# Development of a camless engine valve actuator system for robust engine valve timing control

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**Abstract:** In this paper, a camless engine valve actuator (CEVA) system for robust engine valve timing control is presented. A particularly challenging control obstacle, shown in hydraulic actuation system for camless engine valve actuators, is the sluggish response of valve actuators at a cold operating condition. This is mainly due to the characteristics of oil viscosity with respect to temperature changes. At a low temperature condition, CEVA shows the very sluggish response. The retarded valve opening and closing timing, by CEVA's slow response at low temperature, cause an increase in pollutant emission and cylinder temperature during engine operation. In order to avoid these adverse effects by retarded timing, the new valve timing controller is proposed to control opening timing and closing timing, which is robust against temperature variations. A proposed valve timing controller is for guaranteeing the valve timing repeatability without using expensive position sensors. Experimental results indicate the feasibility of a fully variable valve timing control and CEVA system which can be constructed at a low cost.

**Keywords:** camless engine valve actuator; CEVA; valve timing control; hydraulic snubber; parameter identification; cost effective system.

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# 1 Introduction

In conventional internal combustion engines, the engine valve's position profiles are fixed according to engine crank angle. The engine valves including an intake and an exhaust valve are actuated by mechanically driven cams whose shape is decided by considering engine performance in various operating conditions. This causes a trade-off among engine speed, torque output performance, fuel consumption, and emission. In recent years, a significant amount of research on engine valve controls (Crane and Theobald, 1991; Ahmad and Theobald, 1989) has been conducted to demonstrate the advantage of variable valve actuation over the traditional cam-based valve actuation in both gasoline and diesel engines. The variable valve actuation can be realised by mechanical cam-based, electro-magnetic, electro-hydraulic, and electro-pneumatic valve actuation mechanisms. The mechanical cam-based variable valve actuation is achieved with additional actuators for continuously changing valve timing phase shift (Schneider et al., 2008; Moriva et al., 1996). The electro-mechanical valve actuation has been studied in terms of actuator design (Parlikar et al., 2005) and control (Nagaya et al., 2006; Chladny and Koch, 2008). In electro-mechanical valve actuation systems, control difficulties related in valve seating velocity and cost problems with expensive position sensors were presented by Montanari et al. (2004) and Butzmann et al. (2000). The variable valve actuation with electro-hydraulic actuators (Sun and Kuo, 2010; Allen and Law, 2002; Anderson et al., 1998) is achieved with digital or proportional valves to control oil flow into actuator's cylinders. The potential problems with electro-hydraulic actuation systems are energy consumption, valve seating velocity control, valve timing repeatability at the different operating temperatures.

In this paper, the hydraulic snubber was designed to achieve soft valve landing without an impact on mechanical parts (e.g., piston and cylinder) and was validated through simulations and experiments (Battistoni et al., 2007). Camless engine valve actuator (CEVA) with hydraulic snubber is shown in Figure 1. The detailed specifications and parameters are described in Table 1 and nomenclature respectively. The hydraulic snubber for the purpose of the soft valve landing was designed using AMESim software and its effectiveness was validated through experimental bench tests. In order to achieve a cost effective valve actuation system, a novel valve timing controller was proposed based on opening and closing timing detection without using expensive position sensors. The proposed valve timing control algorithm was verified through experiments.

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The paper is organised as follows. First, a model for describing CEVA system is presented in the system modelling section and operating principle of hydraulic snubber is explained. Next, robust valve timing control strategies are presented. Third, the experimental results of a proposed control system are shown in the experiments section. Finally, a conclusion is given.





Table 1Specifications of CEVA system

Supply hydraulic pressure	100 MPa	
Maximum valve lift	7.1 mm	
Piston diameter	12 mm	
Valve spring constant	25 N/mm	
Preloaded spring force	50 N	
Piston mass	0.05 kg	
Desired valve opening time	6 msec	
Desired valve closing time	5 msec	
Sampling time (Ts)	1 msec	
Position sensor	Micro-Epsilon OptoNCDT 1700 CCD laser sensor	
Controller	Micro AutoBox1401 PCI board	

# 2 Design and modelling of CEVA system

#### 2.1 Design of novel hydraulic snubber

A hydraulic snubber design is required for ensuring endurance and reliability of the CEVA. Mechanically designed hydraulic snubber controls the engine valve's landing velocity in order to reduce impact stress and thereby actuator noise without additional

active control efforts. The performances of hydraulic snubber design are directly related to energy consumption and manufacturing cost. A smooth valve landing but, rapid deceleration without bouncing is considered a significant requirement at a design stage. A dynamic simulation and analysis for verifying the performance of the snubber design are performed by engineering softwares, AMESim and MATLAB/Simulink. Experimental implementations through a prototype test bench are also performed for the purpose of snubber design validation. The snubber, used with an actuator piston, is an orifice flow control type which controls piston landing velocity with small sizes orifices.

An orifice type is illustrated in Figures 2(a) to 2(d) are the 2D-drafts of the actuator piston at a closing stage and at an opening stage respectively. Although ideal cylinder snubbers generally use a hyperbolically shaped plunger, the orifice type snubber design is chosen due to a limitation of the manufacturability.



Figure 2 Design illustration of hydraulic snubber, (a) and (b) at closing stage, (c) and (d) at opening stage (see online version for colours)

The objectives of the velocity snubber design are to reduce impact stress and have a rapid deceleration without severe bouncing. There are three significant things to evaluate the actuator performance with a hydraulic snubber.

- maximum valve landing velocity
- maximum valve opening and closing time
- valve bouncing just prior to a fully closed state.

The target value of the maximum valve landing velocity is approximately 0.3 m/sec. The hydraulic snubber engagement begins when the valve is at 1 mm from the landing position. The valve velocity during hydraulic snubber engagement must be less than 0.3 m/sec without or minimum fluctuation.

The simulation and experimental results are shown in Figure 3. The valve bouncing prior to be fully closed can be confirmed by the valve lift profile shown in Figure 3. Through hydraulic snubber design, engine valve is closed with soft landing as shown in Figure 3(b).









#### 2.2 Modelling of CEVA system

A CEVA system consists of a high speed servo valve, CEVA, and a hydraulic power unit including a hydraulic pump, a motor, a pressure relief valve, and filters. The schematic of CEVA system is shown in Figure 4.

The differential equation governing the dynamics of the CEVA is represented based on free-body-diagram shown in Figure 5.

$$m\ddot{x}_{valve,p} + b\dot{x}_{valve,p} + kx_{valve,p} + F = P_1A_1 - P_2A_2 \tag{1}$$

where F is a preload spring force by compressed engine valve spring. The instantaneous oil pressures in cylinder chambers can be expressed as follows (Kanghyun and Seibum, 2008; Kosmidis et al., 2006):

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$$P_{1} = \frac{\beta_{e}}{V_{1}} \int \left( Q_{1} - A_{1} \dot{x}_{valve, p} - C \left( P_{1} - P_{2} \right) \right)$$
(2)

$$P_{2} = \frac{\beta_{e}}{V_{2}} \int \left( -Q_{2} + A_{2} \dot{x}_{valve, p} + C(P_{1} - P_{2}) \right)$$
(3)

where  $\beta_e$  is the bulk modulus of the hydraulic fluid,  $V_1$  and  $V_2$  are instantaneous volume of the chambers, *C* is the leakage coefficient. The oil flow rate is calculated using the orifice equation (Merritt, 1967).

$$Q_{1} = C_{d}A_{0}\sqrt{\frac{2}{\rho}(P_{s} - P_{1})}, \quad Q_{2} = C_{d}A_{0}\sqrt{\frac{2}{\rho}(P_{s} - P_{2})}$$
(4)

where  $Q_1$  and  $Q_2$  are flow rates,  $\rho$  is the mass density of oil,  $P_s$  is the supply pressure,  $A_0$  is the valve orifice area. In this study, the applied oil pressure in each cylinder is calculated from valve command and supply hydraulic pressure instead of using the non-linear orifice equations. A non-linear hydraulic model considering oil flow dynamics is a too complicated model which requires much computational time and thereby it is difficult to be implemented in a real time controller. Hence, a linear second order model with oil temperature dependent parameter *b* as described in equation (1) is suggested. The net hydraulic force, which is an input to the system, can be simplified to be a pulse force based on supply hydraulic pressure and duration of valve activation as follows:

Actuating force = 
$$P_1 A_1 - P_2 A_2 = F_{\text{supply}}(t) - F_{\text{supply}}(t - \tau_{on})$$
 (5)

where  $\tau_{on}$  is the duration of valve activation,  $F_{supply}$  is a hydraulic force. The operating principle of the valve actuators can be described with Figure 6. In order to open an engine valve, a CEVA is connected to the high pressure oil pump by activating the valve opening command. When the pressure of forward chamber overcomes the preloaded spring force, it accelerates the engine valve downwards. In order to close engine valve, a forward chamber of the CEVA is connected to the oil tank by activating the valve closing command, which significantly reduces the oil pressure. Hence, the high pressure in a return chamber accelerates the engine valve upward. The pure time delay  $\tau_{delay}$  shown in Figure 6 can be easily obtained from the valve command and valve opening detection sensors. Considering that the control objective of the proposed system is not to perfectly track the desired valve position profile, this pure dime delay between a valve command and actual valve responses is not such a significant factor. The information on pure time delay can be considered when generating command in a next control cycle. A simplified CEVA model for controller design is compared with a non-linear model including oil flow equations through simulations. Figure 7 shows the simulation results for model comparisons. It indicates a relatively good agreement between linear and non-linear models. In a previous study (Kanghyun and Seibum, 2008), the model validation test using the hydraulic analysis software, AMESim, has been performed.



Figure 4 Schematic of CEVA system (see online version for colours)

Figure 5 Free body diagram of CEVA



Figure 6 Schematic of the CEVA operation (see online version for colours)





Figure 7 Model comparison for step input (see online version for colours)

# **3** Robust valve timing control

#### 3.1 Parameter adaptation and sensitivity analysis

The schematic of model reference parameter identification is shown in Figure 8.

Figure 8 Model reference parameter identification (see online version for colours)



In this study, the purpose of the parameter identification is to account for the relationship among oil temperature, valve velocity, and damping coefficient and to reflect its relationship in a controller design. Due to a wide range of operating oil temperature at which the CEVA system works, the parameter that drastically varies with respect to oil temperature conditions can be utilised for the adaptive controller design.

This study considers a gradient method to design an online parameter estimator. The gradient method was first suggested and utilised by Whitaker et al. (1958). The change in a system parameter is defined as a function of system output error and gradient of the system output error with respect to the system parameter is defined. The gradient of the error is the partial derivative of the error with respect to the parameter. The desired system response is specified by a nominal model whose out is  $x_p[kT, \phi]$ . The error between the plant and nominal model outputs is expressed as  $e[kT, \phi] = x_p[kT, \phi] - x_{p,n}[kT, \phi]$ . This parameter identification method is based on minimisation of the quadratic performance index and is expressed as:

$$J(kT,\phi) = \frac{1}{2}e^{2}(kT,\phi)$$
(6)

In order to minimise the performance index, the parameter  $\phi$  has to change in the opposite direction from gradient of performance index with respect to the parameter. Thus the updating law for  $\phi$  is obtained as:

$$\phi(kT+T) = \phi(kT) - \alpha \frac{\partial J(kT,\phi)}{\partial \phi} = \phi(kT) - \alpha e(kT,\phi) \frac{\partial e(kT,\phi)}{\partial \phi}$$
(7)

where partial derivative  $\frac{\partial e(kT,\phi)}{\partial \phi}$  is called the sensitivity derivative of error with respect

to parameter and  $\alpha$  is a positive constant and called the adaptation gain. The implementation of equation (7) requires real time generation of sensitivity derivative. The sensitivity derivative can be obtained by applying Laplace transform to plant and nominal model dynamics respectively. The error E(s) in Laplace domain can be expressed as follows:

$$e(s) = x_{valve}(s) - x_{valve,p}(s) = \left(\frac{1}{P(s)} - \frac{1}{P_n(s)}\right)u(s)$$
(8)

$$P(s) = ms^2 + bs + k \tag{9}$$

$$P_n(s) = m_n s^2 + \hat{b}_n s + k_n \tag{10}$$

The sensitivity derivative can be calculated by differentiating equation (8) with respect to  $\phi$  and taking the inverse Laplace transform as follows

$$\frac{\partial e(s)}{\partial \phi} = \frac{\partial}{\partial \phi} \left( \frac{1}{P(s)} - \frac{1}{P_n(s)} \right) u(s) = x_{valve,p} \left( \frac{s}{P_n(s)} \right)$$
(11)

$$\frac{d}{dt}\left(\frac{\partial e(t)}{\partial \phi}\right) = \frac{x_{valve,p}}{m} - \frac{\phi}{m}\frac{\partial e(t)}{\partial \phi} - \frac{k}{m}\int\frac{\partial e(t)}{\partial \phi}$$
(12)

where parameter  $\phi$  is a damping coefficient in the nominal model. From equation (8) to (12), the temperature dependent parameter  $\hat{b}_n$  can be identified in real time. The parameter identification algorithm was implemented on a CEVA test bench using three different temperature values ranging from -3°C from 40°C due to limited test environments. The experimental result showing relationship between identified damping coefficient  $\hat{b}_n$  and valve velocity according to oil temperature variations is described in Figure 9. Figure 9(a) is an experimental result when a CEVA works at  $-3^{\circ}$ C. The valve opening and closing velocity are 600 mm/sec and -800 mm/sec respectively and an identified damping coefficient is about 80N-sec/m. These are well below than desired velocities that are calculated from a maximum valve lift and desired valve opening/closing time as described in Table 1. Figure 9(b) is an experimental result when a CEVA works at 20°C. The valve opening and closing velocity are 900 mm/sec and -1,100mm/sec respectively. Figure 9(c) is an experimental result when a CEVA works at 40°C. The valve opening and closing velocity are 1,200 mm/sec and -1,500 mm/sec respectively and an identified damping coefficient is about 40 N·sec/m. The black dotted line shows that the damping coefficient and valve velocity has a monotonous relationship with respect to oil temperature variations.





The experiment for oil temperature sensitivity analysis has been carried out using three different temperature values. As shown in Figures 9 and 10, the response of a valve is the most sluggish at  $-3^{\circ}$ C. In other words, the valve's opening velocity is decreased due to increased oil viscosity at a low temperature.







# 3.2 Design of a valve timing controller

A block diagram of the proposed valve timing control system is shown in Figure 11. The goal of a valve timing control is to track the desired valve timing calculated by an engine control unit (ECU). The valve timing generator makes a desired valve command based on ECU information including driver's intention. Opening and closing time detectors are for measuring elapsed time until fully open and close using cheap sensors.

Figure 11 Block diagram of camless engine valve timing control (see online version for colours)



A proposed valve timing controller is based on advancing the valve timing by a timing compensation factor K. Considering that the supplying hydraulic pressure is a constant and servo valve dynamics is very fast, the control input to a CEVA described in equation (5) can be expressed as follows:

$$u = F_{\text{supply}} \left[ v(t) - v(t - \tau_{on}) \right]$$
  

$$\delta(t) = v(t) - v(t - \tau_{on})$$
(13)

where  $\delta(t)$  is a voltage pulse input to servo valves. In this study, the voltage pulse input is redefined as a function of the timing compensation factor to control the valve timing by the timing compensation factor as follows:

$$\delta(t) = v \left( t + KT_s \right) - v \left( t - \tau_{on} + KT_s \right) \tag{14}$$

The timing compensation factor K is defined as a discrete number 0, 1, 2, 3 which are correspond to the value of an identified damping coefficient and measured elapsed time until fully open and close. As described in equation (14), the valve timing is advanced dependent on sampling time  $T_s$  multiplied by K. The maximum advancing time  $(KT_s)_{max}$  is bounded by valve's resoponses according to oil temperature variations from experimental data. The advancing time contributes to lead an valve command anywhere between 0 to  $(KT_s)_{max}$  at 1,000 rpm or below. It is noted that small volume of operating oil to move the piston is required. This means temperature of the oil in a actuator chambers is increasing drastically as the engine room temperature increases. Therefore, the control conditions defined from experimental tests such as  $(KT_s)_{max}$  and 1,000 rpm are reasonable. Here, the timing compensation factor K is updated based on a following equivalent mapping table of a identified parameter and measured timing error  $E_{\text{timing}}$  every cycle. The equivalent mapping table shown in Table 2 is based on a monotonous relationship between a damping coefficient and valve velocity. The reliable correlation between a dominant parameter and measured data enables the control system to be achieved robustly without using expensive position sensors. The timing error  $E_{\text{timing}}$  is defined as follows:

$$E_{\text{timing}} = \Delta t_{desired} - \Delta t_{measured} \tag{15}$$

where  $\Delta t_{desired}$  is desired value opening and closing time described in Table 1,  $\Delta t_{measured}$  is a measured opening and closing time.

Compensation factor	With position sensors	Without position sensors (proposed system)
Κ	$\hat{b}_n (N \cdot sec/m)$	$E_{timing}(sec)$
0	Desired value (≈40)	Desired value (≈0)
1	60	1
2	80	3
3	$(\hat{b}_n)$ : Limit	(E <sub>timing</sub> ): <i>Limit</i>

 Table 2
 Equivalent gain mapping for adaptive control system

As shown in Table 2, the valve timing compensation factor is updated according to  $E_{\text{timing}}$  in a proposed control system without using position sensors.

# 4 Experiment

## 4.1 Experimental setup

To prove the performance of the proposed CEVA system, an experimental apparatus was constructed and well integrated into a control system as shown in Figure 12. The experimental test bench is comprised of computer controlled dSPACE/MicroAutoBox to implement a proposed control algorithm, a hydraulic power supply unit, a CEVA system, DC power suppliers to provide the electrical power for both the laser sensor and PCI board, a time detection sensor, a laser position sensor to measure valve position, and measurement devices including an oscilloscope to display control command. The laser position sensor is mounted on a high precision sensor stage with an angle such that the laser beam from the emitter of the laser sensor is perpendicular to the surface of the end of the valve stem.

Figure 12 Configuration of CEVA test bench (see online version for colours)



# 4.2 Experimental results and discussion

The experiments were conducted with an open loop control and also a proposed valve timing control at  $-3^{\circ}$ C and 40°C respectively. In this experiment, the valve position and velocity profiles, obtained from conducted at 40°C with an open loop control, is assumed as desired valve position and velocity profiles. Considering that the desired engine room temperature is about 50°C with air ventilated equipment and oil viscosity characteristics during normal engine operation, the above assumption can be reasonable.

Figure 13 shows experimental results performed at  $-3^{\circ}$ C and 40°C when a proposed valve timing control is not activated. The maximum supply voltage to the servo valve is 12 V and is a pulse wave signal. Figure 13(a) shows the different valve position profiles of tests at  $-3^{\circ}$ C and at 40°C. The valve response at  $-3^{\circ}$ C is relatively sluggish due to increased oil viscosity that causes also increase in damping coefficient. The differences in valve's opening and closing velocity, shown in Figure 13(b), well express the effect of operating oil temperature. Figure 14 shows experimental results conducted with a proposed valve timing controller at the same temperature condition. The valve opening and closing timing are compensated to guarantee repeatability of the valve operating timing as shown in Figure 14(a). Through advancing valve opening and closing timing, adverse effects such as increase in cylinder temperature and pollutant emission can be avoided. In a proposed adaptive valve actuation system, the valve timing control can be easily realised based on cheap valve open/close detection sensors without using expensive linear position sensors.

Figure 13 Experimental result with an open loop control, (a) valve position, (b) valve velocity (see online version for colours)









# 5 Conclusions

In this study, a CEVA system for robust engine valve timing control is proposed for application to a camless engine valve system. A significant amount of research on a camless valve actuation system has been carried out in terms of fully valve lift and timing controller design with expensive position sensors. In the camless valve actuation system with continuous valve lift controls, the control performance totally depends on precise oil flow control to ensure accurate positioning of the engine valves. Hence, it requires high performance flow control valves, and position sensors that cause increase in system production cost.

A proposed adaptive valve actuation system that is robust to oil temperature variations can be realised by a simple valve timing controller based on valve opening and closing timing detection. In order to minimise adverse effects by retarded valve timing, a proposed control system is desirable in terms of system cost and endurance with no sacrifice of the control performance.

#### References

- Ahmad, T. and Theobald, M.A. (1989) 'A survey of variable-valve-actuation technology', SAE Transactions, Vol. 98, SAE Technical Paper 891674, 1989, doi:10.4271/891674.
- Allen, J. and Law, D. (2002) 'Production electro-hydraulic variable valve-train for a new generation of I.C. engines', *SAE*, paper, SP-1692.
- Anderson, M.D., Tsao, T.C. and Michael, B. (1998) 'Adaptive lift control for a camless electrohydraulic valvetrain', *SAE*, Vol. 107, SP-1346.
- Battistoni, M., Foschini, L., Postrioti, L. and Cristiani, M. (2007) 'Development of an electro-hydraulic camless VVA system', *SAE*, pape2007-24-0088
- Butzmann, S., Melbert, J. and Koch, A. (2000) 'Sensorless control of electromagnetic actuators for variable valve train', SAE, Technical paper, 2000-01-1225.
- Chladny, R.R. and Koch, C.R. (2008) 'Flatness-based tracking of an electromechanical variable valve timing actuator with disturbance observer', *IEEE Trans. Control Systems Technology*, Vol. 16, No. 4, pp.652–663.
- Crane, M. and Theobald, M. (1991) 'Variable valve actuation', SAE, Vol. 99, No. 10, SP-1599.
- Kanghyun, N. and Seibum, C. (2008) 'Dynamic simulations and an adaptive valve timing control for a camless engine valve actuator', *Korean Society of Automotive Engineers Annual Conference*, pp.189–194.
- Kosmidis, V., Panga, A., Sakizlis, V., Charles, G., Kenchington, S., Bozinis, N. and Pistikopoulos, E.N. (2006) 'Output feedback parametric controllers for an active valve train actuation system', *Proceedings of the 45th IEEE Conference on Decision & Control*, pp.4520–4525.
- Merritt, H.E. (1967) Hydraulic Control System, p.81, John Wiley & Sons, New York.
- Montanari, M., Ronchi, F., Rossi, C. and Tonielli, A. (2004) 'Control of a camless engine electromechanical actuator: position reconstruction and dynamic performance analysis', *IEEE Trans. Industrial Electronics*, Vol. 51, No. 2, pp.299–311.
- Moriya, Y., Watanabe, A., Uda, U., Kawamura, H., Yoshioka, M. and Adachi, M. (1996) 'Newly developed intelligent variable valve timing system continuously controlled cam phasing as applied to a new 3 liter inline 6 engine', SAE, 96579.
- Nagaya, V., Kobayashi, H. and Koike, K. (2006) 'Valve timing and valve lift control mechanism for engines', *Mechatronics*, Vol. 16, pp.121–129.
- Parlikar, T.A., Chang, W.S., Qiu, Y.H., Seeman, M.D., Perreault, D.J., Kassakian, J.G. and Keim, T.A. (2005) 'Design and experimental implementation of an electromagnetic engine valve drive', *IEEE/ASME Trans. Mechatronics*, Vol. 10, No. 5, pp.482–494.
- Schneider, M., Krueger, K. and Ulbrich, H. (2008) 'Experiments and simulation of hydraulic cam-phasing systems', SAE World Congress, SP-2174.
- Sun, Z. and Kuo, T.W. (2010) 'Transient control of electro-hydraulic fully flexible engine valve actuation system', *IEEE Trans. Control Systems Technology*, Vol. 18, No. 3, pp.613–621.
- Whitaker, H., Yamron, J. and Kezer, A. (1958) 'Design of model reference adaptive control systems for aircraft', MIT, Cambridge, MA, Tech. Rep. R-164, MIT Press.

# Nomenclature

m	Sum of the piston mass and engine valve mass (kg)
b	Damping coefficient (N ··sec/m)
k	Spring constant (N/m)
$A_1$	Upper area of piston (m <sup>2</sup> )
$A_2$	Lower area of piston (m <sup>2</sup> )
$P_1$	Pressure of forward chamber (Mpa)
$P_2$	Pressure of return chamber (Mpa)
$F_p$	Preload of engine valve spring (N)
$x_{valve,p}$	Valve position (m)
$\dot{x}_{valve, p}$	Valve velocity (m/sec)
$\ddot{x}_{valve, p}$	Valve acceleration (m <sup>2</sup> /sec)