Frequency response enhancement of variable valve system by employing peak and hold method

Xie Yingjun[a]*, Gao Zhen[a], Jin Bo[a] & Liu Jinrong[b]

[a]The State Key Laboratory of Fluid Power Transmission and Control, Zhejiang University, Hangzhou 310027, P R China

[b]No.704 Research Institute, CSIC, Shanghai 200031, P R China

A control method based on peak and hold technique is applied to variable valve system to prompt its response rate. The proposed method is characterized by a significantly higher response rate compared with the standard control method. Different from the standard control method, which provides the coil with a constant voltage proportional to the target lift, the control system based on PH method provides the coil with higher voltage during the rising stage of the valve to prompt its response rate. The mathematical model of the system is derived to establish the transfer function of the system. The optimal duration of peak voltage is obtained by simulink models. The experimental results indicate that much higher frequency response can be guaranteed by employing PH method. The maximal frequency that the system can achieve is 51 Hz, which is greatly enhanced from 25 Hz by employing the standard control method.

Keywords : Peak and hold method, Frequency response, Variable valve system, Proportional pressure reducing valve

Variable valve technique is currently being utilized in automobile engines to meet the demands for high engine performance and environment protection[1]. The technique based on electro-hydraulic system is more and more popular because of its capability of flexibly changing valve timing, opening duration and valve lift[2]. However, the response rate of hydraulic system is usually slow, restricting its application to high speed engines.

Generally, there are two approaches to improve system response of hydraulic systems. The first approach is to optimize the profiles of grooves in control valves to obtain optimal internal flow field. The system response rate can be enhanced once the precision and promptness of the valve is obtained. The other is to employ faster control strategies and also to design faster electronic peripherals[3]. Amiranti et al.[4,5] improved the internal profile of the valve to prompt the response rate. Yang et al.[6] utilized nonlinear PD control strategy to achieve fast-tracking of an electro-hydraulic force system. Sampson et al.[7] studied the influences of different controllers on the performance of electro-hydraulic actuator system. The two approaches can be coupled to enhance the system response.

This paper proposes a new control system for variable valve system that is capable of providing a PWM signal with variable duty cycle. The proposed control strategy is based on a peak and hold (PH) method, widely employed to control diesel fuel injection[8]. This method allows the system to obtain a very fast response compared with standard control methods. Some experiments were carried out to verify the advantage of the proposed control system.

Principle of Variable Valve System and the Test Rig

The variable valve system developed is based on a three-way proportional pressure reducing valve. The actuator is a single rod hydraulic cylinder controlled by the valve. Proportional pressure reducing valve is a vital part in the entire system, as it controls the flow rate and pressure of the fluid which act on the system.

We choose a positive opening valve for the sake of system stabilization. The top of the valve is equipped with a proportional electromagnet, which drives the spool to move. The valve is directly mounted on the top of hydraulic cylinder. The prototype of the variable valve system is shown in Fig.1.

The proportional electromagnet is the electro-mechanical converter that converts control signals to mechanical force and displacement. Its output force is proportional to control signal within specific displacement range. The three-way proportional pressure reducing valve controls the flow rate and output pressure that flow into the cylinder chamber.

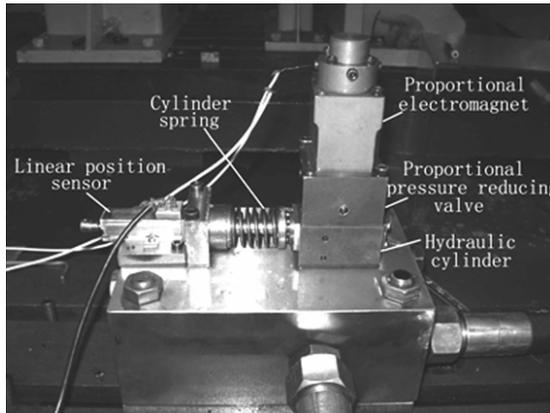*Corresponding author (E-mail: xyj@zju.edu.cn)

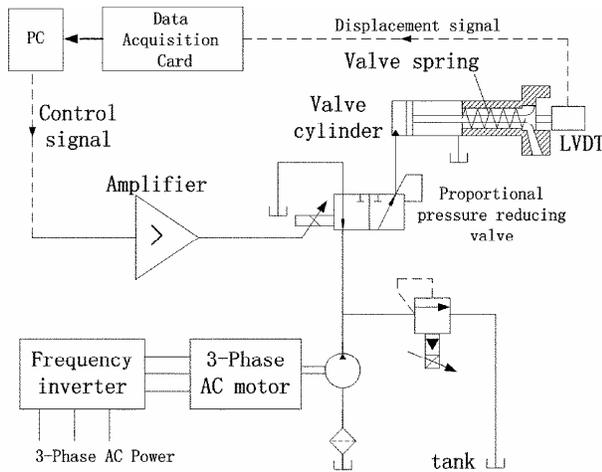Fig. 1– Photograph of the electro-hydraulic variable valve system

Fig. 2– Layout of the experiment system

Fig. 3– Electronic circuit of the variable valve system

A single rod hydraulic cylinder is used to simulate the movement of intake valve or exhaust valve. At the end of the cylinder rod, a linear position sensor (PYZC-25, full scale error: ±0.2%) is directly mounted to it to detect its instant displacements. Besides, cylinder spring acts as the return force to push cylinder rod to its initial position.

When the proportional electromagnet is in initial state, the input voltage is zero. The pressurized hydraulic oil from inlet Port is blocked. When the electromagnet is energized, the electromagnetic force pushes the valve spool to move downwards, connecting port P and port A. Hence, pressurized oil from the pump flows into the cylinder chamber, pushing the cylinder rod to rise. The valve spring acts as load force to compel the actuator to return to the initial position when control signal is cut off.

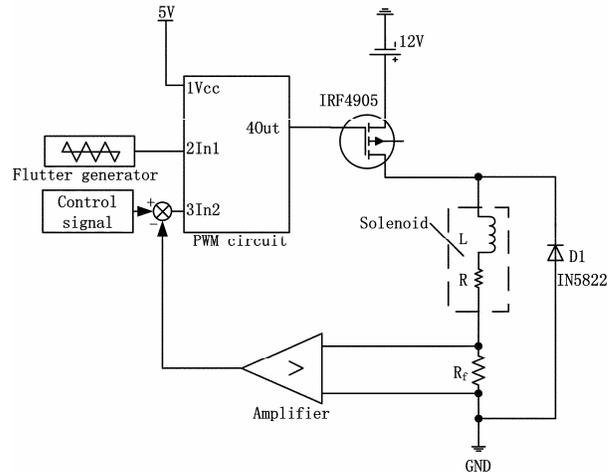Figure 2 shows the schematic diagram of the experimental set-up. The frequency inverter allows variable system flow rate while the electronically controlled variable relive valve allows variable system pressure. Therefore, the test system is able to work under a constant pressure and flow rate environment.

The displacement signal is actually analog voltage signal, ranging from 0 V to 12 V, and is sent to a data acquisition card. PC reads the digital signal which is converted by DAQ card, records and displays all the measured quantities. The control signal is generated by program codes and can be easily changed from one signal form to another, i.e., from square waves to triangular waves.

The electronic circuit shown in Fig. 3 employs PWM techniques. Flutter signal is actually triangular wave with tiny amplitude and a frequency of 200 Hz. It is used to reduce frictional hysteresis loop, hence to improve the dynamic performance of the proportional electromagnet. The output signal, whose duty cycle is proportional to control signal, directly drives the gate of power MOS-FET. Thus, the open duration of power MOSFET can be controlled by the input signal. The ultimate result is that the voltage which acts on the coil is proportional to the control signal.

In our apparatus, the control signal is continuously adjustable between 0 V and 5 V to change the amplitude of current passing through the coil. The open loop gain of the circuit is 3.43 and the closed loop gain is reduced to be 2.4 as negative feedback is utilized to stabilize coil current.

Mathematical Model of the System

Dynamic model of the electro-hydraulic variable valve system

For simplicity and conciseness of the equation, valve leakage, compressibility of the oil and the back

velocity electromotive coefficient of the coil is not considered.

As PWM technique is utilized in the system, the voltage acting on the magnet, namely $U_m(s)$, is proportional to the control signal. The proportional relationship is indicated by Eq. (1), where K_1 is the duty cycle factor, s is the Laplacian operator and $U_c(s)$ is the Laplacian form of the control signal. In this equation, delay of electronic peripherals is not taken into consideration, as it is very small compared with mechanical delay.

$$U_m(s) = K_1 U_c(s) \qquad \dots (1)$$

The dynamic current passing through the coil can be defined as Eq. (2):

$$U_m(s) = LsI_m(s) + RI_m(s) \qquad \dots (2)$$

Where L is the equivalent inductance, R is the coil resistance and $I_m(s)$ is coil current.

The output force of the proportional electromagnet can be described as Eq. (3):

$$F_m(s) = K_2 I_m(s) \qquad \dots (3)$$

Where $F_m(s)$ the output is force of electro-magnet and K_2 is the displacement-force gain of the proportional electromagnet.

The dynamic performance of the valve and hydraulic cylinder are described by Eqs (4) and (5):

$$F_m(s) = m_v s^2 X_v(s) + B_v s X_v(s) + P_A(s) A_v \qquad \dots (4)$$

$$P_A(s) - F_N = m_p s^2 X_p(s) + B_p s X_p(s) + K X_p(s) \qquad \dots (5)$$

Where, B_v and B_p represent the viscous damping coefficient of the valve and the viscous damping coefficient of the cylinder respectively. m_v and m_p represent the equivalent mass of valve spool and cylinder rod respectively. $P_A(S)$ is the pressure in cylinder chamber; F_N is the pre-tightening force; K is spring stiffness; A_v is the area of valve spool and A_p is the area of hydraulic cylinder. X_v is the displacement of the valve and X_p is the displacement of the cylinder rod.

The continuity expression of fluid in the working chamber of the hydraulic cylinder can be described by Eq. (6):

$$A_p(s) X_p(s) = K_q X_v(s) - K_c P_A(s) \qquad \dots (6)$$

Where, K_q is the flow rate coefficient and K_c is the flow-rate pressure coefficient.

For positive opening valves, the flow rate gain coefficient K_q and flow rate-pressure coefficient K_c can be defined as follows[9,10]:

$$K_q = C_d w \left(\frac{2(P_s - P_A)}{\rho} + \sqrt{\frac{2P_A}{\rho}} \right) \qquad \dots (7)$$

$$K_c = \frac{\sqrt{2}}{2} C_d w Z \left(\frac{1}{\rho(P_s - P_A)} + \frac{1}{\rho P_A} \right) \qquad \dots (8)$$

Where C_d is the flow rate coefficient; w is the area gradient; Z is the positive opening size; ρ is the oil density; and P_s is the system pressure.

From above equations, we get the transfer function between $U_c(s)$ and $X_p(s)$:

$$X_p(s) = \frac{A_p K_q K_2 K_1 U_c(s) - R A_v K_q F_N}{\begin{array}{c} K_c m_v m_p s^4 + (K_c B_v m_p + K_c B_p m_v + A_p^2 m_v) s^3 \\ + (K_c B_v B_p + K_c K m_v + A_v K_q m_p + B_v A_p^2) s^2 \\ + (K_c B_v K + A_v K_q B_p) s + A_v K_q K \end{array}} \times \frac{1}{Ls + R}$$

$$\dots (9)$$

Linear relationship between cylinder stroke and control signal in steady state

Considering the steady state, we take the limit on both sides of the transfer function. The cylinder stroke versus the control signal in the time domain is describes by Eq. (10).

$$\lim_{t \to \infty} x_p(t) = \lim_{s \to 0} X_p(s) = \frac{A_p K_2 K_1}{A_v KR} \lim_{s \to 0} U_c(s) - \frac{F_N}{K}$$

$$= \frac{A_p K_2 K_1}{A_v KR} \lim_{t \to \infty} u_c(t) - \frac{F_N}{K}$$

$$\dots (10)$$

The critical control signal can be obtained by setting the cylinder stroke to be zero:

$$U_{cr} = \frac{A_v R}{K_1 K_2 A_p} F_N \qquad \dots (11)$$

It is obvious that the actuator keeps still when control signal is low. This is because load force surpasses the driven force. Hence, the steady state of the cylinder can be rewritten as the following equation:

$$\begin{cases} x_p = 0 & ; when \; u_c \leq \dfrac{A_v R}{K_1 K_2 A_p} F_N \\[3mm] x_p = \dfrac{A_p K_2 K_1}{A_v KR} u_c - \dfrac{1}{K} F_N & ; when \; u_c > \dfrac{A_v R}{K_1 K_2 A_p} F_N \end{cases}$$

$$\dots (12)$$

The Eq. (12) shows that cylinder stroke is proportional to the control signal in steady state. The linear relationship makes it easier to design control strategies.

Experimental verification of the mathematical models

Experiments under different input signals have been carried out to test the linear relationship. The input quantity is the amplitude of the control voltage while the output quantity is cylinder stroke. As is shown in Fig. 4, cylinder stroke is proportional to the control signal and the maximal stroke that the actuator can achieve is about 14.5 mm. Besides, the critical control voltage is approximate 0.8 V. Figure 4 demonstrates that the experimental results well agree with the theoretical results.

Results and Discussion
The presentation of PH method and simulation analysis of the system

From above analysis, it is clear that the larger the control signal, the larger the electromagnetic force and the higher the output hydraulic pressure of the control valve. Hence, larger control signal will finally lead to faster rising speed and larger stroke of the cylinder.

For specific cylinder stroke, control signal with constant amplitude can meet the requirement. However, the frequency response of hydraulic system is usually low due to friction, inductance of the coil
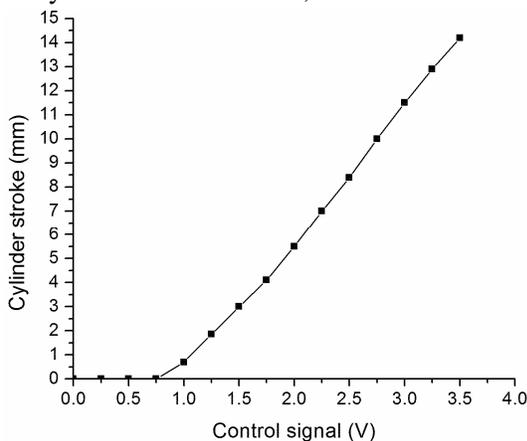
and the compressibility of the fluid[11,12]. By employing peak-hold method, the system frequency response can be significantly enhanced.

Figure 5 shows the basic principle of PH method. During the rise period of the cylinder stroke, the control signal is relatively high and the holding voltage which is demanded by target lift is relatively low. Obviously, larger electromagnetic force and hydraulic driving force will be generated to push the cylinder rod to move faster. Therefore, it costs less time for the cylinder to get to the target stroke.

The duration of peak voltage is very important in order to maintain a good performance of the system. Longer duration of the peak voltage leads to an overshoot of the cylinder stroke while shorter duration cannot maximally reduce the time for cylinder to rise.

The relationship between T_{peak} and U_{hold}

The transfer function deduced previously indicates a high order relationship between cylinder stroke and control signal. Generally, T_{peak} increases while the demanding cylinder stroke increases. The analytical solution could not be deduced from system transfer function. However, we can utilize numerical computation to figure out the optimal duration of peak voltage.

The optimal value of T_{peak} with the corresponding U_{hold} can be obtained by establishing the simulation model of the electro-hydraulic variable valve system. The main parameters of the simulation are shown in Table 1.

The simulink model is shown in Fig. 6.

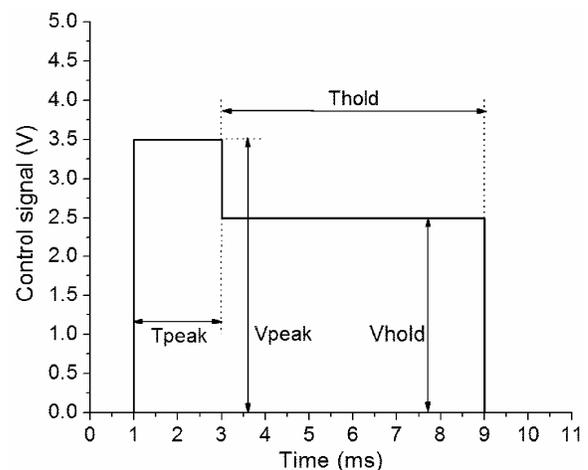By changing the duration of peak voltage in simulink model, a set of the simulation curves of

Fig. 4– Proportional relationship between cylinder stroke and the control signal

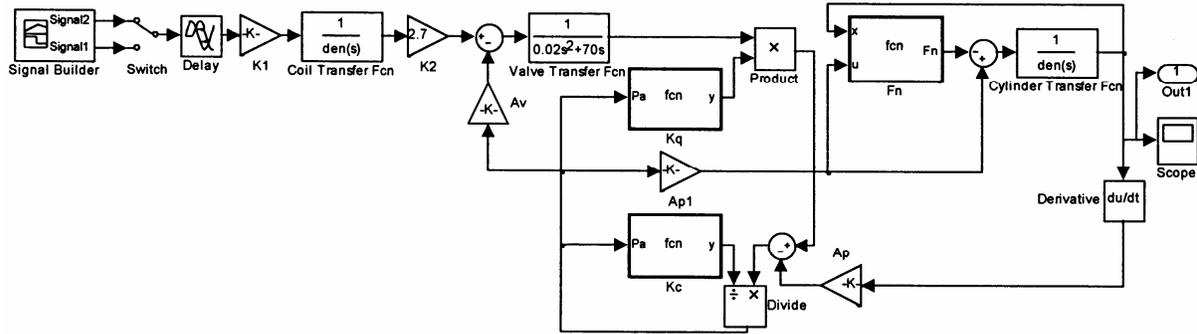Fig. 5– Description of peak and hold method

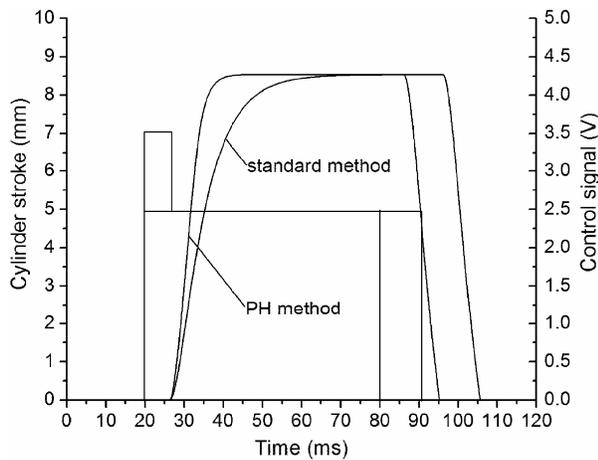Fig. 6– Simulink model of the electro-hydraulic variable valve system

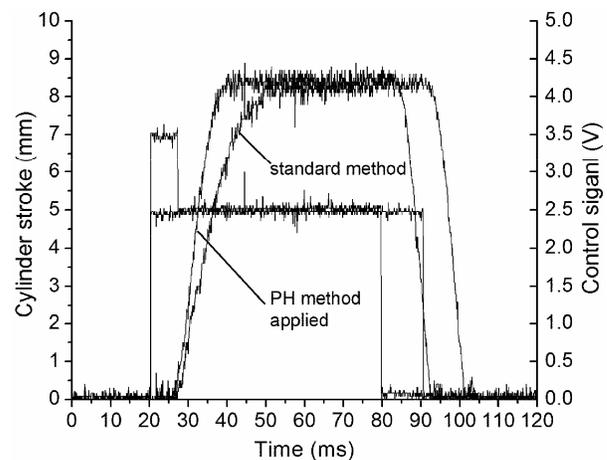Fig. 7– Simulation results of system response by PH method under optimal T_{peak}

Fig. 8– System response comparison of PH control method and stand method

Table 1—Main parameters of the electro-hydraulic variable valve system

Para-meters	Value	Unit	Para-meters	Value	Unit
F_N	48	N	ρ	850	kg/m^3
K	8000	N/m	B_p	75	N/(m/s)
A_p	1.1304×10^{-4}	m^2	B_v	70	N/(m/s)
A_v	2.826×10^{-5}	m^2	m_p	0.17	kg
P_s	6	MPa	m_v	0.02	kg
Z	0.2	mm	U_{hold}	2.5	V
L	0.15	mH	U_{peak}	3.5	V
R	0.45	Ω	K_1	2.4	/
w	1.884×10^{-2}	m	K_2	2.7	/

system responses can be obtained. The optimal value is selected among the curves that the rising time can be maximally reduced while no overshoot is generated.

The simulation results of PH method and standard method is shown in Fig. 7.

The peak voltage is 3.5 V and the target stroke is set to be 8.5 mm with the corresponding U_{hold} 2.5 V.

The optimal duration of peak voltage T_{peak} is found to be 6.8 ms. It can be seen from Fig. 7 that rising time of the cylinder can be maximally reduced and no overshoot is generated.

Experimental validation of PH method in electro-hydraulic variable valve

In order to validate the proposed control method and the correctness of the simulation, the following experiments are carried out.

In our experiments, the optimal duration of peak voltage is found to be 8 ms which is close to the simulation result 6.8 ms. The system response to a single square wave is shown in Fig. 8. The observed vibration is due to the dithering signal overlapping the PWM signal. It takes cylinder 25 ms to reach the target stroke of 8.5 mm under standard control method.

By employing PH method with optimal duration of peak voltage, it only takes cylinder 12 ms to achieve 8.5 mm. The response time is less than 50% of that required for standard control method. When the control signal is cut down to zero, the return speeds of
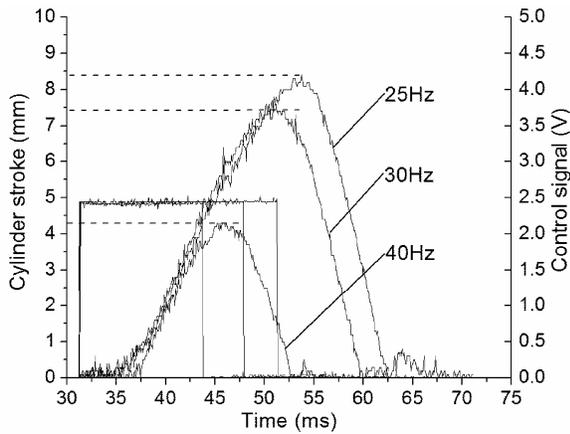
Fig. 9– Cylinder strokes under different signal frequencies by standard method
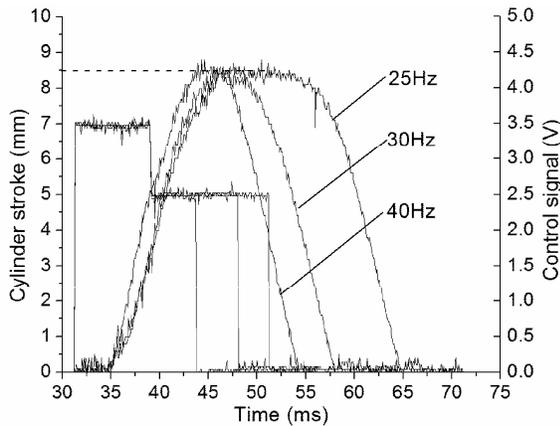

Fig. 10– Cylinder strokes under different signal frequencies by PH control method

both the two methods are the same. This is because cylinder spring load force (i.e. stiffness of spring and pre-tightening force) determines how fast the cylinder rod returns to its initial position.

System frequency response comparison of the two control methods

The frequency response of the system is tested by inputting control signals with different frequencies. Figure 9 shows the experimental results. Control signals are triggered at the same time point but vary in signal durations. It can be seen From Fig. 9 that when the frequency of the control signal is 25 Hz, the valve lift is 8.5 mm. When the frequencies of the control signals is 30 Hz and 40 Hz, the cylinder stroke falls to 7.3 mm and 4.3 mm respectively.

This is because the rising period of the cylinder costs too much time. When the control signal falls to zero, the valve spring compels the actuator to return to its initial position. Therefore, the maximal
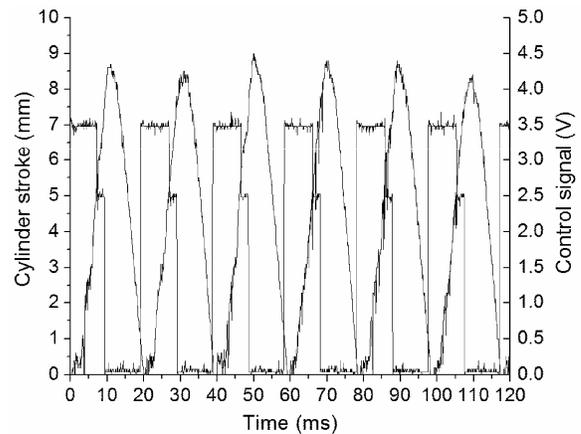

Fig. 11– Frequency response of PH control method with frequency of 51 Hz

frequency that the system can achieve is 25 Hz by standard control method.

The same experiments are carried out to verify the correctness and advantage of the proposed method. The peak duration of the control signal is set to be 8 ms obtained previously. The holding duration of the control signal decreases while the frequency of the signal increases. The experimental results are shown in Fig. 10. It can be seen that valve lift remains to be 8.5 mm even when the frequency of the control signal increases from 25 Hz to 40 Hz. Therefore, the maximal frequency that the system can achieve by applying PH method is much higher than the system utilizing standard method. This is because the rising stage of the cylinder is much faster by employing PH method. The maximal frequency response that the system can achieve is tested by inputting continuously squared wave signals with higher frequencies. Figure 11 provides the experimental results of system response to peak and hold control signal with the frequency of 51 Hz. We noticed that the valve lift is still kept as approximate 8.5 mm.

Therefore, the maximal frequency that the system can achieve is 51 Hz by PH control method. The results demonstrate that the system frequency response can be greatly enhanced by employing peak and hold method.

Conclusions

This paper presents the application of peak and hold control method to the control of a variable valve system. The system is based on a single rod hydraulic cylinder which is controlled by a three-port proportional pressure reducing valve. The conclusions are as follows:

(i) The results of the experiments show a very good linear dependency of the holding voltage versus the valve lift. Different valve lift can be obtained easily by changing the amplitude of the holding voltage acting on the proportional electromagnet.

(ii) The valve lift decreases while the frequency of the input control signal increases. When the frequencies of the standard control signal are 25 Hz, 30 Hz and 40 Hz, the maximal valve lifts are 8.5 mm, 7.5 mm and 4.5 mm respectively.

(iii) The optimal duration of peak voltage can be obtained by establishing the simulink model. The optimal value of T_{peak} is found to be 6.8 ms by simulation while the value is found to be 8 ms by experiment. The experimental results agree with the simulation results and demonstrate the correctness of the simulink model.

(iv) By employing peak and hold control strategy, the response rate of the variable valve system is much higher than employing stand control technique. The time for hydraulic cylinder to rise to target position (i.e. 8.5 mm) can be reduced from 25 ms to 12 ms. So, even the frequency of the control signal is 40 Hz, valve lift can be still kept as 8.5 mm.

(v) The experimental results demonstrate the advantage of the peak and hold control method. The maximal frequency the system can achieve is 51 Hz while keeping the valve lift to be 8.5 mm. The system is supposed to be applied to low speed engines.

(vi) The variable valve system is very inexpensive as it is base on a high speed proportional pressure reducing valve. The peak and hold control strategy is also easy for the implementation of system frequency response enhancement.

Acknowledgement

References

1 Dresner T, *SAE 890674.*
2 Chen Q X, *Vehicle Engine*, 3 (2002) 1.
3 Amirante R, Innone A & Catalano L A, *Energy Convers Manage,* 49 (2008) 2225.
4 Amirante R, Catalano LA, Moscatelli PG, *Energy Convers Manage*, 48(2007)42.
5 Amirante R, Bruno S, Del Vescovo G & Ruggeri M, *Improvement of a proportional valve dynamics by means of a peak & hold technique*, paper present at the 9th Scandinavian Int Conf on Fluid Power, Linköoping, Sweden, 2005.
6 Huayong Yang, Yiming Xu & Wei Sun, *New strategy of nonlinear PD controller for hydraulic force system under large variation of load stiffness*, paper present at IEEE/ASME Int Conf on Advanced Intelligent Mechatronics, Xian, China, 2008.
7 Sampson Eric, Habibi Saeid, Burton Richard & Chinniah Yuvin, *Int J Fluid Power*, 5 (2004) 2.
8 Bianchi GM, Pelloni P, Filicori F, Vannini G, Optimization of the solenoid valve behavior in common-rail injection systems, *SAE*, 1 (2000) 2042.
9 Liu J R, Jin B, Xie Y J, Chen Y & Weng Z T, *Int J Automot Technol*, 10 (2009) 27.
10 Liu Jin-rong, Jin Bo, Xie Ying-jun, Chen Ying & Weng Zhen-tao, *Indian J Eng Mater Sci*, 16 (2009) 7.
11 Merritt H E, *Hydraulic Control System*, (Wiley Press, New York), 1967.
12 Wu G M, Qiu M X & Wang Q F, *Electro-hydraulic Proportional Technique in Theory and Application*, Zhejiang University Press, China, 2006.