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Electrohydraulic Valvetrains for Direct Injection Compression Ignition Camless Engine
Operations – Phase I Project Final Report

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1. BACKGROUND AND TECHNICAL OBJECTIVES

Most existing variable-timing systems, which perform a camshaft phase shift or a switch to another cam lobe profile, offer measurable improvements at some engine operating points. A microprocessor controlled camless valvetrain, which continuously adjusts the parameters of valve motion with changing operating conditions, offers much more flexibility and potential benefits as they allow more complete optimization of engine events over the entire operating speed range. However, such devices can be complex, bulky, and relatively expensive, so their applications tend to be limited. The goal of this project is to investigate the technical challenges, and develop the enabling valve design and control capability for the camless operation of small compression ignition direct injection (CIDI) engines.

In this project the feasibility of such type of valvetrain has been investigated. The operation of the system to fire a single cylinder in a camless 2.0 liter 4 cylinder SI engine has been demonstrated with wide opened throttle. Groundwork on analytical modeling and initial experimental testing has been conducted to address some of the known critical issues and discover new technical challenges for the electrohydraulic valvetrain. This report presents the results obtained from this Phase I investigation.

Most piston-type internal combustion engines use mechanically driven camshafts for intake and exhaust valve operation. Such conventional mechanical valvetrains generally have fixed values for valve lift and valve event timing and duration. These fixed valve events represent a compromise among the conflicting requirements for various operating conditions. Until recently, this was considered an acceptable compromise. Simple, reliable, and relatively inexpensive mechanical valvetrains satisfied most basic engine requirements, and there was no incentive to seek a better, but costlier solution.

In recent times, the growing need to improve fuel economy and reduce the harmful exhaust emissions forced the engine designers to seriously consider alternative methods of valve operation, which could lead to an engine with less compromised performance. Numerous camshaft-based variable-valve mechanisms have been developed [1, 9, 10, 11, 14, 18, 26, 27, 29, 30, 33, 38, 47, 54]. A recent search on U.S. patent data base with the phrase "variable valve timing" resulted in 196 patents. Add-on devices, which perform a camshaft phase shift or a switch to another cam lobe profile, offer measurable improvements at some engine operating points, but it is only a fraction of what could be achieved if the valve timing, lift, event duration, and other parameters of the valve motion were selectively optimized for each operating condition. For this, a system offering a total valve motion control is needed.

A microprocessor controlled camless valvetrain, which continuously adjusts the parameters of valve motion with changing operating conditions, offers much more flexibility and potential benefits than the above mentioned adding-onto camshaft types of variable valve timing devices as they allow more complete optimization of engine events over the entire operating speed range. CIDI engines in particular, can show significant improvements, as variable valve timing can be used as a surrogate for variable compression ratio under certain operating conditions [7]. However, such devices can be complex, bulky, and relatively expensive, so their applications tend to be limited. Figure 1 [5,14,25,37] lists the works known to the applicants with comparisons on features and capabilities. It is noted that the Ford design has more features than others, but has not advanced to full engine operations. A detailed discussion on the advantages is given in a later section.

The springless electrohydraulic camless valvetrain (ECV) first developed at Ford Research Laboratory [37] has more advantages than other ones but does not yet have a fully functional running engine. The major technical barrier has been the generation of consistent and robust controlled valve motion for a full 16-valve system under various operation modes and environmental conditions. This

project proposes to conduct a systematic research effort that will address this problem and other critical technical barriers to improve this system to the fully functional state.

The goal of this research effort is to develop a prototype variable valve timing system for camless operation of small compression ignition direct injection (CIDI) engines. A systems oriented research effort that incorporates both analytical and experimental components will be conducted. Phase I of the project aims at answering two questions: Can and how can an electro-hydraulically driven springless valvetrain generate desirable variable valve timing performance consistently over a wide range of operating conditions? Can and how can such actuating system be integrated into camless CIDI engine operations to achieve potential benefits? Specific objectives are

- Development of a system dynamics (electro-mechanical-hydraulic) model and simulation for the proposed actuating system.
- Development of electronic control methods for valve timing, lift, soft seating, and multiple valve operations.
- Integration of an instrumented variable valve timing system on an engine head with electronic drivers and control interface software. The 16-valve system shall be fully electronically driven via triggering from the crank angle of an engine running in parallel.
- Characterization of the system performance envelope, energy consumption and sensitivity to environmental conditions through both simulation and experiment.
- Development of a system and control oriented dynamic model of CIDI engines and feasibility investigation of the control configurations and strategies using the proposed actuating system.

	Springless Electrohydraulic (Ford)	VVT With Camshafts (Honda, META)	Spring return electrohydraulic (Toyota, Caterpillar)	Electro- mechanical (FEV, Aura)
Variable lift	✓	✓	✓	
Low energy consumption	✓			✓
Low height and weight	✓			
Packaging flexibility	✓			
Skip firing flexibility	✓		✓	✓
Running engines		✓	✓	✓

Figure 1 Comparison of several variable valve timing systems

2. POTENTIAL TECHNICAL AND ECONOMIC BENEFITS OF ECV FOR CAMLESS ENGINE OPERATION

The electrohydraulic camless valvetrain (ECV) proposed in this project offers a continuously variable and independent control of all aspects of valve motion. This is a significant advancement over the conventional mechanical valvetrain. It brings about a system that allows independent scheduling of valve lift, valve open duration, and placement of the event in the engine cycle, thus creating an engine with a totally uncompromised operation. Additionally, the ECV system is capable of performing selective valve deactivation, and varying the activation frequency. It also offers advantages in packaging. Freedom to optimize all parameters of valve motion for each engine operating condition without compromise is expected to result in better fuel economy, higher torque and power, improved idle stability, lower exhaust emissions and a number of other benefits and possibilities. A review of expected improvements in both SI and CI engines associated with optimization of each aspect of valves operation is given below [37].

Variable Intake Timing

Increased Torque - Camshaft-driven engines usually produce a torque curve with a distinctive peak value in the mid-speed range. The one-value compromise for valve timing and lift causes performance deterioration for both low and high speeds. Optimization of valve timing yields a flatter torque curve due to improvement in volumetric efficiency. Dresner et al [9] cites a 50% increase in low speed torque and an average torque increase over the entire speed range of 10%. Urata et al [52] and Hatano et al [15] report 10% and 15% improvement, respectively, in the low speed torque. Kreuter et al [24] obtained 33% higher IMEP at 1000 rpm due mainly to higher volumetric efficiency and a minimized residual gas fraction.

Engine torque at low speeds can be increased, when the intake valve closing is delayed to take full advantage of the turbocharger dynamics. At low speeds, when compressor and turbine are not efficiency, the closing delay should be eliminated to retain the maximum effective compression ratio. In conventional engines, with fixed valve timing, the timing of the intake valve closing is a fixed compromise between the two contradictory requirements for low and high speeds. In an engine with variable valve timing there is no need for such a compromise. An electronic control system senses the changes in the engine speed and continuously adjusts the valve closure to achieve the best compromise between turbocharger dynamics and compression ratio at each speed. With proper tuning of intake closing timing for best turbocharger characteristics, the engine turbo-lag can be considerably reduced.

Variable Compression Ratio. Varying the timing of the intake valve closing varies the effective compression ratio without a corresponding change in the expansion ratio. This can be beneficial in turbocharged and supercharged engines. A conventional turbocharged engine can utilize only a fraction of the usable energy contained in its exhaust at high speed. In that case, a substantial part of exhaust gas must bypass the turbocharger to prevent excessively high inlet air pressure and temperature, which could lead to engine knock. This is called wastegating. With camless operation a precisely controlled late or early intake valve closing can be used to reduce the in-cylinder air compression and prevent knock. The reduction in the volume of air trapped in the cylinder is compensated for by an increase in the inlet air pressure. The benefit comes in the form of improved cycle efficiency, since the piston performs a smaller fraction of the total compression work. Varying the effective compression ratio can also be useful in Miller cycle engine and especially in diesels, where it is often desirable to operate at less than maximum compression ratio.

Variable Exhaust Timing

Optimized Expansion Ratio. The effective expansion ratio is determined by the timing of the exhaust valve opening. Usually, the exhaust valve begins to open well in advance of the bottom dead center (BDC), to provide enough time for the cylinder blow-down at high engine speeds. This makes the effective expansion ratio less than it could be if the timing was close to BDC. At low speeds, however, there is much more time for the blow-down and the excessively early exhaust valve opening is wasteful. With variable exhaust valve timing this deficiency can be eliminated by retarding the exhaust valve opening at low speeds, and, in general, optimizing the timing as a function of the engine speed. Much faster movement of engine valves in a careless engine permits a retard of exhaust valve opening even at high speeds. The low speed opening can occur almost at the bottom dead center. Increased expansion stroke work increases torque and, especially at low engine speeds, improves the engine fuel efficiency.

Internal EGR Exhaust gas recirculation (EGR) directly affects the residual gas fraction in the cylinder. Higher quantity of residual gas lowers the peak combustion temperature and thus reduces the quantity of nitrogen oxides produced during combustion. Ability to vary the timing of the exhaust valve closing eliminates the need for an external recirculation.

Advancing the exhaust valve closing to before the top dead center permits to retain the last portion of gases leaving the cylinder. On the other hand, when exhaust valve closing is substantially retarded, a certain quantity of exhaust gas is sucked back into the cylinder from the exhaust port by the downward moving piston on its early part of the intake stroke. The later the valve closure, the more exhaust gas is sucked in. Thus, varying the timing of closure, controls the quantity of the residual gas in the cylinder. Figure 3 illustrates the various changes in the exhaust valve timing.

Dresner et al [9] quote other researchers, who achieved near 90% in NO_x reduction with 29% of residual fraction. Gould et al [13] also report 90% NO_x reduction with tuning of exhaust valve dosing and the valve overlap using camless valve actuation.

Stein et al [44] used simultaneous retard of intake and exhaust camshafts at part loads to delay valve overlap for increased residual gas fraction, late intake closing for reduced pumping work and late exhaust opening for increased expansion work. He projected reductions in HC and NO_x emissions of up to 12 and 5% respectively, with unconstrained cam retard for a 4.6 L engine in a 4000 lb vehicle.

Variable Valve Lift and Timing

Reduced Energy Consumption. In an engine with a conventional cam-driven valvetrain the lifts of the intake and exhaust valves are selected to assure good volumetric efficiency and minimum exhaust stroke pumping during high speed operation. At lower engine speeds, lower valve lifts could be adequate and, sometimes, even desirable. In a camless engine with a variable valve lift the lift of both intake and exhaust valves can be reduced with reduction in engine speed (see Figure 4). Since the energy consumed by the valvetrain goes down with reduction in the valve stroke, varying the valve lift as a function of the engine speed can improve fuel efficiency at lower speeds.

Increased Inlet Air Velocity Reduced intake valve lift at low engine speeds has another potential benefit: it increases the inlet air velocity. This leads to faster burn rate and improved idle stability. Kreuter et al [24] showed 8- 9% reduction in fuel consumption due to increased intake-air velocity realized by deactivating one of the two intake valves.

Air & Start of Injection Control - In a camless engine the pattern of air motion in the cylinder can be influenced by independently varying the lifts of the two valves. If the two intake valves have unequal lifts, the air flows into the cylinder in two unequal streams, which, when mixed, can provide a variety of in-cylinder flow patterns, depending on the relative strength of each stream. Independently varying the lift of each intake valve, varies the distribution of the air flow among the valves. This provides a

subtle means of fine tuning the air flow in the cylinder. Wilson et al [53] completely mapped the tumble, swirl and flow characteristics of a four-valve (per cylinder) engine. Special camshafts were used to obtain unequal lifts of paired intake valves (including a total deactivation of one of the valves). Testing revealed that unequal lifts had potentially greater influence on the in cylinder charge motion, than the cylinder head design. The fuel efficiency improvements of up to 3%, as well as better combustion stability, improved idle and lower emissions were cited. Kreuter et al [24] also state that by closing one of the two intake valves, combustion delay and combustion duration can be considerably reduced, especially for high residual gas fractions and lean air-to-fuel mixtures. The shorter combustion delay lowers cyclic variability, and faster combustion results in better efficiency.

Improved Stability Idle stability is adversely affected by residual gas in the cylinder. Although presence of certain amount of residual gas in the cylinder at the beginning of intake is unavoidable, its quantity can be minimized. Adjusting the timing of exhaust valve closure and the corresponding opening of the intake valve (i.e. valve overlap) for minimum residual gas quantity contributes to better idle stability.

Improved Volumetric Efficiency. In an engine with a conventional cam-driven valvetrain the valve opening and closing is a gradual process lasting the entire valve event. The valve velocity is rigidly related to the crankshaft speed, so that the valves move faster only when the crankshaft turns faster. In contrast to this, the valve movements in a camless engine are very fast governed only by hydraulic forces that are independent of crankshaft speed. The opening and closing occupy only small portions of the entire valve event. Consequently, the valve lift profile, as plotted versus crank angle, has a nearly rectangular shape at low speeds. Even at high speeds the profile can look trapezoidal, if adequate hydraulic pressure is used. Such form of the intake valve event, in which the valve remains at maximum opening during a substantial part of the event, favors better volumetric efficiency at high speeds. This contributes to a higher torque. Asmus [4] discusses in detail the flow losses associated with sluggish opening and closing of cam-driven valves. His analysis supports the anticipated advantages of quicker motions of valves enabled by the camless operation.

Reduced Energy Consumption. As the engine speed is reduced, the need for faster opening and closing is diminished. The hydraulic pressure driving the valves can be reduced to save energy and improve the powerplant overall efficiency. Further energy savings are achieved with energy recovery feature of the hydraulic pendulum concept described later.

Valve Deactivation

Variable Engine Displacement. Deactivating some of the engine cylinders forces the remaining cylinders to operate at a higher load to maintain a given engine output. The higher load reduces specific fuel consumption. The camless engine control system can selectively deactivate any pair (intake or exhaust) of hydraulically coupled valves at any time by simply interrupting the electric signals to the respective control solenoids. This also applies to fuel injectors. Deactivation of valves and fuel injectors selectively deactivates individual cylinders and effectively varies the engine displacement. Only small quantity of residual gas is trapped in the deactivated cylinder, which contributes to low in-cylinder pressures and, thus, to low friction. Deactivation of some of the engine cylinders leads to significant improvements in fuel economy and hydrocarbon emission. Dresner et al [9] quote one researcher who reported 25% improvement in fuel consumption at low-speed cruise and 40% at idle. In another case significant reductions in emissions were achieved: 40% in HC and 47% in CO at idle.

Variable Air Motion Patterns. In a camless engine with two intake valves per cylinder the pattern of the in-cylinder air motion can be switched by deactivating one of the two valves. Operation with two valves can favor a tumble-like motion while one-valve operation favors a swirl. Selective valve deactivation may be used to prevent the damage. If a camless engine is equipped with misfire

detection sensors, such damage can be avoided by preventing exhaust valve opening, whenever there is a misfire, and thus retaining the unburned mixture in the cylinder. Subsequent fuel injection and intake valve opening is also prevented, and only after the mixture ignites, in the next or any subsequent cycle, is the normal operation resumed. Such misfire damage control can prevent premature catalyst burnout.

Reduced Cranking Torque. Cranking the engine with some of the cylinders deactivated (open valves, inactive fuel injectors) will substantially reduce the power requirement for an electric starter and, possibly, allow to combine a starter and an alternator into one reversible electrical machine.

Firing Order Change. An engine crankshaft design allows more than one firing order. The selection is based on reducing the engine vibration and may not be an optimum one for all operating speed/load regimes. For example, in a four-cylinder engine the crankshaft design puts two of the four pistons in the TDC position, when the other two are at the bottom dead center. This permits two choices in the firing order: 1-3 4-2 and 1-2-4-3. Varying the firing order can not be done in a camshaft-based engine, but it can be accomplished in a camless engine.

Variable Activation Frequency

A camless system can deactivate valves and cylinders for periods as short as one cycle. Ability to selectively skip individual firing cycles in individual cylinders can be used to improve fuel consumption and exhaust emissions during part-load operation. The effect is similar to that achieved with continuous deactivation of some of the engine cylinders, except that all cylinders are firing, and uneven cooling-off of the deactivated cylinders is avoided. This would prevent a hydrocarbon emission spike during each re-activation of the cold cylinders.

Skip Cycle Operation. Ability to selectively skip individual firing cycles in individual cylinders can also be used during part-load operation. This would increase the IMEP during each firing cycle, with potential benefits for fuel economy and exhaust emissions.

Four-Stroke into Two-Stroke. If the normal frequency with which the valves and injectors operate in a four-stroke engine is suddenly doubled, the engine, in principle, can switch to a two stroke operation. Thus, a possibility exists that a camless engine capable to alternate between four-stroke and two-stroke operations can be created.

Regenerative Braking. Camless engine can provide a very effective dynamic braking of the vehicle. This can be done via a combination of increased activation frequency and valve deactivation. To perform dynamic braking, the exhaust valves and the fuel injectors are deactivated while intake valves are opened during each downstroke of the piston. The engine performs compression work during each upward stroke of each piston. When the intake valve opens, a blow-down of the compressed air back into the intake manifold takes place, thus dissipating the energy of the compressed air charge. Since most of the blow-down takes place during early stages of the piston downstroke, very little expansion work is performed on the piston, and a very efficient energy dissipation can be achieved. Instead of being dumped into the atmosphere, the compressed air can be pumped into a reservoir and then used for engine supercharging during vehicle acceleration. This provides regenerative braking which improves fuel economy.

3. TECHNICAL APPROACH AND RESULTS

In the past, electrohydraulic camless systems were created primarily as research tools permitting quick simulation of a wide variety of cam profiles. For example, Dopson et al [8] and Richman et al [35] described systems with precise modulation of a hydraulic actuator position in order to obtain a desired engine valve lift versus time characteristic, thus simulating the output of different camshafts. In such systems the issue of cost in quantities and energy consumption are often unimportant. The Ford ECV system has been conceived for use in production engines. It was, therefore, very important to minimize the system hardware and component complexity and minimize the hydraulic energy consumption. Based on these constraints, the designed ECV device is compact in size and consists of only a few components. Figure 2a shows the components of a disassembled ECV unit, where one engine valve unit, one solenoid valve unit, and one check valve unit. Figure 2b shows the present experimental system set-up.

System Description

Description of ECV Operation Principle. ECV provides continuously variable control of engine valve timing, lift, and velocity. It uses neither cams nor springs to actuate the engine valve. It exploits the elastic properties of a compressed hydraulic fluid, which, acting as a liquid spring, accelerates and decelerates each engine valve during its opening and closing motions. This is the principle of the hydraulic pendulum. Like a mechanical pendulum, the hydraulic pendulum involves conversion of potential energy into kinetic energy and, then, back into potential energy with minimal energy loss. During acceleration, potential energy of the fluid is converted into kinetic energy of the valve. During deceleration, the energy of the valve motion is returned to the fluid. This takes place both during valve opening and closing. Recuperation of kinetic energy is the key to the low energy consumption of this system.

As shown in Figure 3, a small double-acting piston is fixed to the top of the engine valve that rides in a sleeve. The volume above the piston can be connected to a high- or a low-pressure source. The volume below the piston is constantly connected to the high-pressure source. The pressure area above the piston is significantly larger than the pressure area below the piston. The engine valve opening is controlled by a high-pressure solenoid valve that is open during the engine valve acceleration and closed during deceleration. Opening and closing of a low-pressure solenoid valve controls the valve closing. The system also includes high and low-pressure check valves.

During the valve opening, the high-pressure solenoid valve is open, and the net pressure force pushing on the double-acting piston accelerates the engine valve downward. When the solenoid valve closes, pressure above the piston drops, and the piston decelerates pushing the fluid from the lower volume back into the high-pressure reservoir. Low-pressure fluid flowing through the low-pressure check valve fills the volume above the piston during deceleration. When the downward motion of the valve stops, the check valve closes, and the engine valve remains locked in open position.

Figure 2. (a) ECV Units Disassembled, (b) Experimental System Set-Up

The process of the valve closing is similar, in principle, to that of the valve opening. The low-pressure solenoid valve opens, the pressure above the piston drops to the level in the low pressure reservoir, and the net pressure force acting on the piston accelerates the engine valve upward. Then the solenoid valve closes, pressure above the piston rises, and the piston decelerates pushing the fluid from the volume above it through the high-pressure check valve back into the high-pressure reservoir.

The hydraulic pendulum is a springless actuator. Thanks to the absence of springs, the valve moves with constant acceleration and deceleration. This permits to perform the required valve motion with much smaller net driving force than systems with springs. The advantage is further amplified by the fact that in the springless system the engine valve is the only moving mechanical mass.

Valve Motion Control. It is necessary to precisely control the solenoid activation sequence to insure proper valve motion. In all, there are four parameters that determine the timing sequence (see Figure 4). First, the occurrence (T_{open}) of the opening pulse determines the valve opening timing. The opening pulse width ($int1$) determines the amount of high pressure fluid admitted into the control chamber, which ultimately determines the peak lift value (see Figure 4). The control of this parameter is referred to as *peak lift control*. Next, the occurrence (T_{close}) of the closing pulse determines the valve closing timing. The closing pulse width ($int2$) determines how far the valve moves in its closing descent; this determines the valve 'undershoot'. As the undershoot determines the end of cylinder gas exchange and partially determines valve seating velocity, its control is also of importance and is referred to as *closing control*. Next, the seating delay time ($int3$) determines how far the valve will rebound after the undershoot and the amount of energy recovery by the high pressure check valve. The rebound will also affect the final seating velocity. As the seating velocity determines both noise and seat wear, it is also of control interest and is referred to as *seating control*. Finally, the seating pulse ($int4$) finishes the closing of the valve, and insures that the valve is completely seated with a proper closing force. It is desired to minimize the electrical energy consumption of the system; therefore this seating pulse should be no longer than necessary.

Figure 3. Single valve actuator schematic diagram

Figure 4. Valve Motion Control Sequence.

The open loop tuning of these solenoid pulse variables is quite labor intensive and may change with the environmental conditions and valve units. For high energy efficiency, low cyclic variability, and low valve seat wear, the camless valvetrain system will require the use of a closed-loop electronic control to all pulse variables.

Technical Results

ECV Dynamic Model and Simulation. The ECV system was first conceived as digital switching devices, where the transient response of valve and pressure switching were neglected in the design analysis. During the course of development it was found that the transient dynamic response has substantial impact on component sizing, valve operation and control. A dynamic model for a single valve was developed [19] and simulated by SIMULINK.

The model takes the solenoid valve openings as input and generates the control chamber pressure and valve lift response. Figure 5 shows test data of a single valve (with one of the paired-valve deactivated) for 6mm valve peak lift, where the each solenoid pulse parameter (see. Figure 4) were manually tuned. Figures 6 and 7 show the close agreement of experiment and simulation results on control pressure and valve lift. The test was taken at a simulated engine speed of 1500 RPM for event duration typical for that speed. The system parameters were: $P_{hi} = 4.14$ MPa (600 psi), $P_{low} = 0.69$ MPa (100 psi).

The results for the 2 mm lift are shown in Figures 8 and 9, as the simulations mirror the experimental data. The responses are less oscillatory when compared to the 6 mm case. Figure 9 suggests that for the smaller lifts the check valves are less likely to open because the solenoid valves

Figure 5. Input solenoid valve lifts for 6 mm EV lift.

Figure 6. Engine valve lifts for 6 mm: experiment (solid) and simulation (dot).

Figure 8. Engine valve lifts for 2 mm: experiment (solid) and simulation (dot).

Figure 7. Control chamber pressure for 6 mm EV lift: experiment (solid) and simulation (dot).

Figure 9. Control chamber pressure for 2 mm EV lift: experiment (solid) and simulation (dot)

close slower. This occurred because the solenoid valves available for these experiments did not open fully at small flows. On the other hand the overall energy consumption will be lower, hence less need for recovery.

The experimental and the simulated data in this test indicate little or no possibility for recovery during closing of the engine valve. Two factors account for somewhat lower recovery during the closing part of the lift compared to the opening part for a symmetric pendulum (a pendulum with the net opening force equal to the net closing force). First, the closing part of the lift is always less due to the overshoot during the opening, which results in a difference between the maximum and the settled-down values of lift. Second, depending on how well the tuning of the quiet seating of the engine valve is done, some part of the lift is always used for the seating.

Figure 10. System integration overview.

To achieve maximally possible energy recovery from both opening and closing, the component parameters, such as the solenoid and check valves spring stiffness, and the pulse timing parameters must be analyzed extensively using the developed dynamic model to arrive at optimal settings.

Figure 10 shows the overall ECV engine control concept. The inner loop ECV device control makes the valve timing and lift follow commands from an outer loop engine control system. The inner loop valve lift and timing should have much faster response than the outer loop engine control, which uses slower feedback sensors like MAP and MAF.

As mentioned earlier, the closed loop device control is necessary due to the variations with the environmental conditions, different valve units, and valve aging/wears, and performance requirement for high energy efficiency and low cyclic variability. In the present state, only the peak lift control is implemented and fixed ratio of the closing pulse *int2* to the opening pulse *int1*, and fixed seating pulses *int3* and *int4* are used. An adaptive lift control scheme was developed and implemented on a single valve actuator [2]. Adaptive control was deemed necessary, as compared to fixed feedback control, because of the desire to maintain consistent performance over wide range of conditions.

The first step of system modeling is to identify the static gain from the plant input to the plant output. The input of the system is the first opening pulse width. The output of the system is the maximum value of engine valve lift. By running the ECV Simulation Model, we can get the static relationship between the system input and system output (see figure 11). The simulation result shows three different regions: Region I, a nonlinear but predictable region; Region II, an extremely unpredictable region; and Region III, a linear region. The reason why the input-output relationship is not purely linear is that there is a maximum displacement limitation for the high pressure solenoid valve spool. The plant input-output relationship is nonlinear before the solenoid spool hits its maximum limit, and linear after the solenoid spool hits its maximum limit and the pulse width is sufficient large. During the transient period, however, the relationship is almost unpredictable. From

Figure 11. Simulation and Experimental Results of System Identification

Figure 12. Dynamic Plant Input-Output Relationship

experimental system we also got similar results for both paired-engine-valve and single-engine-valve tests. In this test the system was run at a 5.52/2.28 MPa (800/330 PSI) high/low pressure combination to drive a pair of valves sharing the same control chamber. Pulse widths from approximately 3.95 to 4.60 msec were used, and the peak lift of the valve pair was measured in each case. The data were plotted comparing peak lift with high pressure solenoid pulse width, as shown in Figure 11. Only the EV lift that is within Region I is considered in this investigation.

The dynamic behavior of the cycle-to-cycle transient responses is shown in figure 12. First of all, the system itself is a stable system. One major dynamics is the three-step delay, which is due to the slow speed of the serial connection between the PC and microprocessor. Other dynamics can be simplified as a first-order differential function.

Nonlinear Adaptive Lift Control of ECV. The main goal of this lift control is try to drive the engine valve lift to follow up the desired lift profile in a short time and with small overshoot. With this in mind, an inverse plant model feedforward controller was exploited. At the same time, we want good performance in terms of disturbance rejection. There are two major disturbance sources in this experimental system: oil temperature and engine speed changes. The former one is a slowly time-varying variable, which can be handled by an online adaptive estimation algorithm. The latter one, however, could be very fast and is difficult for the adaptive controller to handle. Therefore, a separate engine speed feedforward controller was utilized with the adaptive feedforward controller to improve transient response when engine speed changes. A series of tests were completed on different engine valves to achieve the engine speed feedforward gain, K_ω (see figure 13). The slope of each curve on the figure indicates the speed feedforward gain we should choose in each case. The experimental results show that the feedforward gain is not sensitive to the engine valve lift, although they are not the same for different valves. Meanwhile, a fixed feedback controller with a relatively small gain was applied to keep the robustness of this control system [46].

There are several parameters in the dynamic plant model, including k , θ , and a_0 . The first two

Figure 13. Test of Engine Speed Effects for (a) Intake #2, and (b) Exhaust #2

parameters, i.e. k and θ that decide the shape of plant nonlinearity, are found crucial to the controller design. Due to lack of persistent excitations, the estimation value may drift and diverge if both parameters are online estimated by projection or RLS method. We tried to reduce the number of parameters need to be estimated online by a 2^{3-1} Design Of Experiment method, testing the sensitivities of each parameter of the plant model with respect to various operating and experimental conditions such as hydraulic fluid temperature, engine speed, and manufacturing variability of valve assemblies. It turns out that one parameter, k , is not sensitive to oil temperature and engine speed changes. This result leads to a single-parameter estimation of θ instead of two-parameter estimation algorithm, which works successfully for this system.

Figure 14 demonstrates that the adaptive control has faster transient response (within 5 cycles) than PID (proportional-integral-derivative) control (about 10 cycles in the best case); and the adaptive control maintains consistent transient responses under various conditions, while the PID control will lose its designed performance as the setpoint changes due to the dramatic nonlinearity of the system.

Figure 14. Adaptive control vs. non-adaptive type of control
(The PID controller was tuned at setpoint of 2.0 mm)

Figure 15 and 16 show the transient lift responses for the adaptive linear feedback controller (before Phase I) and the nonlinear adaptive feedforward with feedback (after Phase I) respectively. The overshoot in Figure 15 in the step setpoint changes is not likely to be acceptable in engine operation, considering possible valve and piston interference. This non-acceptable overshoot problem has been resolved by the adaptive feedforward controller developed in Phase I.

Figure 15. Adaptive Feedback Control (before Phase I)

Figure 16. Nonlinear Adaptive Feedforward with Feedback Control (after Phase I)

Figure 17. Steady-state valve parameters and pulse width vs. temperature at 2.0mm/1000 RPM.

Figure 18. Hydraulic Pressure Test

Figure 17 demonstrates the ability of the adaptive lift control to track the variability due to temperature changes. The steady state values of the real-time estimated valve parameters and the pulse width to generate a constant lift are shown in terms of the fluid temperature. As expected, the higher temperature has lower fluid viscosity and hence requires smaller pulse width to achieve the same lift.

Figure 18 demonstrates the ability of the control algorithm to compensate for the effects of hydraulic pressure. In this test, the minimum and maximum hydraulic pressure in the high pressure reservoir was 600 psi and 1000 psi respectively, while the low pressure was held at 320 psi. The steady-state parameters and pulse width values at the 2.0 mm/1000 RPM are shown versus pressure. As expected, as the high pressure of the hydraulic system increases, the required pulse width decreases.

Figure 19 demonstrates the capability of adaptive control to handle the manufacturing variations on different valvetrains. The transient performance has been maintained consistently, indicating proper functioning of the adaptive control.

Figure 19 Experimental test of adaptive lift control for two different paired valves under speed and lift changes.

Closing And Seating Control Of ECV System The aims of this control are: fast closing; quieting seating; and more energy recovery. Currently two closing pulses are utilized to meet these specifications (see figure 20). By controlling the two solenoid pulses and the interval between them, we may be able to control the seating speed as well as energy recovery. In order to implement this control, three feedback signals are need, i.e. undershoot, 2nd overshoot, and seating delay. The disadvantage of this idea is that because we are relying on snubber chamber to reduce the seating speed, the seating speed and seating delay are dependent on hydraulic oil viscosity, which is very sensitive to oil temperature. This makes the seating speed and seating delay difficult to control. Another problem may be that the snubber engagement has to be big enough in order to absorb the potential energy of the engine valve. However, a big snubber engagement may cause closing slowly. Sometimes we have to take a tradeoff between seating speed and seating delay [46].

Figure 20. Two Pulses with Large Snubber Engagement

Figure 21. Three Pulses with Small Snubber Engagement

Another idea to control the closing and seating of engine valve is to use three control pulses (see figure 21). The first pulse drags the engine valve down to some lift value. Then the second and the third pulses control the valve seating quietly. A big advantage of this idea is that instead of choosing a big snubber engagement, we can use a very small engagement value. Since the snubber engagement is very small, the seating speed is still mostly decided by the two control pulses. The purpose of keeping a small snubber engagement is only to reduce the sensitivity of seating speed versus control variations. The disadvantage of this control algorithm is that more feedback signals may be necessary, which will increase the load of analog/digital signal processing. Figure 21 shows that by using three pulses, we can achieve a seating speed of less than 25 mm/s within 1ms. Figure 22 shows that we can still get energy recovery by three-pulse control. Figure 23 shows that the small snubber engagement can reduce the sensitivity of seating speed versus control variations.

Figure 22. Control Chamber Pressure by Three Pulses w/ Small Snubber Engagement

Figure 23. Sensitivity of Seating Speed versus Control Variables

Engine Control Using ECV. The majority of research work in the area of valvetrain design is divided in two categories: (i) hardware development, and (ii) engine optimization and management[6]. This project combines recent work by the PI [2,19] and the Co-PI [3,43] by addressing the actuator development together with the engine management system.

A CIDI engine model has been developed with a turbocharger, and the ECV unit. This model is capable of representing cylinder-to-cylinder operation, variable compression, and internal exhaust gas recirculation. Analysis of the interactions between the turbocharger, exhaust gas recirculation, and variable valve timing mechanism are in progress.

Findings from the model include that the naturally aspirated VVT/CIDI engine breathing dynamics are unstable in response to a load disturbance. In response to this it has been found that the fuel limiter commonly used in CIDI engines to prevent visible smoke during acceleration needs to be incorporated and used to stabilize the VVT engine. Through the analysis of different VVT methodologies that in-steady-state achieve large levels of internal Exhaust Gas Recirculation (iEGR) it has been found that currently used external EGR (eEGR) systems may be eliminated. Current eEGR mechanisms rely on a positive pressure drop between the exhaust and intake manifold compromising fuel economy. The simulations have shown that significant charge cooling and reduction in NO_x generation may be possible (approximation of the Miller cycle). This would alleviate problems associated with intercoolers and thermal efficiency. It may even reduce the need for sophisticated exhaust after treatment, such as DeNO_x catalytic converters or urea-based catalysis.

To enable fast torque response and address the largely unresolved issues associated with the cylinder air charge balancing in SI engines, the co-PI presents in [3] a feasibility analysis of an adaptive controller that coordinates intake valve lift and duration by using conventional engine measurements. This controller is currently being fit to a CIDI engine.

The controller design will be based on a multi-cylinder phenomenological CIDI engine model which functions with unthrottled engine operation. The model-based indirect-adaptive controller

shown in Figure 24 consists of a feedforward algorithm, an on-line estimator of parameter variability, and an adaptation scheme. The feedforward controller with adaptive capabilities ensures cylinder-to-cylinder balancing and fast tracking performance while it minimizes pumping losses. An engine controller architecture that is based only on feedback would limit the speed of response due to the inherent delays of the measurements. Robustness of the feedforward controller is ensured by the feedback controller that detects parameter variations and compensates for modeling errors. The feedback controller adapts the nonlinear feedforward scheme that is based on nominal engine characteristics to match then and adaptation during component variability and wear. This engine modeling and control method has also been successful applied to other powertrain control problems by the co-PI [17, 22,23,39,40,41].

Figure 24 Small Block diagram of the camless engine control scheme.

A/F control will allow the optimization of the intake valve timing to increase the air charge in the cylinder. This will increase the full power of the engine. The exhaust valve opening timing will be optimized to improve the transient performance of a turbo-charger as well as its steady state performance.

Also the ECV unit allows the use of internal Exhaust Gas Recirculation. The use of this ability may allow the removal of the external EGR devices (including an EGR throttle and EGR cooling system). The removal of this equipment will increase the overall efficiency of the engine. The EGR transient may also be reduced or even eliminated.

System Hardware/Software. Previously due to the limitation of the hydraulic pump capacity used in the experiment, the system could not run in a consistent and stable manner at speeds higher than 3000 rpm. Also lift over 5 mm for a paired valve could not be controlled consistently due to the lack of an accumulator in the hydraulic system to supply transient flow demand [2].

The limitations of the pump capacity resulted in pressure dynamics in the high and low-pressure reservoirs. This can result in decreased efficiency at high valve lifts and crank speeds (requiring longer pulse widths) as well as decreased repeatability. In the latter case, the supply pressures are not consistent from one cycle to the next, causing incomplete closing or fluctuating opening. Figure 25 shows a common example of operation in the unstable regime. In this case, the system is being operated at 3000 RPM with an attempted lift of 4.5 mm, with both the intake of cylinder four and the exhaust of cylinder three in operation. In this case with the old pump and hydraulic system the pressure fluctuations, with a constant open loop opening and closing pulse width, cause the lift to be too high one cycle, then too low the next cycle. As can be seen from Figure 26, the pump used in the experiment is a fixed-speed, fixed-displacement pump with pressure regulation by relief valve. The electrically driven pump is not connected to an engine shaft and does not increase flow capacity with

Figure 25 Pump Capacity and Dynamics Affect Valve Stability (Experimental Result).
an increase in speed.

A new variable displacement pump was installed during Phase I. This pump is driven by a variable speed DC motor. The use of the variable displacement pump removes the use of the high-pressure regulator and thus increases energy efficiency. The high-pressure setting is also adjustable. This feature may be utilized to optimize the hydraulic energy consumption and provide lift velocity shaping capability. This motor has an encoder attached to the ECV controller. The ECV controller uses the encoder signal to drive the ECV unit. The speed of the DC motor is controlled to simulate the speed of the engine. Thus the pump is supplied with more power when at higher speeds. See Figure 26.

The present ECV system is pulsed by an 16 bit microcontroller. The pulse timing, as it relates to engine speeds, is emulated by a software timer in the controller. The adaptive feedback control loop is implemented by a Personal Computer, which changes the pulse width values stored in the microcontroller memory through a serial communication link. In real-time control the communication delays are uncertain with respect to the engine cycles. A higher engine speed tends to have more delay cycles. The delays present a problem in achieving fast closed loop response. The present closed loop lift control settling time is about twelve engine cycles while it is desirable to achieve settling time less than five cycles for engine operations. Improvement on the ECV controller speed is even more critical when other valve pulse variables are taken into closed loop control. Due to hardware limitations this controller can not control the valve closing.

Figure 26 Controllable hydraulic power supply.

Figure 27 New ECV controller

The new ECV controller will use a 32 bit DSP board and an interface board to connect to the ECV sensors and actuators. The controller will be driven by interrupts from an encoder attached to the engine crank shaft or pump DC motor and thus from crank angle. This controller uses hybrid digital and analog electronics to control both the closing and opening of the engine valves. See Figure 27.

Physical characteristics: Sensitivity to environmental conditions - In order to understand the effect of changing the temperature of the ECV environment, fluids of different viscosities were used as the hydraulic fluid. See Figure 29 for the relative viscosities of the fluids. The resulting information will be

used to determine what physical properties the hydraulic fluid must have in order to operate. The goal is to either use diesel fuel, engine oil, or antifreeze as the hydraulic fluid to power the ECV. The fluids tested and their viscosities are listed below. The tests showed that the system

	Temp of test (deg C)	Viscosity (cSt)	Leakage rate (mL/min)
Antifreeze	(not tested)	2 to 4	
Diesel Fuel	40 to 50	1 to 3	610
Hydraulic Fluid	40 to 50	14 to 10	184
Engine Oil, 5W-30	40 to 50	52 to 36	(not measured)
Synthetic Lubricant, Mobilgear SHC 22M	50 to 100	15000 to 700	0

could operate over a range of fluid viscosity. But the tests conducted with the synthetic lubricant need to have the temperature controlled so that an upper limit on the required viscosity can be found. The ECV unit has long tubes and the head itself is an unheated aluminum block. Thus the heated synthetic lubricant cooled down as it entered the head at an uncontrolled rate. The results from this test were not consistent. Because of the wide range in temperature and the resulting range in viscosity the limiting viscosity could not be determined. We did determine that the system will not operate with a fluid whose viscosity is above 15,000 cSt.

Hydraulic Media Tests with Cold-Start Consideration: Sensitivity to environmental conditions - In order to understand the effect of changing the temperature of the ECV environment, fluids of different viscosities were used as the hydraulic fluid. See Figure 29 for the relative viscosities of the fluids. The resulting information will be used to determine what physical properties the hydraulic fluid must have in order to operate. It is our goal to either use diesel fuel, engine oil, or antifreeze as the hydraulic fluid to power the ECV. The fluids tested and their viscosities are listed below. The tests showed that the system could operate over a range of fluid viscosities. But the tests conducted with the synthetic lubricant need to have the temperature controlled so that an upper limit on the required viscosity can be found. The ECV unit has long fluid delivery tubes and the head itself is an unheated aluminum block. Thus the heated synthetic lubricant cooled down as it entered the head at an uncontrolled rate. The results from this test were not consistent. Because of the wide range in temperature and the resulting range in viscosity the limiting viscosity could not be determined. We did determine that the system will not operate with a fluid whose viscosity is above 15,000 cSt.

	Temp of test (deg C)	Viscosity (cSt)	Leak rate (mL/min)
Antifreeze	(not tested)	2 to 4	
Diesel Fuel	40 to 50	1 to 3	610
Hydraulic Fluid	40 to 50	14 to 10	184
Engine Oil, 5W-30	40 to 50	52 to 36	(not measured)
Synthetic Lubricant, Mobilgear SHC 22M	50 to 100	15000 to 700	0

Figure 28 Fluid Viscosity Vs. Temperature

The temperature of Mobile engine oil 5W-30 or 0W-30 at -40 deg C is about 17,000 cSt. This fluid viscosity is not useable for the ECV, but with a block heater the engine block and the ECV could be warmed up to allow the fluid's viscosity to decrease. Also Texaco has an engine oil 'Ursa SS' which has a viscosity of 7000cSt at -40 deg C. This fluid may work in the ECV unit. Again more tests need to be run to determine the upper limit of the fluid viscosity. There is also a factor of leakage. Because of its low viscosity the diesel fuel has a much higher leakage than the other fluids considered. Using diesel fuel requires tightening up the valve clearances and manufacturing tolerance.

Figure 29 shows the valve lift and control chamber pressure data for the three different fluid. Interestingly, the lower viscosity fluid has less oscillatory behavior. This is because the fluid damping effect of the valve leakage outweighs the effect of valve piston viscous friction damping. The low viscosity fluid has larger leakage but less viscous friction. Our model simulation has confirmed this phenomenon. Inspecting the control chamber pressure (dashed) hydraulic energy recovery is also evident since it reaches high- and low-pressure.

Also noticeable is that when the valve is closed the control chamber pressure increase to about 500PSI due to the net balance of solenoid and check valve leakage. This must be addressed by redesigning the valves with high precision fabrication tolerance in the future development. The long seating tails in the plots also suggest improperly designed hydraulic snubber orifice, which should be addressed in the valve redesign.

Diesel Fuel (1500 RPM & 5mm lift)

Hydraulic Fluid (1500 RPM & 5mm Lift)

Figure 29 Valve Profile and control chamber pressure using different fluid.

Finally, the valve profile using the highly viscous fluid could barely work. The oil on the pump side was heated, but it is difficult to measure the fluid temperature inside control chamber. However, the oil viscosity surrounding the engine valve piston area could be calculated for the constant valve velocity regime. This value was calculated to be about 20000 cSt, suggesting that the heated fluid quickly dissipated the heat to around 40C in the engine head. This result also suggests that the ECV is

not likely to cold-start using the regular engine oil unless the engine head and pump are heated effectively.

ECV Power Consumption Measurement and Analysis To make use of ECV's potential benefits, the power required to run the system must be at least comparable to the camshaft frictional loss. The ECV power consumption was calculated with optimized lift for maximum fuel efficiency with emission constraint. The energy recovery for the ECV system was measured to be about 10% for the present solenoid valves. The electrical power required to operate the solenoids was also measured. Using these figures, the power consumption comparison in term of Mean Effective Pressure of ECV is shown in Figure 30. The worst case (maximum lift for full load) ECV power is similar to that of the finger rocker arms, but larger than the rolling finger followers. Fast solenoid closing time would generate high hydraulic energy recovery, as predicted by our model prediction shown in Figure 31. Considering partial load conditions and the fact that cam friction loss is a small portion of fuel consumption, such level of ECV power should be acceptable. To further reduce the power consumption, the valve leakage must be reduced and the solenoid response must be increased.

Figure 30. Power Consumption of ECV and Camshaft Valvetrains

Figure 31 Expectation of Energy Recovery Improvement by Increasing the Solenoid Speed (based on model simulation)

ECV head running conditions: As a demonstration of the camless engine operation, the ECV was mounted on a SI engine to test single cylinder start-up and firing. This test consisted of a single cylinder used in a four cylinder engine. The other three cylinders were removed. Although the engine was not balanced after the cylinders were removed, it ran stably at 1200RPM. The stock camshaft was used to run the valves and then the ECV unit was optimized with late intake valve closing for three different lift heights. The torque required to run the ECV, using 70% pump efficiency and an 8% energy recovery, was removed from the useable engine output torque. This modified torque was then compared to the camshaft's torque using brake specific fuel consumption as shown in Figure 32.

SI engine Late Intake Valve Closing BSFC vs Torque (1200 RPM)

Figure 32 BSFC for an SI engine with a standard camshaft and the ECV unit.

At an engine torque of 25 Nm the ECV unit using a 6mm lift has BSFC that is 5% less than the stock camshaft. This torque is only a partial load for the engine, thus some of the increased fuel efficiency is from the removal of throttle pumping losses by using the valves to control the air flow. Also, during these tests the torque of the engine was controlled with the ECV with the throttle fully open, as Figure 33 shows.

Note: 'o': Simulation; '+': Experimental

Figure 33 Camless SI Engine Torque Regulation Test by Variable Intake Closing (Experimental Result)

During these tests it was noticed that the cylinder pressure had a substantial effect on the engine valve opening and closing (see Figure 34). The solenoid pulse-width required to open the exhaust valve to 5 mm with combustion is 7.2 ms, without combustion the pulse-width is 3.7 ms. This effect is likely to be more significant for diesel engines.

Figure 34 Cylinder Pressure Effect on Engine Valve Motion (Experimental Result)

4. SUMMARY OF DEGREES TO WHICH PHASE I OBJECTIVES HAVE BEEN MADE

The degree to which the Phase I project objectives have been made is summarized as follows.

- The ECV system dynamics model have been developed and the simulation agrees with experimental data.
- Adaptive nonlinear feedforward control for valve lift has been developed and implemented to successfully improve transient response time, eliminate transient overshoot, and enhance robust performance against operating condition changes. Soft seating and closing control methods have been developed and numerically simulated based on the ECV dynamic model. 8-valve (due to lack of lift sensors) ECV subsystem has been demonstrated operational to 5500 rpm.
- The ECV system has been demonstrated to be fully electronically driven via triggering from the crank angle of an engine running in parallel. Furthermore, the ECV has been demonstrated to start up and run single cylinder SI engine combustion.
- The ECV system's energy consumption has been measured. The ECV has also been shown operational when replacing the hydraulic oil with engine oil or diesel fuel. Model simulation matches with the experimental data for the different range of fluid viscosity.
- A system and control oriented dynamic model of CIDI engines has been developed and used to discover several important engine control and sensor/actuator speed requirement issues to be addressed when using the ECV for engine operation.

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