R. Seethaler, M. Mashkournia, R. R. Chladny, J. Zhao, and C. R. Koch. Closed loop electromagnetic valve actuation motion control on a single cylinder engine. In *SAE Paper 2013-01-0594*, page 8, 2013

13PFL-0582

Closed Loop Electromagnetic Valve Actuation Motion Control on a Single Cylinder Engine

R. Seethaler, M. Mashkournia, R. R. Chladny, J. Zhao, and C. R. Koch
University of Alberta, Canada
University of British Columbia - Okanagan, Canada

Abstract

In an effort to improve the efficiency of internal combustion engines, much focus has been put into variable valve actuation technologies in recent years. Electromagnetic solenoid valves can provide the cycle-by-cycle flexible valve timing needed for throttleless engine control or high efficiency combustion modes such as Homogeneous Charge Compression Ignition. One challenge with electromagnetic solenoid intake and exhaust valves is the robust control of the motion to achieve smooth landing under a variety of operating conditions. Promising algorithms have been demonstrated under test-bench conditions, but no work to date has demonstrated a robust electromagnetic valve-train on a functional engine that also satisfies soft landing and transition timing criteria. In this work, two previously developed valve motion controllers are experimentally tested on a single cylinder test engine. The controllers are compared for the opening transition of the exhaust valve with large variations in combustion pressure. A new control algorithm that combines favorable aspects of both methods is also presented. The new algorithm is shown to operate reliably under a wide range of operating conditions. An analysis indicates that the electrical energy consumed by the camless valve system is comparable to that of an equivalent conventional low friction cam-based valve train.

INTRODUCTION

BACKGROUND ON VVA

The emphasis on increasing efficiency and lowering the emissions of internal combustion engines is becoming an increasing area of focus due to the increase in fuel prices and stricter emissions regulations. The ability to alter the valve timing for the intake of air and exhaust of combustion products shows promise in dealing with this technical challenge. Variable Valve Actuation (VVA) has been shown to decrease fuel consumption by as much as 20% [1] and the ability to deactivate cylinders by turning off the valves may increase fuel efficiency further [2]. VVA has also been shown to decrease emissions of NOx by 20% [3] and decrease CO emissions by 66% [4]. Different technologies to modify valve timing exist and strike a compromise between complexity and the amount of control over valve timing. The goal of all variable valve systems is to improve the fuel economy of the vehicle. This means that both the system mass and power consumption requirements are important considerations. Note that all valve trains have some power requirements even a camshaft has a frictional component.

Mechanical valve trains which use VVA vary in complexity but almost always are limited in their flexibility to vary valve timing events independently. Some mechanical VVA systems incorporate variable lift [5], phasing [6] or have separate valve lobe profiles for different engine speed [7]. These systems add mass, do not offer full cycle-by-cycle control of the valve timing events except at low engine speed, and there is often a fixed relationship between the valve events. To fully utilize the capability of a variable valve train with both existing engine technology and future engine technology such as Homogeneous Charge Compression Ignition [8] cycle-by-cycle control of the charge in each individual cylinder for fast transients and combustion mode-switching is needed. Camless VVA systems provide full control of valve timing events.

One such camless system uses hydraulics to actuate each valve in an engine by the use of a hydraulic pump and proportional valves that direct the fluid. The disadvantage of these VVA systems is that they typically have high power consumption, are temperature sensitive, and require expensive servo valves [9][10].

VVA is also possible using electric motors that convert the rotational motion into linear motion. These systems can be temperature and wear compensating and can provide variable valve lift [11][12]. However, the motors required to compensate against gas forces add weight to such systems.

Electro Magnetic Valve Systems (EMVS) are another camless VVA system that uses independent electromagnets for each valve in order to achieve individual cycle-by-cycle control of each cylinder. As one example of this active area of research, a novel electromagnetic valve train has been developed by [13][14] with continuous lift and timing capabilities. Closed-loop control of the valve motion has been accomplished with different methods including: H-inf [15], sensorless [16], iterative leraning [17], repetitive [18]. Recent control strategies include an Energy-Based Key-On controller [19] or an inverse system method [20] of a linearized model which ultimately uses state feedback to control the valve motion [21]. An energy based soft landing controller has been developed by [22] and experimentally validated.

In this paper, the implementation of an EMVS on a single cylinder research engine and associated motion control strategies of the valves is presented. The actuators consist of two pre-compressed springs attached to an armature and valve with two electromagnets at each end of the armature travel as shown in Figure 1. The electromagnet has an attractive force to bring the armature in and hold the armature (and thus valve) at one end of the travel. The EMVS system has the following requirements: 1) soft landing to minimize engine acoustic noise; 2) ability to respond to large combustion pressure variations due to cycle-by-cycle load and valve timing changes; 3) power requirements that are comparable to conventional camshaft friction losses. The first two requirements require the use of closed loop motion control of the valve motion. To evaluate the third requirement, power consumption of the exhaust valve at different exhaust gas back pressures, representing engine load, is measured.

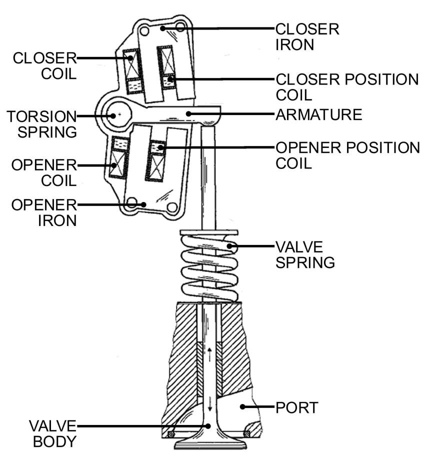

Figure 1: Conceptual illustration of an electromagnetic actuator [23]

In previous work [24][25], the authors have previously developed two EMVS valve motion controllers and tested each of them individually on a testbench that emulates an engine. The goal of this paper is to evaluate these two algorithms on a real engine. As a measure of the control performance, the range of exhaust backpressure for which the control provides reliable landing is examined.

This paper is divided into five sections. The preceding text consists of a brief background of variable valve timing with an emphasis on recent advances made in the area of EMVS. The next section describes the control problem for EMVS. The experimental engine setup is then described in the subsequent section. Experimental results and a discussion thereof are presented next and finally conclusions are reached.

ELECTROMAGNETIC VALVE CONTROL

Many authors have described different control strategies to achieve consistent low seating velocities for solenoid driven electromagnetic valve actuators. However, only [24] and [25] have attempted to cope with rapid changes in combustion forces on the valves without prior knowledge of the size of these forces and without cylinder pressure feedback. For a typical engine with fixed valve timing, the largest cycle-by-cycle pressure variation occurs on the exhaust valve. Exhaust pressure in an SI engine at valve opening typically varies from about 1 bar at low load to about 7 bar at full load (engine load variation of 1 to 14 bar IMEP) [23]. In addition, the flexibility of variable valve timing allows for the valve timing to be varied to achieve negative valve overlap to create internal residual gas. This means that both the intake and exhaust valve will be subject to a wide range of pressures that vary from cycle to the next as the valve timing is changed. However in this paper, the performance of the exhaust valve is used to test the robustness of the valve motion control while subject to cylinder pressure disturbances. Specifically, the control performance is evaluated over a range of different operating pressures during steady engine operation as well as during switching between two different pressures within one combustion cycle. Details of the two algorithms which were developed in simulation and then tested on a testbench simulating cylinder pressure are provided in [24][25].

The challenges of solenoid motion control for intake/exhaust valve actuation system can be summarized as follows:

- The large motion required by the valve (8 mm) results in a large air-gap during the valve travel which corresponds to a large gradient of available magnetic solenoid force and sufficient magnetic force to substantially influence the valve motion is only available close to the end of the stroke.
- Correspondingly, the inductance of the solenoid is relatively large in the region of control authority. That is, the mechanical time constant is of the same order of magnitude as the electrical time constant making landing control difficult. A large inductance gradient is also

responsible for a large back EMF which may also lead to current control instability.

- Each of the two coils for opening and closing a valve can only pull. Thus, if the valve approaches the end position with excessive speed, then the landing solenoid cannot slow the valve down. To ensure robust valve motion, particularly at low combustion pressures, the release solenoid needs to slow down the valve at the start of the motion.
- Modern engines have hydraulic lash adjusters to adjust for manufacturing tolerances and wear of the valve seat during the life of the engine. In the case of a solenoid valve train, this means that the valve and the armature are connected through the lash adjuster and this further complicates the control since both the armature and the valve landing must be controlled.
- Valve motion control requires an accurate estimate of the position and velocity.

In view of these challenges, tracking controllers with fixed trajectories are not feasible at large air gaps because of the low magnetic forces available. At small air gaps, sufficient magnetic force is available for tracking controllers, but due to the coil inductance, limited coil voltage (42V), and induced back EMF current, the catching coil needs to be initialized carefully and the reference trajectory needs to be chosen in such a fashion that it does not require fast changes in magnetic force. To accomplish valve motion control, the motion is divided into four distinct control stages of holding, release, approach and landing – as shown in Figure 2.

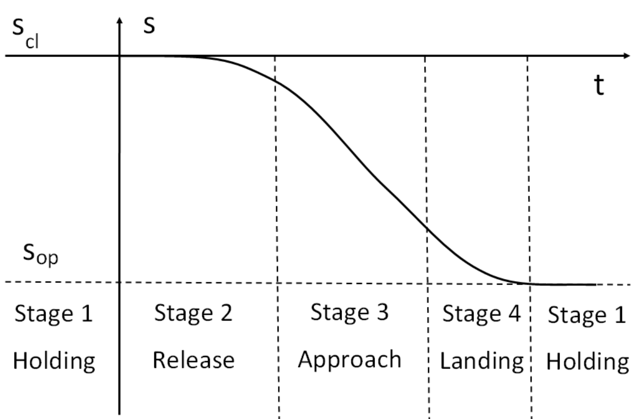

Figure 2: Valve motion control stages – s is valve position

The holding algorithm is responsible for keeping the valve in the end position. Typically, the solenoid current is held at a fixed value. The release control algorithm attempts to keep the maximum valve velocity at a level that leads to a compromise between fast and robust valve motion. Typically, a fixed feed-forward current trajectory is used for this purpose. The approach control algorithm compensates for friction and combustion losses. It also initializes flux of the catching solenoid for the landing stage. In both [24] and [25], energy based methods that are able to compensate for large changes in combustion forces are detailed. A disturbance observer is used by [24] to estimate the size of the combustion forces. The catching solenoid is then used to provide a restoring magnetic force that mirrors the estimated combustion force around the mid stroke position. The approach by [24] requires a sophisticated disturbance observer that is based on an analytical model of the combustion forces. The work of [25] uses a different technique, where an estimate of the gas energy lost is done by comparing potential and kinetic energies during approach to the potential energy at the end of the stroke. Then a magnetic force in the catching solenoid that restores the lost energy and simultaneously leads to a valve trajectory with no oscillations is used. A Lyapunov criterion is used in [25] to show that this approach strategy is stable.

The landing controller guides the valve along a reference trajectory into the valve end position – low valve seating velocity (< 0.1 m/s) are required to reduce acoustic noise and wear of the valvetrain. Since the magnetic force varies approximately with the inverse of the air-gap squared, a wide variety of nonlinear control strategies have been applied to this problem. A flatness based controller that uses a nonlinear magnetic model and a fixed end-trajectory is demonstrated in [24]. A controller with linearizing feedback that uses a lumped parameter magnetic model and a real-time generated trajectory that takes its starting values from the end values of the approach controller is demonstrated in [25].

Comparing the two algorithms presented in [24] and [25], one expects that the algorithm for the approach stage in [25] as well as the real time trajectory for the landing stage in [25] should be more robust than the corresponding algorithms in [24] since they are derived using strict stability criteria and they are both designed to adapt to large changes in combustion pressure. However, the flatness based landing controller in [24] should be more reliable than the landing controller using linearizing feedback approach in [25], since it guarantees exponentially asymptotic tracking and it is based on a more exact solenoid valve model.
In the following sections, details of the experimental implementation and performance of each of these two algorithms on a single cylinder engine are provided. Improved performance with a new third method which combines the most favorable portions of [24] and [25] is also demonstrated.

EXPERIMENTAL SETUP

An experimental single cylinder Ricardo Hydra Mark III single cylinder engine block is combined with a Mercedes cylinder head fitted with four electromagnetic valves [26] [23] and a custom piston [6]. A schematic of the actuator is shown in Figure 1 with one actuator for the intake and one actuator for the exhaust valve. Each valve actuator has a closer and opener magnet above and below the armature. The armature is connected to a torsion bar which pushes the valve open and a spring sits inside the cylinder head pushing back. Without activating the magnets, the armature is at equilibrium in the middle position of a total valve travel of 8 mm. To deduce the position of the armature, secondary coils that experience an induced current during changes in magnet path flux (as a result of armature movement) are used [27].

The actuators are mounted on a Mercedes head with a centrally mounted spark plug used for tests in SI. A Kistler 6061B in-cylinder piezoelectric pressure transducer is used to measure pressure on a 0.1 crank angle basis from an optical encoder as shown in the schematic in Figure 3. The pressure trace is also recorded by both the intake and exhaust valve motion controllers (dSPACE 1103 with a sample time of 50 kHz). The pressure is not used in the control algorithm, as a production engine would not have this signal, but it is recorded so that the performance of the valve controller can be examined. The desired valve timing in the four stroke engine cycle and the fuel energy injected is set by the engine controller (dSPACE 1401) and recorded for 200 engine cycles. All other engine operating parameters such as engine speed, intake air temperature, exhaust temperature, throttle position, and exhaust emissions are recorded at 10 Hz by the dynamometer test cell system. A detailed description of the experimental setup can be found in [28].

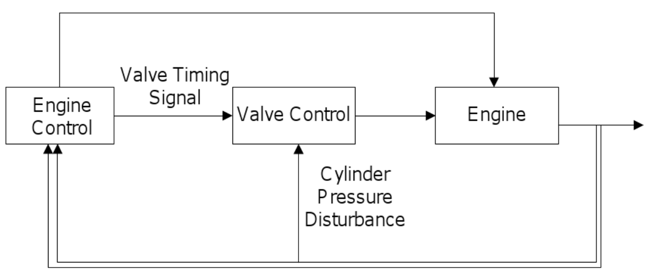

Figure 3: Schematic for the single cylinder engine and valve control

The engine parameters and operating conditions are listed in Table 1. The values of valve timing are given as the base timing which is used to start the engine and then kept constant unless otherwise specified. The engine operating conditions are given as the range of values used in the tests conducted comparing the valve algorithms.

VALVE MOTION CONTROL IMPLEMENTATION ON A SINGLE CYLINDER ENGINE

The two valve control methods in [24][25] demonstrated valve motion control on an experimental testbed that simulates combustion pressure but tests on functional engine were not performed. Implementation on an operational engine is expected to require additional considerations and constraints, some of which have been noted above. One aspect that is needed on a functional engine is an interface between the engine controller and the valve controller so that the valves are synchronized with respect to the correct crank angle in the engine cycle. This was accomplished by implementing a logical signal in the engine control to trigger each of the exhaust and intake valve controllers to open or to close. Robustness to real engine temperatures and combustion pressures on the valve controller is one of the main aspects that will be examined. Additionally, a number of changes are needed to be made to the mechanical valve setup, and consequently, the implementation of the control strategies. One aspect is that, since the valves do not have a hydraulic lash adjuster, it was essential to mechanically set the middle position of the valve so that control is possible. The lack of valve lash means that it is not possible to have a soft landing of both the valve and the armature so the acoustic noise of the valve train could not be fully minimized.

Table 1: Engine configuration and operating conditions for the Ricardo single-cylinder engine

Parameters	Values
Bore $\times$ stroke [mm]	97 $\times$ 88.9
Compression Ratio	13.9
Displacement [l]	0.653
Valves	2
Valve lift [mm]	8
Fueling	Port injection
IVO [bTDC]	300
IVC [bTDC]	180
EVO [bTDC]	-180 to -50
EVC [bTDC]	-300

Operating Condition	Values
Engine speed [rpm]	823-1594
Intake Temperature [C]	28
Intake Pressure [kPa]	69
Lambda [-]	0.94-1.10
Fuel Energy [kJ]	1.007
Octane Number [PRF]	100
External EGR [%]	0
Throttle [%]	0-23
Oil temperature [C]	50
Coolant temperature [C]	53

OPENING OF EXHAUST VALVES IN THE PRESENCE OF COMBUSTION PRESSURE

As discussed previously, the main valve motion disturbance occurs while opening against a variable combustion pressure. This pressure acts on the valve area and creates a force opposing the opening and can significantly change the velocity of the valve. As the valve opens, the pressure difference across the valve equalizes so the pressure at the instant the valve is commanded to open is used as a measure of the pressure. Experimental testing on the engine of the methods previously developed show that the fixed landing trajectories originally proposed by [24] works up to approximately 2.6 bar pressure at the start of the exhaust valve opening transition. This is remarkable since the control method is identical to the method developed using only the simulation model [23].

This indicates that the simulation model developed is a useful tool to develop motion control algorithms for these valves.

A real-time landing trajectory that starts at a fixed position s_{tr}, using a measured starting velocity v_{tr} as described by [25] is also tested on the single cylinder engine. However, the valve velocity deduced from the flux position sensor and a Kalman filter is too noisy under release conditions for establishing a reliable starting velocity, v_{tr}. To solve this problem, the transition velocity to the time of flight, t_{tr}, between the start of valve motion and the beginning of the landing trajectory is correlated and used as shown in Figure 4.

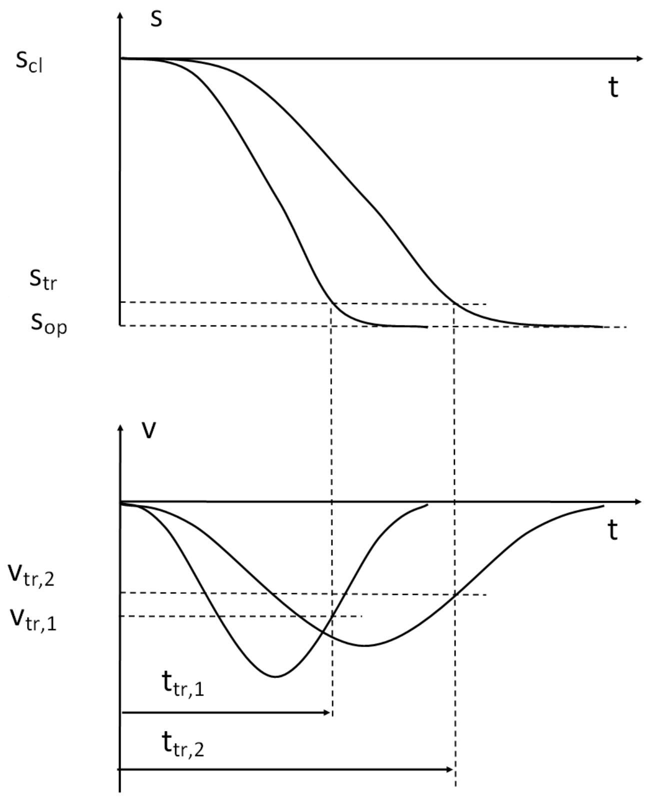

Figure 4: Schematic of transition velocity – v is velocity

The relationship between v_{tr} and t_{tr} can be established by numerically integrating the equations of motion for the actuation system with different combustion pressures. However, since there is considerable uncertainty in the disturbances present, a second order polynomial function was determined experimentally:

$$v_{tr} = -21.01 + 10495(t_{tr}) - 1418000(t_{tr})^2 \qquad (1)$$

EXPERIMENTAL RESULTS

This paper compares the control strategies proposed by [24] and [25] on a single cylinder engine. The robustness of the valve motion control to variations of combustion pressure on the exhaust valves and engine speed changes are examined. Also, the energy consumption of the actuation system is measured and compared to literature values of a conventional mechanical valve train.

COMPARISON OF CONTROLLER ALGORITHMS

Three different types of control methods are tested and are summarized in Table 2. The first row, denoted as Method 1, uses the method described in [24] for all of the approach, trajectory and landing stages. It is found that the reference and measured trajectories drift apart with increasing combustion pressure and above 2.6 bar (at exhaust valve opening) the method is not stable. The second row, denoted as Method 2, uses the method described in [25]. Here the control error is much smaller and combustion pressures up to 5.3 bar are possible. The third row blends the best stages of the two previous methods and is denoted Method 3. Method 3 combines the flatness based landing controller [24] of Method 1 with the approach strategy, and real-time landing trajectory [25] of Method 2. Method 3 is the most robust to exhaust pressure variations and operates up to 6.7 bar combustion pressure. On a conventional engine with standard valve timing this is close to the exhaust opening pressure of full load.

Table 2: Control Method Comparison

Method	Approach	Trajectory	Landing Controller	Max Pressure
1	1	1	1	2.6 bar
2	2	2	2	5.3 bar
3	2	2	1	6.7 bar

Figure 5 shows landing velocities during the opening transition at low combustion pressures and 800RPM for the three different control methods. Landing velocities improve for higher combustion pressures, since the valve approaches the end-position with a lower velocity and there is more control reserve available to guide the valve. Landing velocities are similar for all three methods, with Method 3 showing slightly lower values. At 1 bar back pressure, the landing velocity is approximately 0.4 m/s and at higher gas pressures the landing velocity reduces to close to 0.1 m/s. Valve seating velocities in the closing transition are expected to be lower at low gas forces then the ones shown here for the opening transition, since the energy imbalance between the open and closed position due to valve lash helps to slow down the valves and increases the control reserve.

Energy consumption between the three algorithms also was equivalent within the expected experimental error. Figures for energy consumption are presented in the last section of the experimental results.

COMPENSATING COMBUSTION PRESSURES

In this, and all subsequent sections, Method 3 in Table 2 is used. Figure 6 shows the results for the exhaust valve opening at 800RPM for steady state engine operation at combustion pressures between 1 and 6.7 bar. These pressures are obtained by operating the engine at an IMEP of approximately 4 bar and varying the

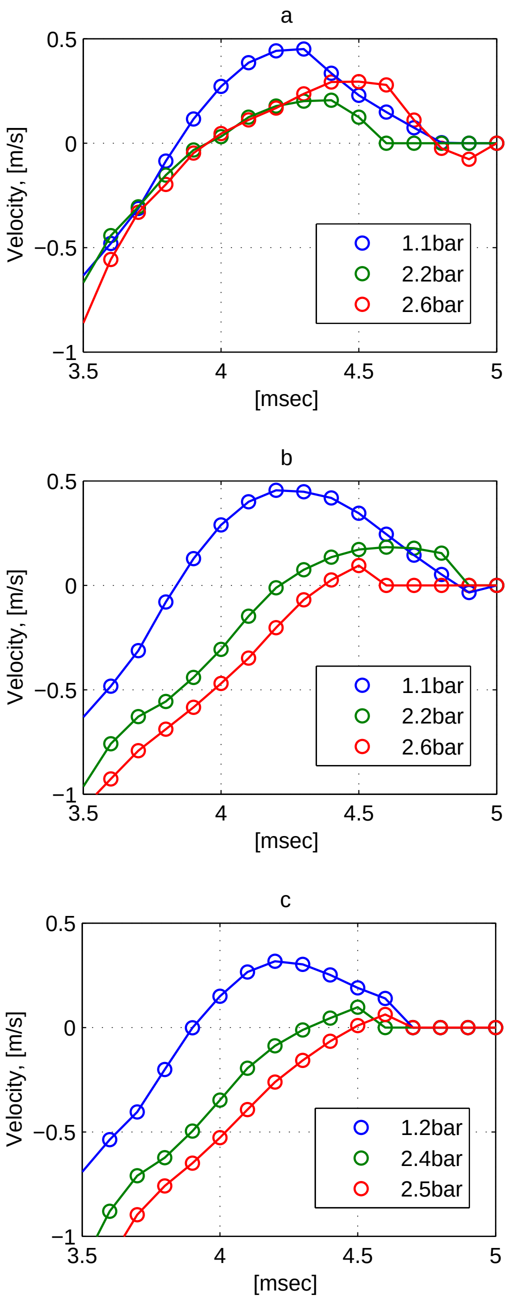

Figure 5: Exhaust valve landing velocities during opening (a) Method 1, (b) Method 2, (c) Method 3

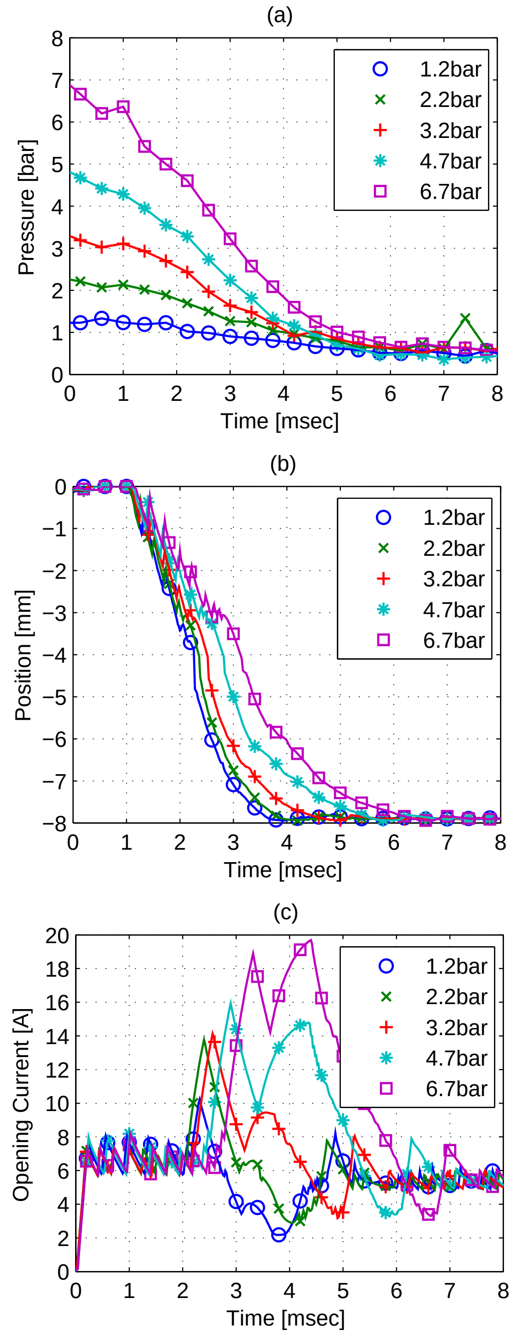

Figure 6: Exhaust valve opening performance for back pressures from 1.2 to 6.7 bar at 800RPM: (a) cylinder pressure, (b) valve position, (c) coil current

exhaust valve opening timing between 120 deg after top dead center (1.2 bar) and 50 deg (6.7bar). The measured in-cylinder pressure is shown in Figure 6(a) as the valve opens and the pressure in the cylinder decreases to the pressure in the exhaust system. Figure 6(b) and (c) show the valve position and armature current as a function of time and show that as the backpressure increases, the valve requires more time to open and considerably more coil current. The valve travel time increases from 4ms up to 6 ms when the back pressure is increased from 1 bar to 6.7 bar and the current required to pull the valve open increases from around 5A to 20A. This has a considerable effect on the energy consumption as detailed later.

The performance of Method 3 while subject to cycle-by-cycle exhaust pressure changes is shown in Figure 7. This figure show the in-cylinder pressure as a function of time in Figure 7(a) which is alternating between a starting pressure of approximately 4.9 bar and 2 bar. This is achieved by alternating the start of exhaust valve opening between 120 deg and 70 deg crankangle. The valve position as a function of time is shown in Figure 7(b) indicating that Method 3 can reject cycle-by-cycle combustion pressure disturbances of this size. This is an important result as it indicates that large cycle-by-cycle load change or valve timing change can be implemented for exhaust valve opening.

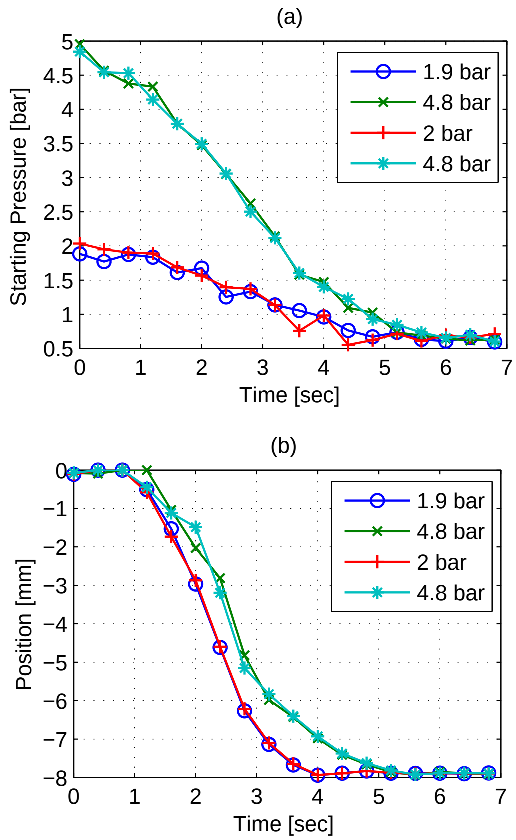

Figure 7: Two cyclic changes of back pressures from 2 to 4.8 bar at 800RPM to test exhaust valve open performance: (a) cylinder pressure, (b) valve position

Since the system is successfully able to compensate for this large change in back pressure, it is also concluded that the control system would be able to compensate for unpredictable drastic events such as misfire.

OPERATION AT VARIABLE ENGINE SPEEDS

Since the valve motion always takes approximately the same amount of time for a given opening pressure, it is critical to check the response at different engine speeds. The dynamometer test cell system is able to operate reliably up to 2000 rpm over which, control Method 3 showed no deterioration in performance.

ENERGY CONSUMPTION COMPARED TO CONVENTIONAL MECHANICAL VALVE TRAINS

Overall energy consumption of the actuation system was calculated by integrating the product of PWM voltage times the measured current. This calculation includes all losses in the actuator, but it neglects switching losses in the H-bridge driving the actuator. Ohmic losses are obtained by integrating the product of squared current and resistance. Comparing the two energy figures, one finds that ohmic losses are responsible for approximately half the energy consumption of the actuation system.

In a first set of experiments, energy consumption as a function of engine speed for two bar gas pressure was investigated. Exhaust valve opening was controlled using Method 3 and all other valve transitions were controlled with the original algorithm in [24]. The results are summarized in Table 3. It is found that intake and exhaust valves require substantially more energy to close than to open. This is attributed to the fact that the valve lash has moved the middle position of the spring assembly towards the open position and as a result, more energy is required to compress the closing spring than the opening spring. It should also be pointed out that the exhaust valves require substantially less energy than the intake valves in our setup. This is likely due to larger valve lash in the intake valves and also less time was spent parameterizing the controller for the intake valves. It is also interesting to note that the energy required to hold the valves in place is quite large compared to the energy required for transitions, especially at low rpm. However, the energy required for holding the valves reduces at higher rpm, since less time is spent for holding. In contrast, the opening and closing energies do not scale with rpm. The overall power consumption at low gas pressures then scales linearly with engine speed. This is shown in Figure 8, a plot of power consumption vs engine speed at 2 bar combustion pressure at three different engine speeds.

Table 3: Valve energy versus engine speed

	Intake		Exhaust	
RPM	800	1600	800	1600
Holding Energy [J]	3.4	1.6	2.0	1.0
Opening Energy [J]	0.8	0.8	0.5	0.5
Closing Energy [J]	1.4	1.4	1.1	1.1
Energy/Cycle [J]	5.6	3.8	3.6	2.6
Power/cycle/valve [W]	37	51	24	35

In a second set of experiments the gas pressure was varied at 800 rpm. Again Method 3 was used for controlling the opening of the exhaust valve and the original algorithm in [24] for all other transitions. The results are summarized in Table 4. Only the exhaust valve opening was affected by the change in combustion pressure. The most interesting observation in these experiments relates to the increase in energy required to open the exhaust valve against the combustion pressure. Combustion pressure slows down the valve during the opening transition, which actually helps to offset the energy imbalance between the open and closed position due to valve lash. As a result, the valve actuators only need to recover part of the work that the exhaust valves perform against the combustion pressure.

Figure 9 shows the linear relationship between energy consumption and combustion pressure that was observed in the second set of experiments..

In a typical 2.0-L, 16-valve four-stroke IC engine, the energy loss attributed to a single intake valve is between 2.1 and 3.35 J/cycle [29]. At fully open throttle, exhaust valves require an additional 0.8 J/cycle due to combustion

Table 4: Valve energy versus exhaust back pressure

	Intake		Exhaust	
Gas pressure [bar]	2	6	2	6
Holding Energy [J]	3.4	3.4	2.0	2.0
Opening Energy [J]	0.8	0.8	0.5	1.1
Closing Energy [J]	1.4	1.4	1.1	1.1
Energy/Cycle [J]	5.6	5.6	3.6	4.2
Power/cycle/valve [W]	37	37	24	28

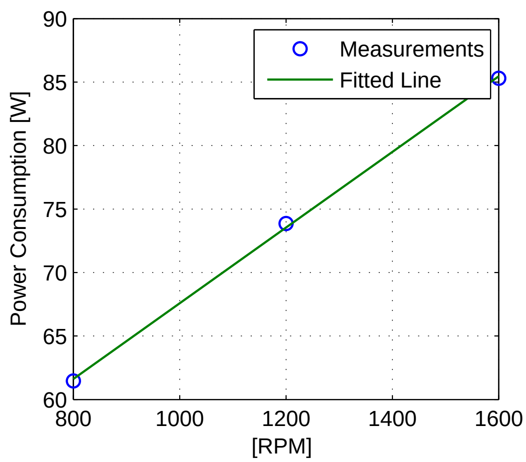

Figure 8: Measured power consumption as function of engine speed at a constant load of 2 bar

forces [30]. To compare the current electromagnetic system to these values, one can extrapolate the linear dependency of energy consumption both with speed and combustion pressure to obtain Figure 10 for a single cylinder with two exhaust valves and two intake valves. This linear assumption is likely only true for engine speeds up to about 4500RPM, where the increased pressure due to piston motion is relatively small and the pressure traces similar to the ones shown in Figure 6a are expected.

Overall Energy consumption with the EMVS is comparable to a low friction conventional valve train. However, a significant portion of the energy is consumed while holding the valve in the open and closed position. This loss could be substantially reduced by adapting these values. Although the energy is comparable, the payback in volumetric efficiency is a clear benefit of a EMVS system.

CONCLUSIONS

The valve control strategies proposed by [25] and [24] have been compared for exhaust valve opening transitions on a single cylinder test engine at the University of Alberta. Both algorithms showed robust valve control for large variations in combustion pressure. However, the best results were obtained by combining the approach controller and the real-time landing trajectories presented by [25] with the tracking controller in the landing stage proposed by [24]. This combination has been shown to compensate for 6.7 bar back pressure and cycle to cycle

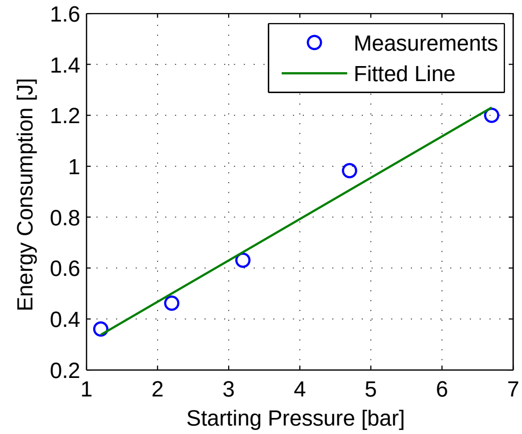

Figure 9: Measured exhaust valve energy consumption as function of combustion pressure at constant engine speed of 800RPM

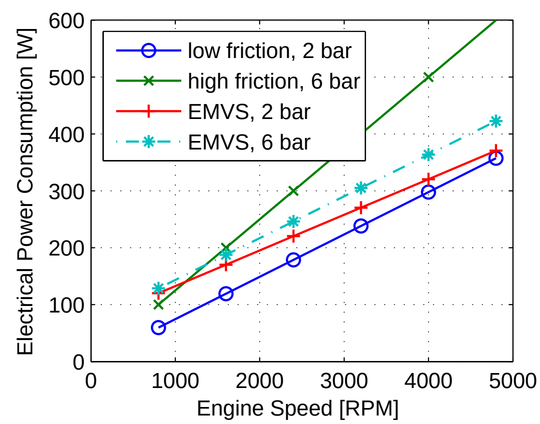

Figure 10: Cylinder power consumption at low and high valve pressure compared to a low and high friction conventional camshaft for varying engine speeds and four valves per cylinder

changes in back pressure of 2.7 bar. In addition, the energy consumption of the actuation system is measured for various engine speeds and loads and found to be comparable to a low friction mechanical valve train.

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