Progress in Camless Variable Valve Actuation with Two-Spring Pendulum and Electrohydraulic Latching

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ABSTRACT

Camless Variable Valve Actuation (VVA) technologies have been known for improving fuel economy, reducing emissions, and enhancing engine performance. VVA can be divided into electro-magnetic, electro-hydraulic, and electro-pneumatic actuation. A family of camless VVA designs (called LGD-VVA or Gongda-VVA) has been presented in an earlier SAE publication (SAE 2007-01-1295) that consists of a two-spring actuation, a bypass passage, and an electrohydraulic latch-release mechanism. The two-spring pendulum system is used to provide efficient conversion between the moving mass kinetic energy and the spring potential energy for reduced energy consumption and to be more robust at the operational temperature than the conventional electrohydraulic actuation; and the electrohydraulic mechanism is intended for latch-release function, energy compensation and seating velocity control. This paper presents the prototype design of a variable valve-time and two-lift LGD-VVA with bench and engine test results. The designed actuator is able to achieve 3 ms opening and closing response time with satisfactory valve seating velocity and low energy consumption. This is all achieved with a cost-effective design and open-loop control.


INTRODUCTION

For traditional internal combustion engines, engine intake and exhaust valves operate with a fixed lift and timing and some valve systems are capable of dual-lift and variable valve timing [1, 2, 3]. Engines with Variable Valve Actuation (VVA) systems are capable of continuously variable lift and timing at any given operational condition to minimize the engine pumping loss with optimized combustion to improve engine performances in fuel economy, emissions, and torque delivery [4, 5, 6].

Engine valvetrain systems can be generally grouped into cam-based and camless. A cam-based valvetrain system is based upon the traditional cam-system to drive the engine intake and exhaust valves with limited control over valve timing and/or lift [7]. A camless system drives individual engine valve directly with electromagnetic [8, 9, 10, 11], electro-pneumatic [12-13], or electrohydraulic [14] VVA systems. Without the restriction of the cam system, a camless system can control both valve lift and timing to achieve any desired target level that can be varied cycle-by-cycle. In addition, it provides control variations among engine valves and among engine cylinders. For example, it is able to provide asymmetric opening for two intake valves for a single cylinder, resulting in better air/fuel mixing. It can selectively deactivate one or more cylinders under low load conditions. Camless systems thus offer more control freedom and greater performance benefits.

Camless systems are the key technical enabler for 2/4-stroke-switch gasoline engine technology and air hybrid vehicles [15]. They are cost-effective with significant improvement of fuel-economy. Air hybrid vehicles have many advantages over electric hybrid vehicles. Also, a camless VVA system alone and combined with a direct injection fuel system is capable of extending the operation range of Homogenous Charge Compression Ignition (HCCI) [16]. Even high efficient diesel engines can benefit from
A family of VVA systems, called LGD-VVA, was designed by LGD Technology, LLC. They consist of a two-spring actuation, a bypass passage, and an electrohydraulic latch-release mechanism. The two-spring pendulum system provides efficient conversion between the moving mass kinetic energy and the spring potential energy, and its resulting force-displacement curve is in agreement with an ideal curve needed for a sinusoidal motion. The springs, for example, are able to absorb most of the kinetic energy right before the engine valve being seated so that the snubber does not have to accommodate as much energy as in a traditional electrohydraulic VVA system. Most of the actuation energy comes from the two-spring system, which is more robust than traditional hydraulic actuation, especially under a wide range of temperatures [18]. The electrohydraulic mechanism in the LGD-VVA system is used primarily for latch-release function, and it is also flexible enough in its design to offer different levels of energy input to overcome engine cylinder pressure variations, which indicates that the LGD-VVA system can be used for actuating exhaust valves. In addition, LGD-VVA does not rely on a position sensor to perform engine valve seating control, thus reducing the cost and improving reliability.

The LGD-VVA designs include three types of lift-control: fixed-lift, two-lift, and continuously-variable-lift. The engine valve release and actuation is triggered by one switch action of a four-way directional valve, and the engine valve completes the rest of the stroke, including seating, without active control [18]. The combination of the two-spring pendulum and bypass design is able to reduce fluid flow during most of the engine valve travel, thus achieving actuator power consumption comparable to that of a conventional cam system. For instance, with 8.0 mm valve lift, up to 64% of the kinetic energy is converted to the spring potential energy before the snubber is engaged, and the energy conversion continues even after the snubber is engaged. This combination makes it possible to achieve a “short-tailed” seating without the need for closed-loop control [18].

In an earlier study described in [18], a numerical model of the LGD-VVA designs was developed and validated in simulations to demonstrate the benefits of the LGD-VVA design principle. This paper presents the prototype results of the LGD-VVA system with both bench and dynamometer tests results. And this paper mainly focuses on the two-step LGD-VVA design.

**LGD-VVA DESIGNS**

**Basic Design Philosophy**

As reported earlier in [18], LGD-VVA design is based upon a two-spring pendulum structure, widely used in electromagnetic VVA (EMVVA) technologies [10-11], with an electrohydraulic latch-release mechanism. With the help of the two-spring pendulum actuation, the springs provide most of the actuation and efficient conversion between the potential and kinetic energy. Since the mechanical system is stable, the spring operation provides good displacement repeatability. Similar to the electromagnetic system, the LGD-VVA system reduces electrical-power consumption with the help of its electrohydraulic latch-release mechanism and stroke control mechanism; and it also provides robust valve-seating control due to its favorable force-displacement characteristics of the two-spring pendulum system. At last, the LGD-VVA system benefits from the effective energy conversion offered by its two-spring pendulum mechanism and the limited required hydraulic energy during the snubbing process, resulting in short seating duration and low seating velocity.

**Two-Lift Design**

The LGD-VVA family includes fixed-lift, two-lift, and continuously-variable-lift designs [18]. In the continuously-variable-lift design, the longitudinal position of the bottom of the actuator cylinder is controlled according to the target lift while the flow distribution is accomplished through a more elaborate flow port-and-circuit design to maintain proper pendulum actuation and flow bypass functions. This paper presents the two-lift design (as shown in Figure 1) and its experimental validation results. The details of its operation principle have been reported earlier in [18] through the system modeling and simulations. Briefly, the actuator consists of a piston, associated top and bottom piston rods, a hydraulic cylinder with a bypass and flow distribution mechanism that blocks external fluid flow during most of the stroke, and top and bottom springs that, together with all the moving mass, form a spring-mass pendulum driving mechanism for efficient energy conservation and consistency. The flow distribution mechanism also control fluid flow between the top of the cylinder and top port (Pt), and fluid flow between the bottom of the cylinder and the bottom port (Pb).

The actuator further includes top and bottom snubbers, each of which consists of an orifice, a check valve, and an optional relief valve. The top and bottom snubbers are designed to control the engine valve seating velocity and the opening terminal velocity, respectively. Fluid supply to the top and bottom ports are controlled by a fast-acting main valve, a two-position four-way switch valve (not shown in Figure 1), that is in turn actuated by a pilot valve (a two-position three-way switch valve, not shown in Figure 1). Both main and pilot valves are on-off valve. Since they are neither a proportional nor a servo valve, the cost of the VVA system can be reduced significantly. One set of pilot and main valves can be used to control either one VVA actuator for one engine valve or two VVA actuators for either two intake or two exhaust valves of the same combustion cylinder. The disadvantage of using one set of pilot and main valves to control two intake or two exhaust valves is that these two valves would have to be operated in the same pattern.
The actuator also includes a spring controller, connected to the spring control port $P_s$, which is supplied by a spring controller switch valve (not shown in Figure 1). Depending on the system architecture or control strategy, one spring controller switch valve may control a whole bank of VVA actuators. Under no or low pressure, the top and bottom springs are under lightly compressed state, thus the moving mass is in the neutral position. The moving mass includes the piston, piston rods and the engine valve assembly; see Figure 1 with the engine valve being closed. The switch action of the actuation valve (the pilot and main valves) causes the actuator to perform low-lift open and close actions; see [18] for details. Under high pressure, the top and bottom springs are under highly compressed state, thus the neutral position of the moving mass is such that the engine valve is half-way between the open and close positions. The switch action of the actuation valve causes the actuator to perform full-lift open and close actions; see [18] for details.

Four generations of prototype actuators have been designed and tested, and the last two designs are noted as VS3 (as shown in Figure 2) and VS4. All of them are intended for intake valve application. One of the key differences between VS3 and VS4 is the placement of the top spring and associated spring controller. For VS4, they are placed at the top of the actuator as illustrated in Figure 1. However, they were placed at the bottom of the actuator between the bottom piston rod and the bottom spring; see [18]. The key design parameters of the VS3 and VS4 are listed in Table 1. To be more specific, VS3 and VS4 have a peak stroke of 10 mm and 8 mm, respectively, which is by choice only. Each design is capable of lower or higher peak stroke value.

Table 1. Design Parameters

<table>
<thead>
<tr>
<th></th>
<th>VS3</th>
<th>VS4</th>
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<tbody>
<tr>
<td>Actuation piston diameter</td>
<td>11.55 mm</td>
<td>11.55 mm</td>
</tr>
<tr>
<td>Piston rod diameter</td>
<td>8 mm</td>
<td>8 mm</td>
</tr>
<tr>
<td>Engine valve mass</td>
<td>40 g</td>
<td>40 g</td>
</tr>
<tr>
<td>Piston and rods</td>
<td>33 g</td>
<td>26 g</td>
</tr>
<tr>
<td>Total moving mass (including spring dynamic mass)</td>
<td>126 g</td>
<td>115 g</td>
</tr>
<tr>
<td>Total effective spring stiffness</td>
<td>86 N/mm</td>
<td>76 N/mm</td>
</tr>
<tr>
<td>Natural frequency</td>
<td>131.5 Hz</td>
<td>129.4 Hz</td>
</tr>
<tr>
<td>Stroke</td>
<td>10 mm</td>
<td>8 mm</td>
</tr>
</tbody>
</table>

BENCH TEST RESULTS AND ANALYSIS

The bench test results described in this section were obtained on a VVA bench test system shown in Figure 3, which includes a test bench with associated hydraulic fluid supply system, a clamping fixture, a set of control and data acquisition system, and a laser displacement sensor.

The engine valve displacement is measured by a Laser triangulation displacement measuring device (LTC-050-20-SA) from MTI Instruments Inc. It has a linearity of $\pm 5 \mu m$, a resolution of $\pm 2.5 \mu m$, a frequency response of 20 kHz and a sampling frequency of 40 kHz. The valve velocity and acceleration are derived from the displacement data through differentiation.

The fluid supply system is capable of regulating hydraulic fluid to a tank temperature between 0°C and 100°C. On the other hand, the test fixture is directly exposed to the room
temperature. Under steady state, the fluid temperature through the prototype is approximately 10°C above the tank fluid temperature. The tank fluid temperature is used when the bench test results are presented and discussed in the rest of the paper. The related temperature sensitivity study should then be taken as directional, rather than precise.

![Figure 3. VVA bench test system](image)

**Engine Valve Opening and Closing Time**

Figures 4 and 5 illustrate, respectively, the valve opening and closing displacement profiles of a VS3 prototype, with SAE 0W30 engine oil as the working hydraulic fluid regulated at 55 °C and 14 MPa pressure. The valve opening and closing times, defined as the time duration between 1 and 99% of the stroke are 2.9 and 3.1 ms, respectively. Note that these valve opening and closing times are fast enough for most gasoline engine applications, and therefore, for diesel engine applications.

![Figure 4. The valve opening displacement curve (VS3 prototype and SAE 0W30 engine oil)](image)

![Figure 5. The valve closing displacement curve (VS3 prototype and SAE 0W30 engine oil)](image)

**Engine Valve Seating Velocity**

Engine valve seat velocity is the valve travel speed when the valve is seated against the engine valve seat. For gasoline engines, the seating velocity is required to be less than 0.5 m/s at normal engine speed and lower than 0.05 m/s at idle speed. This requirement is due to the requirement of engine valve durability and noise constraint. It is also required that the valve seating velocity should not be sensitive to temperature and it is controllable under different engine speed within the limit of the durability consideration. The seating velocity is normally increased as engine speed increases in order to indirectly shorten the seating time to ensure adequate valve open window for proper intake and exhaust valve operations.

The seating velocity presented in this paper was defined as the slope of the valve displacement near the displacement of 0.1 mm. In Figure 5, the VS3 prototype has a seating velocity of 0.27 m/s under the conditions shown in the figure. In Figure 7, the VS4 prototype demonstrates almost a constant seating velocity over a wide temperature range, from 0°C to 90°C, which is great for an electrohydraulic VVA system. In Figure 7, the displacement traces under various temperatures are adjusted in time to have the same starting time for easy comparison of the seating velocities. It can also
be observed from test data that the engine valve opening/closing delay for a fixed control switch signal increases as the hydraulic fluid temperature decreases. This is due to the increased fluid viscosity as temperature decreases. In practical applications, this temperature induced delay variation can be properly compensated through calibrations.

Figure 7. VS4 prototype seating velocity under different fluid temperatures (SAE 0W30 engine oil under a system pressure of 8 MPa)

It is important to be able to adjust the seating velocity at different engine speed since it affects the valve closing period between 0.1 mm and the fully closed position. Figure 8 shows that the seating velocity can be well controlled by the hydraulic fluid pressure. For the VS4 prototype the valve seating velocity increases with the system pressure and the seating velocity is 0.12 m/s and 0.5 m/s under a system pressure of 7 MPa and 11 MPa, respectively. With this controllability, it is feasible to achieve a low seating velocity for the low-noise idle operation and sufficient short closing period for high speed operation to ensure adequate air exchange. Our goal is to optimize the VVA design so that the seating velocity is close to 0.05 m/s at low supply hydraulic fluid pressure, which could be achieved, for example, by reducing the size of the snubber orifice.

Figure 8. VS4 prototype seating velocity under different system pressures (SAE 0W30 engine oil under a fluid temperature of 40 °C)

Valve Displacement Repeatability

In order to have smooth engine operation with improved fuel economy and reduced emissions, the valve opening and closing repeatability shall meet certain requirements, for instance, less than ± 1.0 to 1.5 crank degrees in the phase angle. Although the current design did not consider the repeatability requirement, a limited number of repeatability tests were performed using VS3 design. As shown in Figure 9, the VS3 prototype test was repeated 20 times, and the valve opening and closing time variations were between 0.04 ms and 0.11 ms, which is equal to 0.23 and 0.67 crank degrees in phase angle domain for an engine operated at 2000 rpm; and equals to 0.69 and 2.01 crank degrees for an engine operated at 6000 rpm. Repeatability performance is therefore satisfactory.

Figure 9. The VS3 prototype repeatability test (20 consecutive tests with 0W30 engine oil at 55°C and under 10 MPa)

As shown in Figure 10, similarly, the VS4 prototype actuator was repeated 20 times, the valve opening and closing time variations were between 0.046 ms and 0.075 ms, which equals to 0.26 and 0.46 crank degrees in phase angle domain for an engine operated at 2000 rpm, and equals to 0.79 and 1.37 crank degrees for an engine operated at 6000 rpm. The VS4 sample is therefore even better than the VS3 sample for the repeatability.

Figure 10. VS4 prototype seating velocity under different fluid temperatures (SAE 0W30 engine oil under a system pressure of 8 MPa)
Figure 10. The VS4 prototype repeatability test (20 consecutive tests with 0W30 engine oil at 55°C and under 10 MPa)

The valve displacement curves shown in Figures 9 and 10 are split between opening and closing, leaving out generous dwell time (defined as the time between the end of opening and the beginning of closing) for these particular tests. Among four generations of LGD-VVA prototypes tested so far, the shortest dwell-time varies between 0 ms to 2 ms, which depends mainly on response times of the pilot and main hydraulic valve.

VVA System Energy Consumption

Table 2. VVA System Energy Consumption (VS4 sample one thermal cycle, 0W-30 engine oil at 55°C)

<table>
<thead>
<tr>
<th>Lift</th>
<th>8 mm (under 7 MPa)</th>
<th>1 mm (under 2 MPa)</th>
</tr>
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<tbody>
<tr>
<td>Fluid Flow Energy</td>
<td>2.88 J</td>
<td>1.76 J</td>
</tr>
<tr>
<td>Pilot Valve Electrical Energy</td>
<td>0.20 J</td>
<td>0.23 J</td>
</tr>
<tr>
<td>Total Energy</td>
<td>3.08 J</td>
<td>1.99 J</td>
</tr>
</tbody>
</table>

Depending on design details and operating conditions, a traditional cam-driven valve system of a conventional gasoline engine consumes between 1.5 and 4.0 Joule energy for each open-close cycle (including one opening action, one closing action, and the dwell time). As shown in Table 2, the VS4 prototype energy consumption is 3.08 Joules and 1.99 Joules for high lift (8 mm) and low lift (1 mm), respectively. This is fairly comparable to the traditional cam system. Furthermore, it is also possible to reduce the VVA system energy consumption by half, for example, under certain operating conditions, by operating one, instead of both, of the two valves for intake or exhaust valves. The energy losses in shown in Table 2 include the electrical power consumption of the pilot valve and the internal leakage flow without hydraulic pump loss. However, they do not include system losses, such as accumulator energy costs, hose compliance and dissipation, temperature effects, etc.

ENGINE INTEGRATION AND DYNO TEST

A VS4 prototype assembly is integrated on to the cylinder head of a Ford Mondeo engine, a 2.0L Duratec, where the cylinders 1 to 3 are operated by conventional cam driven valvetrain and for cylinder 4 the exhaust valves are driven by the cam shaft and both intake valves are fitted with the LGD-VVA actuators to drive two intake valves independently. The VS4 prototype assembly is first tested on a hydraulic bench (Figure 11), with the engine valves fitted on to partial engine head.

Figure 11. Hydraulic test bench for VS4 actuator

The cylinder head of the test engine is then modified for the VS4 prototype assembly to integrate the two intake valves onto the number four cylinder; see Figure 12. The corresponding intake cam shaft segment was cut off. The camshaft position sensor, which is normally located on that cam shaft segment, is relocated to the exhaust cam shaft. A Hall-effect position sensor was employed to replace the original sensor, and the magnet was installed on the timing gear while the sensor was integrated with the camshaft bearing.

While LGD-VVA does not require close-loop displacement control, it is helpful to have a displacement or position sensor in the experiment since it is important to measure the valve seating velocity on an engine. It is also desirable to measure the open and close status of the engine valves for diagnostic purposes. Since the Laser triangulation displacement sensor cannot be used for displacement measurement after the VS4 prototype assembly is integrated onto the engine, the non-contact differential variable reluctance transducers (Model NC-DVRT-1.5 from MicroStrain, Inc.) are integrated into the actuator body to measure the displacement of one of the spring retainers in the LGA-VVA actuator. Due to the space limitation, the sensor was arranged in such a way that the sensor displacement output is not a monotonic function of the displacement,
certain interpretation is required in the control software to obtain the measured displacement.

A separated hydraulic supply station was built to provide necessary fluid flow and pressure control for the VS4 prototype assembly. The engine is installed on a motoring engine dynamometer. To avoid unnecessary complication during the engine start-up, the engine is motored to a target speed before the fuel is injected.

The VS4 prototype assembly is controlled by a Gongda developed VCU (VVA Control Unit), which communicates through CAN with a customer ECU (Engine Control Unit) based upon a Mototron engine control system. The OEM ECU is therefore not used. The VCU contains various calibrations and control functions particular to the operation of the VS4 prototype actuators. One of features is to convert calibrations and control functions particular to the operation designed to be able to operate up to 6000 rpm. Due to safety during the engine start-up, the engine is motored to a target speed before the fuel is injected.

The VS4 prototype assembly was successfully operated with combustion in all cylinders under engine speed between idle to 3500 rpm. In fact, the VVA system is designed to be able to operate up to 6000 rpm. Due to safety reason, the engine operation was limited to 3500 rpm. Even though the test data is not available for turbocharged (or boosted) engines, we are confident that LGD-VVA system can be used, with proper spring pre-load, in boosted engines; and it can also be used to accomplish traditional valve operations such as Miller and Atkinson cycles, early and later intake valve closing. With its camless nature, the system would perform favorably in dynamic engine control, especially with large cycle-to-cycle timing change.

**FUTURE WORK**

Even though significant progress has been made for the LGD-VVA system, work remains to move the LGD-VVA system to a production ready status. The following is a list of planned work:

- Design optimization for repeatability, reliability, packaging, manufacturing, and cost with intensive repeatability and durability tests.
- Development of exhaust valve actuators and continuously-variable-lift actuators.
- Engine packaging and integration study, including integrated hydraulic system and control, and full engine VVA assembly, with and without exhaust VVA actuators for both gasoline and diesel applications.
- Control strategy development and fuel economy validation either on a single-cylinder or multi-cylinder engine.

**CONCLUSION**

A camless variable-valve-timing and dual-lift LGD-VVA actuator was designed, prototyped, and bench validated. During the bench tests, the prototype actuator, with its on-off open-loop control (without help of the displacement sensor), achieved 3 ms opening and closing time. More significantly, consistent seating velocity was achieved over a hydraulic fluid temperature range between 0°C and 90°C, furthermore, the seating velocity can be controlled from 0.12 m/s to 0.5 m/s by regulating the supply hydraulic fluid pressure between 7 MPa and 11 MPa. The low energy consumption is also validated during the bench tests, and the test result show that the actuator consumed, over an opening and closing cycle, 3.08 joules and 1.99 Joules of energy for a lift of 8 mm and 1 mm, respectively. This compares favorably to a conventional cam-driven valve, which consumes between 1.5 and 4.0 Joules of energy. The actuator also showed reasonable repeatability in displacement. Finally, two LGD-VVA actuators were integrated into a Mondeo 2.0L Duratec engine to drive both intake valves. Smooth engine operation was demonstrated at engine speed between idle to 3500 rpm.

**REFERENCES**


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DEFINITIONS/ACRONYMS/ABBREVIATIONS
ECU - Engine control unit
EHVVA - Electrohydraulic VVA
EMVVA - Electromagnetic VVA
LGD - LGD in LGD Technology, LLC
OEM - Original equipment manufacturer
Pt - Top port
Ph - Bottom port
Ps - Spring control port
VCU - VVA control unit
VVA - Variable Valve Actuation

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