

An Efficient Lift Control Technique in Electro-hydraulic Camless Valvetrain Using Variable Speed Hydraulic Pump

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ABSTRACT

Significant improvement in fuel consumption, torque delivery and emission could be achieved through flexible control of the valve timings, duration and lift. In most existing electro-hydraulic variable valve actuation systems, the desired valve lift within every engine cycle is achieved by accurately controlling of the solenoid-valve opening interval; however, due to slow response time, precision control of these valves is difficult particularly during higher engine speeds. In this paper a new lift control strategy is proposed based on the hydraulic supply pressure and flow control. In this method, 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 non-linear differential equations of the mechanism, a novel average model of the system is developed based on energy conservation equations. A non-linear sliding mode controller (SMC) is also designed based on the developed average model and the boundary layer method is used to eliminate the chattering problem. The performance of the proposed controller is then examined through some simulations. 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.

INTRODUCTION

Although a significant numbers of engine valve-actuation systems including cam-based and camless mechanisms have been already introduced by several researches and companies, only few types of these systems (mainly cam-based) have been employed on commercial vehicles due to the liability, durability and cost issues. Although cam-based valve systems offer more reliable and durable functionality, the camless valvetrains can vary valve lift and timings to a greater extent comparing to the cam-based types [1]. Among various categories of camless mechanisms, the electro-hydraulic valve actuation system is the most repeatable and durable one. This type of camless valvetrain was first developed by Ford Motor Company in 1994 [2] and it was improved later by Sturman (1997) [3] and Lotus (2004). The basic electro-hydraulic valve system, presented by Schechter and Levin [2], consists of two pressure sources, a spring-return single-acting actuator and two solenoid valves. The first solenoid actuated spool valve (HPSV) is located between the high pressure hydraulic source and the engine valve actuator and is responsible for controlling the submission of the high pressure oil into the actuator during engine valve opening interval. The second solenoid valve (LPSV) is located between the engine valve actuator and the low-pressure hydraulic source and is responsible for evacuating the

actuator during engine valve closing stage. In this type of camless valvetrain, the engine valve timings and duration can be adjusted by controlling the opening and closing timings of the two hydraulic solenoid valves.

In addition to controlling the engine valve timings, the engine valve lift can also be controlled using this type of valvetrain. Throttle-less engine control can be achieved through accurate control of the valve lift during high engine speeds where short valve opening duration is not feasible due to the slow solenoid valves response time [4]. Moreover, a precise lift control is crucial in electro-hydraulic valve systems where the engine valve lift is highly influenced by the upstream hydraulic pressure, valve operating speed and valve opening interval.

In most of the current electro-hydraulic valvetrains, the valve displacement is controlled by precisely controlling the high pressure spool valve opening interval [5][6][7]. An adaptive pole-placement controller was designed by Anderson et al. based on the simplified model of the system to control the maximum lift [5]. They controlled the pulse-width of the signal transmitted to the high pressure solenoid valve to adjust the hydraulic flow to the engine valve actuator, and consequently, controlled the engine valve lift within each cycle. A combination of adaptive feedforward and feedback controllers were employed in [7] to control the engine valve displacement. A non-linear inverse model of the system was developed to design the feedforward controller. The valve displacement, hydraulic supply pressure and the spool-valve position were measured precisely as feedback signals to the controller.

In this paper, a new technique is proposed to control the engine valve lift in the basic electro-hydraulic valvetrain. In this method, instead of precisely controlling the high pressure solenoid valve opening interval, the hydraulic oil pressure and flowrate are controlled through adjusting the pump speed. This method is efficient where drastic changes in the reference engine valve lift is not necessary from engine cycle to cycle. Based on energy conservation law, an average model of the system is developed [8] based on which a non-linear sliding mode controller is designed. The performance of the designed lift controller is then studied through both simulations and experiment.

MECHANISM DYNAMICS

Similar to existing electro-hydraulic valve systems, the proposed mechanism consists of a hydraulic pressure unit and a valve actuation mechanism. The valve actuation mechanism has the basic configuration which was proposed by [3]. Based on this configuration, the actuation mechanism consists of two on/off solenoid valves (HPSV and LPSV) which repeatedly connect and disconnect the valve actuator cylinder to the high and low pressure sources, and consequently, open

and close the engine valve. The control signals to the solenoid valves are determined based on the desired valve timings and duration. The hydraulic pressure unit consists of a positive displacement pump, an oil tank and an air-accumulator.

In contrast to existing lift control technology in electro-hydraulic valve systems in which the desired valve lift is achieved by precisely controlling the high pressure solenoid valve (HPSV), in the proposed mechanism, the desired valve lift is achieved only by controlling the pump speed, and the solenoid valves are only responsible for controlling the engine valves opening/closing timings. As shown in Figure 1, the hydraulic pump is driven by a gearbox mechanism with two inputs: one from the engine and one from the variable speed electric motor. A transmission mechanism such as belt or chain is employed to transfer the power from the engine crankshaft to the gearbox; thus, at zero electric motor speed, the pump to engine speed ratio (r_{pe}) remains constant at all operating conditions. This ratio is chosen such that the maximum required engine valve lift ($lift_{max} \approx 10\text{mm}$) can be obtained at every engine speed. In the positive displacement pump, the hydraulic pump flowrate (Q_{pump}) is determined as follows:

$$Q_{pump} = N_{pump} V_{disp} \quad (1)$$

where, N_{pump} and V_{disp} are the pump speed and displacement volume, respectively. Considering that the engine valve operating speed is half of the engine speed, the required pump flowrate to achieve the maximum required lift ($lift_{max}$) can be determined as follows:

$$Q_{pump} = \frac{N_{engine}}{2} A_p lift_{max} \quad (2)$$

where, A_p and N_{engine} are engine valve actuator piston area and engine speed, respectively. Therefore, equating the equations (1) and (2), the desired r_{pe} at zero electric motor speed is determined according to the following equation:

$$r_{pe} = \frac{N_{pump}}{N_{engine}} = \frac{A_p lift_{max}}{2V_{disp}} \quad (3)$$

Using the proposed mechanism, at zero electric motor speed, the engine valve lift is as high as the maximum required valve lift ($lift_{max}$); however, as the electric motor starts rotating, the pump speed will be a function of both crankshaft and electric motor speeds. As the pump speed deviates from its original speed, the pump output flowrate is varied, which consequently, alters the engine valve lift. Hence, using this mechanism, the engine valve lift can be flexibly changed at

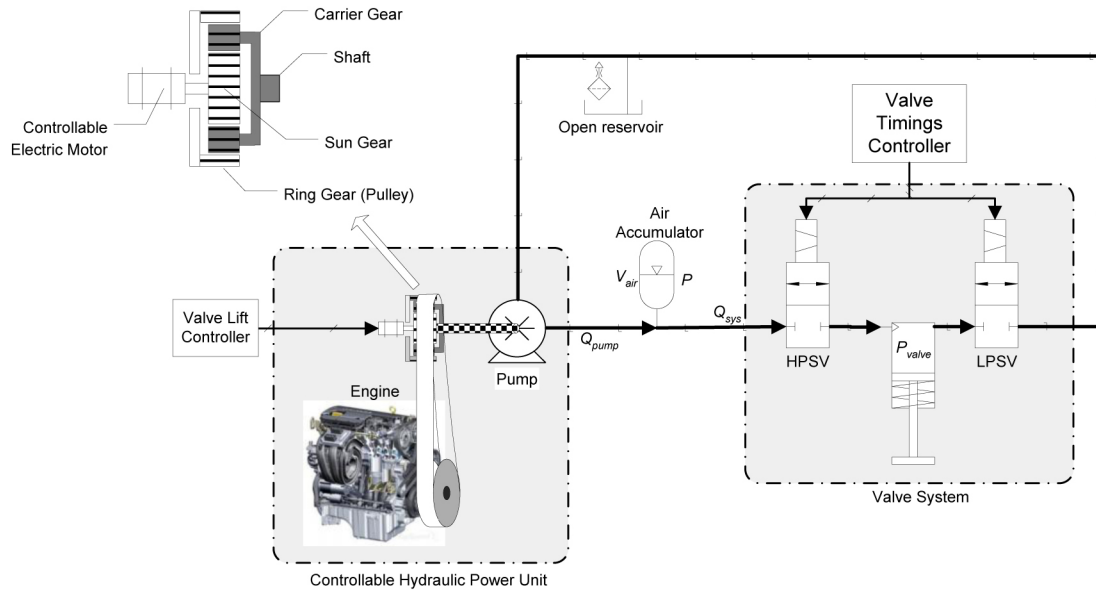


Figure 1. The schematic drawing of the proposed valve lift control mechanism

every operating condition. Moreover, in the case of having leakage in the system (eg. pump internal leakage), the electric motor can increase the pump speed to compensate for the leaked flow.

An air accumulator is used to reduce the supply pressure fluctuations and at the same time store the pump energy while the engine valve is standstill and there is no flow through the high pressure solenoid valve. The restored energy in the air accumulator can be reused again during the subsequent engine cycle and this could reduce the system power consumption considerably.

SYSTEM MODELING IN AMESIM

To study the performance of the proposed lift control mechanism for a conventional electro-hydraulic valvetrain, the system described in the previous section is modeled using AMESim8.b[©] software. As shown in Figure 2, the model consists of several sub-models: engine, positive displacement pump, engine valve actuator, air accumulator, HP and LP solenoid valves, valve timing controller and lift controller.

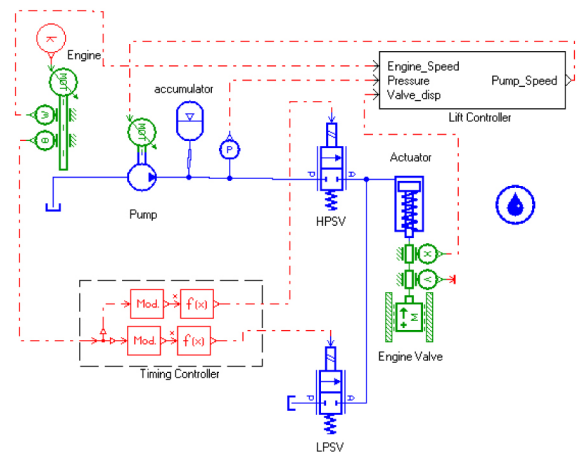


Figure 2. AMESim model of the electro-hydraulic valve system with the proposed lift controller

In this model, it is assumed that the electric motor is fast enough and the pump can easily follow the pump speed signal commanded by the lift controller. Therefore, for the sake of model simplification, the dynamics of the electric motor and gearbox is neglected. The HPSV and LPSV are modeled with two solenoid operated two-position, two-port hydraulic valves. A second order lag is also employed as follows to model the solenoid valve dynamics:

$$A_{sv} = \frac{A_{sv,max} \omega^2}{s^2 + 2\zeta\omega s + \omega^2} u_{sv} \quad , \quad u_{sv} \in \{u_{HPSV}, u_{LPSV}\} \quad (4)$$

where, ω , ζ and $A_{sv,max}$ are solenoid valve bandwidth, damping ratio and maximum port area respectively. u_{HPSV} and u_{LPSV} are control signals from timing controller to HPSV and LPSV.

The timing controller is a simple digital controller which switches the HP and LP solenoid valves based on the crankshaft angular position and desired engine valve opening and closing timings.

$$u_{HPSV} = \begin{cases} 1 & \text{for } \theta_{\text{opening}} < \theta < \frac{\theta_{\text{closing}} + \theta_{\text{opening}}}{2} \\ 0 & \text{else} \end{cases}$$

$$u_{LPSV} = \begin{cases} 1 & \text{for } \frac{\theta_{\text{closing}} + \theta_{\text{opening}}}{2} < \theta < \theta_{\text{closing}} \\ 0 & \text{else} \end{cases} \quad (5)$$

where, θ , θ_{opening} and θ_{closing} are crankshaft angular position, desired engine valve opening and closing respectively.

The valve actuator is also modeled with a single-acting spring-assistance hydraulic actuator. For the sake of simplicity, the engine valve and hydraulic piston masses are lumped into a single linear mass. Due to the large heat transfer from the accumulator casing to the ambient, the air compression and expansion processes in the accumulator are considered as isothermal processes ($PV=P_0V_0$). The lift controller will be discussed later in this paper. The model parameters are set based on the existing experimental setup specifications and are given in Table 1.

Table 1. Simulation parameters

K_{spring} [N/m]	F_{preload} [N]	F_{friction} [N]
12000	400	50
V_{disp} [cc/rev]	$A_{\text{sv,max}}$ [mm ²]	$P_0 V_0$ [Pa.m ³]
2.1	30	8
A_p [mm ²]	θ_{opening} [CA°]	θ_{closing} [CA°]
200	0	120
ρ [kg/m ³]	ω [Hz]	ζ
850	400	1

SYSTEM AVERAGE MODEL

In controlling the valve lift in variable valve systems, the aim is to achieve the desired maximum valve lift at each cycle rather than tracking a desired valve trajectory within each cycle. Therefore, it is more valuable if an average model of the system correlating the maximum valve lift to the control inputs is developed. To this end, an average model of the system is proposed based on the system energy conservation. Assuming a steady-state steady-flow condition, the power produced by the hydraulic pump is either reserved in the air

accumulator or used to power the valve-system. Therefore, the energy conservation equation for the studied system could be written as below:

$$\frac{d}{dt} E_{\text{pump}} = \frac{d}{dt} E_{\text{accumulator}} + \frac{d}{dt} E_{\text{system}} \quad (6)$$

Assuming that the hydraulic energy transferred to the valve system is dedicated to compress the returning spring and also overcome the friction forces, the total energy consumed by the valve system within each cycle could be determined as follows:

$$E_{\text{system}} = \frac{1}{2} K_{\text{spring}} x_{\text{max}}^2 + (F_{\text{friction}} + F_{\text{preload}}) x_{\text{max}} \quad (7)$$

In the four-stroke internal combustion engine with the engine speed of N_{engine} , the engine cycle duration is $(120/N_{\text{engine}})$. Thus, the average power consumption of the engine valve system can be approximated as follows:

$$\frac{d}{dt} E_{\text{system}} = \frac{N_{\text{engine}}}{120} \left(\frac{1}{2} K_{\text{spring}} x_{\text{max}}^2 + (F_{\text{friction}} + F_{\text{preload}}) x_{\text{max}} \right) \quad (8)$$

In the above equation, the effects of stiction and viscous friction are neglected and it is assumed that the dominant friction in the hydraulic cylinder is of Coulomb type and is constant. The pump power consumption can be determined based on the following equation:

$$\frac{d}{dt} E_{\text{pump}} = P Q_{\text{pump}} \quad (9)$$

where,

$$Q_{\text{pump}} = N_{\text{pump}} V_{\text{disp}} \quad (10)$$

Assuming isothermal air compression and expansion processes ($PV_{\text{air}}=\text{constant}$) in the accumulator, the rate of energy storing in the air accumulator could be approximated as follows:

$$\frac{d}{dt} E_{\text{accumulator}} = - \frac{P_0 V_0}{V_{\text{air}}} \frac{dV_{\text{air}}}{dt} \quad (11)$$