A comparison of inlet valve operating strategies in a single cylinder spark ignition engine

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ABSTRACT

This experimental work was concerned with comparison of inlet valve actuation strategies in a thermodynamic single cylinder spark ignition research engine equipped with a mechanical fully variable valvetrain on both the inlet and exhaust. The research involved study of the effects of the valvetrain on combustion, fuel economy and emissions when used to achieve variable valve timing alone and when applied together with early inlet valve closing for so-called unthrottled operation. The effects of such early inlet valve closure were examined using either fully variable events or by simulating two-stage cam profile switching. While fully variable operation enabled the maximum fuel savings over the widest operating map, it was apparent that two-stage switching mechanisms can provide an attractive compromise in terms of cost versus CO_2 benefit on engines of moderate to large capacity. However, from speed-load maps obtained in the current study it would appear that a wide range of inlet valve durations would be necessary to obtain fuel savings sufficient to warrant a system any more sophisticated than current variable valve timing mechanisms in future aggressively downsized gasoline engines.

ABBREVIATIONS

BMEP: Brake Mean Effective Pressure
BSFC: Brake Specific Fuel Consumption
c.a.: Crank Angle
DI: Direct Injection
EGR: Exhaust Gas Recirculation
EIVC: Early Inlet Valve Closing
EOI: End of Injection
EVC: Exhaust Valve Closing
EVO: Exhaust Valve Opening
FSI: Fuel Stratified Injection
IVC: Inlet Valve Closing
IVO: Inlet Valve Opening
IMEPg: Gross Indicated Mean Effective Pressure
IMEPn: Net Indicated Mean Effective Pressure
ISFCn: Net Indicated Specific Fuel Consumption
MBT: Maximum Brake Torque
MFB: Mass Fraction Burned
MPT: MAHLE Powertrain
NA: Naturally Aspirated
OEM: Original Equipment Manufacturer
PFI: Port Fuel Injection
RON: Research Octane Number
SI: Spark Ignition
TDC: Top Dead Centre
THC: Total Unburned Hydrocarbons
ULG: Unleaded Gasoline
VLD: Variable Lift and Duration
VVA: Variable Valve Actuation
VVT: Variable Valve Timing
WOT: Wide Open Throttle
λ: Relative air-to-fuel ratio
σ: Standard deviation

1 INTRODUCTION

The ability to vary the inlet and exhaust valve events of the spark ignition engine is well known to facilitate improved compromise between performance, fuel economy and emissions. Variable Valve Timing (VVT) is certainly an established technique for improving the fuel economy of the gasoline engine (1-3). There are several mechanisms by which VVT influences fuel consumption:

- Increasing or delaying valve overlap, which increases trapped residuals and reduces pumping work
- Late Inlet Valve Closure (IVC), which decreases pumping losses
- Late Exhaust Valve Opening (EVO), which can increase expansion work

As a result, fuel economy can typically be improved by up to ~5% over the European drive cycle. The choice of optimum VVT strategy is highly dependent on exhaust manifold design, compression ratio, cam phasing limits due to clash, part-load residual dilution tolerance and the importance of Wide Open Throttle (WOT) performance relative to part-load fuel consumption and emissions. A dual-independent VVT strategy offers high overlap potential and reasonable compromise between maximising WOT torque and minimising part-load fuel consumption and emissions. This is particularly true for hardware combinations that benefit from high overlap at both WOT (for scavenging) and part-load (to increase trapped residuals). An alternative strategy that could be used is high overlap retard, which can potentially give even lower fuel consumption than a maximum overlap strategy (4). However, WOT torque can be compromised, particularly at low speed unless very wide range, fast response cam phaser units can be used (5).

In more recent years, there has also been considerable interest in fully variable valvetrain systems for additional improvements, such as:

- Further reduction in throttling losses via load control directly at the inlet valve(s)
- Increased thermal efficiency through greater expansion ratio
- Reduced tailpipe emissions under cold-start (direct fuel injection) conditions via late Inlet Valve Opening (IVO)

Numerous fully variable valvetrain strategies exist depending on application, but the currently reported work is most concerned with those SI engine strategies claiming to enable improved part-load fuel efficiency. During his useful study in a similar vein, Tuttle compared the effects of Late Inlet Valve Closing (LIVC) and Early Inlet Valve Closing (EIVC) in a single cylinder research engine. In early experiments (6), IVC was delayed through 60-96° crank. Fuel consumption was observed to reduce by up to 6.5% and was accompanied by lower engine-out emissions of NOx (~24%) but similar hydrocarbon levels. Regardless, Tuttle concluded that 96° was the maximum delay that could be tolerated due to loss of effective compression ratio, which would significantly limit the attainable speed-load map (assuming no external compression was available). In subsequent work (7), it was concluded that the EIVC strategy was favourable for part-load, allowing de-throttled operation over a wider speed-load window, albeit reliant on 200° crank range of inlet valve closing.

To date, there still remains significant interest in the thermodynamic effects of EIVC operation. For example, in a recent study Cleary and Silvas (8) investigated EIVC at part-load (1300rpm/3.3bar net IMEP) in a single cylinder PFI engine. It was observed that such a strategy led to reduced in-cylinder turbulence, associated with the increased time under closed valve conditions. In turn, this resulted in prolonged combustion duration, reduced in-cylinder gas temperatures, reduced engine-out emissions of NOx (up to 25%) and increased values of unburned hydrocarbons (also 25%). The deterioration in burn rate could be reduced by switching to a LIVO strategy but, under the reduced valve duration conditions tested, the pumping losses were worse than the conventional throttled benchmark case.

The benefits achieved by EIVC operation are highly dependent on the valvetrain system employed. At one end of the spectrum, various electro-magnetic and electro-hydraulic systems have been proposed. Such camless systems have been reported to allow the greatest potential for reduction in SI engine breathing losses (9, 10,) and/or can be used to realize advanced modes of operation such as Controlled Auto Ignition (11, 12). However, in general these systems often still have significant issues to overcome including packaging, noise and cost. The majority of fully variable valvetrain systems entering production have been mechanically based as these arguably present a viable step on a cost-benefit basis. For example, Unger et al. (13) recently summarised the design and development of the second generation Valvetronic system, capable of providing up to ~12% reduction in fuel consumption at part-load. Improvements claimed in the second-generation iteration included more aggressive valve lift profiles, re-optimization of the inlet port masking, friction reductions and integration of the control system within the main engine control unit. Elsewhere, Shimizu and co-workers (14) recently reported on the benefits of the new production Valvematic system, a well-packaged continuously variable valve lift system offered in conjunction with dual VVT. These workers reported up to ~10% improvement in fuel economy could be achieved during typical part-load operation, with similar percent improvement in power. Another mechanical fully variable system recently introduced is the Nissan VVEL (15, 16), which enables relatively aggressive inlet lift profiles to be used with fuel economy benefits similar to those above. Elsewhere, Sellnau and co-workers (17) have demonstrated how two-stage mechanical valve actuation can allow viable further compromise on a cost-benefit basis, achieving 5.5% improvement in fuel economy and ~46% reduction in NOx over an EPA drive cycle.

The above systems were all used with port fuel injection. However, the combination of EIVC with homogeneous Direct Injection (DI) has also begun to warrant interest, with the potential for further fuel savings if used together with increased compression ratio. For example, workers on the "Hotfire" collaborative project recently examined such effects in both optical and thermodynamic single cylinder SI engine assemblies (18, 19). During this study, greatest fuel consumption benefits could be achieved if just one of the two inlet valves was actuated. However, the swirl dissipated quickly once the valve closed and the fuel economy benefits recorded varied substantially depending on which of the two inlet valves was activated.

The main objective of the currently reported work was to assess the fuel economy benefits of various valvetrain systems of different complexity. To this end, a single cylinder engine has been prepared with a fully variable valvetrain on both the inlet and exhaust to allow the effects of variable valve timing, two-stage cam profile switching and finally fully variable unthrottled operation to be quantified using a common hardware platform.

2 EXPERIMENTAL SETUP

2.1 Engine Assembly

The experiments were performed in a naturally aspirated single cylinder four-valve per cylinder spark ignition research engine. Some general details of the unit are presented below in Table 1.

Table 1: Basic engine characteristics

No of cylinders	1
Bore (mm)	82.5
Stroke (mm)	88.9
Geometric compression ratio	9.8:1
Variable Valve Timing	Fully Variable (Inlet & Exhaust)
Fuel injection	Interchangeable (port fuel injection used in this study)
Spark plug	NGK single electrode
Ignition coil	Bosch coil-on-plug
Fuel	95 RON unleaded gasoline

The base engine assembly is illustrated in Figure 1. The engine was designed to provide a low-cost single cylinder assembly. As such, the bottom-end was based on a modified industrial Lister-Petter diesel sub-assembly, re-fitted with a prototype water-cooled barrel and bespoke con-rod and piston assemblies. Sections through the prototype cylinder head assembly are illustrated in Figure 2. This head was designed to allow study of port, side direct or central direct fuel injection using a common platform. The ports and combustion chamber geometry were based on those of an Audi FSI production engine.

This modular single cylinder assembly also allows straightforward conversion between the existing thermodynamic setup and fully optical operation, with the latter demonstrated recently under both port and side direct fuel injection operation (20, 21). The currently reported work has so far only been concerned with thermodynamic operation. The port fuel injection setup used throughout this study included a production injector and prototype flat-topped piston, with the fuel spray targeting the back of the inlet valves as per a typical production engine application. The valvetrain valve lift profiles had been designed to maintain acceptable dynamic loads at engine speeds up to 6500rpm. Prior to running the test engine, the valve lift versus duration was measured by spot facing the valves and fitting the cylinder head sub-assembly to an in-house motoring rig. A laser differential vibrometer system was then used to obtain direct valve velocity measurements via the laser Doppler interferometry technique (results are shown later). During these tests, the cylinder head oil circuit was connected to an oil conditioning rig that controlled oil temperature and pressure to engine-like motoring conditions (90°C±2°C, 3bar gauge).

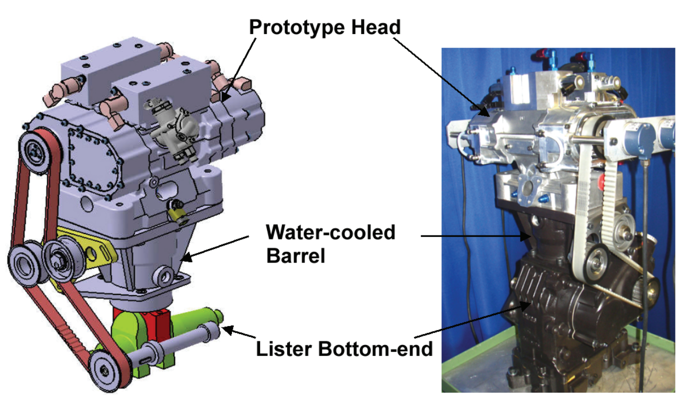

Figure 1: Base engine assembly (thermodynamic setup)

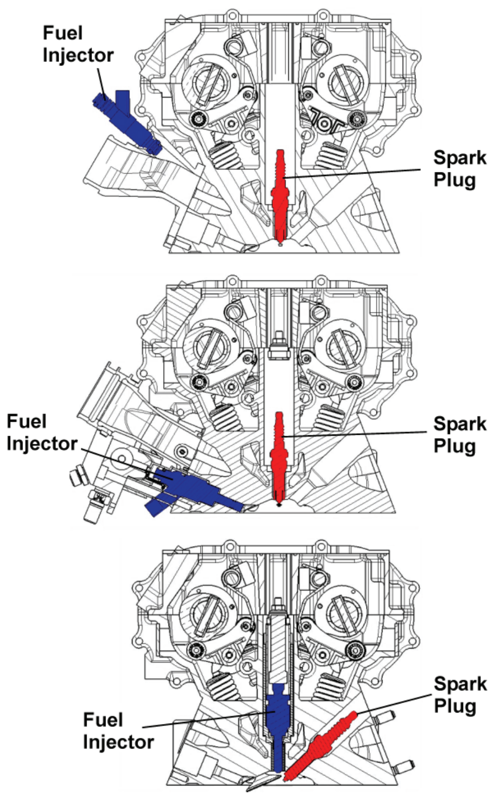

Figure 2: Cylinder head assembly showing port, side direct or central direct fuel injection using a common hardware platform

The cylinder head included a mechanical fully variable valvetrain assembly, fitted to both the inlet and exhaust. The system used was an evolution of the MAHLE Variable Lift and Duration (VLD) mechanism, previously introduced in detail elsewhere (22, 23). In brief, the system is based on a shaft-in-shaft cam operating principle. An example of an inlet VLD mechanism is illustrated in Figure 3. The two opening cams are pressed on to the outer shaft. These cams are equivalent in profile and must open the two inlet valves in a synchronous fashion via the lever assembly. The opening cam contours were designed so that, if no closing cam were available, the inlet valves would remain open at maximum lift for a prolonged period before eventually closing in a safe manner. However, such operation is hypothetical as the closing cam is available and pinned to the inner shaft. The phase of this closing cam, relative to the opening cam, can be advanced so as to close the valve earlier and reduce the valve lift and duration in proportion. The profiles of the cams were designed so as to ensure acceptable dynamic forces were produced regardless of phase.

In summary, the opening and closing cams act in tandem to produce a mean cam (and hence valve) lift curve. By advancing the phasing of the inner shaft relative to the outer shaft, the closing of the valve is advanced, hence allowing reduced lift and duration to be achieved. The phasing of the closing cam was controlled using a prototype wide range (140° crank) hydraulic cam phaser, denoted in Figure 4 as the "VLD Phaser". In order to then achieve fully variable valvetrain operation, a second hydraulic VVT cam phaser was fixed to the outer tube, providing 40° crank timing range. Similar ranges were available on the exhaust, however during the current study the exhaust VLD phaser remained locked in the maximum lift setting.

An additional point to note regarding the adoption of this valvetrain was that the cylinder head used in the current work did not include any form of inlet port masking (sometimes alternatively referred to as port shrouding). Most mechanical VVA systems entering production are known to include such masks, which effectively reduce the flow area in to the cylinder at the back of the inlet valves and hence increase forward tumble ratio. However, during this initial work it was decided necessary to omit such masks and obtain a baseline data set. An alternative less common approach might be to attempt to invoke increased swirl but unfortunately one limitation of the VLD system available was that symmetrical valve lifts had to be used. Furthermore, the effects of such swirl during EIVC operation are arguably less well understood.

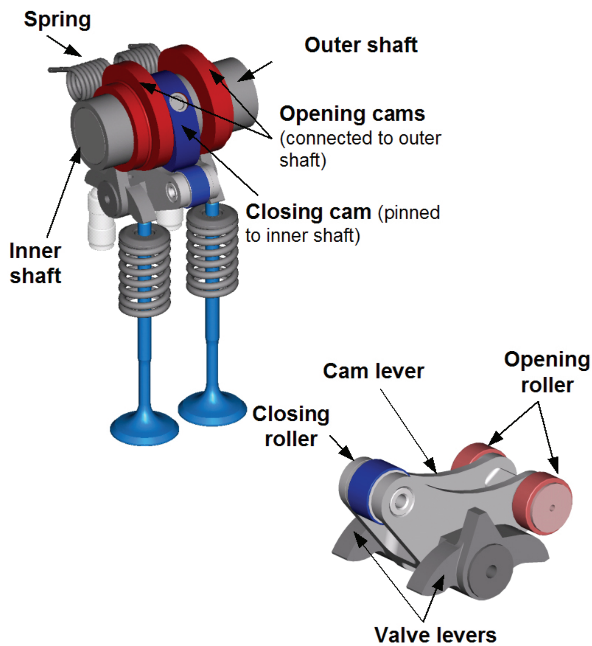

Figure 3: Key VLD components (example shown for inlet VLD operation)

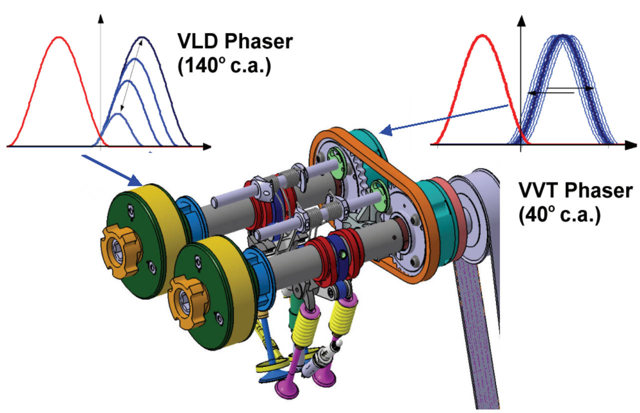

Figure 4: Single cylinder VLD assembly with the VLD and VVT cam phasers of the inlet camshaft identified

2.2 Test Apparatus

During the experiments, the cylinder head was fitted with a Kistler 6041A water-cooled in-cylinder pressure transducer, the face of which mounted flush with the combustion chamber walls. Pressure data acquisition was performed using a SMETEC Combi system. The corresponding pressure data analysis was performed using AVL Concerto Version 3.8. Corresponding thermodynamic parameters were evaluated as the average of values compiled over 300 engine cycles. The engine-out emissions were sampled using a Horiba MEXA 9100 analyser. Fuel flow measurements were performed using a coriolis fuel flow meter assembly, calibrated in-situ to provide a maximum reading error of <0.5% at the minimum fuel flows reported.

3 RESULTS

3.1 EIVC & VVT Effects

The objective of these first tests was to study EIVC when combined with internal EGR, such as if the engine were to be fitted with a fully variable valvetrain on the inlet and fixed valvetrain on the exhaust. Shown in Figure 5(a) are corresponding maps produced at 1500rpm/3.2bar IMEPn (MBT spark timing, λ=1, 95RON pump gasoline). This net IMEP was equivalent to that generated in a suitable multi-cylinder production engine at a reference brake load of 2.62bar BMEP (a mapping site used by some OEMs). The "baseline" data point marked on the map indicates the valve timing settings for a typical non-VVT multi-cylinder engine, with overlap fixed to allow acceptable performance across the speed-load map. During the experiment, the exhaust valve lift remained locked at the maximum value of 8.9mm. EVC was also held fixed at 9°aTDC. The fuel injection timing also remained constant and was set for best fuel economy at the "baseline" condition (EOI=400°bTDCF). The most advanced IVO setting shown was governed by valve-to-piston clash, with 1mm clearance maintained.

Shown in the corners of the valve timing map are four valve lift cartoons that help illustrate the valve events occurring in different areas of the map. In the top right-hand corner the engine inlet was throttled, with maximum valve lift and minimum valve overlap applied. In the bottom right-hand corner, IVO was advanced for increased overlap and internal EGR. Moving horizontally from right to left, in the bottom left-hand corner the engine was operating with the highest possible valve overlap and also reduced inlet duration and lift for an unthrottled EIVC strategy. Finally, in the top left-hand corner, the unit was operating with minimum valve overlap while maintaining EIVC operation. It is important to note that the inlet plenum pressure could be increased to between 0.99-1.0bar absolute across the entire left-hand side of this map, with full unthrottled operation hence achieved at all IVO timings studied.

Observing Figure 5(a), up to 11% improvement in net Indicated Specific Fuel Consumption (ISFCn) was possible at this site when combining EIVC and inlet VVT. Of this benefit, ~3% was attributable to VVT as seen in the bottom-right hand corner of the map. The remaining ~8% was associated with EIVC operation. The inlet valve duration and lift at the best ISFC site were 112°crank/2.23mm. The corresponding standard deviation in gross IMEP (σ IMEPg) remained well within acceptable limits (0.06bar). It was therefore apparent that higher amounts of internal EGR could be tolerated in this engine at this site. Otherwise, it was interesting to note the island of worst ISFC occurred with IVC around 180°. When IVC occurs after 180°, the motion of the piston can push air back up in to the inlet system. When throttled, this resulted in slightly higher inlet plenum pressures and hence marginally lower pumping losses than incurred with IVC at ~180 °.

The next tests were concerned with combining a fully variable inlet valvetrain with exhaust VVT (a slightly higher cost solution but still considered to be feasible for production). As such, shown in Figure 5(b) are percent ISFC measurements made at equivalent test conditions to those employed in Figure 5(a), but now with the exhaust event retarded by 34° and EVC fixed at 43°aTDC. An exhaust phase shift of 34° is well within the capability of typical production cam phasers. Up to ~13% improvement in fuel economy was possible c.f. a fixed valvetrain solution, with ~6% benefit attributed to dual independent VVT and ~7% associated with throttleless effects. Advancing the inlet event beyond this point soon resulted in degradation in stability and hence fuel consumption; the dashed line superimposed on the map illustrates where the stability limit of σ=0.12 was reached. The 0-10% and 10-90% Mass Fraction Burned (MFB) periods were arguably more strongly influenced by increasing internal EGR than EIVC. However, this was not always the case as discussed in more detail later on.

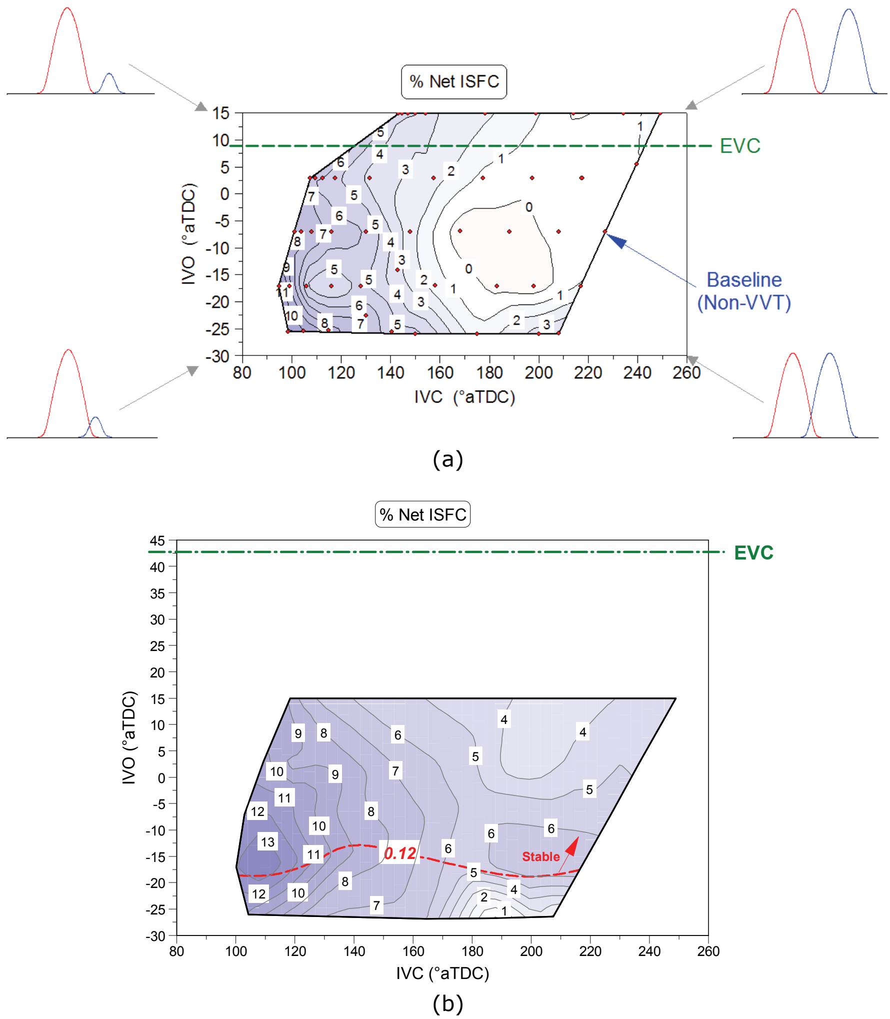

Figure 5: Percent ISFC maps when combining EIVC with (a) inlet VVT and (b) exhaust VVT at 1500rpm/3.2bar IMEPn. Dashed line superimposed on (b) shows combustion stability limit of σ IMEPg=0.12.

In order to understand why VVT further improves part-load fuel economy when full unthrottled operation has already been achieved it is useful to consider the inlet valve lift curve shown in Figure 6(a). As previously discussed, this curve was measured on a motoring cylinder head valve motion rig. This characteristic S-shaped curve is typical of many production mechanical variable valvetrains; where valve lift must be reduced at the lowest valve durations in order to maintain acceptable kinematic loads on the valvetrain across the engine speed operating range. Superimposed on the figure are dashed lines denoting the values of inlet valve duration and lift required when operating firstly using EIVC operation without variable valve timing (solid line) and secondly when combining EIVC with the maximum valve overlap possible (dashed line). This data was produced at the typical part-load operating site of 2000rpm/2.7bar IMEPn (MBT spark timing maintained, λ=1). In brief, the adoption of variable valve timing in addition to EIVC operation resulted in increased trapped residual mass until the acceptable combustion stability limit of σIMEPg=0.12 was approached. In order to then maintain engine load, it was necessary to increase the valve duration and lift from 107°/1.97mm to 125°/2.99mm as indicated. In turn, this resulted in reduced throttling at the inlet valve itself, with operation forced much nearer to the point of inflection on the S-shaped curve and hence a more aggressive valve lift profile event achieved. The corresponding reduction in the pumping loop area of the log p Log V diagram is illustrated in Figure 6(b).

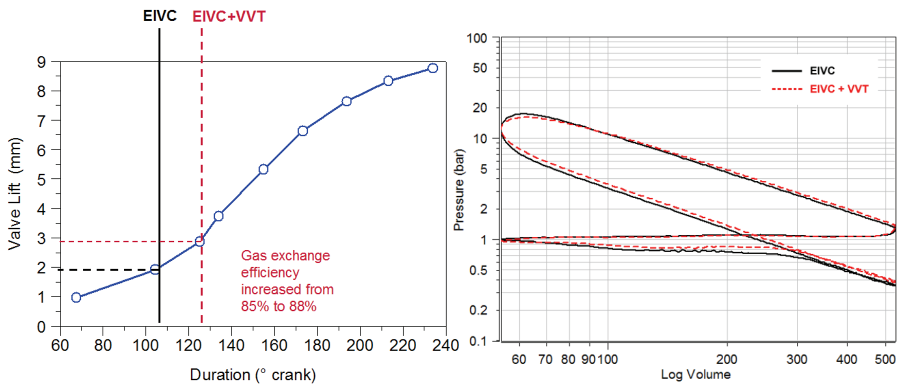

Figure 6: a) Values of valve lift for EIVC operation with and without VVT at 2000rpm/2.7bar IMEPn and (b) corresponding pumping loop diagrams

In conclusion, most mechanical Variable Valve Actuation systems must endure low valve lift at low valve duration, which can lead to throttling at the valve itself. Under such circumstances, it seems the more EGR tolerated the better. Such benefits might of course be reduced if, for example, camless operation with trapezoidal wave valve lift profiles and/or faster rates of valve actuation could be practically achieved. Otherwise, in terms of maximising lift at low duration, the current VLD system performed relatively well for a mechanical system, with reasonably aggressive lift profiles achieved as seen in Figure 6.

3.2 Speed-Load Maps

With the above observations understood, the effects of combining dual independent VVT with EIVC could be mapped across a speed-load window. The low-cost bottom-end available at the time of this work could only be used at speeds up to 2000rpm. Nonetheless, a reasonable portion of the drive cycle could still be studied. Therefore, shown in Figure 7 is a comparison of VVA effects over such a window. The engine was operated at MBT spark timing, λ=1 and with EOI=400°bTDC at all shown sites. The left-hand column of graphs was produced using VVT alone. The right-hand set was generated using VVT in combination with EIVC operation. The overlap used at each site was governed by the EGR tolerance of the engine. Beyond the overlap settings shown no additional improvements in fuel consumption could be made. Observing the maps, the overlap varied from 26° to 65°, increasing with speed and load. At higher engine speeds there is usually less time for the EGR to be re-breathed from the exhaust manifold and hence higher overlap is required. It is interesting to note that similar levels of overlap were tolerated with and without EIVC. However, it is important to note that less internal EGR will have been trapped in the VVT+EIVC case for a given level and phase of overlap due to the less favourable EGR pressure ratio from exhaust to inlet during the breathing event. This is also perhaps reflected in the differences in engine-out NOx.

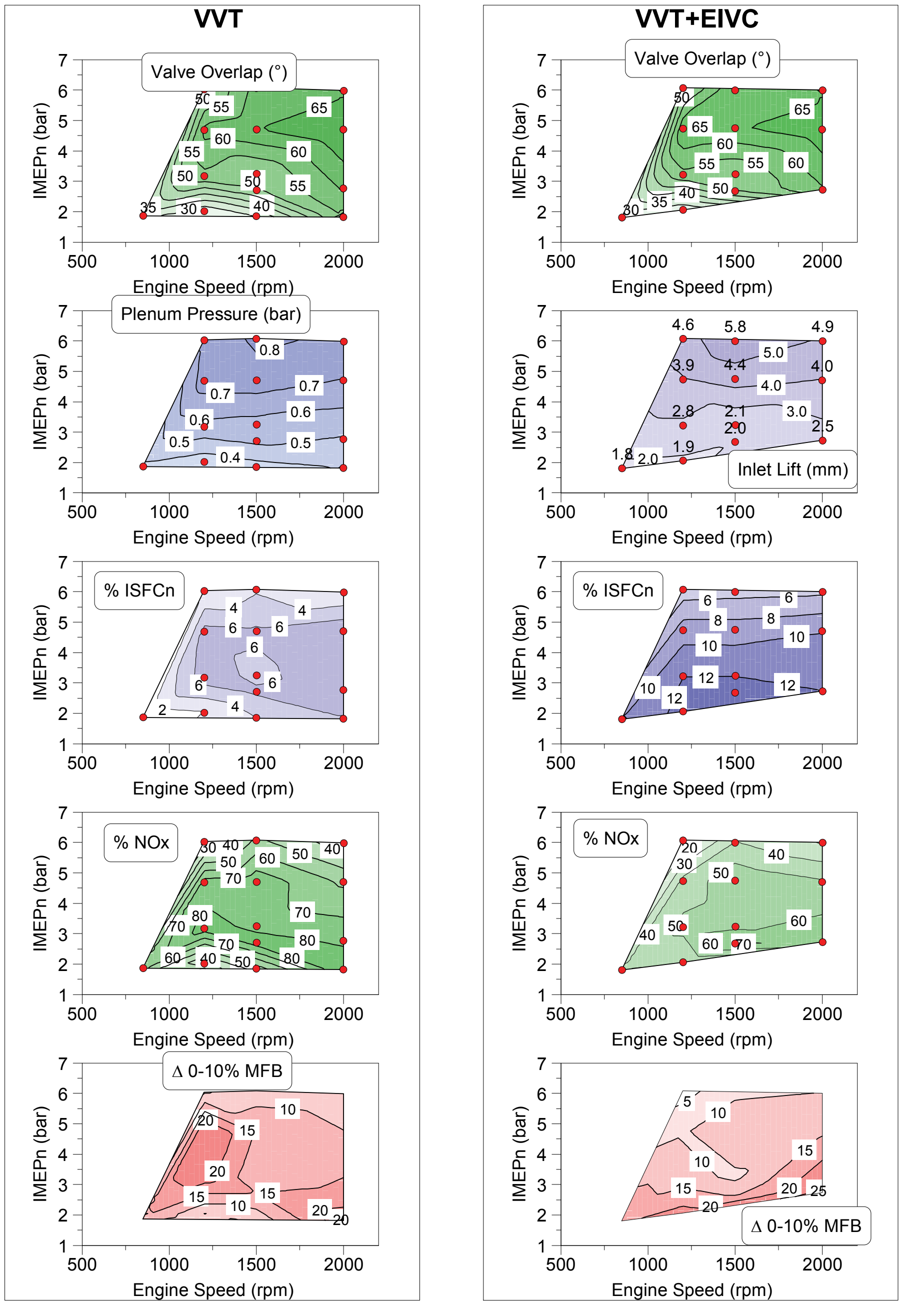

Figure 7: Speed load maps for a) VVT and b) EIVC+VVT operation

During the VVT-only tests, the engine was still operating under part-throttled conditions, with corresponding values of inlet plenum pressure shown. When adopting VVT+EIVC, atmospheric inlet plenum pressures were achieved at all shown sites. Corresponding values of inlet valve lift are shown, varying from 1.8-5.8mm across the mapped regime and increasing with speed and load. As output increases under unthrottled conditions, a longer inlet duration is required to inhale an increased mass of fresh charge. Assuming at least an equivalent percentage of EGR to that tolerated at lower load is still required, a higher mass of residual would be needed and hence additional valve overlap would be necessary. The assumption of at least equivalent EGR rate is deemed sensible, given EGR tolerance is well known to increase with load in the SI engine (4). However a larger step change in overlap than observed here would be required to claim any significant improvement in such tolerance occurred with such relatively small change in output over the map.

The values of percent ISFC shown in Figure 7 are expressed relative to the "baseline" valve timing case (conventional throttled non-VVT operation). The effects of VVT on fuel economy are well understood and the improvements recorded are not too dissimilar to those reported in the literature (2-5). The additional use of EIVC allowed further improvements in ISFC of 6-8% at lower loads. At higher output, the benefit of EIVC operation dropped to ~2%. It is important to note that these benefits were recorded under stable conditions, with the highest value of σIMEP across each map equal to 0.08 (VVT) and 0.075 (VVT+EIVC) respectively. When greater overlap was attempted the combustion stability soon deteriorated. Overall, these results of course neglect any cylinder-to-cylinder breathing effects in real engines and the friction losses of such a valvetrain. Unfortunately a single cylinder cannot provide a representative friction baseline due to the high degree of bottom-end friction and parasitic losses acting on the one cylinder alone.

At 2000rpm, it can be seen that it was not possible to combine VVT with EIVC operation below 2.7bar IMEP. Each technology could be applied alone but not together and this shall be discussed further later on. Another key point to note at this speed was that the net IMEP values at two of the mapped sites (2.7bar and 4.7bar respectively) had been selected to match the indicated load produced at 2bar and 4bar BMEP in a multi-cylinder production engine. Comparison of these two sites arguably gives some insight in to the indicated fuel economy gains available if such a valvetrain was to be adopted on an aggressively downsized engine. When capacity is reduced by 50%, the brake output will approximately double for a given road-load requirement. By testing the engine at output equivalent to both 2bar and 4bar BMEP, it may be argued that some of the effects of halving an engine's capacity can be uncovered. This simplistic approach of course neglects differences in compression ratio, in-cylinder flow, EGR tolerance, friction or in-vehicle gearing which will be necessary in many cases to best utilise the downsizing principle. Nonetheless, the data still highlights how the benefits of unthrottled EIVC operation are significantly reduced in proportion to any degree of downsizing employed. In summary, if capacity was to be halved, the additional benefit in indicated fuel economy from fully variable EIVC operation would drop from 4.7% to 2.2% at the example 2000rpm cruising point.

Finally values of the difference in 0-10% MFB period are also shown c.f. a "baseline" throttled non-VVT case. When employing VVT+EIVC, the corresponding isolines in the bottom right-hand figure indicate that the 0-10% MFB period was heavily influenced by load at very low engine speeds. However, these isolines became more diagonally inclined at the highest engine speed studied. Comparing the 1500rpm/2.7bar and 2000rpm/2.7bar sites, the 0-10% MFB period increased by 12° crank. The associated 0-10% MFB time period increased from 3.3ms to 4.65ms from 1500rpm to 2000rpm, clearly demonstrating the combustion period was prolonged. As a result, it was not possible to combine VVT and EIVC with the hardware available at the highest speed and lowest loads, which may be unacceptable in any practical application. In conclusion it seems efforts are required in future work to increase the flame speed under light loads and also moderate engine speeds.

3.3 Cam Profile Switching

Set out in Figure 8 are inlet valve duration/lift sweeps performed at the speed-load sites in the corners of the VVT+EIVC speed load map. Two data sets are shown at each site. The low overlap data was obtained with the inlet and exhaust cam timing fixed as per the non-VVT engine baseline setting. The high overlap data set was gathered at the maximum possible overlap setting at each site, where combustion stability could be maintained below the acceptable limit of σ IMEPg>0.12 across the sweep. The dashed lines superimposed on the figure illustrate the best possible fuel economy benefit with the inlet valve duration/lift set to a fixed value of 125°/3mm in an attempt to simulate the effects of two-stage cam profile switching (CPS). At higher loads it was found that, when combined with VVT, fully unthrottled operation could not be achieved with a 3mm lift CPS system. Only by reducing the valve overlap back to as far as the non-VVT setting was it still possible to achieve load. Under such conditions, the improvement in fuel consumption was not too far away from that obtained with the fully variable valvetrain, with 3.9% saving c.f. 5.7% for fully variable actuation at 2000rpm/6bar IMEP. This 3.9% represented the best fuel economy available from this CPS setting at this site; no other combination of inlet or exhaust valve timing achieved load with 3mm lift. The CPS system performed better with VVT at 2000rpm and lower load, with 11% improvement in fuel economy still achieved c.f. 12% with the fully variable system. Such a speed-load site is often quoted to be a typical cruising condition for moderate-to-large capacity engines and hence such substantial savings are not without significant merit over typical drive cycle assessments. However, the major limitation of such a CPS system was selecting a valve duration/lift large enough to capture the high load sites (and hence avoid excessive transitions) but also still achieve fuel savings at the lowest speeds and loads visited during the drive cycle. In the 3mm case studied here it can be seen that the improvement at the idle site was 4.7%, which compares to 10% via fully variable operation.

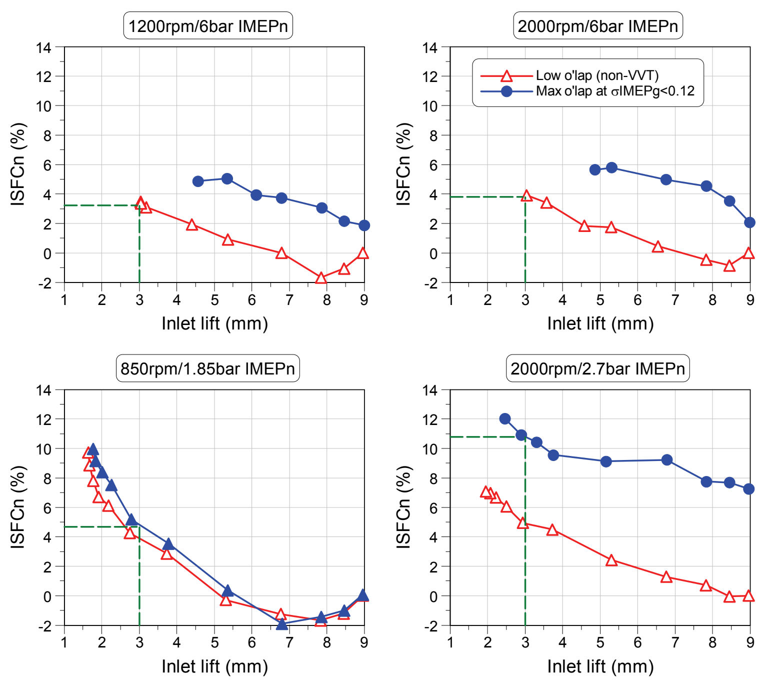

Figure 8: Comparison of inlet valve lift sweeps performed at the speed-load sites in the corners of the VVT+EIVC speed-load maps

Set out in Figure 9 is a cost-benefit analysis of various spark ignition engine technologies based on experimental fuel consumption measurements and cost information available to the authors at the time of publication. The datum case used in the analysis was a four cylinder 2.0litre PFI engine operating with a fixed valvetrain. With specific respect to the current study, inlet cam profile switching combined with dual VVT would appear to present a very attractive compromise. However, such an analysis must be treated with caution in terms of generic application. For example, the data shown is compared to a naturally aspirated baseline and therefore does not readily account for combination of such valvetrain technologies with engine downsizing or down-speeding, both of which are widely foreseen to become dominant gasoline engine technologies within key markets in the short-to-medium term. From the speed-load maps obtained in the current study it would appear that a wide range of inlet valve lifts would be required to obtain an operating map sufficient in size to warrant a VVA system any more sophisticated than VVT in such an engine. The relative cost-benefits of other emerging (and arguably competing) technologies such as stop-start must also be considered. However, the possible benefits of fully variable inlet valve operation at moderate and higher loads should also not be neglected (Miller cycle operation, for example) and this should be the subject of future work.

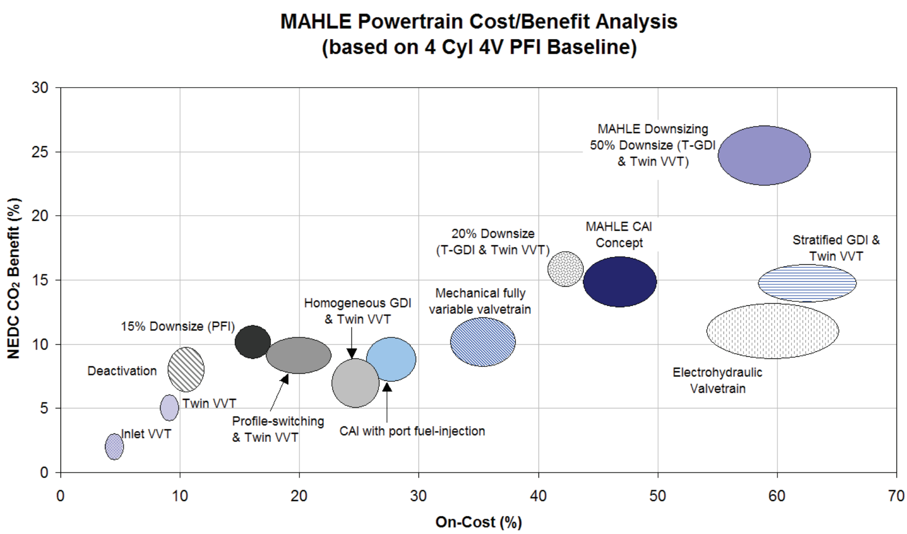

Figure 9: Cost-benefit diagram for various engine technologies

4 CONCLUSIONS

The effects of combining VVT with EIVC have been studied in a thermodynamic single cylinder research engine under port fuel injection conditions. During full unthrottled (EIVC) operation the following conclusions were made:

- At the typical part-load cruising site of 1500rpm/3.2bar IMEP, EIVC operation allowed fuel consumption reductions of up to 8%. The addition of inlet VVT allowed further savings of up to 3%. Exhaust VVT then enabled yet more small improvements of 2%.
- The maximum fuel economy benefits achieved were limited by increased 0-10% mass fraction burned periods and the onset of unstable combustion. This

effect seemed most limiting at low speed and very low loads (very early IVC with the lowest valve lifts) but also moderate loads at higher speeds.

- During unthrottled EIVC conditions at constant speed and load, increasing the valve overlap and hence internal EGR rate allowed greater inlet valve duration and lift to be used. In turn, this reduced the throttling across the inlet valves and enabled higher gas exchange efficiency.
- Most mechanical EIVC systems must endure relatively low valve lifts at low valve durations, which can lead to throttling at the valve itself. It would therefore appear that the more EGR tolerated, the better.
- The effects of EIVC on NOx emissions were typical of those reported elsewhere, with engine-out values reduced by up to ~40%. Unburned hydrocarbons were increased by up to ~70% at the lowest speeds and loads but the lack of port masking must be considered. These differences in emissions decreased as the valve duration was increased at higher loads.
- Under 50%-downsized conditions, the fuel economy benefit of EIVC operation fell from 4.7% to 2.2% at a typical 2000rpm cruising site.
- Two-stage cam profile switching is an attractive compromise in terms of cost-benefit but only provided a large enough operating window can be achieved, which is highly dependent on both engine capacity and vehicle transmission ratios.

Future work shall be concerned with study of the effects of inlet port masks, so far omitted from any engine test work, on flow, combustion, performance and emissions when operating in such valvetrain modes. Synergy between such VVA and alcohol-blended fuels shall also be examined at both low and higher loads.

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