iVT, Intelligent Valve Technology: Development of a 16 Valve Cylinder Head

Dr Roger **Stone**, Technical Director
Mr Calum **Douglas**, Senior Design Engineer
Camcon Auto Ltd, Cambridge, UK

Summary

Camcon's iVT is an electro-magnetic system using rotary actuators to control engine poppet valves. The system provides near limitless, independent variability of lift, timing, period and even lift curve shape. The advantages arising from such a valve train are well known. However, further potential arises if port and chamber designs are modified to maximise benefits from this unprecedented valve control capability.

An inlet only system has previously been described [1]. However, the design of a 4 valve/cylinder iVT cylinder head permits a wider ranging study of induction, combustion and exhaust. The purpose of this paper is not to provide solutions but firstly, to suggest possible opportunities, to explore the mechanisms by which they may function. The potential to improve the CO2 performance of ICE and hybrid vehicles is significant.

1 The iVT (Intelligent Valve Technology) System

The iVT system uses a compact, torque dense electro-magnetic actuator, coupled with a high-speed electronic controller to provide inlet and exhaust valve events that are synchronised with, but mechanically independent of, the crankshaft.

These events are variable to an extent never before seen outside specialised systems dedicated to single cylinder research engines. The controllable variables are:-

1. Valve Timing or Phasing - the crank angle at which the valve starts to lift
2. Valve Period - the number of crank degrees that the valve remains open
3. Valve Lift
4. "MOP Shift" - a conventional lift curve shape but featuring a skewed position for the maximum opening point which can be "early" or "late"
5. Event Shaping - similar in principle to MOP shift but permitting a variety of less conventional lift curves
6. Event Frequency - events can be omitted or added within a 720˚ cycle. This can apply to individual valves or to pairs of valves. This can allow extra exhaust valve events for EGR events in the inlet stroke or permit easy cylinder de-activation.

These are shown in Figure 1, all are infinitely variable, between limits where appropriate, and, with the exception of double events within a 720 degree cycle, are independent of succeeding and preceding events. Events 4 and 5 as shown in Figure 1 are more limited in the case of exhaust valves at high load because of the higher δP across the valve at the start of the valve event.

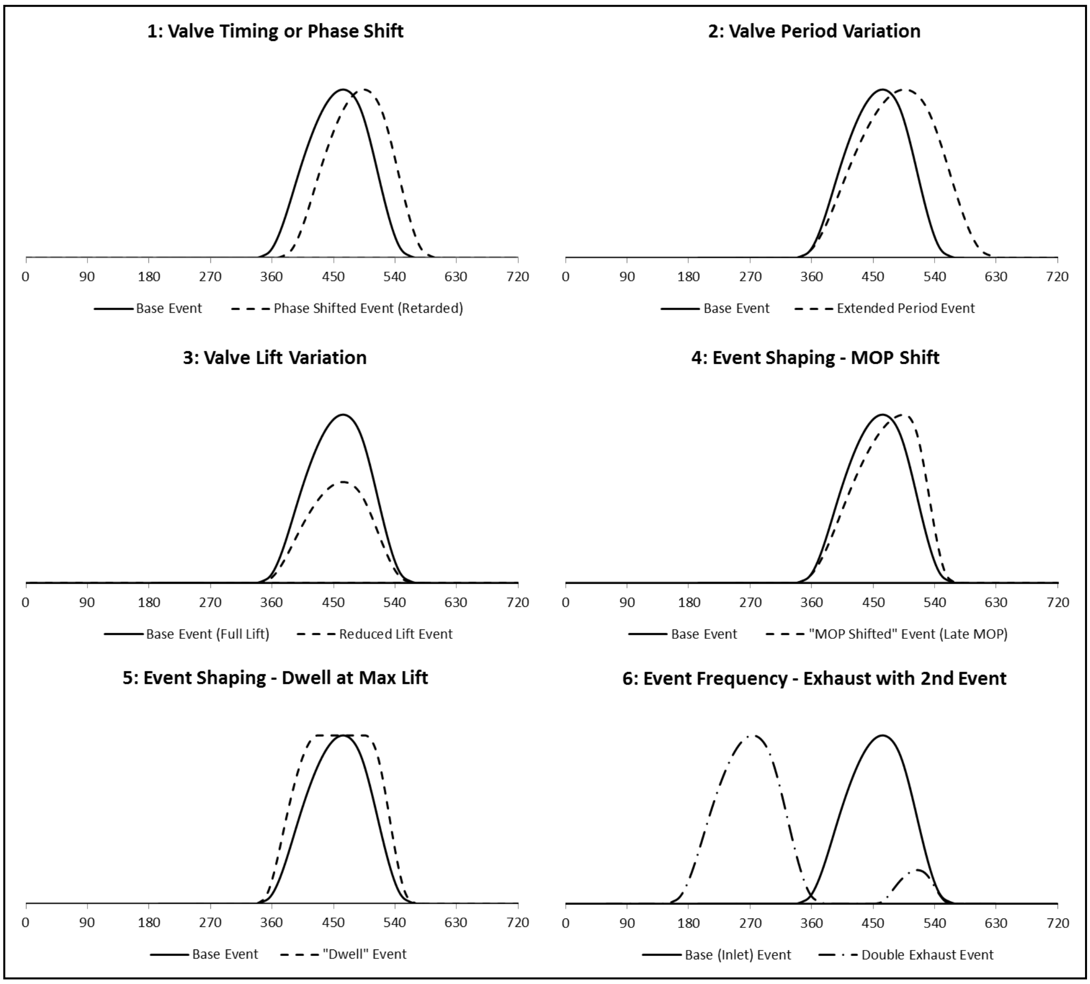

Fig. 1: Diagram Showing Examples of iVT Event Variability

2 The 4 Valve/Cylinder Heads

The base engine for the 16 valve 4 engine is, as reported in previous work, the current Jaguar Land Rover "Ingenium" 4 cylinder spark ignition engine. The cylinder head design has been modified to remove the existing valve gear and the head re-cast in a form that allows the Camcon iVT equipment to be fitted.

A section through the 4 cylinder head assembly is shown in Figure 2. Note that the modifications are confined to the upper part of the head to provide for the iVT modules and electronic Valve Control Units (VCU). Changes to the ports and combustion chamber have been confined to eliminating the integrated water cooled exhaust manifold in order to allow individual exhaust ports to be provided for each valve. A number of exhaust manifolds have also been designed (both with and without water cooling), in order to allow a number of options to be evaluated.

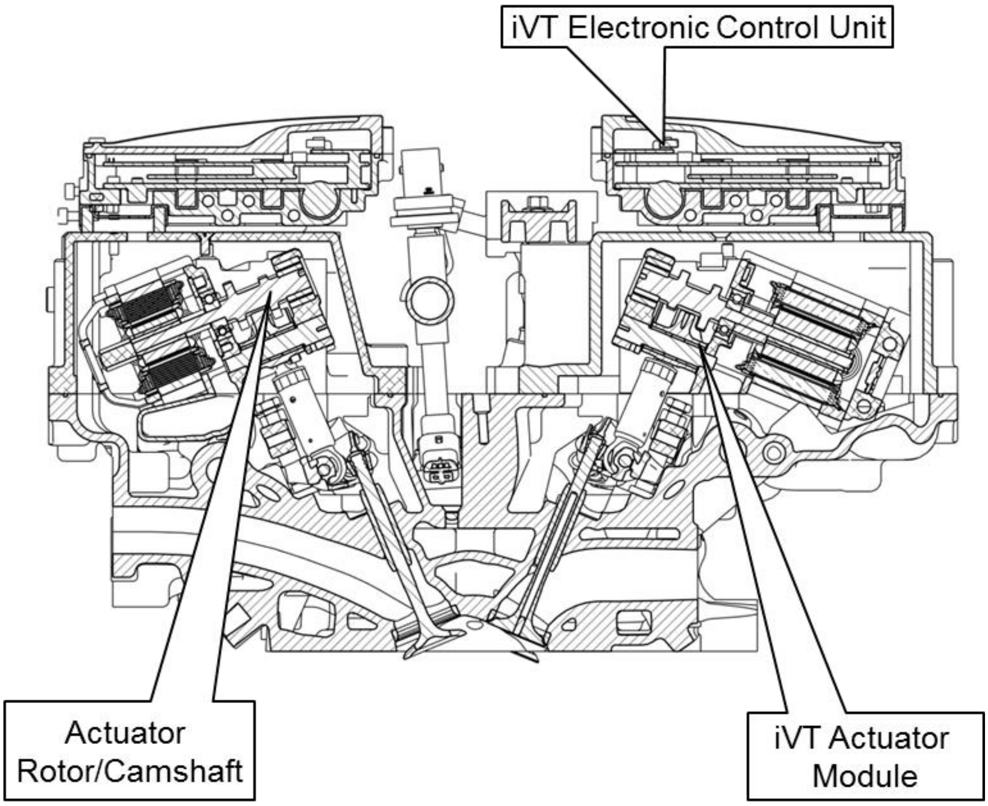

Fig. 2: Section through the 16 valve cylinder Head

The combustion system has been left unchanged at this stage to allow a direct comparison between the results obtainable from the original, standard, specification engine, the inlet only iVT arrangement described previously [1] and the new, 16 valve version. However, Camcon suggest that there are very considerable opportunities for improving the combustion performance over the speed:load map by developing modifications to the inlet ports, the combustion chamber and, indeed the exhaust valve operation and downstream gas routing/EGR arrangements. Some of these opportunities may be employed simultaneously, others would be mutually exclusive.

As an example of the possible opportunities, it was demonstrated on the iVT inlet only Ingenium engine that, at lower engine speeds and loads, operation on a single inlet valve could result in significant reductions in burn rate - changes in the 10-90% combustion time of as much as 19˚ crank were observed. It is upon the basis of this observation that a preliminary study of ideas has been conducted and is reported here. To date no empirical work has been conducted and indeed, to fully explore all these ideas on a fired engine would be an intimidating task! Therefore, our first review is on the basis of a desktop and spreadsheet study as follows.

In order to aid the study of the ideas that are expounded here, a single cylinder arrangement has also been developed. This uses larger "Generic" actuators which can be applied easily to existing single cylinder research engines rather than the more constrained production like geometry of the multi-cylinder case. A view of the cylinder head with its iVT system is shown in Figure 3.

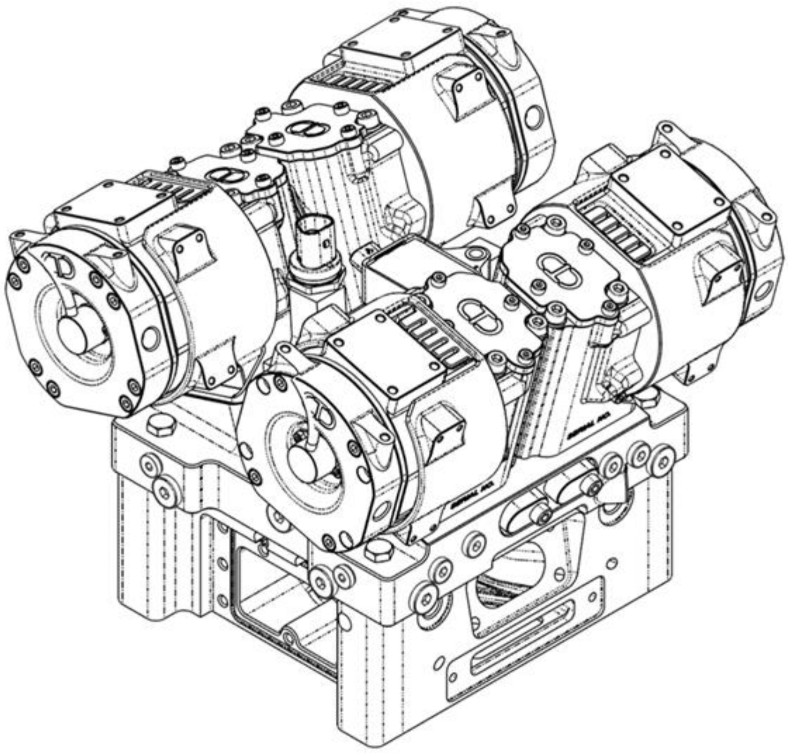

Fig. 3: A View of the Single Cylinder Conversion kit as applied to a Ricardo Hydra at Brunel University

It is accepted that many of the following arguments are speculative. However, in the past, the absence of any valve train system possessing both the necessary capability and affordability in production has rendered the questions asked irrelevant. Therefore, it is the purpose of this paper to ask questions rather than to answer them!

3 Combustion Optimisation Opportunities

3.1 LIVC and EIVC for Effective Compression Ratio Control

Much work has been completed over the years in this area and the application of these approaches is well understood. The main opportunity afforded by iVT with respect to employing them is the ability to permit optimisation of the specific event and to switch between these modes at will. 1-D modelling is the obvious tool to apply in this case although the opportunity is arguably one of calibration rather than combustion.

3.2 Combustion Chamber Optimisation to Exploit Single Valve Operation

The combustion chamber of the donor engine has been optimised to function best on the basis of a conventional high tumble gas motion regime which is what the standard port geometry is designed to provide. However, we have seen that under low speed, light load conditions, that the combustion performance improves if one inlet valve is disabled and the entire flow is through a single port. It is considered that this is the result of higher gas velocity and therefore higher kinetic energy within the charge which contributes to turbulence and possible organised gas motion at the point of ignition.

If we examine the single valve/dual valve situation in more detail it is useful to consider the mean inlet gas velocity (MIGV). This may be a relatively crude tool but it provides a first level of understanding whilst eliminating details of limited relevancy at this stage, such as the cam profile. Conventionally, MIGV is, of course, linearly dependent upon engine speed. Now, inevitably, when selecting valve and port sizing, the high speed, high load condition dominates because it is essential that volumetric efficiency is adequate under full load conditions throughout the speed range. This leads to lower than optimal gas velocities at lower loads. Figure 4 shows a plot of MIGV under a range of conditions.

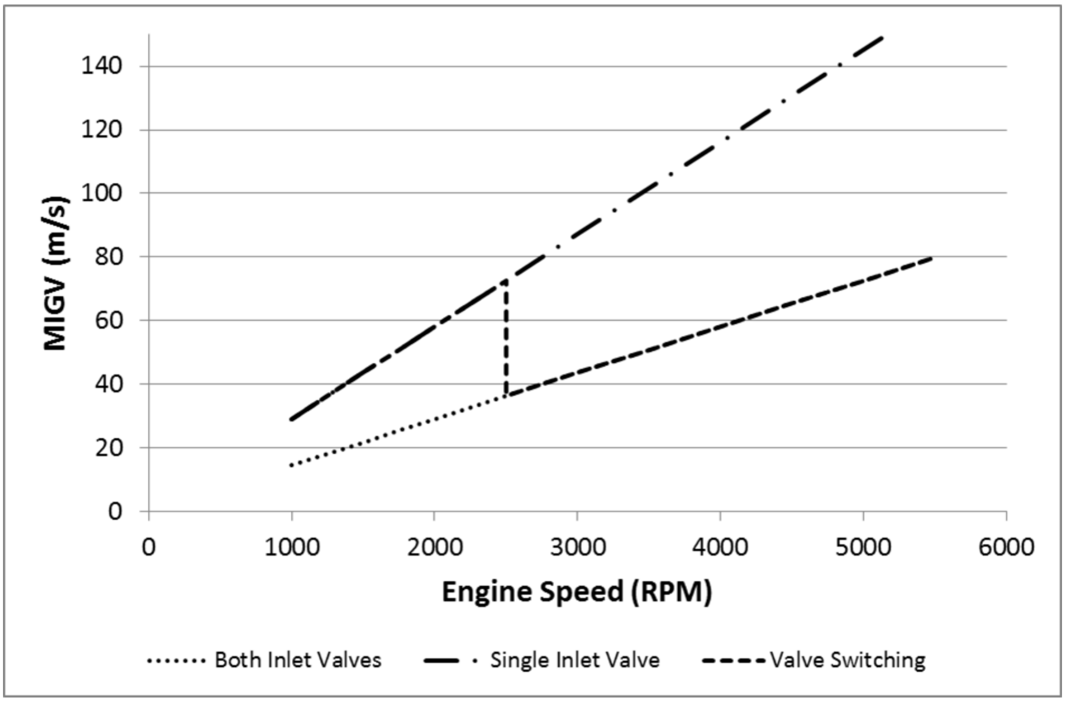

Fig. 4: Plots of Mean Inlet Gas Velocity under Single and Dual Valve conditions

The bottom curve shows a typical arrangement where both inlet valves operate in lockstep and the typical design maximum MIGV is seen close to peak power speed. If we run the engine using a single valve the MIGV is doubled and we have the opportunity to switch into or out of this mode at whatever speed gives the best results - and which may vary with load. Importantly, however, we can generate useful high gas velocities at modest engine speed and thus generate unusually high in-cylinder gas motion at low engine speeds. This does introduce a discontinuity - not unlike a cam profile switching mechanism in many existing variable valve arrangements - and this can produce calibration challenges, which will be expanded upon later in this paper.

However, this high intensity gas motion is produced by a high tumble port design operating at a significant offset from the cylinder centre line and is therefore a mixed swirl and tumble motion for which the chamber was not designed. This swirl element can potentially be sustained later in the cycle and thus be especially valuable for low speed conditions as are the higher shear rates in the jet streams entering the cylinder. Relatively small changes to take advantage of this swirl, which is likely to persist in the piston bowl, may yield significant benefits. A relocation of the sparking plug by a matter of a mm or two towards the exhaust valve side of the chamber, even if it requires some compromise on exhaust valve size, may improve single valve combustion disproportionately relative to any adverse effect on full load, dual inlet valve conditions. Similarly, minor changes to the height to diameter ratio and positioning of the piston crown dish may yield an improved compromise across the load:speed map, especially combined with re-calibration of the direct fuel injection settings.

Improved detonation characteristics were, as would be expected given the faster combustion, also observed on the test engine under single valve operation.

Changes of this type generally require a fired engine tests to allow proper quantification and optimisation. However, they do lend themselves particularly well to single cylinder testing. Ideally, such testing would use heads fully machined from billet including the ports and combustion chamber, this eliminates any errors arising from casting tolerances and enhances the integrity of the results because it is so much easier to be confident that improvements are entirely due to the experimental parameter and not to any random variation.

3.3 Individual Inlet Port Shape Asymmetry

The experimental work to date has been conducted using inlet port and chamber geometry which is mirrored across the combustion chamber so in single valve operation either port could be selected without any effect on the results. However, if we can have asymmetrical valve events, why not asymmetrical ports? New opportunities may be realised if the geometry controlling swirl and tumble were re-optimised in each "leg" of the inlet port to suit different areas within the load map (whilst also remembering the need for the "legs" also to work together at higher loads).

Such an approach may permit further improvement over the results attainable by simply using symmetrical ports. Clearly, there will be a cost to such an approach and high load test points are likely to suffer some detriment. It is probable that the degree of geometry adjustment may be limited in the interests of full load performance. Nevertheless, improvements of a few per cent in important areas of the map are valuable and it should be possible to minimise the adverse effect whist enjoying significant benefits in those areas of the map that are most important under real world driving conditions.

A great deal of useful preliminary work to explore these possibilities could be completed using flow bench work with swirl and tumble measurement plus CFD analysis. In this manner, different approaches could be compared and rated before a small number of promising candidate geometries were selected for fired engine testing.

3.4 Inlet Valve Size Asymmetry

If the ability to switch from single to two valve operation is valuable could further improvements be achieved by reducing the size of one inlet valve whilst increasing the other? For example, if one inlet valve diameter is reduced by ~10% whilst the other is increased to maintain a similar overall flow area, a double advantage may be gained. Firstly, at very low speed, gas velocities are further increased whilst remaining firmly within conventional limits, then, as gas speeds approach the limiting value a switch to the larger valve allows the speed range over which the benefit applies to be extended before switching to two valve operation at the highest speeds. Figure 5 shows a diagram of such a chamber whilst in Figure 6 mean inlet gas velocity is plotted against engine speed for just such an arrangement.

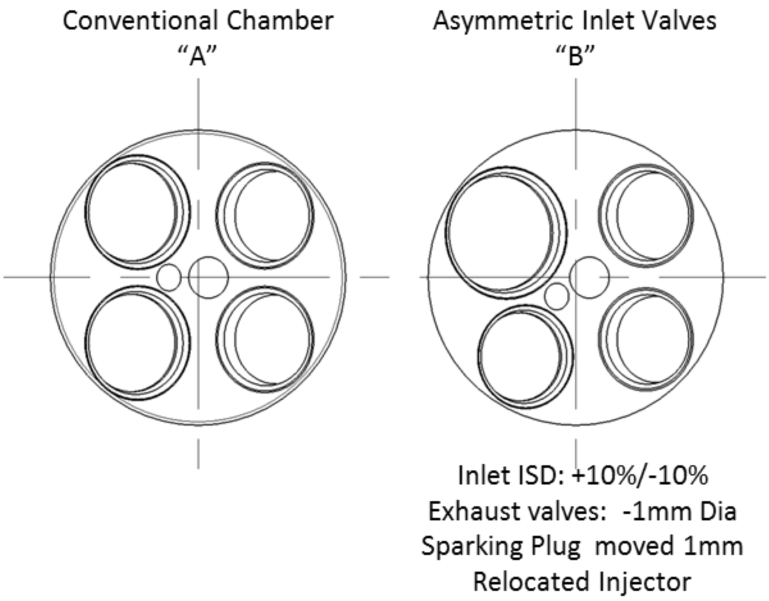

Fig. 5: Comparison of a Conventionally Laid out Combustion Chamber with a Chamber using Asymmetric Inlet Valve Sizes

In the case of a throttled engine discontinuities of this kind are, at least to some extent, damped out and this reduces the challenge from a calibration/driveability viewpoint. However, in the case of an engine using the valve events themselves to control trapped mass, such a discontinuity would be much more significant. Furthermore, a tool considerably more subtle than MIGV is required to understand the extent of the discontinuity and derive a means to address the issue. In order to achieve this, a spreadsheet was developed in which the gas exchange process is modelled but combustion is represented very crudely simply by using empirical data for cylinder temperature and pressure at a fixed point after TDC. Conditions at the valve seat are modelled to reflect mass flow under both choked and sub-sonic conditions. The instantaneous mass flow is determined at small intervals and the corresponding kinetic energy of each gas "packet" is also established. Care is taken to ensure that the direction of the flow is properly modelled and taken into account. In this way both the trapped mass and the charge energy arising from the induction process at the point of Inlet Valve Closing (IVC) can be established. The former can be then be compared with test results where available for validation purposes. The kinetic energy of any reverse inlet flows are excluded from the calculation as this energy will not contribute to the organised gas motion at IVC. Clearly, both inertia and wave ram effects are neglected but this omission is regarded as helpful at this point because their inclusion would add an overlay to the results that would serve to obscure or even distort the underlying trends rather than to clarify them.

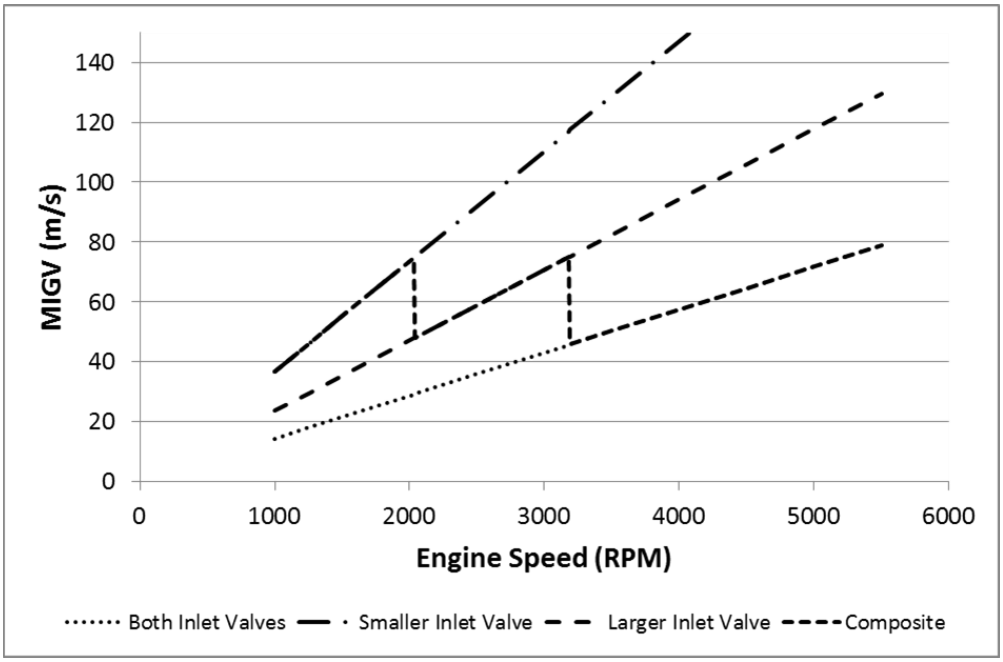

Fig. 6: Plots of Mean Inlet Gas Velocity using Differential Inlet Valve Sizing

By using this model it was then possible to demonstrate that valve events "equivalent" to the optimised empirical event could be established relatively easily. For any given single valve reference event the trapped mass is determined and the charge air kinetic energy calculated. The spreadsheet is used to explore twin valve events providing the same trapped mass with similar charge air kinetic energy values. These calculations were extended over a speed range of +/- 100 RPM each side of the test speed just to ensure that the effects were approximately linear.

The charge air kinetic energy results for a sample point are shown in Figure 7 - the load/speed condition is the 1800 RPM, 4.72 bar bmep point for which experimental data was available and where full lift on a single valve was being used. Table 1 shows details of the three corresponding valve events.

Note that two cases have been identified which use both inlet valves in place of one and meet the criteria of constant trapped mass and similar charge kinetic energy at IVC. Note also that, in all cases valve timings have been maintained unchanged as this should result in broadly similar levels of residual content for all the events. These two valve events are considered, at a preliminary level, as "Equivalent Events" to the datum event. The first of these, "A" in Table 1, is characterised by employing equal lifts for each valve, whereas event "B" uses differential lifts.

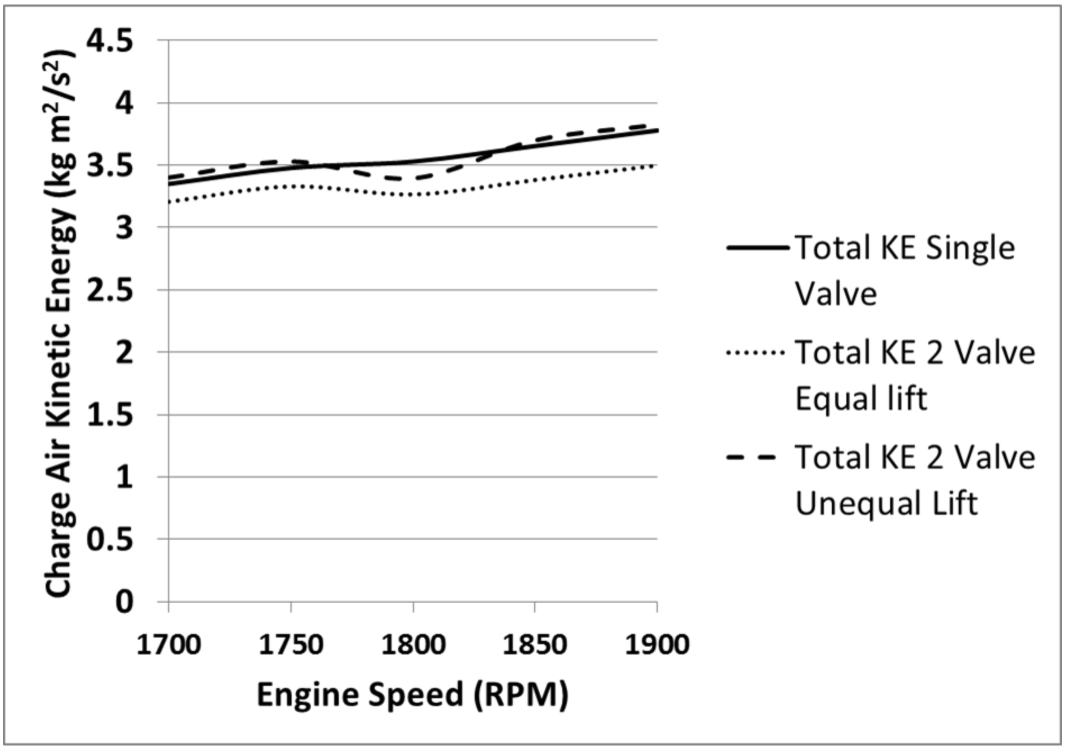

Fig. 7: Charge Air Kinetic Energy at Constant Trapped Mass for Differing Valve Event Configurations

Event	Valve Number	Inlet Open ˚BTDC	Inlet Close ˚ABDC	Valve Lift 1	Valve Lift 2
Datum	1	60	-30	7.8	0
A	2	60	-30	3.3	3.3
B	2	60	-30	5.0	1.9

Tab. 1: Detail of Valve Timings and Lifts for "Equivalent Events"

Of course, this "Equivalence" is only partial because kinetic energy level does not imply any particular organised charge air motion regime and these are inevitably going to be different between the three events shown in the table. The Datum event, for example, will exhibit high swirl as a result of the offset position of the active valve but there will also be an element of tumble behaviour arising from the port shape. Event 2, on the other hand, should show pure tumble behaviour. In order to achieve similar burn rates it will probably be necessary to manipulate the kinetic energy to be different from the datum event. If the general degree of kinetic energy change is known then the spreadsheet method can be used to further refine the "Equivalent" events. Initial study suggests adjusting the valve timings can have a significant effect on the charge energy. It has thus been shown that, by using such a simple method, candidate "Equivalent" events can be identified which can then be further refined using a combination of CFD, 1-D simulation and, ultimately, engine testing.

3.5 Dual Independent Inlet Valve Operation

Inlet events of type "B" where both inlet valves are operational but do not deliver identical events may offer further potential - the timing and period setting for the two valves may differ as well as the lift - giving a huge variety of options for any given load point. The simplest way of looking at this may be to consider the possibility of "infinitely variable swumble" but this may miss the point.

It may be possible to introduce other effects into the induction process and these could be of more significance. For example, it may be advantageous to define a "lead" event that defines the start and end of the charge induction process but to embed within that event a secondary event using the other valve. This may serve to reduce pumping loss by providing more flow area during a period when the instantaneous pumping loss is high. Alternatively, if the secondary event were to close early, there may be a benefit in maintaining high velocity in the "lead" port so that the last part of the charge entering the cylinder before the induction event is completed has a disproportionately high kinetic energy. Figure 8 shows such an event - including the opportunity to skew the shape of the lead event so that the Maximum Opening Point (MOP) occurs late, further increasing the proportion of total charge energy introduced late in the induction process.

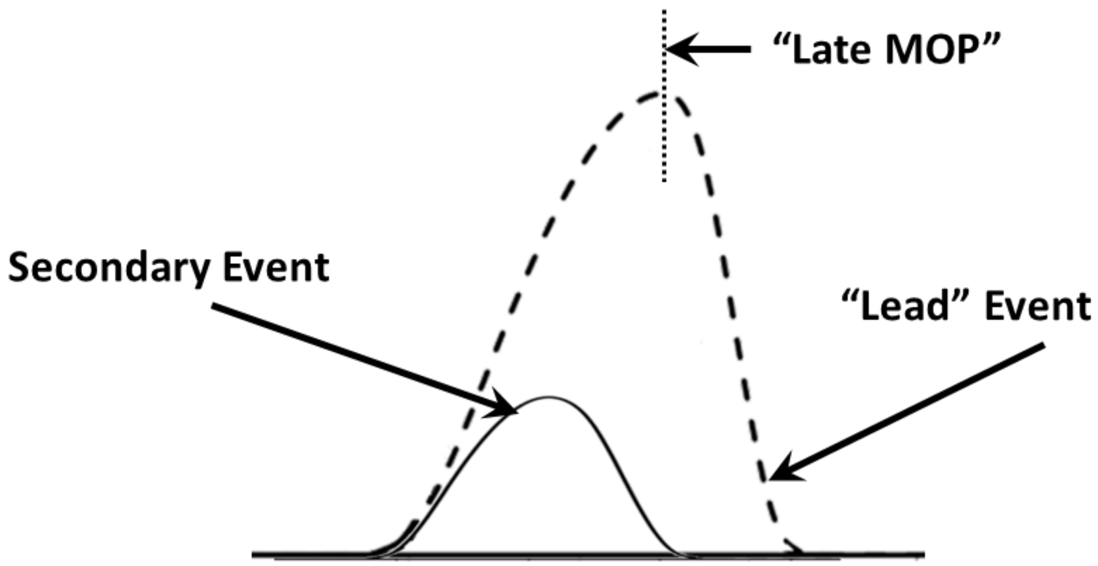

Fig. 8: Possible "Dual Independent" Composite Inlet Valve Event

3.6 Combustion at λ=2?

The appeal of lean burn combustion remains as a longer term goal for combustion development engineers and iVT may offer a tool to assist in achieving this objective. One of the main issues is combustion stability over the speed range and very highly turbulent chambers are required. One of the difficulties is that these high turbulence levels must be maintained at both high and low speed conditions as well as over the entire load range.

In order to achieve such behaviour within the charge air it may be necessary to rethink conventional charge motion regimes quite radically. The ideas suggested above in terms of differential inlet valve sizes and dual independent event selection may be useful steps towards what would be required. However, even combined with quite radical port shape adjustments they may not be sufficient to achieve the desired results. However, a combination of iVT or similar capabilities with the use of valve masks may provide a powerful tool to allow the achievement of the necessary in-cylinder gas motion characteristics required.

Such masks may apply to one or both inlet valves, may apply around part or the entire circumference and may be of uniform or varying height around that circumference. Radial clearance between the mask and the valve head may be set at a practical minimum or more generous and, within the constraints of manufacturing practicability, may even vary circumferentially.

Because valve lift is a controllable parameter, a fixed height mask becomes more or less effective as the selected valve lift increases. This could make lift a very powerful influence over swirl ratio. The obvious tool with which to start an investigation into these opportunities is CFD. The challenge is the sheer number of variables.

3.7 Extending λ=1 over the Full Load:Speed Range

Full load operation at λ1 is a task that must be addressed - is it possible that a combination of the high turbulence measures suggested in section 3.7 combined with LIVC/EIVC could yield useful progress? Assuming trapped mass is maintained by using higher boost so that output is maintained, effective after-cooling plus the reduction in effective compression ratio will reduce Tmax and therefore, gas temperatures throughout combustion and exhaust.

3.8 Exhaust Operation Asymmetry

Under lower speed conditions, full scavenging of the cylinder can be achieved using a single exhaust valve - even under full load running. This has been modelled using the same spreadsheet used for the inlet calculations mentioned above and the results for an engine speed of 2000 RPM are shown in Figure 9. In all three cases the valve timings employed were the same and typical of modern practice. The results showed that the difference between the masses of gas expelled in each of these events was less than 0.14%.

We can, of course see that the mass of exhaust gas expelled during the blow down period to BDC is reduced in the case of single valve operation but this is completely recovered by the time the valve closes. Just in case this reduced flow adversely affected cylinder pressure during the exhaust stroke, cylinder pressures were over-plotted and this is shown in Figure 9 - where we can see that cylinder pressure varies little over this period peaking at between 1 and 2 bar near the dead centres where pressure does not convert effectively into crank work.

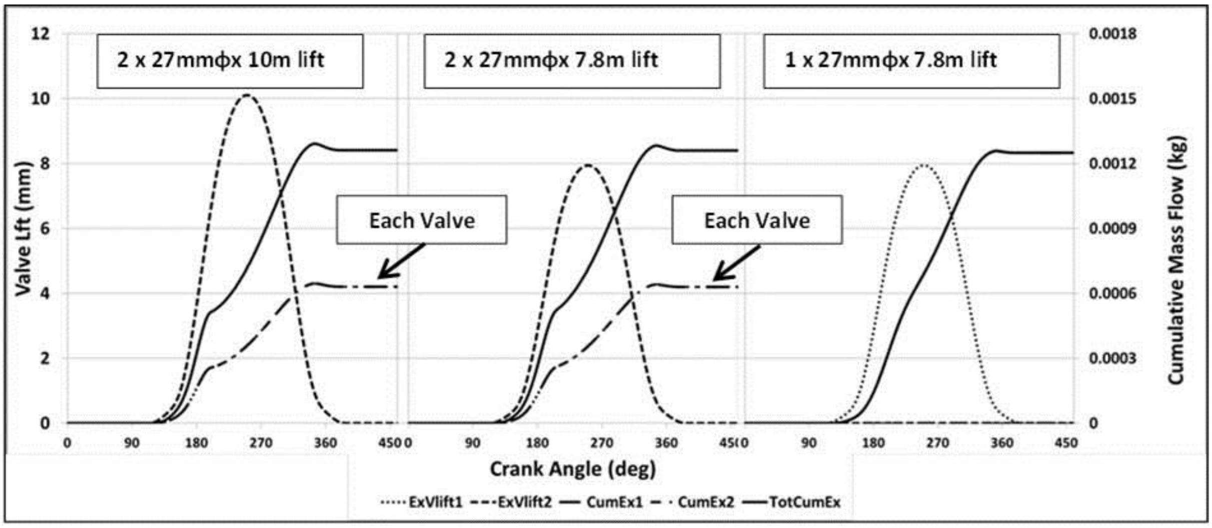

Fig. 9: Comparison of Scavenging Effectiveness for 3 Different Exhaust Valve Configurations - 2x27φx10mm lift, 2x27φx7.8mm lift and 1x27φx7.8mm lift

In the case of cylinder head arrangements where the gas streams from both exhaust valves in each cylinder are mixed, there is no significance which exhaust valve is used for any given engine cycle. This provides an opportunity that may be useful from a combustion point of view, because there is no disadvantage in alternating the exhaust valve on consecutive firing strokes. Quite apart from reduced parasitic losses, such alternation will reduce the running temperature of the exhaust valves in comparison with conventional operation. The mechanism for this temperature reduction is twofold - firstly, the time available for the valve to transfer heat from the valve head to the seat (the main cooling heat path) is increased from some 500 degrees crank to approaching 1200 degrees crank. Secondly, the time over which the valve is exposed to heat transfer from the high velocity, high temperature spent gas flowing past the valve head and stem is halved. Whilst this will not halve the heat input because instantaneous gas velocities and densities will be different, the reduction will be substantial. A simple approach to estimating the effect of these changes using time averaging of gas temperatures and heat transfer co-efficients derived from previous experience suggested a uniform reduction in temperature of the "combustion chamber surface" of a sodium cooled valve of 38˚C. This is probably a minimum figure and may seem modest but, as a comparison, it is useful to bear in mind that sodium cooling may reduce peak valve temperatures by only 80˚C.

Cooler valves are beneficial to classical knock performance in a number of ways: lower heat transfer to the charge and a reduction in the temperature of key hot spots are both useful in reducing knock sensitivity. However, modern, downsized, highly supercharged engines also suffer from a variety of "irregular ignition" or pre-ignition mechanisms as described by Dobes et al [4]. This problem is frequently referred to as "super-knock" or "mega-knock" and seems actually to have more than one cause or mechanism depending upon the engine conditions and speed. Significantly reduced exhaust valve temperature can improve knock performance, to a greater or lesser extent, in each case.

Low Speed Pre-Ignition, LSPI, occurs at engine speeds from the speed at which significant boost is developed to around 2200-2500 RPM and is exacerbated by high temperatures within the chamber - spark plug, exhaust valves, chamber deposits and high gas temperatures. Low engine speed provides the necessary elapsed time for precursor reactions culminating in the undesirable premature ignition of the charge. Reduced exhaust valve temperature is very likely to improve chamber performance in this respect. Spreadsheet modelling of the scavenge efficiency to date suggests that alternating exhaust valve operation is typically viable up to an engine speed between 3000 and 3500 RPM.

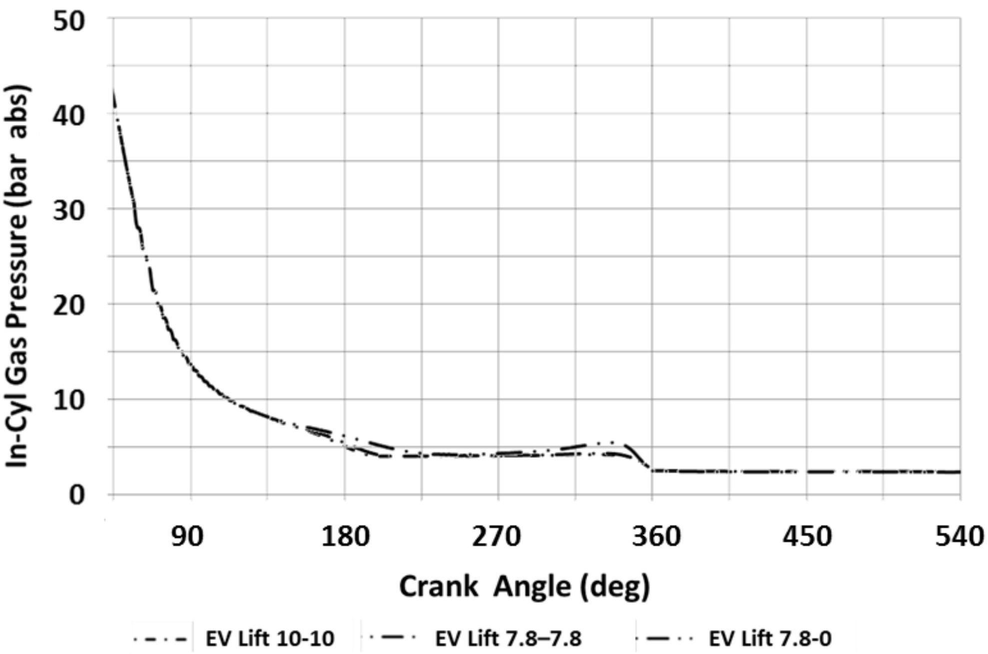

Fig. 10: Overlaid Cylinder pressure Diagrams Showing the Period from around EVO to beyond EVC for the Three Exhaust Configurations of Figure 9

Thermal Pre-ignition - as engine speeds increase, inevitably chamber temperatures increase and, eventually, a hot spot or spots can become hot enough to become an ignition source of itself. This phenomenon starts at approximately 2000-2500 RPM and continues throughout the remainder of the speed range. The exhaust valves are one such potential hot spot and alternating exhaust valve actuation will reduce the probability of thermal pre-ignition at least in the lower to mid speed range.

Sporadic pre-ignition is a rather more complex phenomenon and seems to be the result of hot or burning airborne particles, which may be lubricant sourced or dislodged chamber deposits, from the previous cycle acting as ignitors within the body of the charge. This may be observed from below 2000 RPM to above 4500 and clearly, reducing exhaust valve temperature can have only a secondary effect on this issue.

A further iVT capability is the ability to adjust valve events from one cycle to the next. Therefore, if a superknocking cycle is detected, the opportunity to reduce overlap or make any other suitable change is available on on the very next cycle. This capability is, of course, available on a cylinder-by-cylinder as well as a cycle-by cycle basis so only the knocking cylinder need be addressed.

3.9 Valve Event Modulated Boost

The concept of Valve Event Modulated Boost has been previously described by Roth et al [2, 3]. Essentially, the principle is to use separate ports for each exhaust valve and separate all the flows from one valve in each cylinder into one stream whilst those from the other valve are collected into a separate stream. One stream can then be directed into the turbocharger turbine inlet whereas the second is diverted past the turbocharger and feeds direct into the exhaust catalyst. The circuit is shown in Figure 11.

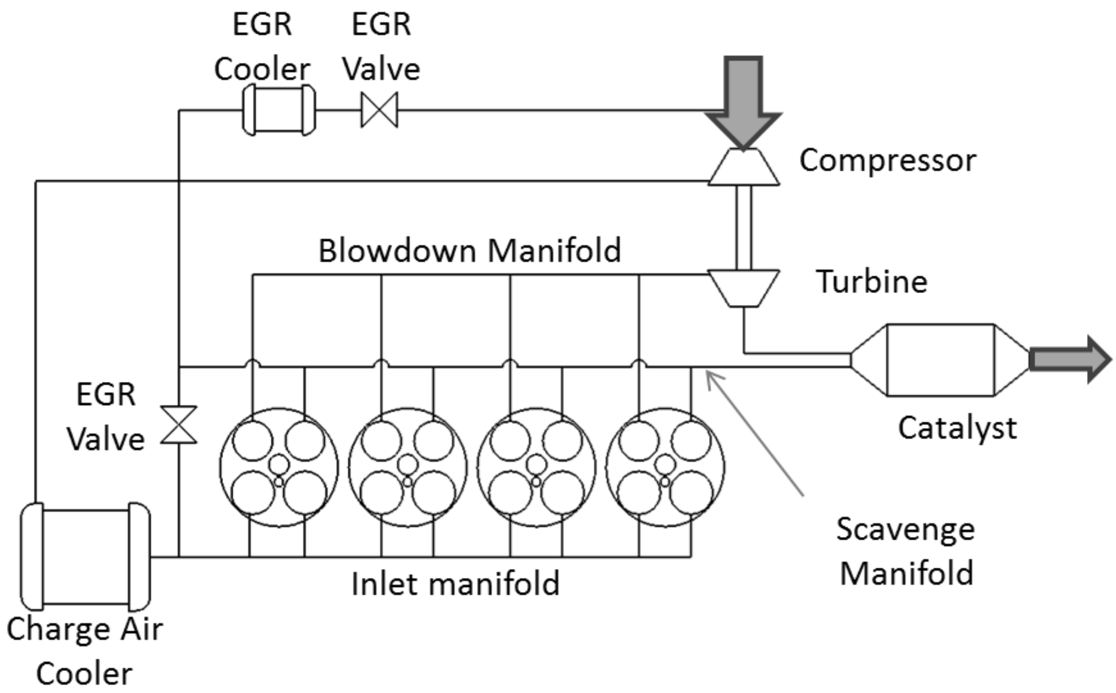

Fig. 11: Diagrammatic Valve Event Modulated Boost Layout

Roth used a concentric, cam-in-cam layout, with fixed cam profiles controlled by a cam phaser fitted to a 2 litre, 4 cylinder, direct injection engine rated at 20bar peak bmep (and therefore broadly similar to the engine used for iVT development). A full re-match of the turbocharger was included, eliminating the wastegate and reducing pumping losses. Using a combination of test bed experiment and simulations some impressive results were demonstrated. Roth's results include:

- Improved part and full load steady state fuel consumption - up to 12% at full load and 2.5-4% at part load.
- Up to 35kPa reduced pumping loss
- Reduced pumping losses incurred whilst achieving high EGR rates
- Reduced engine out HC emissions by using "Scavenge EGR" from the port connected directly to the catalyst

Note however, that this was all achieved using a very simple valve control mechanism with capability only to change the phase angle between the exhaust lobes. Thus the timing of the scavenge valve could be varied relative to the blow down and the phasing of the camshaft as a whole were the only variables. An iVT valve control system would open the possibility of adjusting the lift and period of blow down and scavenge valves independently and even, under cold start conditions, perhaps to reverse the phasing between the exhaust events so that the hottest gas was directed to the catalyst first.

Adapting an iVT system to this concept would probably imply the use of differently sized exhaust valves in order to reduce the gas force on the blow down valve at the valve opening timing and thereby avoid the need for a higher torque actuator for that valve only. This may result in a very advantageous package with differential inlet valve sizing. A diagram showing the how such exhaust valve asymmetry may be combined with the inlet valve asymmetry identified earlier is shown in Figure 12.

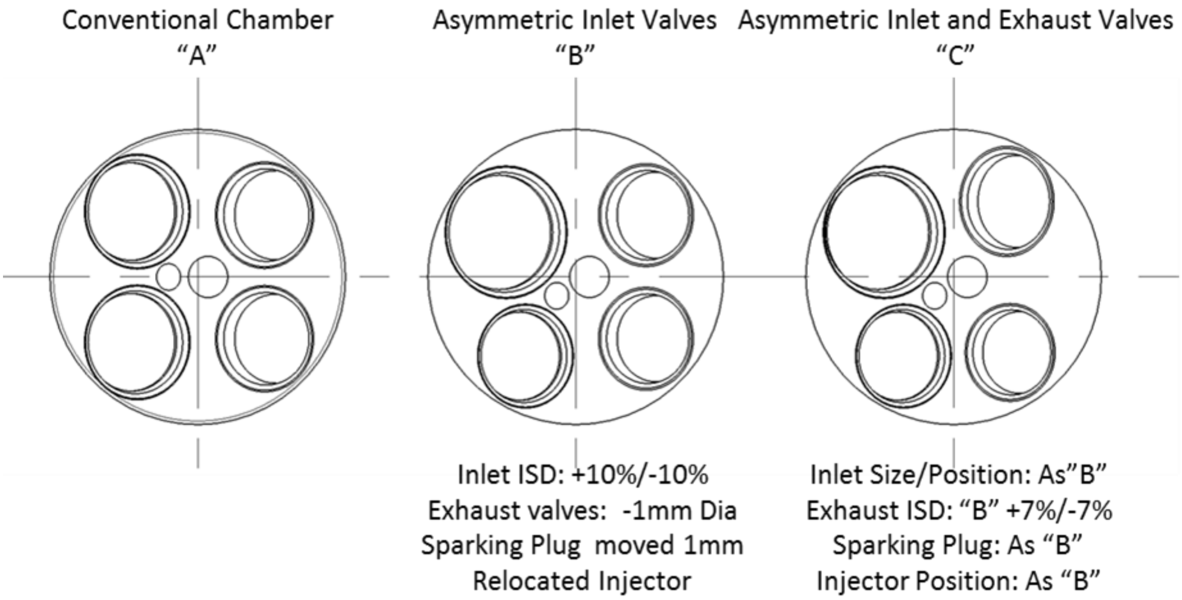

Fig. 12: Increasing Combustion Chamber Asymmetry - a Conventional Chamber, Asymmetric Inlet valves and Asymmetric Inlet and Exhaust valves

Many of these possibilities could be explored in the first instance using 1-D analysis as Roth has so ably demonstrated but multi-cylinder engine testing would be required to validate the results and to develop the control systems.

3.10 Additional Exhaust Events

In some circumstances, it may be of advantage to add a second exhaust valve event within a 720° cycle - this might be done to introduce hot EGR which can effectively be cooled in cylinder as a result of late inlet valve opening or early inlet closing strategies or it may be to extend the area of effective HCCI/CAE combustion over the load/speed range. At low to medium engine speeds iVT has the capability to provide such events. Even greater capability is possible if an alternating exhaust valve strategy is being pursued because the main exhaust event can be completed using one exhaust valve whist the secondary event is provided using the other.

Whilst other, essentially mechanical, systems are available to fulfil such a function, these are primarily aimed at the thermal management of diesel exhaust gas after-treatment systems and have limited flexibility. iVT can provide a similar function plus all the control flexibility inherent in the iVT concept: infinitely variable lift, period, phasing etc. This may extend the applicability of such a system from thermal control of (petrol or) diesel engine after-treatment systems to a rather more sophisticated HCCI optimisation feature. 1-D simulation modelling would be a useful first step in the investigation of such a possibility.

4 Conclusions

Despite the fact that variable valve control systems have been in volume production for over 30 years the opportunities for using them to extract yet more from internal combustion engines remain extensive.

This paper has concentrated on those opportunities that are related to combustion and ignored others, many of which will also offer CO2 reduction benefits. This emphasis upon combustion however, is important: the potential for improved fuel consumption and therefore reduced CO2, in conjunction with further improvements in toxic emissions, is critical to the future of the IC engine as part of our worldwide transport structure.

Significant issues presented by such a wide range of control variable are both the optimisation of an engine employing such technology and the subsequent calibration of the vehicle. This paper attempts to show how some reduction in complexity may be achieved by using the simplest approaches possible to reduce the optimisation envelope both swiftly and cheaply before fired engine testing begins.

Application of such technology to hybrid vehicles offers even greater opportunities as the ability to switch into and out of Miller/Atkinson modes, to reduce (or increase) motoring torque, deactivate cylinders and perform very rapid transient changes can all be exploited to improve CO2 performance.

Much work will need to be done in order to establish which of the potential approaches shown above will yield the most useful real world benefits and how they may be combined to best effect. However, it is clear that the scope for using very capable variable valve train systems such as Camcon's iVT arrangement in order to further the combustion development of petrol engines is unprecedented. There is much to learn and much to gain.

5 Acknowledgements

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