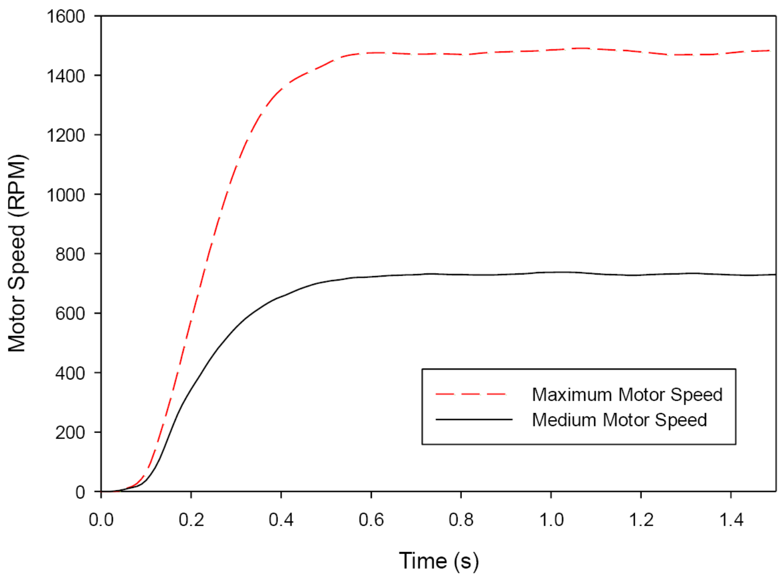


Figure 9. Pump speed open loop step response of the proposed pump flow control system

Figure 10 shows the comparison of velocity open loop step responses of valve and pump flow control systems. The response of valve flow control system was faster than pump flow control system. Delay time and rise time of 0.05 and 0.01 seconds were observed in valve flow control system, whereas delay time and rise time of pump flow control system were 0.15 and 0.28 seconds, respectively. For valve flow control system, pump operated at constant speed discharging a constant oil flow rate during idling, proportional control valve then reacted to the constant step command received at time zero. On the other hand, the pump was at rest during idling for the pump flow control system. The much larger inertia of motor-pump rotor in pump flow control system compared with the inertia of valve in valve flow control system is the reason of the much slower step response of the pump flow control system.

Figure 11 shows open loop square input responses of both valve and pump flow control systems. Pump discharge pressure is also shown in the same figure. For the valve flow

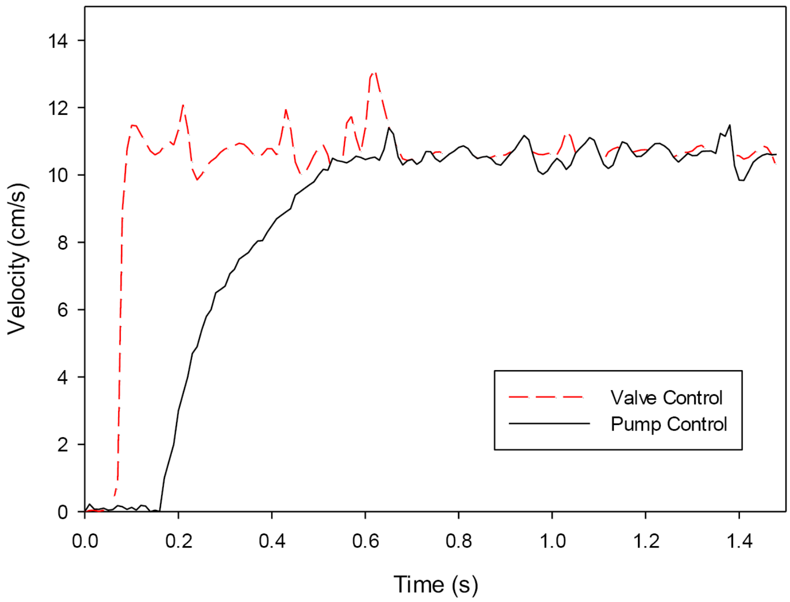
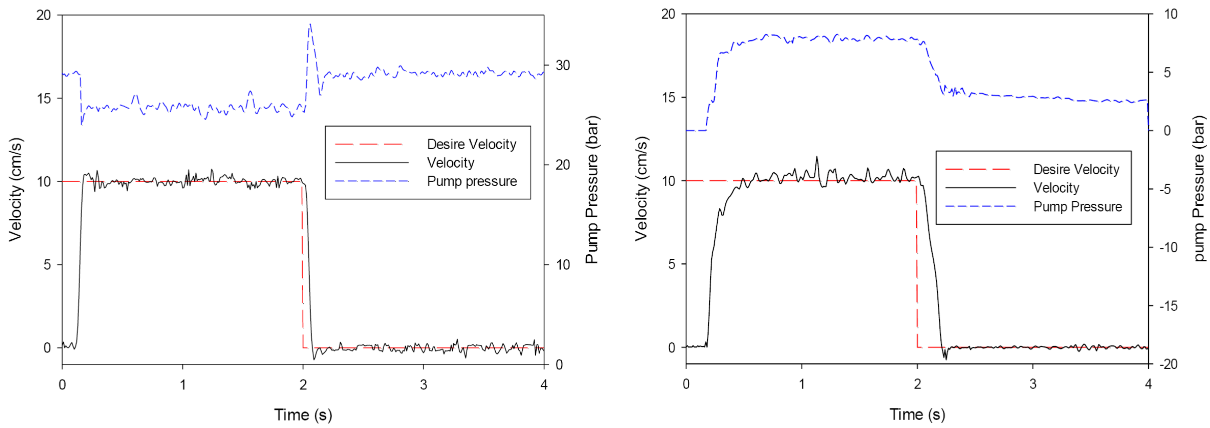


FIGURE 10. Comparison of velocity open loop step responses of both systems



(a) (b)

FIGURE 11. Open loop square input responses of (a) valve flow control system and (b) proposed pump flow control system

control system (Figure 11(a)), oil pressure serge above relief pressure could be observed at the time 2 second, the same instant of valve shuts off, due to the compression of oil before the relief valve is opened. Such surge in oil pressure could not be observed in the case of pump flow control system (Figure 11(b)). The reason, as mentioned earlier in Chapter 2, is the use of open center type directional control valve in the pump flow control system (Figure 2). Oil could flow back to tank freely at the valve shutoff instant. The pump pressure at the time 2 second of pump flow control system did not suddenly come down to zero even the pump command was reduced to zero (Figure 11(b)). The revolving motion of the pump did not rapidly stop at the zero pump command due to the inertia effect of the motor-pump rotor. The pump still continually discharged the oil flow before it completely stopped. The discharged oil flew back to tank via the port opened at center of the directional control valve. Pump pressure after the time 2 second was in the decline trend (Figure 11(b)). It would be a while before the pump stopped and the pump pressure would come down to zero.

Figure 12 shows the open loop sinusoidal velocity tracking responses of both valve and pump flow control systems. Delay of 0.17 second could also be observed in the pump flow control system as in the case of open loop step response (Figure 10). Delay in tracking could not be observed in simulation (Figure 7) since the delay was not included in the mathematical model (Equations (11)-(19)). The RMS velocity tracking errors of valve and pump flow control systems are 1.27 cm/s and 2.40 cm/s, respectively.

Figure 13(a) shows the power consumption of open loop valve control system that was relatively constant at approximately 2302.82 W. The pump pressure was relatively constant at 30 bar, at the relief pressure (Figure 13(a)). For the valve flow control system, the pump outputs a constant flow rate at all the time since the motor speed is kept constant. Not all the oil pump flow would pass through the proportional control valve, the rest of the oil pump flow would divert back to tank through the relief valve. That caused the system pressure to be relatively constant at the relief pressure, and the power consumption to be constant at its maximum value. Figure 13(b) shows the power consumption of pump flow control system that varied periodically according to velocity response and had the average value of 132.41 W. The pump pressure periodically varied

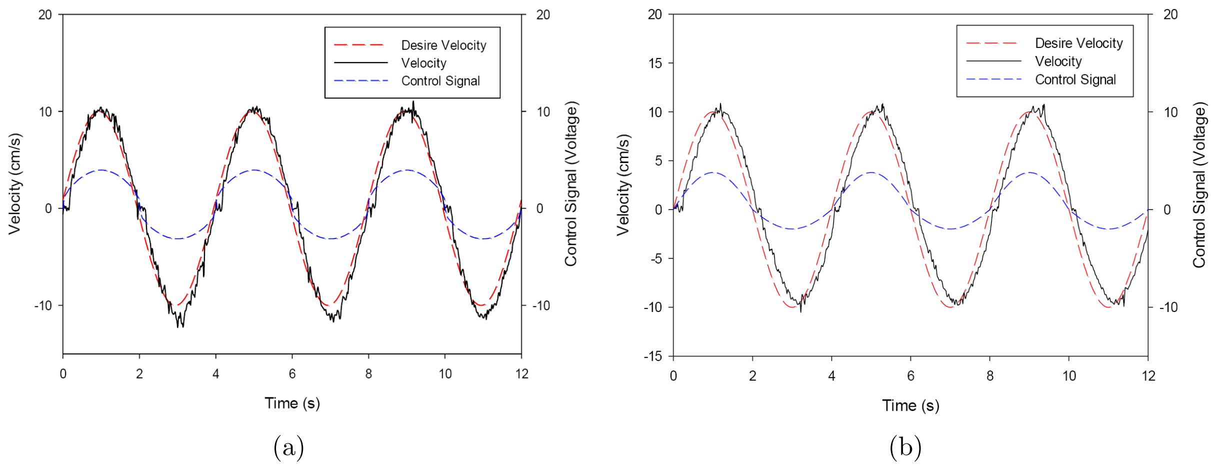


FIGURE 12. Open loop sinusoidal tracking responses of (a) valve flow control system and (b) proposed pump flow control system

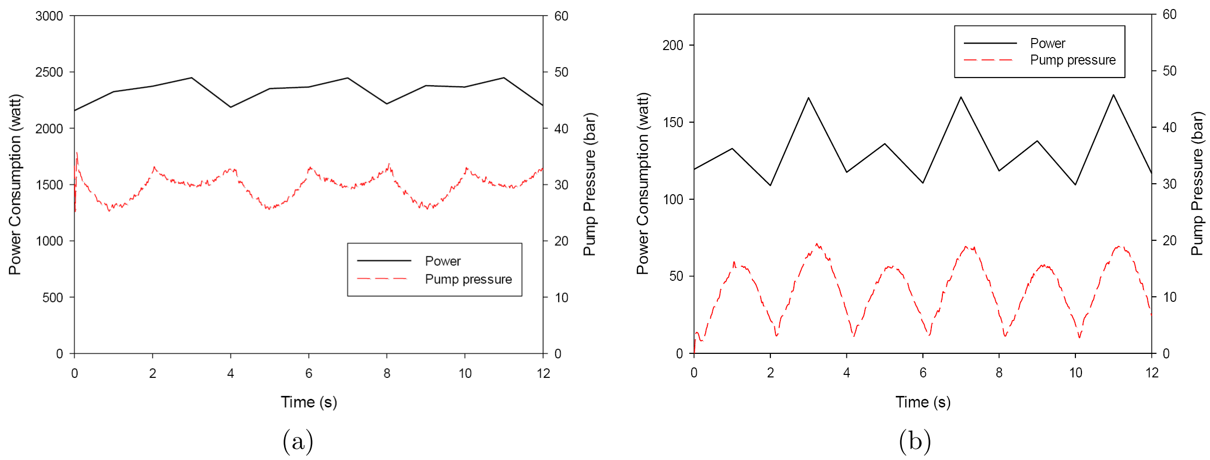
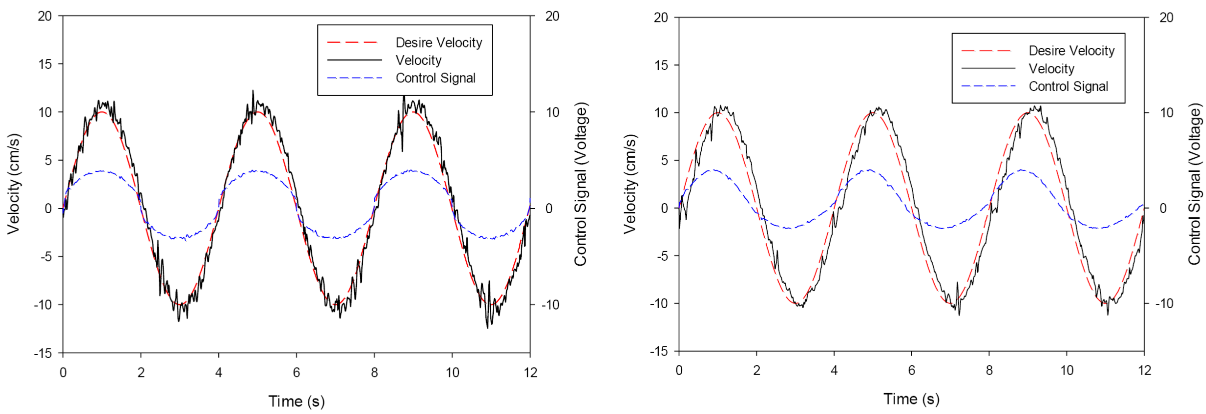


FIGURE 13. Open loop power consumptions and pump pressures of (a) valve flow control system and (b) proposed pump flow control system

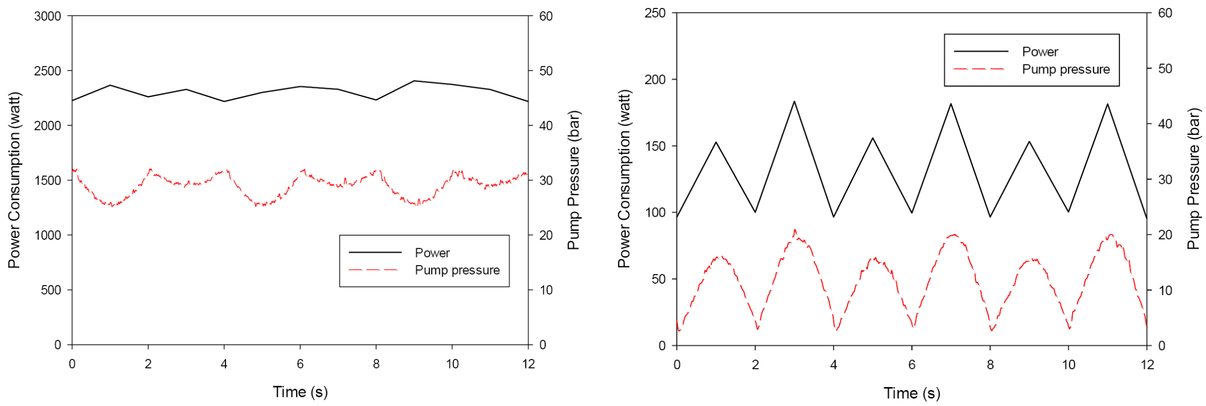
between 1 and 20 bar (Figure 13(b)). For the pump flow control system, pump discharged only enough oil flow rate according to the velocity command, no excess oil would have to divert back to tank through the relief valve. The system pressure was then at the value that is enough for moving the load, not at the relief pressure. The maximum values of the pump pressure obtained from the experiment (Figure 13(b)) were a little higher than ones obtained from the simulation (Figure 8(a)), due to the uncertainties in system parameters. The other reason is that the model of the pilot operated check valves (Figure 2) was not included in the mathematical model (Equations (13)-(22)).

Figure 14 shows closed loop sinusoidal velocity tracking responses of valve flow control and pump flow control systems. The PI controller gains were tuned experimentally using the Ziegler-Nichols method. The controller gains of the valve flow control and pump flow control systems were found to be $K_P = 0.01$, $K_I = 0.0001$, and $K_P = 0.02$, $K_I = 0.00025$, respectively. Closed loop control improved the tracking performance of the pump flow control substantially. RMS tracking error was reduced by 40%, and the tracking delay was reduced to be 0.13 second. RMS velocity tracking errors of valve flow control and pump flow control systems were 1.18 cm/s and 1.45 cm/s, respectively.



(a) (b)

FIGURE 14. PI closed loop control sinusoidal tracking responses of (a) valve flow control system (b) proposed pump flow control system



(a) (b)

FIGURE 15. PI closed loop control power consumptions and pump pressures of (a) valve flow control system and (b) proposed pump flow control system

Figure 15 shows the power consumptions and the pump pressures of both control systems. The same trends could be observed as the open loop case (Figure 13), with little higher values of the power consumptions and pump pressures. The average power consumptions of valve and pump flow control systems were 2328.07 W and 132.48 W, respectively. Table 3 shows the comparison summary of the performance KPIs achieved in all cases of experiments. While tracking performance of the pump flow control system was not up to that of the valve flow control system, its power consumption was better by a big margin (Table 3).

Table 3. Comparison of RMS velocity tracking errors and power consumptions

Experiment	RMS (cm/s)	Power (watt)
Open loop valve flow control	1.27	2302.82
Open loop pump flow control	2.40	132.41
PI closed loop valve flow control	1.18	2328.07
PI closed loop pump flow control	1.45	132.48

7. **The Frequency Response of the EHS.** In order to better understand the limitations of tracking performances of both valve and the proposed pump flow control systems in terms of response speed, frequency responses of both systems were tested. Gaussian noises at a sampling rate of 50 Hz were used as input signals. For valve flow control system, the excitation signal was sent to the proportional valve. For pump flow control system, the excitation signal was sent to electric motor VSD and to directional control valve in order to alter both the oil flow rate and the oil flow direction. The velocity responses of both systems were measured. The transfer function in frequency domain can be calculated from Equation (21).

$$G(j\omega) = \frac{Y(j\omega)}{U(j\omega)} \tag{21}$$

where $Y(jw)$ and $U(jw)$ are the Fourier spectrums of velocity output and valve or pump input signals, respectively. Frequency responses of both open loop and closed loop systems are tested and shown below.

Figure 16 shows frequency responses of open loop valve flow control (Figure 16(a)) and pump flow control (Figure 16(b)) systems. Valve flow control system has a higher bandwidth compared with pump flow control system due to the faster time response of the smaller inertia of proportional valve. The bandwidths of valve and pump flow control systems were approximately 5 Hz and 2.5 Hz, respectively. The dynamics of the open loop pump flow control system is limited by the dynamics of the combined motor-pump components. This limitation in terms of bandwidth could also be realized in time domain by implementing step response test as explained early in Figure 9.

Figure 17 shows the open loop sinusoidal velocity response of both systems at the critical bandwidth frequencies. The velocity amplitudes of both experimental cases were about 7 cm/s which is −3 dB smaller than the 10 cm/s amplitude of the reference signal.

Figure 18 shows the closed loop frequency responses of both systems. Closed loop control improved the bandwidth of pump flow control significantly. The bandwidths of closed loop valve and pump flow control systems were approximately 6.5 Hz and 5.5 Hz, respectively. Figure 19 shows the closed loop sinusoidal velocity response of both systems at the critical bandwidth frequencies. The reductions of velocity tracking amplitudes confirm the value of bandwidths of both systems.

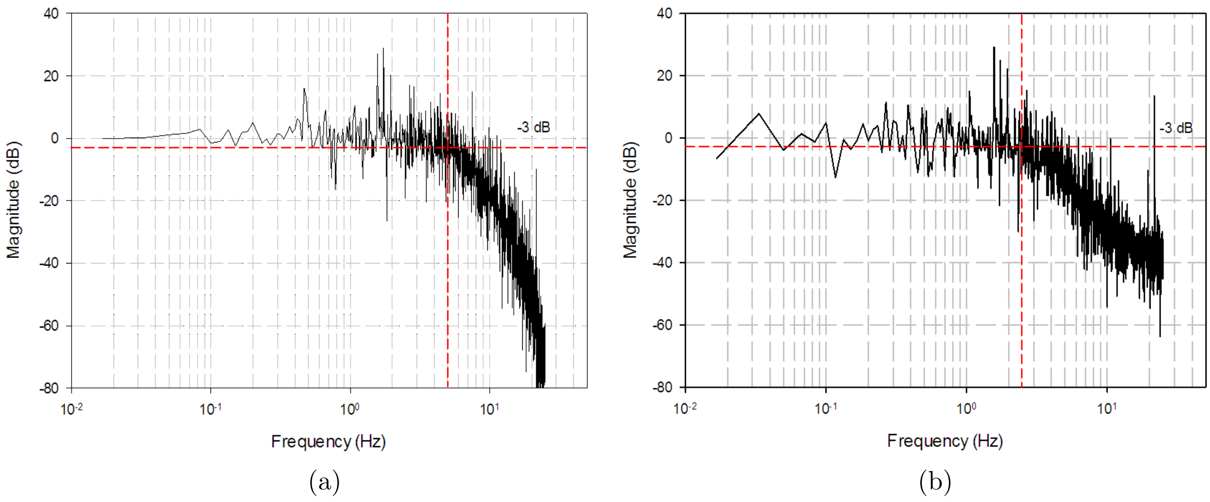


FIGURE 16. Open loop frequency responses of (a) valve flow control system and (b) proposed pump flow control system

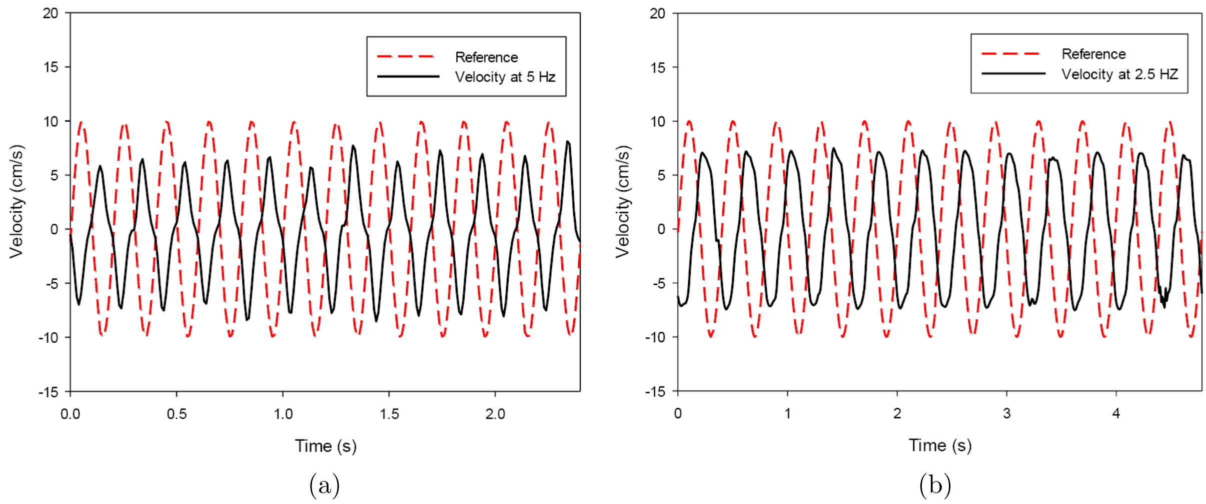


FIGURE 17. Open loop velocity responses at critical bandwidth frequencies of (a) valve flow control system and (b) proposed pump flow control system

8. **Conclusion.** The main contribution of this paper is the proposal of a low cost pump flow control system. Mathematical modeling and performance tests of the proposed low cost pump flow control system were deeply investigated in this article. The results from computer simulations agreed with the actual experiments. The conclusions of the study can be summarized as follows.

1) A simple pump flow control system could be constructed with the use of conventional gear pump, inverter type variable speed drive and a 4/3 directional control valve. Open center type directional control valve should be used in order to avoid oil pressure surge during cylinder sudden stop command. The total cost of the proposed pump flow control system is less than a conventional valve flow control system, and is much less than a pump control system that utilizes variable displacement pump.

2) The dynamics or the bandwidth of the open loop pump flow control system is limited by the dynamics of the combined motor-pump element due to its large inertia. The bandwidth could be vastly improved by implementing closed loop control. The bandwidth

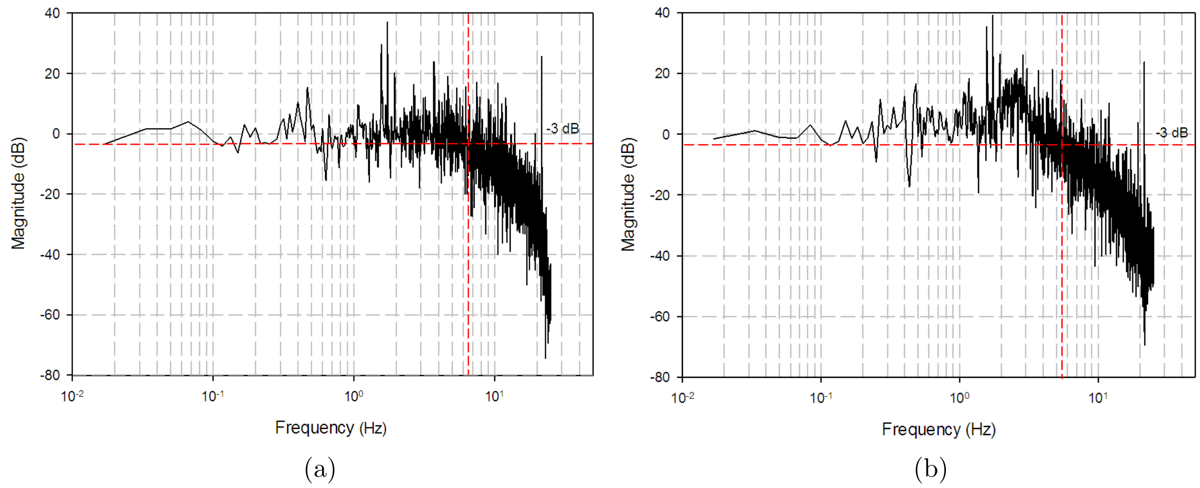


FIGURE 18. Closed loop frequency responses of (a) valve flow control system and (b) proposed pump flow control system

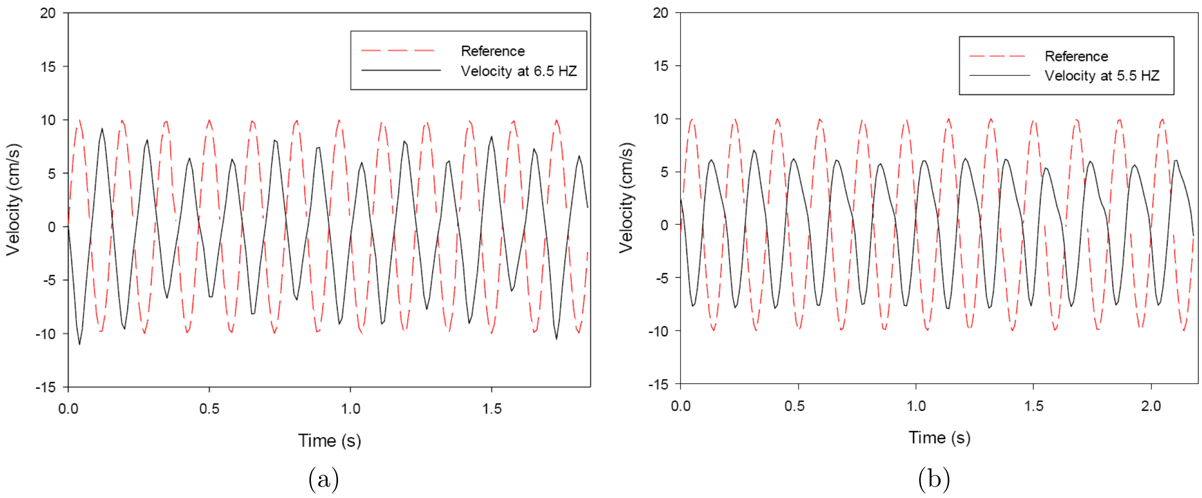


FIGURE 19. Closed loop velocity responses at critical bandwidth frequencies of (a) valve flow control system and (b) proposed pump flow control system

of the open and closed loop pump flow control system achieved in this study was 2.5 Hz and 5.5 Hz, respectively.

3) The bandwidth of valve flow control system, either under open loop or closed loop control, was always higher than pump flow control system due to the smaller inertia of the proportional valve. The bandwidth of the open and closed loop valve flow control system achieved in this study was 5 Hz and 6.5 Hz, respectively.

4) The velocity tracking performances of valve flow control system were better than pump flow control system in all cases of experiments. However, the power consumption of the pump flow control system was always much lower than the valve flow control system.

5) A valve flow control system with proportional valve and fixed displacement pump (Figure 1) could be converted to be a pump flow control system at small cost by adding an inverter type variable speed drive and altering the system control scheme. With the use of existing proportional control valve, pilot operated check valves are not needed as shown in Figure 2. Proportional control valve could be commanded to restrict the oil flow

during the cylinder downward motion. Therefore, proportional control valve, by itself, could prevent gravitational overrunning motion.

6) Our current work emphasizes on performance improvement of the proposed pump flow control system already achieved under PI control. Various adaptive controllers are under intensive investigation. Their performances on the proposed system will be compared with PI controller, and the result will be reported in the near future.

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