SAE TECHNICAL PAPER SERIES

2000-01-1221

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Reprinted From: Variable Valve Actuation 2000 (SP–1523)

The Engineering Society For Advancing Mobility Land Sea Air and Space®
INTERNATIONAL

SAE 2000 World Congress
Detroit, Michigan
March 6-9, 2000

400 Commonwealth Drive, Warrendale, PA 15096-0001 U.S.A. Tel: (724) 776-4841 Fax: (724) 776-5760

ISSN 0148-7191

Printed in USA

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ABSTRACT

Compromises inherent with fixed valve lift and event timing have prompted engine designers to consider Variable Valve Actuation (VVA) systems for many decades. In recent years, some relatively basic forms of VVA have been introduced into production engines. Greater performance and driveability expectations of customers, more stringent emission regulations set by government legislators, and the mutual desire for higher fuel economy are increasingly at odds. As a solution, many OEM companies are seriously considering large-scale application of higher function VVA mechanisms in their next generation vehicles.

This paper describes the continuing development progress of a mechanical VVA system. Design features and operation of the mechanism are explained. Test results are presented in two sections: motored cylinder head test data focuses on VVA system friction, control system performance, valve lift and component stress. Dynamometer results from a four-cylinder test engine focus on fuel consumption, emissions, torque and combustion stability characteristics. The paper finishes with a description of proposed future activities.

INTRODUCTION

Variable Valve Actuation (VVA) has recently attracted significant attention as a viable technology to meet future fuel economy, emissions and engine power requirements. In addition, unlike lean-burn Gasoline Direct Injection, VVA does not require significant technology developments in exhaust after-treatment to meet projected benefits. This paper describes the progress to develop a VVA system. Before presenting motored head and operating engine experimental data results, a brief mechanism description is presented.

Experiments on the VVA system are ongoing. To date, the VVA mechanism has been installed on a four-cylinder engine and tested on both an engine dynamometer and motored head test stand. Aside from the typical instrumentation modifications such as pressure transducers and thermocouples, the engine has only been modified to fit the mechanism to the cylinder head and the cam drive system. No changes were made to the combustion chamber, manifolds, ports and cylinder block.

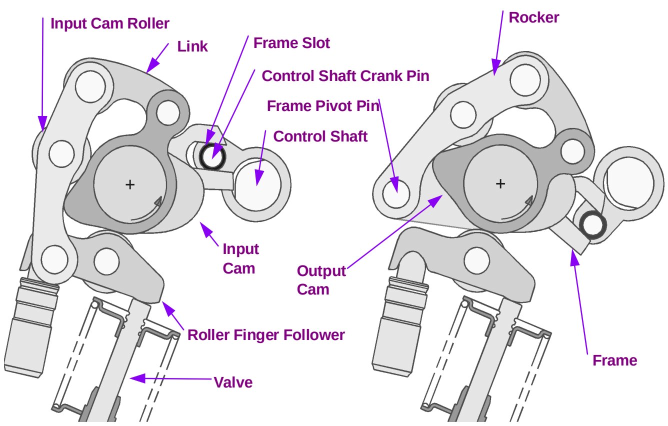
Many of the theoretical benefits described in an earlier paper[1] including increased fuel economy and reduced NO_x emissions have been realized using the first pre-prototype hardware sample. However, HC emissions have increased at some operating points. Also, engine stability (measured as COV of IMEP) at very low loads has degraded somewhat. These results were not unexpected because optimization of other engine design parameters for VVA has not yet taken place.

As other researchers have noted[2], attaining the full benefits of VVA requires re-optimization of engine design parameters such as the family of valve profiles, and combustion chamber and inlet port geometry. Additionally, the flexibility offered by VVA opens up new possibilities for engine control algorithms and software. To this end, activity has begun in these areas, which may be reported on in the future.

MECHANISM DESCRIPTION AND OPERATION

The mechanism is a mechanical, continuously VVA system. The system simultaneously varies lift, duration and phase; i.e., for any single control shaft position, there is a fixed lift, duration and phase relationship. A complete description of the mechanism and system has previously been published[1].

The pre-prototype system provides the following capabilities: 0 - 9 mm lift range, 0 - 290 crank degree duration adjustment and 0 - 80 crank degree peak lift phase change. The VVA mechanism is basically a linkage and cam device that replaces the conventional engine camshaft. Fig. 1 shows end views of the pre-prototype mechanism in the high and low lift positions (note: some parts have been partially removed to clarify the moving components). Fig. 2 shows another perspective of the mechanism configuration (again some parts have been removed to aid clarity) and the resultant kinematic family of valve lift curves.



High Lift Position   Low Lift Position

Figure 1. Roller Finger Follower VVA Mechanism

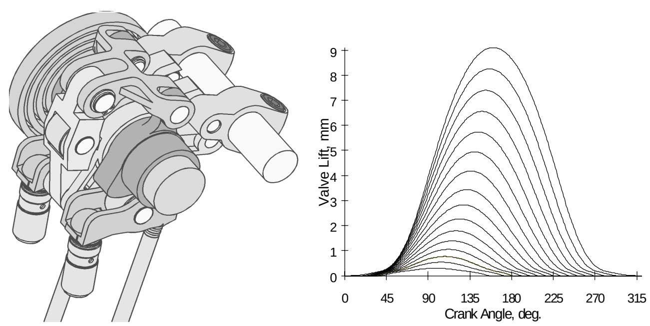


Figure 2. VVA Mechanism Configuration and Lift Curves

The input cam, which is similar to a conventional engine cam in shape and in driving methods, always remains in phase with the engine crankshaft. The input cam interfaces with a roller mounted in the rocker. The rocker oscillates approximately 20 degrees about the frame pivot pin. The opposite end of the rocker is pinned to a link that is also pinned to the output cam. As the rocker oscillates, it pulls the link, which causes the output cam to oscillate. The output cam, which has a cam profile on a portion of the outer surface, oscillates 40 degrees by the linkage mechanism during 360 degrees of input cam rotation, regardless of overall mechanism orientation. As the output cam oscillates, the cam profile interfaces with the roller of the roller finger follower, which opens and shuts the valve. Torsion spiral springs were installed in the pre-prototype hardware but some testing was performed with laboratory spring setups which limited speeds to 3200 rpm.

Variable valve events are achieved by repositioning parts of the mechanism about the input cam centerline. The rocker and input cam roller, link, frames, output cams and all associated pins comprise these re-positionable parts. A control linkage that is positioned by a computer-controlled actuator determines the orientation of these parts about the cam centerline. Since the roller finger follower does not rotate with the mechanism, the output cam must oscillate the same amount the mechanism is rotated from the peak lift position, before the output cam profile moves the roller finger follower and the valve. This produces the variable lost motion effect of the mechanism and allows the mechanism to reduce valve lift and duration. Since the mechanism is designed to oscillate the output cam 40 degrees, the peak lift phase change between maximum and zero lift is also 40 cam degrees (Fig. 2). To elaborate, if the mechanism is rotated 15 degrees clockwise from the maximum lift position (Fig. 1), then 15 of the 40 degrees of the output cam rotation will result in lost valve motion before the valve is lifted. The location of the peak lift for low lift valve events advances because the roller, installed in the rocker, changes phase relative to the input cam.

The pre-prototype family of lift curves was designed based on engine simulation modeling when the project was started. A commercially available software package was used to analyze the dynamic effects of pressure waves, mass flows and energy losses in engine manifolds and cylinders. However, the effects of mixture charge fluid flow on combustion initiation and quality were not available with this software, and thus those effects were not predicted.

The VVA maximum valve lift and duration curves were designed to be greater than the baseline, but the highest valve lift was limited to the same value as the baseline engine lift curves in order to reduce the impact on the hardware design. Higher valve lift events require significant valve train hardware re-design and while possible, were beyond the scope of this first pre-prototype hardware build.

The baseline engine was modeled in the simulation software and the baseline performance determined. Then the potential valve lift curves of the VVA mechanism were entered into the simulation software input data set and the model was re-run. The VVA mechanism capabilities of reduced lift, Early Intake Valve Closing (EIVC) and Late Intake Valve Opening (LIVO) were predicted to provide the best pump work reduction for partial loads.

The VVA hardware is installed on a European market, 1.6 l, four-cylinder engine. This particular engine was selected at the project initiation because it both represented modern engine design and was controlled with a familiar engine management system (EMS). (The particular engine selection does not imply any development program with the engine or vehicle manufacturer.)

Fig. 3 shows a cross-section through the cylinder head and the mechanism. The selected baseline engine utilizes a Direct Acting (DA) valve train, so it was first converted to a Roller Finger Follower (RFF) valve train before VVA mechanism installation. Engine modifications were limited to the valve train portion of the cylinder head and the associated cam drive system; the engine block remained unchanged.

Fig. 3 also illustrates the impact of the VVA mechanism on cylinder head packaging. The original valve train was

DA so the camshaft centerlines were aligned with the valve centerlines. As stated earlier, the DA valve train components were replaced with RFF components and the camshafts were moved inward accordingly. The exhaust valve train (shown on the left side of the figure) could be considered representative of a standard RFF valve train. The VVA mechanism (illustrated on the right side of the figure) increased the end view foot print area, but not so substantially that it becomes a packaging impossibility. The figure also shows that the VVA mechanism only slightly increases the valve train height (though the mechanism is not shown in its maximum extended position). For the selected application engine, the valve cover height was not changed, only widened, to accommodate the VVA mechanism.

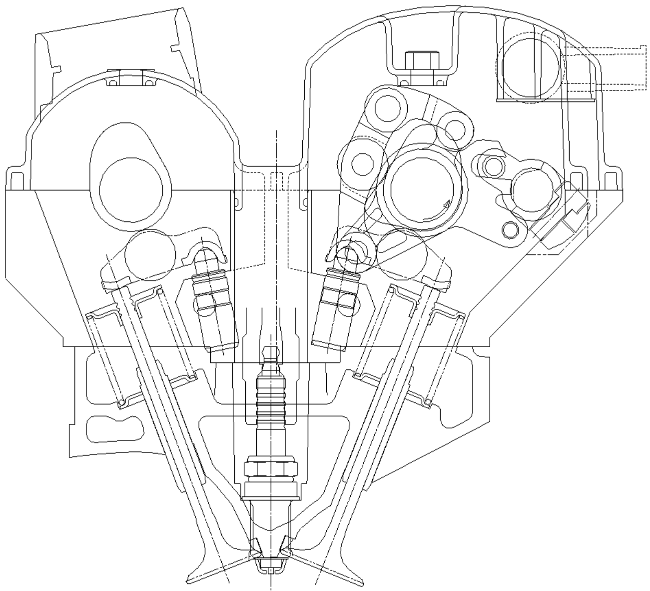


Figure 3. Dynamometer Engine Cross-section

Fig. 4 shows one proposal for an actuator installed in a cylinder head. Ultimately, the actuator may be installed within the cylinder head leaving only the control shaft position sensor extending outside head casting. The actuator is connected to the mechanism control shaft that is installed parallel to the camshaft.

Some key actuator and control sub-system performance criteria are time response, power consumption, position accuracy and repeatability. Fig. 5 shows a typical control system step response at idle speed. The actuation system in this figure moves the peak valve lift from 6.3 mm to 0.1 mm in approximately 100 ms and returns in approximately 120 ms. These response characteristics are a function of the actuator power, required lift change and mechanism speed. A system full-scale displacement time response of 300 ms. is probably adequate for most applications.

The actuator power, in this particular design, constantly fluctuates even when the VVA mechanism position is not changing. The camshaft torque oscillations cause similar (though reduced) control shaft oscillations. The closed loop control system applies motor current to counter the torque oscillations and maintain control shaft position. The actuator consumes more power at high peak valve lifts than at low peak valve lifts due to the greater average control shaft torque at high peak lifts.

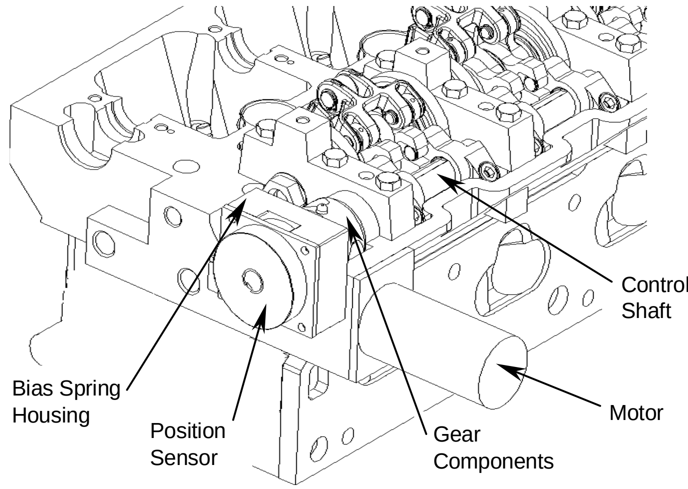


Figure 4. Actuator installed in cylinder head

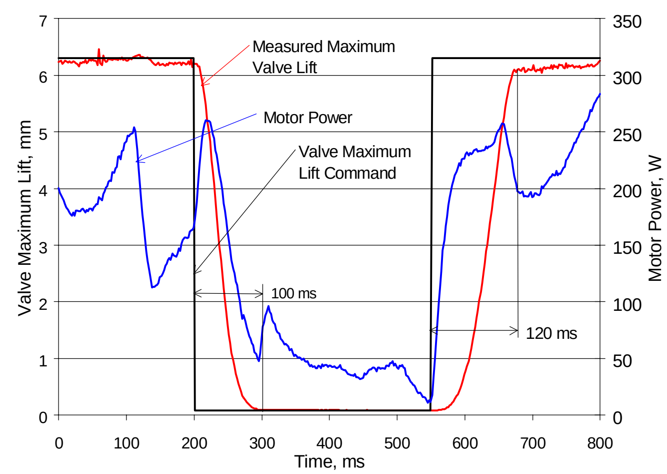


Figure 5. Control System Step Response

All components were stress analyzed and designed for infinite life. Fig. 6 shows the results of a typical Finite Element Analysis (FEA) stress analysis of the output cam. This particular analysis also includes an assessment of the stress at the roller bearing contact.

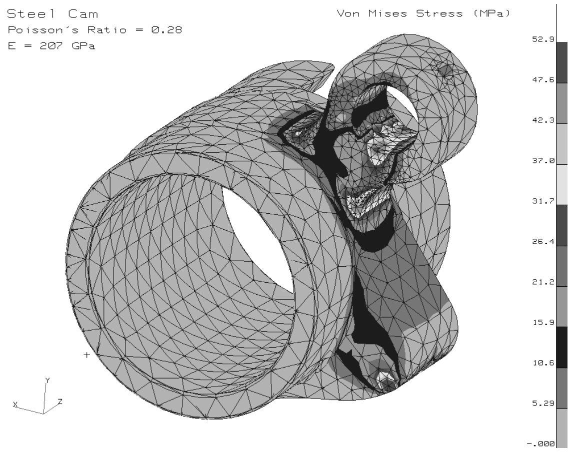


Figure 6. Typical Component FEA Results

CURRENT PERFORMANCE: MOTORED HEAD TEST STAND DATA

SETUP – The motored cylinder head test stand contains a platform to which the cylinder head (without engine block) is affixed. An electric motor drives the intake cam pulley through flexible shaft couplings and a torque transducer. In addition, there are protective transparent safety covers that totally enclose the running hardware, a pressurized heated oil supply system and a variable speed control system. The VVA mechanism is instrumented with camshaft position, valve acceleration, oil pressure, component strain gage and valve lift proximity transducers. The transducer signals are supplied to computer controlled data acquisition systems and later processed in spreadsheet software.

RESULTS – Fig. 7 shows typical valve acceleration data for 3500 rpm plotted with kinematic analysis data. This figure also shows that the natural frequency of the pre-prototype VVA mechanism is about 1000 – 1200 Hz. Likewise, the peak valve acceleration rate was designed to be fairly low at 0.0125 mm / deg^2. When this hardware was designed, the mechanism stiffness and natural frequency were not known, so a conservative valve acceleration value was selected. Future design efforts will concentrate on increasing the mechanism stiffness, natural frequency and peak acceleration levels.

The VVA mechanism friction was expected to be less than the baseline RFF on a standard drive-cycle test procedure. The system, once installed in a vehicle, will operate for a significant percentage of time at low valve lifts, and therefore should require less average valve train power than a production-engine RFF valve train. Fig. 8 shows early measures of the VVA mechanism intake cam torque (friction) values. The cam torque for low lift was less than a baseline RFF for low speed and low lift, but greater for high lifts and high speeds. A significant friction reduction during the pre-prototype mechanism wear-in period was also observed.

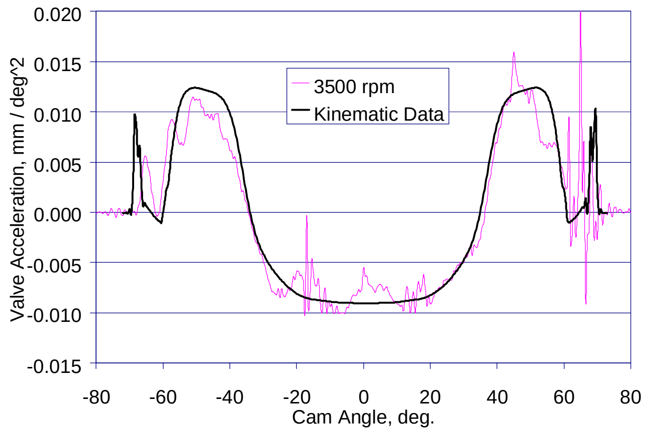


Figure 7. VVA Valve Acceleration at 3500 rpm / 8.9 mm lift

The measured torque (friction) also tended to decrease with increased cam speed for the speed range shown. This trend is attributed to the development of better bearing lubrication conditions between moving parts (which, all but one pin joint, are pressure lubricated).

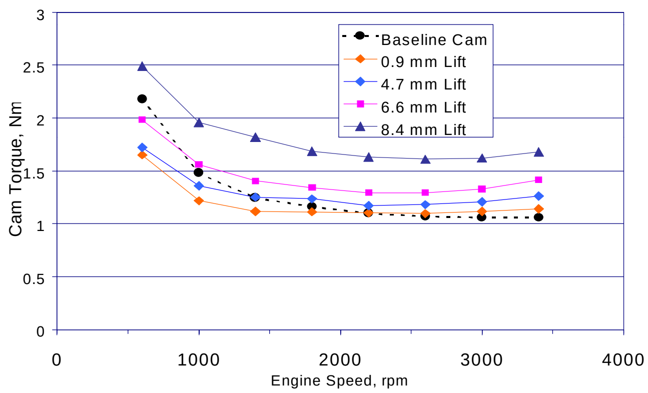


Figure 8. VVA Mechanism Torque

CURRENT PERFORMANCE: FIRED ENGINE DYNAMOMETER DATA

SETUP – The engine was broken-in and baseline tested in its production (DA) configuration. The VVA (RFF) cylinder head with the modified cam drive system was then installed. Because many compromises inherent with fixed cam timing engines were eliminated, it was beneficial to reset the engine's initial cam timing. This was adjusted with vernier cam pulleys to optimize engine performance over a matrix of speed / load points. The VVA cam timing is depicted in Fig. 9; the baseline valve lift curves are shown for reference. For clarity, only a few of the possible VVA lift curves are shown, however, an infinite series of lift curves are possible.

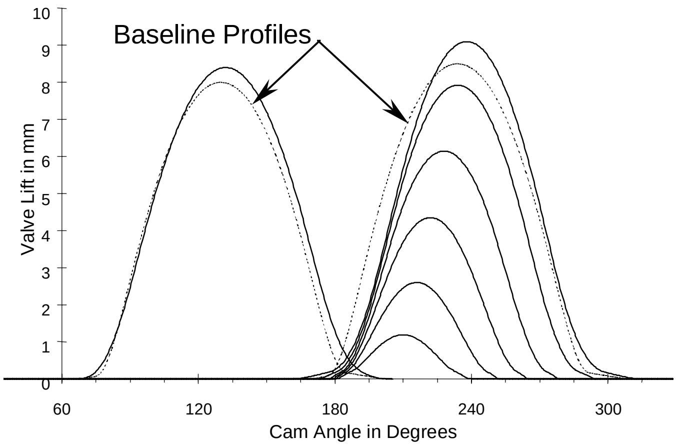


Figure 9. VVA Cam Timing and Baseline Lift Curves

The dynamometer test cell utilized a 375 kW DC dynamometer. The engine was instrumented with in-cylinder pressure transducers, thermocouples, manifold and oil pressure transducers and emission analysis benches. The engine was tested without accessories (alternator, power steering pump, etc.). The test cell supplied temperature controlled engine coolant and oil. Inlet air temperature, pressure and humidity were controlled with a conditioned air supply to promote consistent emissions results. The intake and exhaust systems were of production design, except the catalytic converter was removed and backpressure was set using an orifice valve.

Data acquisition and analysis of combustion related parameters were performed with a DSP-ACAP system. All engine testing was conducted over a minimum of 500 complete engine cycles for each data point. A Pressure - Volume (PV) diagram, displayed in real time on a computer monitor, was used to visually monitor combustion quality. Combustion stability was judged statistically with the COV of IMEP as well as through other stability metrics.

Most testing to date has been without Exhaust Gas Recirculation (EGR) or other dilution control means. Work in this area is currently underway and may be the topic of a future report.

The throttle plate was retained on the test engine so that experiments involving partial throttling would also be possible, if desired. The data presented herein represents that obtained with the throttle blade in its “full-open” condition with an intake manifold absolute pressure (MAP) of approximately 100 kPa.

Before discussing results, it is worthwhile to define some terms and conventions used in this paper.[3] Fig. 10 shows a typical PV diagram for a conventional, throttled engine, enlarged to provide increased intake and exhaust detail . The sum of the areas labeled “A” and “C” define a quantity termed “Indicated Mean Effective Pressure” (IMEP). Area “B” + Area “C” is termed “Pumping Mean Effective Pressure” (PMEP). Most researchers use IMEP (A + C) to denote the positive work performed by the gases upon the piston while PMEP (B + C) denotes the negative work required to accomplish the gas exchange process during the intake and exhaust strokes. Net Mean Effective Pressure (NMEP) is the difference between IMEP and PMEP, and is thus equal to A – B.

In reality, the quantity represented by area C is fictitious, as it reflects a path not actually traversed by the moving piston. Calculating either area A or area B by itself, though, is computationally intensive, so it is more common to include area C in the others. Normally, this does not present a problem, as C is cancelled out when calculating NMEP anyhow.[4] The distinction becomes more relevant when EIVC strategies such as with VVA are employed. These strategies reduce area “B” (which represents the truly lost work), while not appreciably changing spurious region “C”. Thus, the percentage reduction in PMEP (B+C) that results from EIVC is less than the true reduction in lost work. The term “Net Pump Loss” is used to define area B alone and is preferred because it provides a more accurate representation of VVA benefit than PMEP.

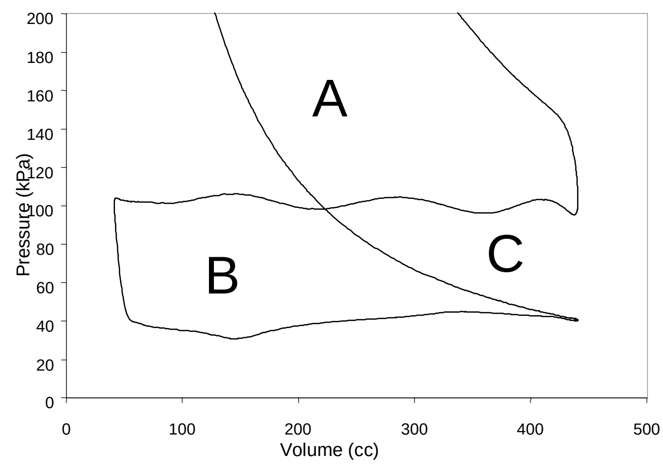


Figure 10. Conventional Engine PV Diagram

Figs. 11 - 13 illustrate typical PV data from mid to low range power operation points at 2000 rpm. This data shows pump loop area (and thus Net Pump Loss) reduced up to 70 - 80% versus the baseline engine, consistent with thermodynamic theory associated with a throttle-less, EIVC strategy. The pumping work improvement increases with decrements in engine load for the VVA engine when compared to the baseline engine. However, relative to the baseline case, the pressure rise after ignition drops and the slope of the pressure trace during the expansion stroke starts to flatten as load decreases. These characteristics, in combination with other quantitative measures of burn rate, indicate that the VVA engine burn rate slows down more than the baseline engine as load decreases.

At very low loads this trend has translated into a burn variability increase and a combustion stability decrease. The primary cause of this is believed to be a decay of cylinder mixture motion following EIVC so that the mixture is quiescent at time of combustion initiation.

Fig. 14 displays the effects of VVA upon idle speed operation. A wide sweep of loads were tested to provide for the range of accessory and transmission loads The data demonstrates that fuel economy improvements of approximately 12% are achieved throughout the range, but the engine stability currently degrades at very low loads faster than the baseline condition.

Figs. 15 – 20 shows the effect of VVA on engine performance over the full range of load at 2000 rpm. This engine speed occurs with high frequency in typical driving and is thus particularly important for emissions performance and pleasibility to the driver. These data in these figures were obtained using stoichiometric fuel / air ratio and 95 Research Octane Number (RON) fuel. Spark advance was set to Minimum for Best Torque (MBT) or to borderline knock, if that occurred first.

Fig. 15 shows that at 2000 rpm, the application of the VVA mechanism to this engine (without additional engine design changes) reduces the Brake Specific Fuel Consumption (BSFC) in the low to middle load regions by 5 – 7%.

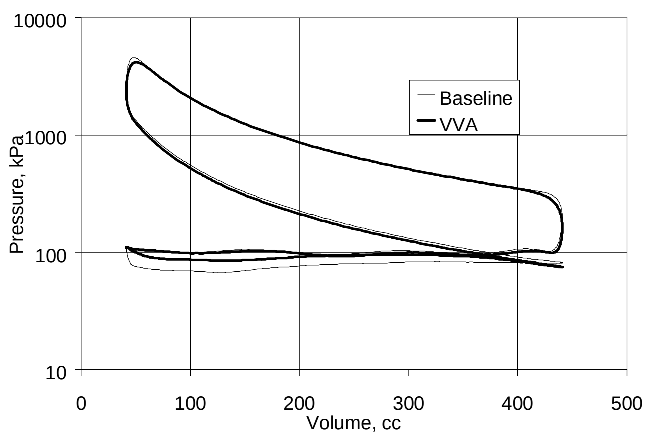


Figure 11. Pressure v. Volume: 2000 rpm / 800 kPa NMEP

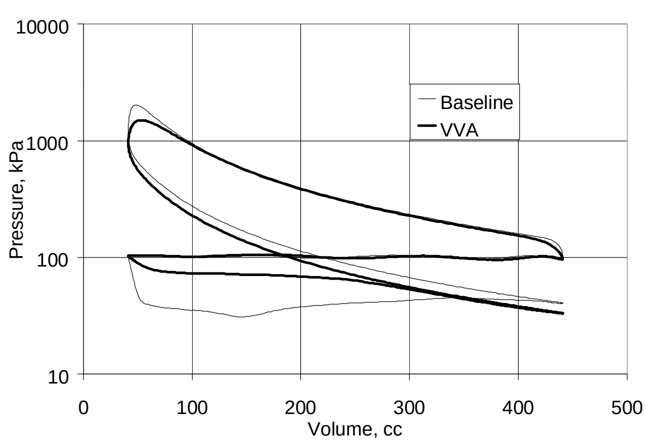


Figure 12. Pressure v. Volume: 2000 rpm / 300 kPa NMEP

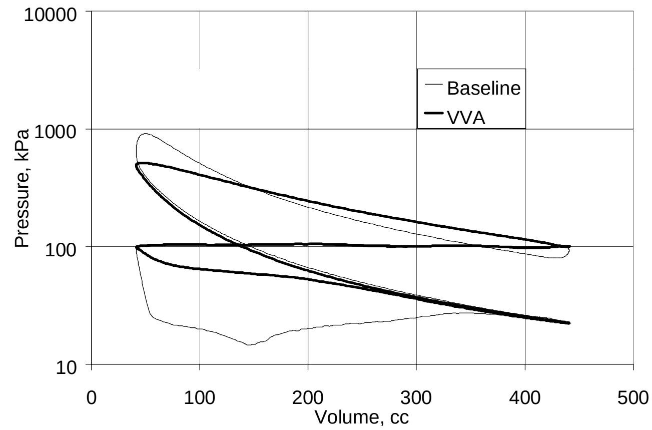


Figure 13. Pressure v. Volume: 2000 rpm / 100 kPa NMEP

Most of the fuel economy benefit is a direct result of reduced pumping losses. Figs. 16 and 17 provide these results in two forms. Fig. 16 is the more traditional, but less accurate, PMEP, which includes both areas “B” and “C” as defined in Fig. 10. It has been included for completeness (and convention), but “Net Pump Loss”, shown in Fig. 17, provides a better measure of the reduction of the *lost* pumping work.

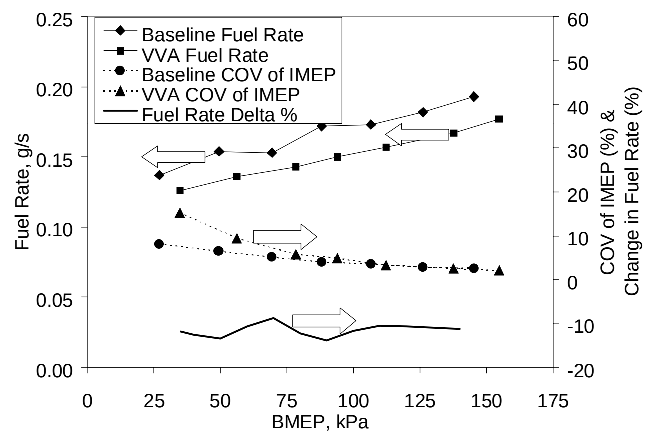


Figure 14. Idle Performance at 750 rpm

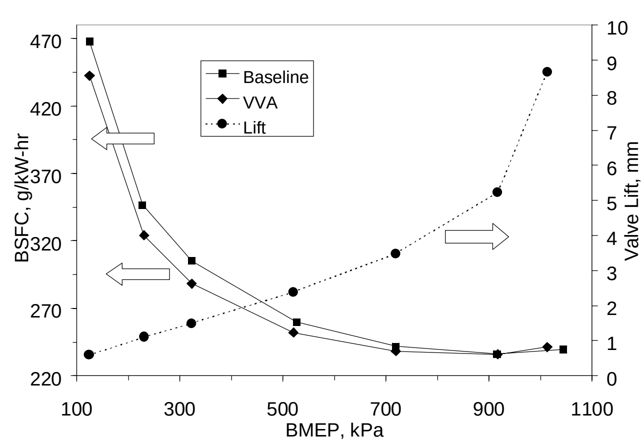


Figure 15. BSFC v. BMEP at 2000 rpm

Fig. 18 compares engine combustion stability (COV of IMEP) between the baseline and VVA configurations. VVA combustion stability is essentially identical to the baseline case at engine loads higher than 300 kPa BMEP. Between 200 and 300 kPa, combustion stability degrades with VVA, though still acceptable (COV of IMEP below 3%). However, at loads lower below 200 kPa BMEP, combustion stability is greater than 5%, which is generally considered unacceptable. Improving this low load performance is one of the targets of future development discussed later.

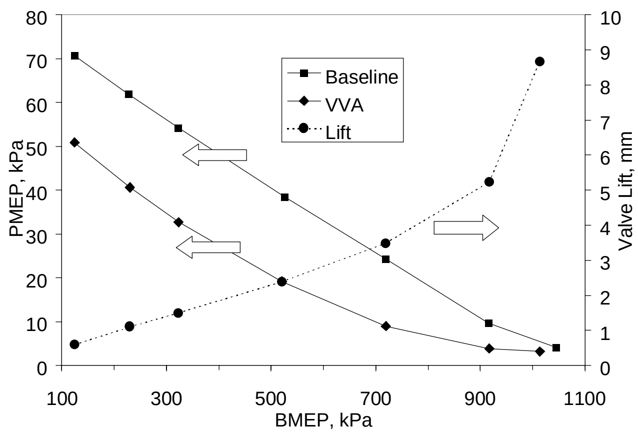


Figure 16. PMEP v. BMEP at 2000 rpm

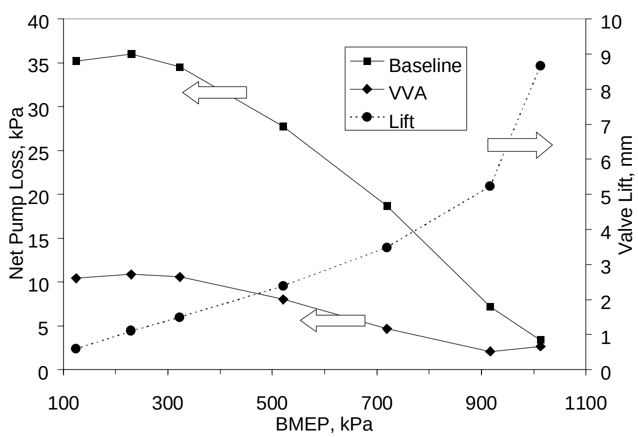


Figure 17. Net Pump Loss v. BMEP at 2000 rpm

Figs. 19 and 20 present engine-out HC and NO_x specific emissions at 2000 engine rpm. These data once again represent the effect of adding the VVA mechanism to the baseline engine without any additional engine modifications. It is not surprising that HC emissions increase under these circumstances since in-cylinder mixture motion has degraded with the EIVC characteristics of the VVA mechanism.

Engine simulation on a similar 2.2 l, four-cylinder engine with RFF profiles for both the VVA version and baseline engine showed up to an 11% gain in torque and NMEP. When these same profiles were run on a 0.6 l, single-cylinder, hydraulic valve research engine[1] (which has a different combustion chamber), a gain of up to 4% was observed.

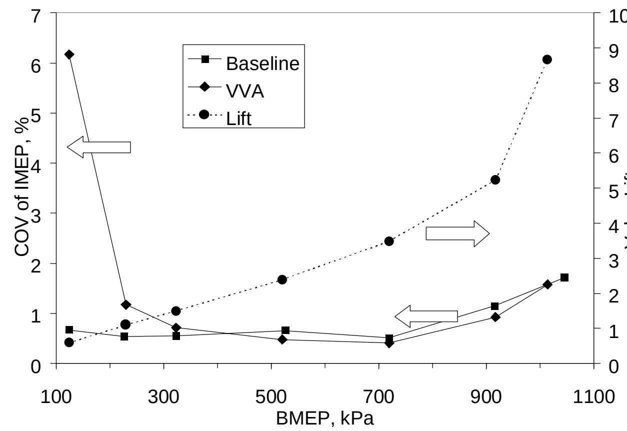


Figure 18. COV of IMEP v. BMEP at 2000 rpm

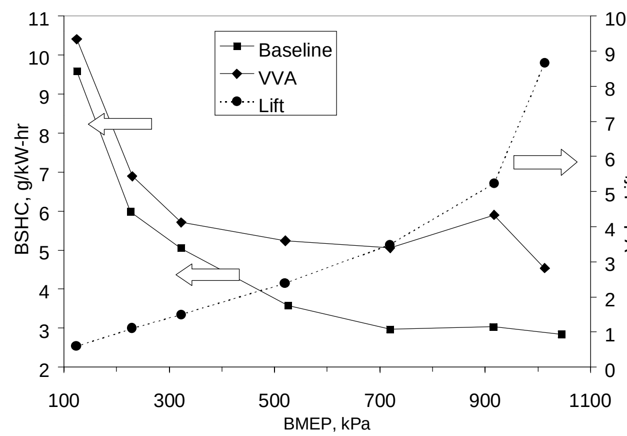


Figure 19. BSHC v. BMEP at 2000 rpm

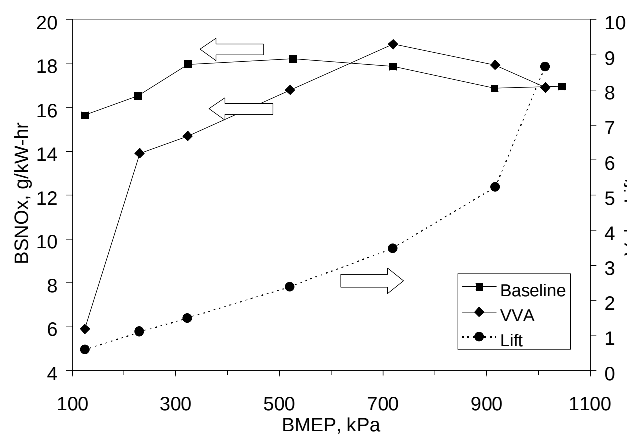


Figure 20. $BSNO_x$ v. BMEP at 2000 rpm

Initial results on the test engine have also shown torque and NMEP improvements within the same range predicted. Figure 21 shows that low to mid rpm (1200 – 3200 rpm) NMEP increased by 1 to 12% in 7 of the 8 points tested, with one point showing a drop of 1%. The +12% occurred at 1600 rpm value, while the –1% occurred at 2000 rpm. Both should be considered atypical as they reflect a shifting of the tuning point downward from 2000 rpm to 1600 rpm with the application of this VVA mechanism. In the absence of this effect, an average 4% increase improvement is to be expected over this speed range. The full load data shown in Fig. 21 was obtained under the following operating conditions: 107 RON fuel, stoichiometric fuel / air ratio, and MBT spark timing.

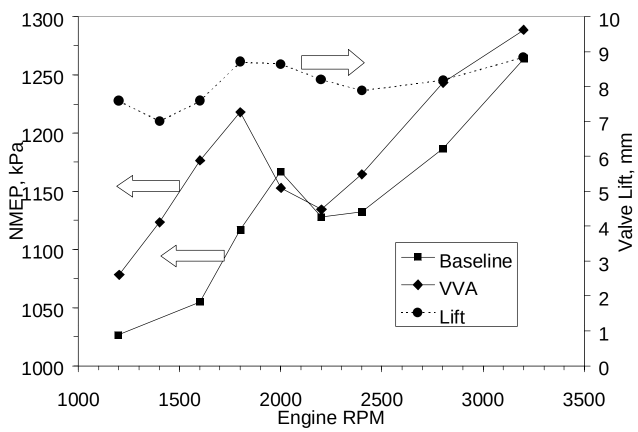


Figure 21. Full Load NMEP v. Engine Speed

CONCLUSIONS

The application of the VVA mechanism to a standard baseline European four-cylinder engine has shown promise and identified areas for further development. The application of this VVA mechanism demonstrated BSFC improvements of approximately 12% at idle, 7 - 10% at low to middle load, and 0 - 3% at middle to high load. Over the low to mid rpm (1200 – 3200 rpm) range, peak torque improved by an average of 3%.

HC emissions performance has generally degraded as a result of reduced mixture motion caused by the EIVC with low valve lift and the as yet non-optimized engine design parameters for this VVA mechanism. NO_X performance (without charge dilution) at low to moderate loads has generally proven much better with VVA. This is however, at least partially due to the lower peak pressures present with the slower combustion that accompanies EIVC.

FUTURE WORK

The development of this VVA technology continues. The goal is to further improve fuel economy and achieve EURO IV (EC 2005) emission levels while (at least) maintaining baseline engine power levels. Future plans include hardware modifications and additional motored-head and dynamometer testing with an expanded scope of speed / load points.

In addition, vehicle system, engine control and combustion system design development is also under way. Several missions are planned for this facet of the program:

- Demonstrate that the results obtained in dynamometer / motored head stand testing can be successfully applied to a vehicle This will take place within the constraints of tailpipe emissions, driveability, packaging, total system cost and reliability.
- Develop optimal alternatives or solutions to traditionally vacuum-dependent ancillary systems such as dilution, evaporative emissions purge, power assist brakes and pneumatically actuated passenger compartment climate controls.
- Develop software controls to optimize the new features provided by this VVA system.
- Explore options to improve combustion performance such as combustion chamber design, mixture swirl and / or tumble, port geometry, low valve lift timing and duration, lift curve shape and ignition system alternatives.

ACKNOWLEDGMENTS

This paper is the product of a team effort. In particular, the authors wish to thank: Mr. Harold Bonisteel, Mr. John Castellana, Mr. Thomas Fischer, Mr. Ryan Fogarty, Mr. Daniel Glueck, Mr. William Haslett, Mr. Nicolas Hendriksma, Mr. Robert Hogeman, Mr. Daniel Kabasin, Mr. Ronald Kunst, Mr. Timothy Kunz, Mr. Jon Lampert, Mr. Timothy Landschoot, Dr. Jongmin Lee, Mr. Jeffrey Rohe, Mr. Mark Spath, Mr. David Trapasso, Mr. Richard Wagner and Mr. James Zizelman.

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