"VLD" a flexible, modular, cam operated VVA system giving variable valve lift and duration and controlled secondary valve openings

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Abstract:

It is becoming more widely understood that advanced VVA systems will be required for future diesel engines to enable advanced combustion strategies such as HCCI. The cost, however, of implementing such systems is commonly seen as a problem.

Affordable VVA systems require low engineering, investment and piece costs to achieve a high cost-benefit ratio.

Engineering and investment costs can be minimised by the use of modular, interchangeable systems that allow maximum commonality between the major engine component assemblies, such as the cylinder head and covers. These systems allow the valve train type to be specified at build time, such that a standard valve train or various levels of VVA can be built into the engine as legislation and product differentiation dictate.

This paper describes a modular, camshaft-operated family of valve train solutions that range from a standard valve train through to advanced VVA systems for both intake and exhaust. Such a family has been built into a common cylinder head package, to give various levels of functionality. Variable valve lift, variable open period and modulated secondary valve openings and the practical combinations of these functions are presented along with their potential uses.

This type of modular design is relatively simple, whilst still meeting all the packaging constraints imposed on modern diesel engines.

This paper describes functional hardware resulting from using this design approach. Dynamic test data is also presented from a motored cylinder head which includes high-speed laser velocity measurements of the valve motion and characterisation of the performance of the VVA actuators under a range of operating conditions.

The hardware will now be used for fired engine testing where the real benefits of exhaust valve re-opening and intake valve duration control can be measured.

Keywords: VVA, Variable valve lift and duration, Exhaust valve re-opening, diesel effective compression ratio, HCCI.

1. Introduction

The application of VVA to gasoline engines has been well documented with many learned papers discussing the operational effects of the various strategies, from simple camshaft phasers, [1] through to advanced variable lift systems [2]. For a variety of reasons, however, this has not been the case with diesel engines.

The valve to piston clearance limitation at TDC in diesel engines has meant that phasing systems have little positive effect and consequently more flexible VVA systems are required. The difficulty is that more flexible VVA systems are more invasive of engine design and more expensive in both prototype and production quantities. Therefore until recently, while progress in fuel injection and turbocharging could provide significant gains within existing engine architectures, there was little incentive to thoroughly investigate more advanced VVA systems.

Consequently, there is little published information discussing the effects of using these systems on diesel engine performance, fuel economy or emissions, but a sample are [3], [4], [5] and [6].

However, recent investigations into alternative combustion systems such as HCCI, along with the requirement for greater control of in-cylinder conditions to improve emissions and increase specific output, have led to a much increased interest in VVA for diesel engines.[7], [8], [9]

Whilst cam phasers do not offer sufficient functionality, variable duration systems, variable lift systems and those capable of generating controlled secondary valve openings do offer desirable functionality for application to diesel engines. However, while these systems avoid any valve to piston contact limitations, they are necessarily more complex than a conventional valve train.

Despite the better understanding of the requirements of VVA for diesel engines and the availability of systems with the required functionality the relative complexity and cost of implementing a suitable system remain barriers to adoption:

The cost of installing a VVA system falls broadly into three areas:

1. Engineering costs
2. Investment costs
3. Piece costs

The crucial factor is the number of parts over which the engineering and investment costs, can be amortised. This is strongly affected by modularity and parts sharing across a range of engine derivatives.

More advanced VVA systems for diesel engines usually add some dynamic mass to the valve train, and result in reduced overall system stiffness. Thus a well optimised design is needed to produce systems with the required functionality that are "well-behaved" dynamically.

The final part of any VVA application is the control system. The major question usually being "how fast does the actuator need to be?" This question is a result of needing to understand the interactions between the VVA system and other engine systems such as the turbo-charger and after-treatment. However, very high speed actuation is typically expensive and may require power sources that are usually not available on automotive engines. Therefore, it is conventional to use hydraulic actuators powered by engine oil to control VVA systems. The question then becomes "how fast is the actuator" over the engine operating envelope as this type of actuator is very dependent upon oil viscosity (temperature) and oil pressure (engine speed.)

These points are discussed below for Mechadyne's VVA system known as VLD.

2. VLD functionality

2.1 Construction of the VLD system

The VLD system is a highly flexible VVA system that produces variable valve lift characteristics by adding together two cam profiles. Figure 1 shows an isometric view of the full system and a section through its main components. The system is controlled by altering the relative phase between the blue and red cam profiles and a continuous range of valve lift characteristics is available between the extreme settings.

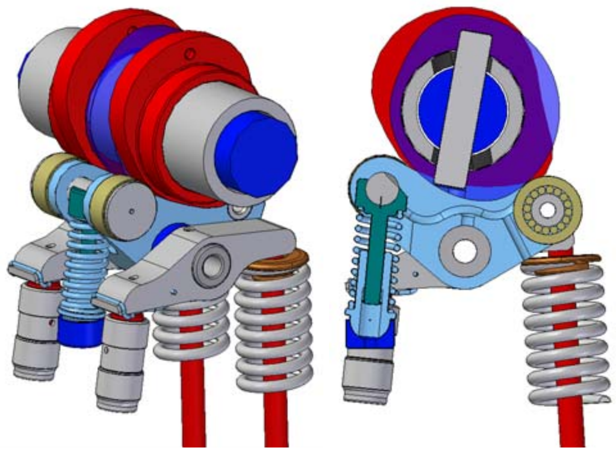

Figure 1: The VLD system – Main components

2.2 Operating characteristics of VLD

The system allows for either fixed opening or fixed valve closing for single lifts as shown in Figure 2, and a large degree of flexibility in the range of valve lift variation, as shown in Figure 3. A variety of different secondary valve lift strategies are also possible, two of which are shown in Figure 4.

Although this paper concentrates on the applications to DOHC light duty diesel engines, this system is also applicable to gasoline engines for valve head load control, and has derivatives that are applicable to both light and heavy duty SOHC diesel engines.

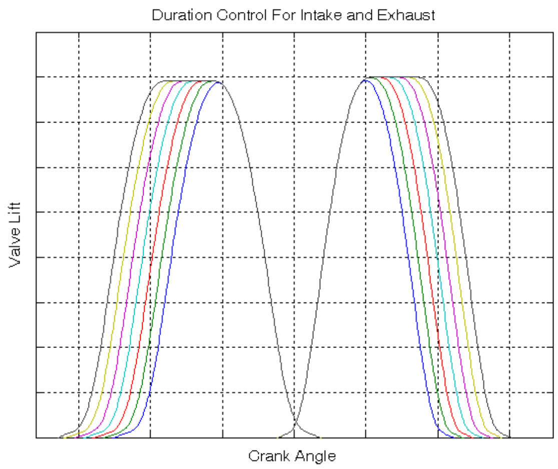

Figure 2: Fixed opening or closing characteristics

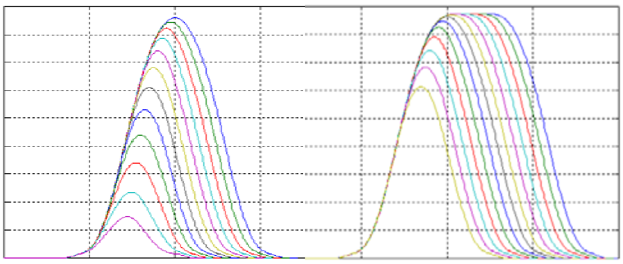

Figure 3: Different lift ranges possible with VLD

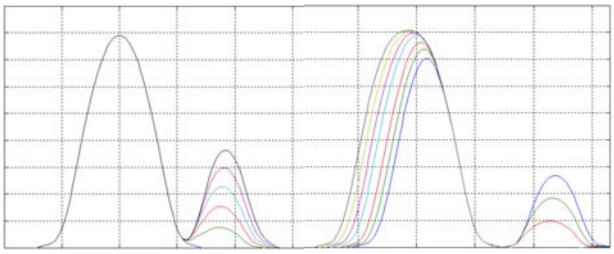

Figure 4: Variable secondary opening strategies

2.3 VVA operating effects in diesel engines

Control of the following parameters will help to enable advanced combustion strategies within diesel engines:

- Internal EGR (I-EGR).
- Intake valve closing (IVC).
- Exhaust valve opening (EVO).

Methods to control these using VLD and their effects on engine performance are discussed below:

Internal EGR: Three strategies for generating I-EGR, in diesel engines, are possible with VLD: See figure 5:

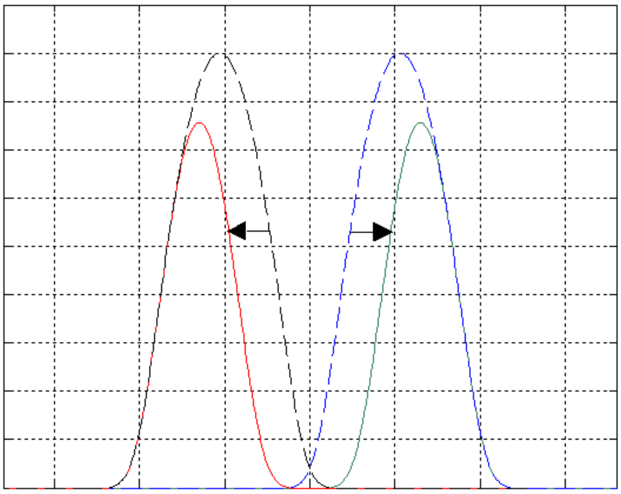

Figure 5a: Negative valve overlap.

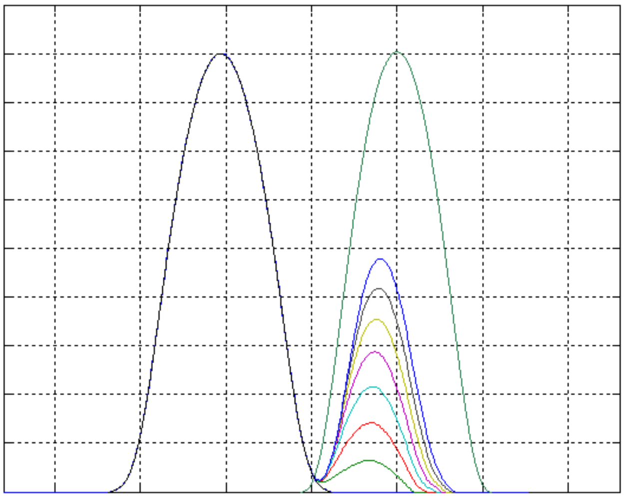

Figure 5b: Re-opening the exhaust valve during the intake stroke.

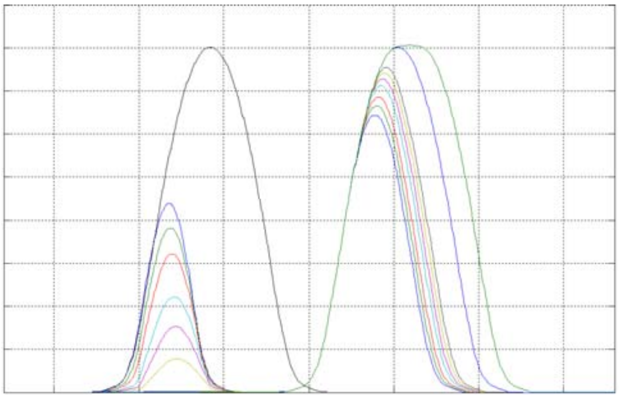

Figure 5c: Re-opening the intake valve during the exhaust stroke (with intake duration control).

I-EGR has both advantages and disadvantages and both are associated with temperature. I-EGR is hotter than external EGR, which is usually passed through a cooler. This higher temperature is helpful in avoiding the need for an EGR cooler bypass for light load operation where cooler fouling can be a problem. It is also helpful in creating higher in-cylinder temperatures at light load when combustion can be improved by raising temperatures.

However, the use of hot I-EGR has two disadvantages: It is not as good at controlling NOx formation because the peak cylinder temperature is higher than it would be with cooled EGR, and I-EGR has a lower density than cooled EGR and thus displaces more fresh charge for a given boost pressure, than would happen with cooled external EGR. Thus to restore a given AFR more boost is needed, leading to higher pumping work.

EVO: Controls the energy balance between expansion work, blow down work and energy flow to the turbine. Retarded EVO offers increased expansion work and can help with low speed, fuel economy at part load. It has also been demonstrated to improve output at full load under low speed conditions. [10], [11]

Advancing EVO has the effect of supplying more energy to the turbine which can help with transient torque rise. Extreme strategies, involving very early EVO are being investigated as a means of controlling the exhaust gas temperature for control of DPF regeneration.

IVC: Aside from the widely know effect of optimising volumetric efficiency across the engine speed range, variation in IVC can also be used to control effective compression ratio (compression ratio calculated with the swept volume from IVC to TDC rather than BDC to TDC). This is usually lower than the geometric compression ratio, but can be equal to the geometric compression ratio if IVC is at BDC, which it can be with VVA. This helps with cold starting, or reduction in compression ratio for control of maximum cylinder pressure.

In addition, late IVC can lead to improved thermodynamic efficiency through providing a compression stroke that is shorter than the expansion stroke.

It can be seen from the foregoing that the VLD system, when utilised to implement variable main and secondary openings, can provide combinations of these functions on a single camshaft, which is a significant factor in the overall economics of the system.

The hardware design presented in this paper uses VLD for controlling exhaust valve re-opening and intake valve duration. The families of curves possible with the physical hardware are shown in figure 6 below. It must be noted, however, that other strategies can be investigated with essentially the same hardware just by changing the cam profiles on the VLD camshafts.

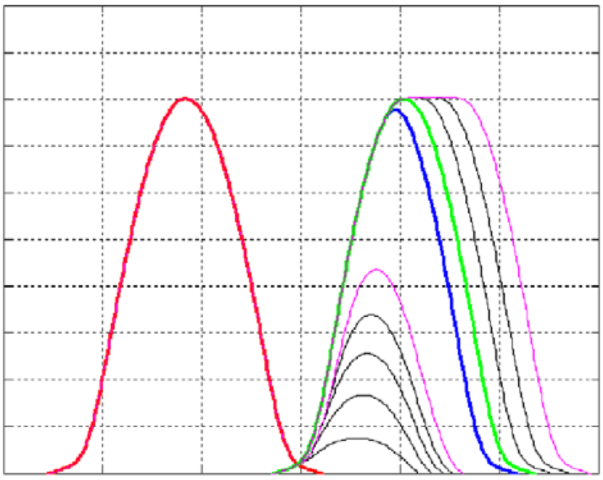

Figure 6 Shows the lift families tested on the presented hardware.

3. Design for modularity

VVA in particular requires careful design for modularity to enable product differentiation (through different specifications at build time) and design protection for future derivatives. The engineering and investment costs can then be amortised over a larger number of engines, leading to a more cost effective implementation. [12]

Design for modularity requires careful identification and consideration of the interfaces between the main systems that make up the cylinder head assembly.

We will now consider a twin cam engine that is to be built in a number of VVA configurations. We will have the option to install two different levels of VVA on both intake and exhaust, with the engine being built without any VVA initially.

The most significant requirement for such a design is that the camshaft axis (common to both VVA and non-VVA designs) is in a conventional position, such that a standard fixed camshaft (with or without cam phaser) or the VVA camshaft(s) can be installed. The position of this axis will be largely dictated by the geometry of the valve stem tips and HLAs.

Once these features are defined the remainder of the interfaces are relatively straightforward to define.

Thus it can be seen that the key to a modular VVA system is a design that has a conventional camshaft position. The VLD system meets this criterion.

VLD does however require that the cam bearings are above the cylinder head bolts. This requires a slightly different approach to the head structure. Both structural cover (with cam bearings integrated to the cover) and cam carrier designs have been implemented, but this paper focuses on the structural cover design.

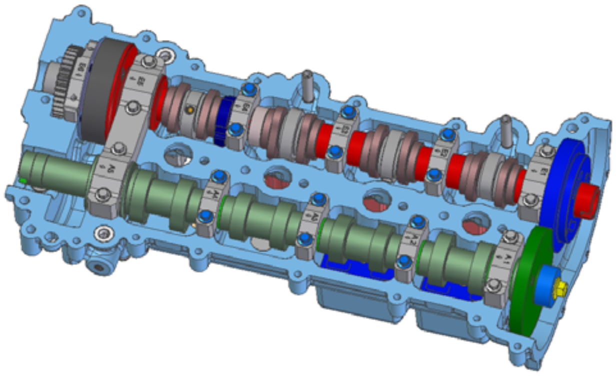

Figure 7: Underside of a structural cover assembly showing a standard cam and a VLD cam.

Figure 7 shows the underside of a structural cover design, with one VLD camshaft and a non-VVA camshaft installed. The design is such that it can be built with VLD on both camshafts, or with an intermediate VVA system that allows one of a pair of valves to be phased relative to the other on each cylinder.

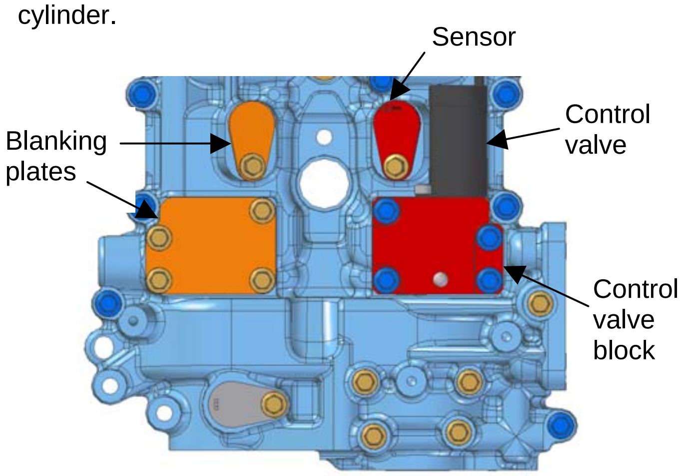

Figure 8: Top of structural cover showing hydraulic control valve and sensors

Figure 8 shows the top of the cam cover for this design. The orange parts are blanking plates needed for a standard valvetrain and the red parts are the sensor and control valve parts necessary for a VLD valvetrain. These parts are fully interchangeable depending on the VVA specification required at build.

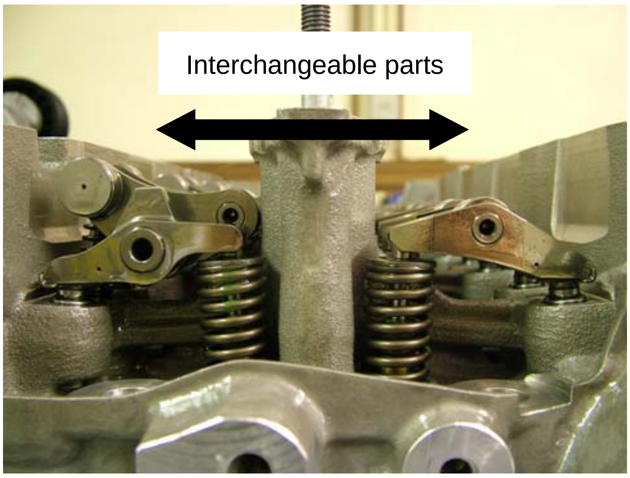

Figure 9: Cylinder head with standard finger follower on one side and VLD on the other.

Figure 9 shows an end view from the front of the cylinder head. VLD is installed on the left hand side and a conventional finger follower is installed on the right. The VLD parts however will fit and run on the opposite side with no modification and the same is true for the standard finger followers.

Thus it can be seen that this engine can be built in different configurations as required by product differentiation (market requirements) and legislation.

Whilst it is clear that each derivative might require a slightly different calibration, this modularity allows the majority of the engineering and investment costs to be amortised over a larger number of engines than would otherwise be the case.

The consequence of using a structural cover is that a very open head structure results. This has the advantage of simplifying the casting and allowing easy access on assembly, as shown in figure 10. Despite this open structure, adequate head stiffness can be maintained relatively easily.

Figure 10: Simple open head structure associated with structural cover design.

4. Valve train dynamics

The following section provides an overview of the VLD valvetrain dynamic behaviour by presenting valve head motion results measured with a laser velocimeter.

The modular cylinder head design was used for this exercise to validate both the standard and VLD valvetrains prior to fired engine testing. The modular structure of this cylinder head enabled us to do a direct comparison between a VLD rocker system and that of a conventional finger follower valvetrain.

The intake and exhaust valve head velocities were measured over the complete operating range of the test engine. The raw data was then post-processed using Matlab® routines to obtain the graphs on the following pages.

Dynamic results are presented for both the intake duration control system and an exhaust second lift system. Both show good stable behaviour over the complete operating range of the test engine.

Both VLD systems use identical valvetrain parts, with the change in lift behaviour resulting only from a change to the profiles on the camshaft. The standard roller finger follower assembly uses the same lash adjuster and cam positions as on the VLD assembly.

Note also that no change was required to the standard valve spring when using the VLD system. The valve lift curve for the standard system is the same as the basis curve for the VLD system.

4.1 Test Installation

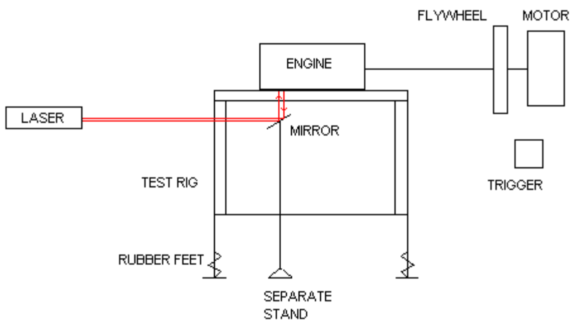

Figure 11. Test installation schematic

Figure 11 shows how the engine was mounted directly onto a test rig stand with a direct drive to the exhaust camshaft from the motor. A large flywheel was used to ensure consistent camshaft speed. The laser beam was directed at the valves on the underside of the engine using a mirror, which in turn was reflected back by reflective tape applied to the valves.

Teeth on the outer diameter of the flywheel were used as a trigger to initiate data capture. The laser was then used to directly measure the velocity of the valves over a 2.5 second duration at 100KHz. A Polytec laser vibrometer unit was used to directly measure valve velocity using the Doppler effect.

Since only a single laser system was available for this testing it must be noted that some of the vibration in the results is due to vibration of the test rig. The level of background vibration can be estimated by looking at the results where the valve is not on lift. This varies with speed as the valvetrain forces increase and as the rig goes through its own vertical resonance modes.

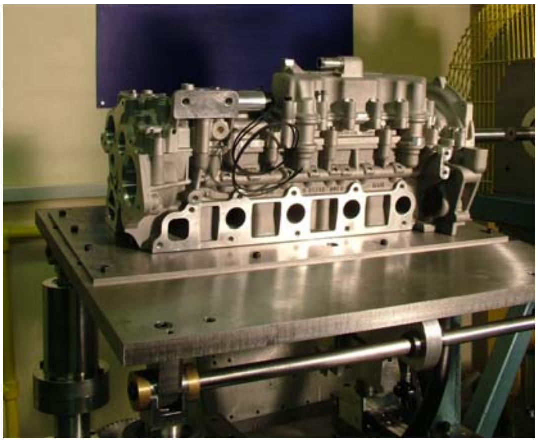

Figure 12. Shows the cylinder head on the rig at Mechadyne

4.2 Results

The results from the laser testing are presented below as a series of graphs, showing the dynamic behaviour of the valvetrain in 500 rpm increments from 500-5000 rpm. Results for the second lift assembly are restricted to 3000 rpm as the second lift closing ramp velocity is higher than that of the standard profile. However, this speed is above the maximum speed at which I-EGR is expected to be needed and it does not therefore compromise the function of the system.

To reduce the numbers of graphs, only minimum and maximum duration is shown, along with minimum and maximum second lift. The curves on each plot have also been offset to enable comparisons to be made between different speeds. In all cases the result for the lowest engine speed is at the top and the highest engine speed is at the bottom.

4.3 Observations

Observations on the lift data (figures 13-15) were as follows:

- Good stable operation of the VLD system over the whole speed range.
- Some loss of lift is noticeable on both the VLD and standard valvetrains. This is likely to be due to compression of the hydraulic lash adjuster.
- Some dynamic compression is evident on valve close for both VLD systems.

Observations from the velocity measurements (figures 16-18) were as follows:

- Good behaviour of the VLD system over the complete rpm range. This is very similar to that of the standard valve train.
- The second lift closing velocity condition is more severe at low rpm. There is more time for the HLA to leak down during lift, especially when we have a second lift and this will create more clearance in the system on valve close. This will generally result in a higher closing velocity.
- Higher frequency oscillations on the standard valvetrain are consistent with the fact that the standard rocker is slightly stiffer than the VLD rocker.

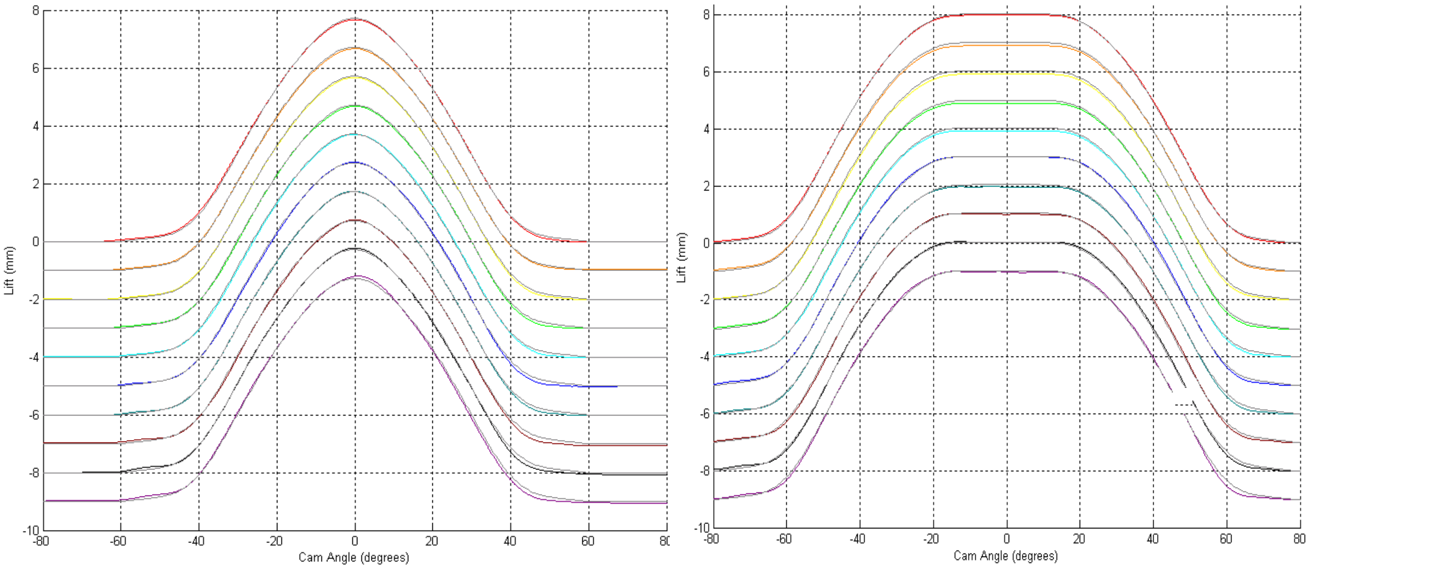

Figure 13 Lift results for Intake VLD (Minimum and Maximum Duration).Target lift curve is overlaid in grey. Max rpm at the bottom (5000), Min rpm at the top (500).

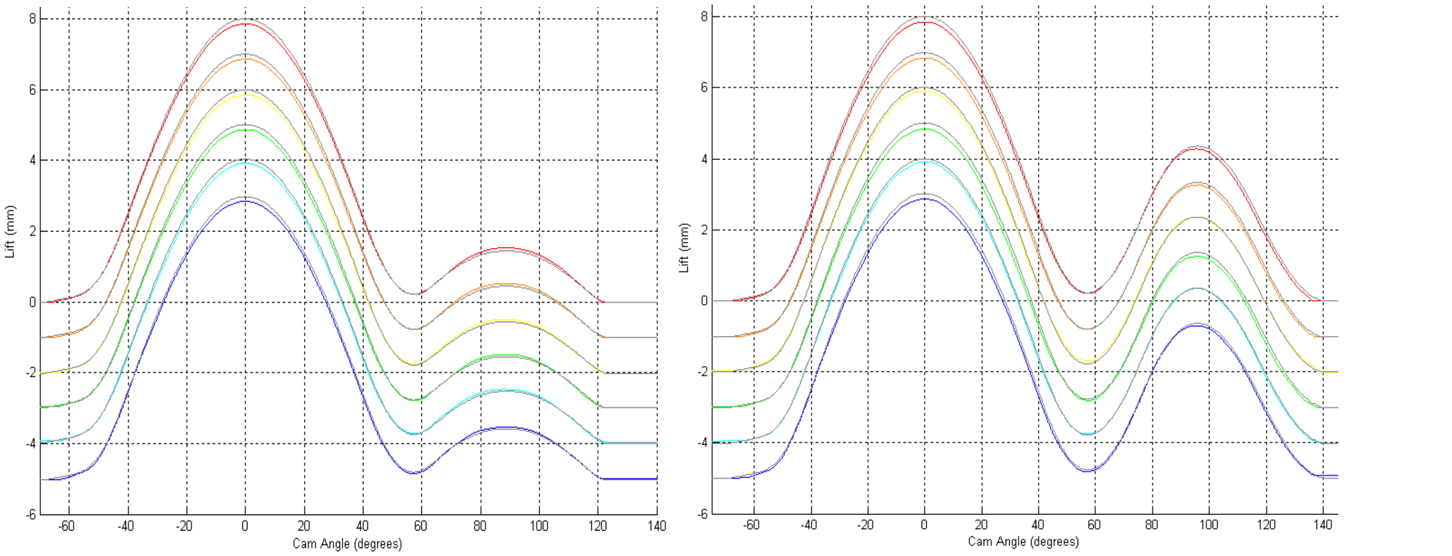

Figure 14 Lift results for exhaust VLD (Two second lift settings shown). Max rpm at the bottom (3000).

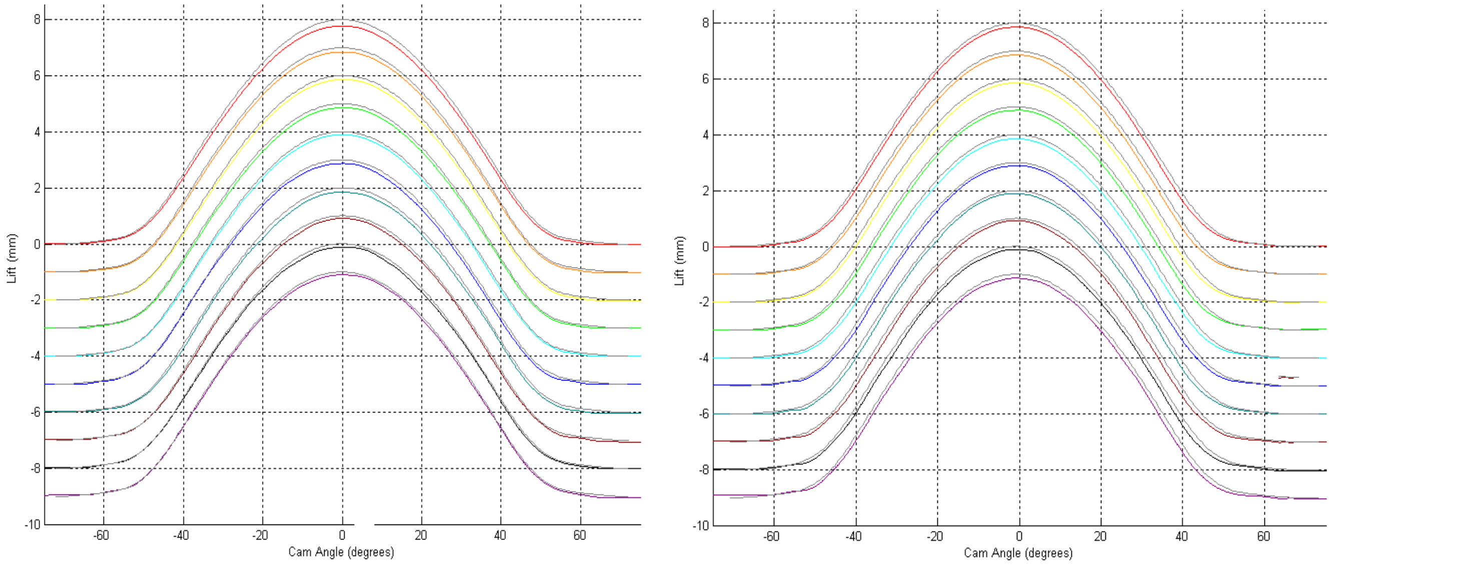

Figure 15 Lift results for standard exhaust (left) and standard intake (right). Max rpm at the bottom (5000)

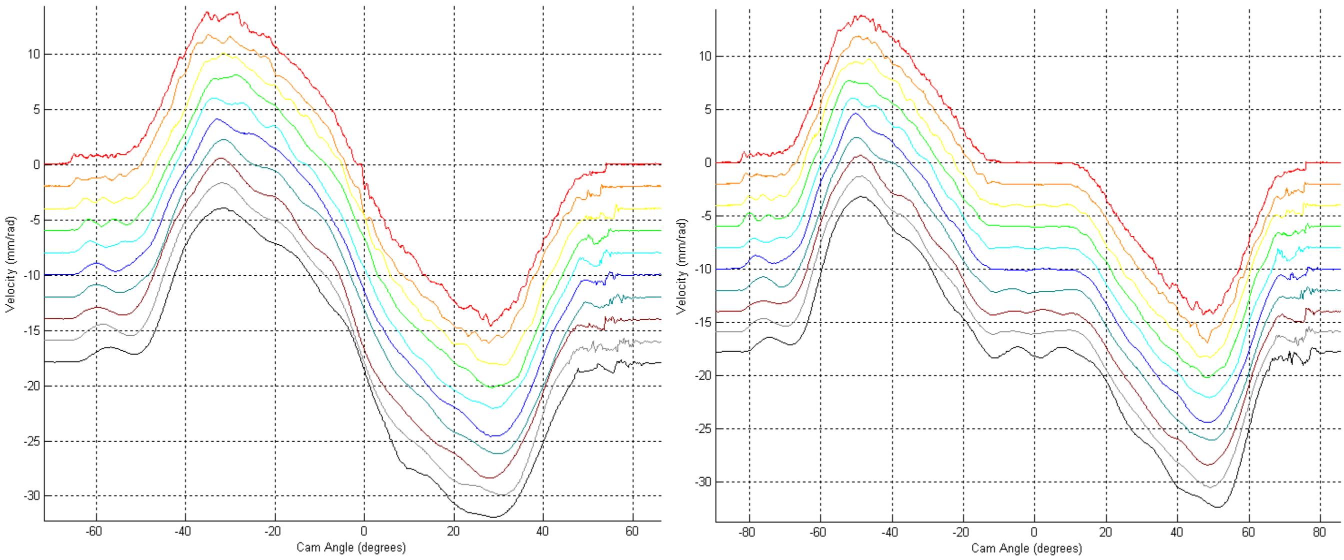

Figure 16 Intake VLD velocity (same two duration settings shown)Max rpm is at the bottom (5000 rpm), Min rpm is at the top (500 rpm).

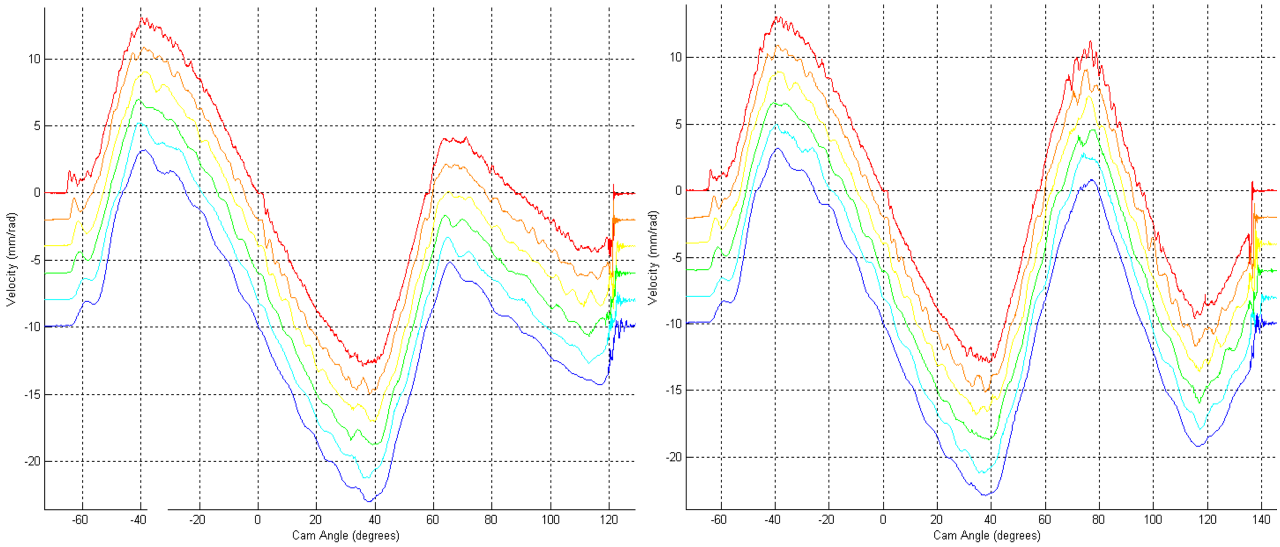

Figure 17 Exhaust VLD velocity (same two lift settings shown). Max rpm at the bottom (3000).

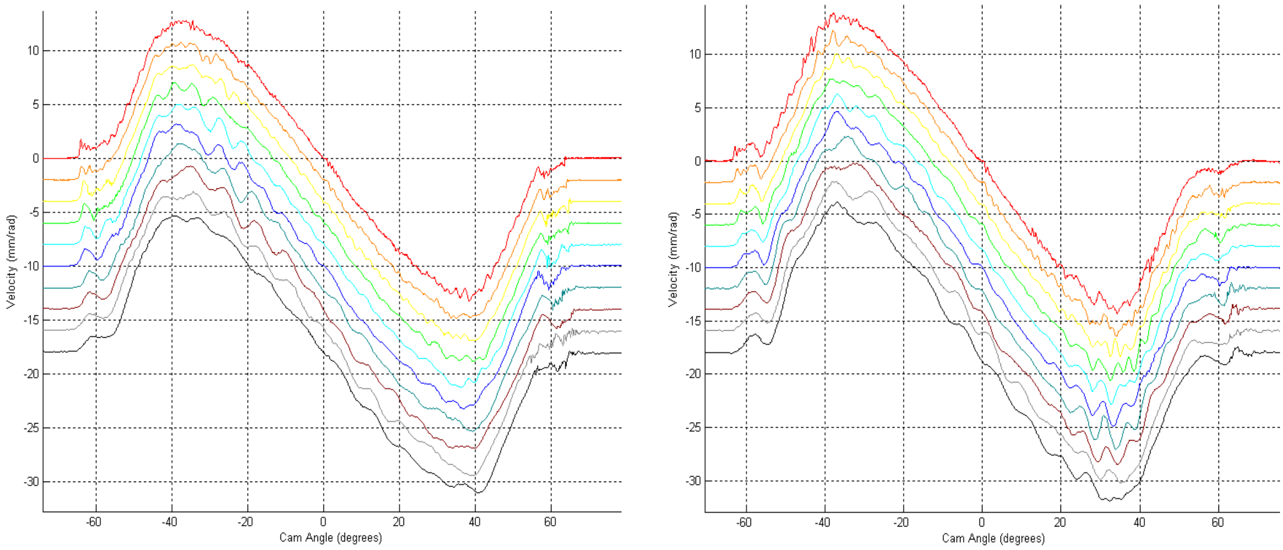

Figure 18 Standard exhaust (left) and standard intake (right) velocity measurements.

5. Actuator response

Mechadyne's own cylinder head rig was used to measure VLD actuator performance over a complete range of speeds, temperatures and oil supply pressures.

Since the cam torques from the intake and exhaust VLD systems are different (one system is for duration control and the other is for second lift), it was thought that this might affect the performance of the actuators. Information was therefore collected for both intake and exhaust for completeness.

It is difficult to show all results for all test points within the confines of this paper, so the results for a single oil temperature (80ºC) are shown to highlight any response trends.

5.1 Results

The following graphs summarise open-loop actuator slew rate at constant oil temperature over a range of supply pressures and engine speeds.

As the engine speed was increased, only the higher pressures were tested to ensure cylinder head durability. Only limited data is therefore available at the higher speeds

Slew rate was calculated using measured phase angle data output from the actuator controller. This is then plotted and two points near the end of the range of travel are chosen (see figure 19). An imaginary straight line is then drawn between these points, the gradient of which is the calculated slew rate.

Observations from the exhaust actuator results:

- Good advance and retard slew rates were achieved over the full operating range.
- Slew rate for the VLD system is far more dependent on oil pressure than on engine speed.
- Advance slew rates are slightly higher than retard slew rates as the torques on the cam phaser from the VLD system are uni-directional.
- The actuator cam torques for a second lift design are much smaller than those for a single lift design. The difference, therefore, between the advance and retard rates is relatively small.

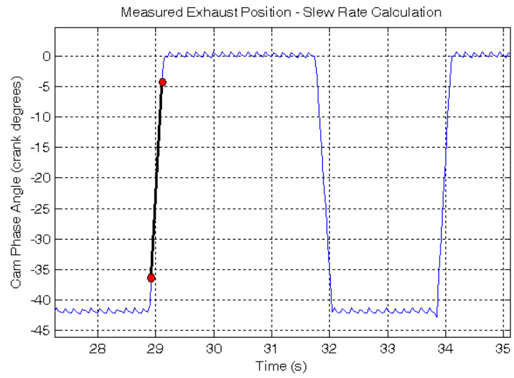

Figure 19 Shows slew rate calculation

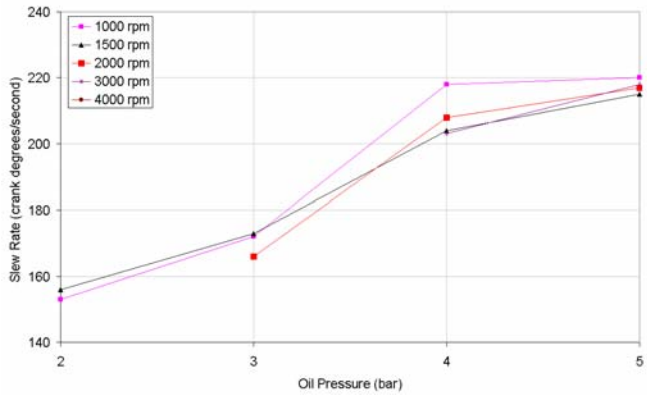

Figure 20a Exhaust Advance slew rates

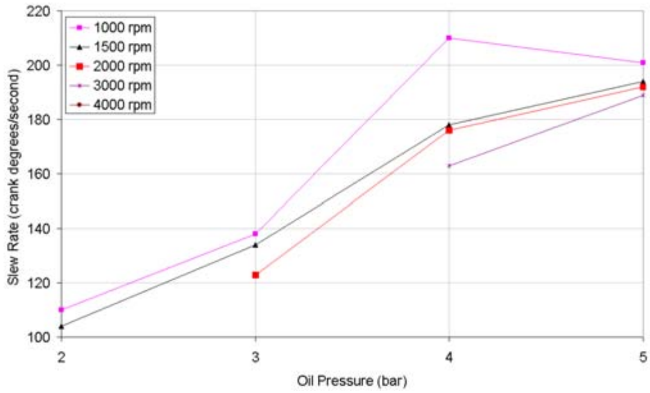

Figure 20b. Exhaust Retard slew rates

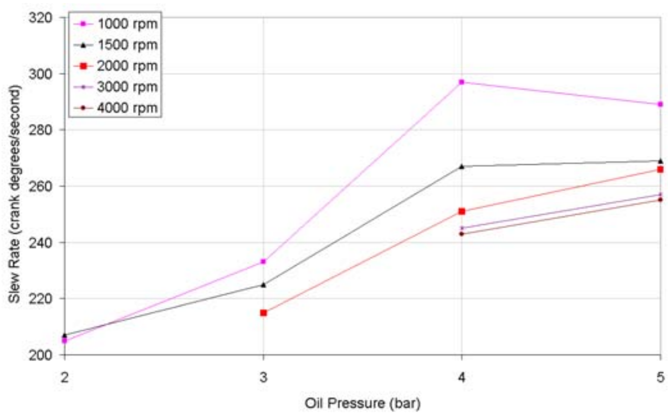

Figure 21a Intake advance slew rates

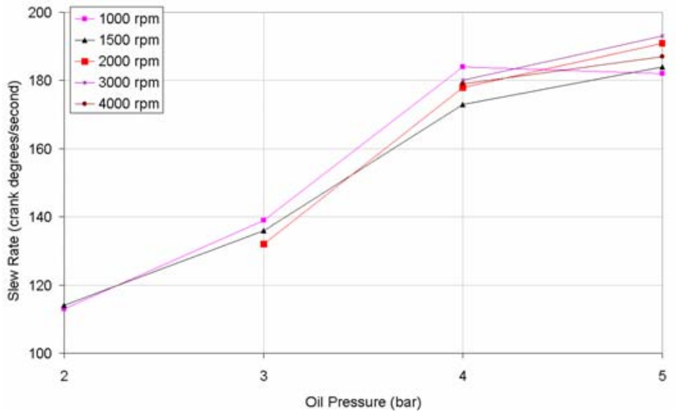

Figure 21b. Intake retard slew rates

Observations from the intake actuator results:

- Good advance and retard slew rates over the full operating range.
- Intake advance slew rate is significantly higher than intake retard. A return spring could be used to balance these two rates out if necessary.
- Again, the difference in slew rates between advance and retard is due to the uni-directional cam torques seen by the actuator. This time, however, these are greater as the system is being used for duration control on the main event.

6. Summary

This paper presents a practical design solution for a modular cylinder head where the VVA specification can be changed via simple part substitution.

This approach can be taken to create engine families with differing levels of VVA functionality, thus future-proofing new production lines.

It will also allow the further development of advanced combustion strategies like HCCI by measuring the effects of controlling exhaust valve re-opening and intake valve closing.

7. Acknowledgement

Our thanks are owed to Renault for allowing publication of the measurement data and to Ian Methley for his helpful comment

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9. Glossary

DOHC	Double overhead camshafts
DPF	Diesel particulate filter
EGR:	Exhaust gas re-circulation
EVO	Exhaust valve opening
I-EGR:	Internal exhaust gas re-circulation
IVC	Intake valve closing
SOHC	Single overhead camshaft
TDC:	Piston "top-dead-centre"
VLD:	Variable Lift and Duration
VVA:	Variable Valve Actuation
AFR:	Air Fuel Ratio
HLA:	Hydraulic Lash Adjuster