Intelligent Valve Actuation – a Radical New Electro-Magnetic Poppet Valve Arrangement

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Summary

Intelligent Valve Actuation (IVA) enables digital control of the last controllable combustion variable – gas exchange. Designed for 12 volt operation, IVA far exceeds the capability of all other variable valve systems and has been designed as a practical, cost effective solution in these days of CO2 constraints. Development work to date has used the state-of-the-art Jaguar Land Rover 2 litre, 4 cylinder, petrol engine already using a very advanced variable valve system. Inlet only IVA steady state fuel economy improvements of up to 7.5% have already been demonstrated. Further work will yield significantly more. Throttle-less operation is inherent, transient response unprecedented and the opportunity for “smart” integration with other systems broad indeed.

1 Concept

IVA is an electro-magnetically operated poppet valve system suitable for both inlet and exhaust valves. It is very different from earlier electromagnetic systems using opposed solenoids, IVA employs a 4 phase rotary actuator using a rotor which is extended to provide a separate camshaft for each individual poppet valve. A desmodromic linkage connects this camshaft to the, entirely conventional, valve. The arrangement is shown in Figure 1 and includes all those capabilities also required in more standard valves trains: providing appropriate valve seating force, compensation for assembly tolerances and wear plus compensation for thermal expansion. The actuator is electronically synchronised with the crankshaft and drives the rotor through the required angular trajectory in order to provide the selected valve event. A full rotation of the rotor will provide a full lift event whilst part rotation and return provides a part lift event. An absolute position sensor is provided for each rotor and this is used as part of a high frequency closed loop control system. Unlike other desmodromic systems there is no “helper” spring occupying the position of a conventional valve spring, both the seating load and the mechanical compliance required to ensure that the mechanism cannot jam are provided in the drop link shown in Figure 7 – this is spring loaded in the direction of a tensile load only – so the cam mechanism can be set to “overclose” the valve, compressing the drop link spring as the valve is pulled onto the seat and thus providing a known seating force. An additional energy recovery mechanism may also be fitted. This reduces electrical power consumption or boosts maximum torque capability by using an additional cam lobe to recycle kinetic energy into and out of a spring arrangement effectively in parallel with the electro-mechanical actuator. From the outset, the system was designed to function using a 12 volt supply and all results presented were obtained using 12 volts. The system could, of course, be modified to use a 48 volt supply but

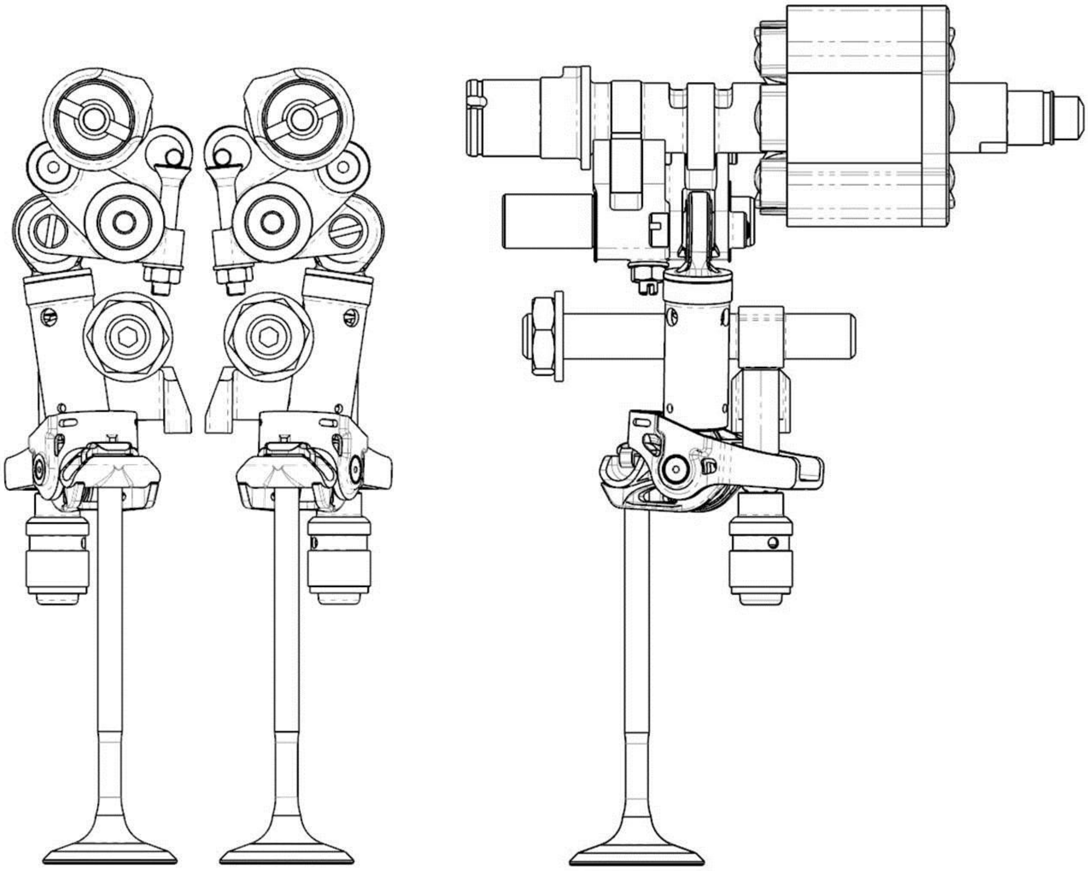

Fig. 1: Front and side views of mechanism without the bearing housing

the advantages would be small. There is absolutely no mechanical link to the crankshaft and yet the danger of piston to valve contact has been eliminated without piston modification. The valve timing, opening period and valve lift are all independently and infinitely variable. Furthermore, the event shape can also be manipulated and the valve is, at all times, under full feedback control. It is this control that allows features such as MOP (Maximum Opening Position) shifting, lift-dwell-return, double events, missed events and more. Roaming cylinder de-activation, swirl, swumble and tumble control are all easily implemented – even short term 2-stroke operation for transient response is possible when IVA is fitted to inlet and exhaust valves. The individual valves can be programmed independently and each valve in a given cylinder can be following different event profiles as can valves in adjacent cylinders. Similarly, the event that a valve is programmed to follow is not dependent upon the previous event so transient response can be exceedingly rapid. The concept has, in conjunction with a Tier 1, been subjected to a significant study on manufacturing feasibility and cost reduction and it has been demonstrated that the technology represents a cost effective measure in terms of cost/gm/km of CO2. Furthermore, other very significant benefits have been demonstrated including, for instance: extraordinarily quiet operation, elimination of torque reserve and other ignition based transient torque calibrations, dramatic reductions in cold cranking torques and more.

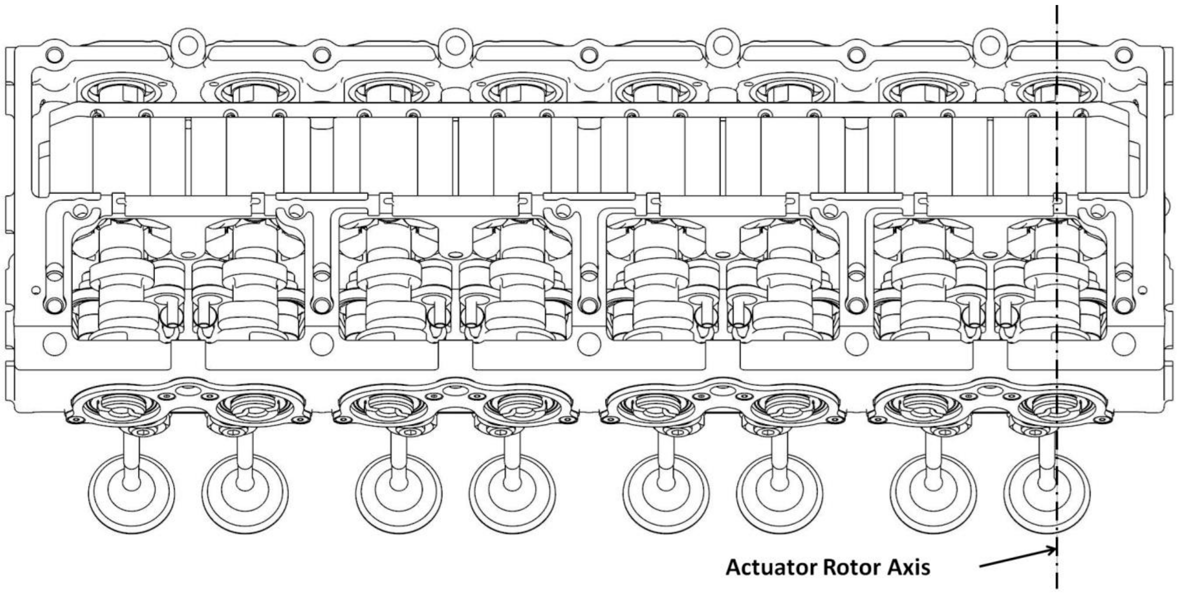

Fig. 2: Plan view of layout

2 Design

2.1 Layout

The base engine chosen by Jaguar Land Rover for the reported phase of the programme was the new, 4 cylinder, 2 litre, Ingenium, petrol engine featuring a bore of 83mm and cylinder spacing of 93mm. The critical dimensions of the actuator sizing and spacing are affected by the cylinder spacing in particular.

The individual actuator and valve train assemblies, which each incorporate a through shaft, or rotor, supported on rolling element bearings and including the cams and the rotating part of the actuator itself, are arranged with their axes perpendicular to the engine crank axis, as shown in Figure 2. A section through a rotor showing how the shaft is supported on a needle roller at the front and a ball bearing providing axial location at the rear is shown in Figure 5. The complementary cam profiles operate through a single rocker assembly using a roller follower for each cam. The rocker output is then connected via a spherical bearing to a drop link which, via a second spherical bearing, connects to the middle of a two part finger follower. This finger follower forms a pincer arrangement, the rear supported between a fixed abutment and an HLA whilst the jaws clamp around the valve retainer and collet assembly, allowing positive control of the valve motion in both directions. The drop link itself is a pre-loaded spring device arranged such that the spring is not compressed by the actuator except when the valve is on the seat and then the spring provides the seating load. It is the drop link and the (one time only) adjustable abutment at the rear of the finger follower that allow tolerance and expansion compensation.

All the actuators are built in a common housing and these modules incorporate the complete valve actuator mechanisms for an entire row of valves. In the case of the 16 valve, 4 cylinder, test engine this means all 8 inlet valve mechanisms are combined into a single module. The electronic valve control unit is mounted directly above the mechanical module as this keeps the phase wire connections short, minimising losses and keeping the high frequency signal cables between the control boards and the gate drivers short. The arrangement is shown in Figure 3.

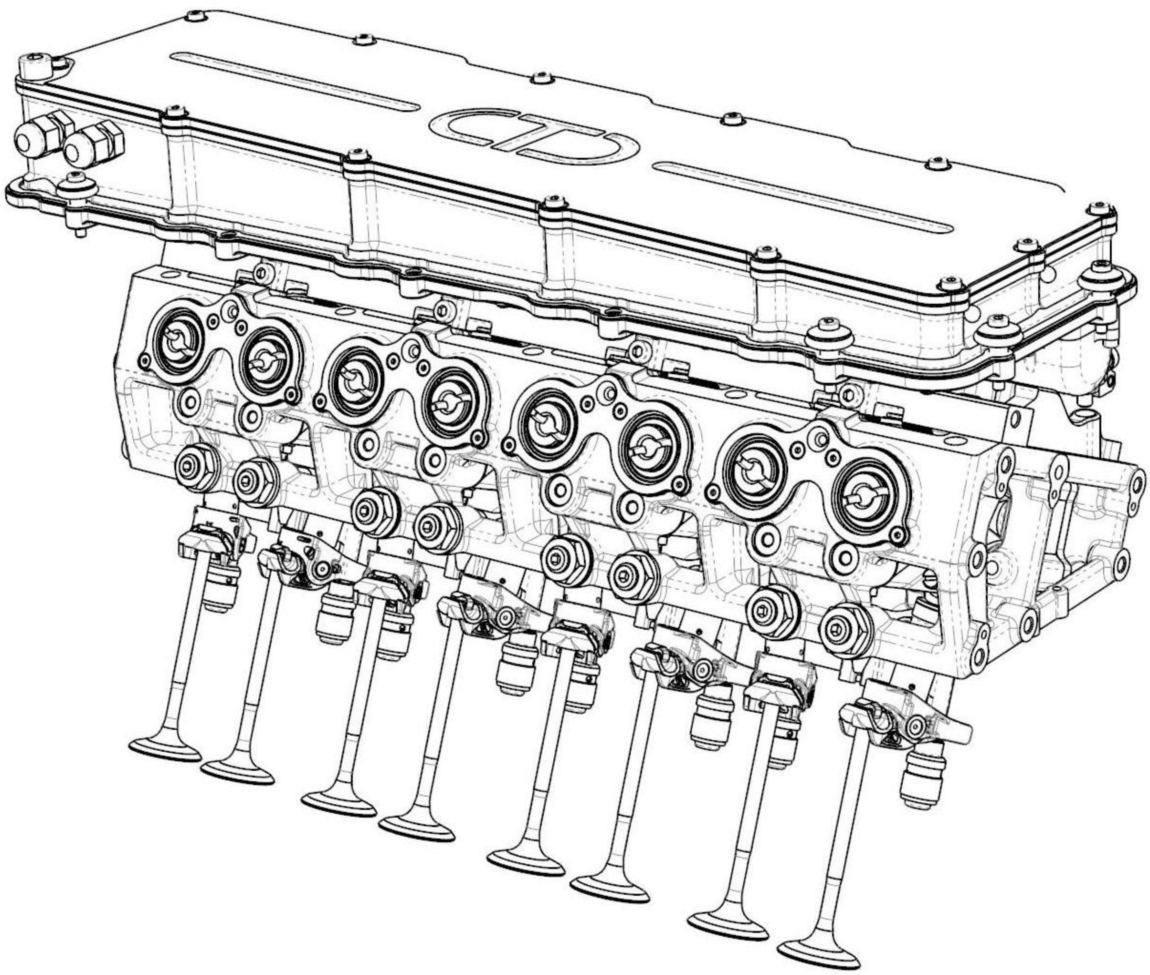

Fig. 3: Isometric view of assembly

2.2 Detail Design and Analysis

Extensive use has been made of modelling and analytical techniques. Electrical machine sizing and detailing was aided by iterative use of MagNet field simulation software plus both Abaqus FE based dynamic models and full system Simulink models to include the control loops. This was based on an initial functional specification that was based on experience with general valve train design plus 1-D modelling. The latter was especially interesting in that it revealed that, given the appropriate valve timings, performance at rated speed could be maintained at constant boost whilst using significantly lower valve lifts than is traditional. This permitted the achievement of a non-interference combustion chamber at standard compression ratio with clearance between the piston and the valve even at TDC/full lift. Lower lift also results in an additional benefit with regard to electrical power consumption.

The cam design is unusual for a number of reasons, the first is that, because there is no longer a mechanical link to the crank and the cam velocity is no longer half the crank velocity, the cam half period can be much longer than is usual which increases instantaneous radii all round the profile. The other is that, because there are no valve spring loads, zero speed cam nose Hertz stress is negligible. These factors allow the base circle to be defined by considerations of shaft bending stiffness and the avoidance of hollow flanks – very different from conventional cams! The absence of valve spring loads also allows the use of very narrow cams.

Non-linear finite element techniques have been used extensively to optimise the system dynamics, to establish the loads within the mechanism and, using those loads, to optimise the sizing of bearings, the stresses throughout the mechanism and the inertia/stiffness characteristics of moving parts. The model was extensively correlated with experimental data both in the form of measured displacements and high speed filming, which proved invaluable. Gas forces on the valves, which could not be simulated on the test rigs, were included in the model long before the first engine ran. There were several examples where the dynamic model allowed an understanding of non-intuitive behaviour to be achieved and counter-measures taken.

In order to achieve the required performance the IVA actuator has to be torque dense, highly efficient and demonstrate extreme dynamic capability. Once the actuator requirements had been derived the detail design of the actuator was completed.

The actuator is a permanent magnet machine using bread-loaf form magnets deployed in a 4-phase layout. An appropriate grade of neodymium magnets are used to give protection against de-magnetisation at high temperatures and high electromagnetic flux conditions.

In order to provide the high torque density and low power losses required for a successful implementation, a segmented stator design was necessary. This allows improved access for winding the coils onto the stator teeth, but package space between cylinders is short and it was necessary to adopt an unusual stator design featuring shared back iron between adjacent actuators. The stator arrangement is shown in Figure 4. Both analysis and experiment have established that the degree of crosstalk between actuators arising from this arrangement is not significant to the independent operation of the actuators.

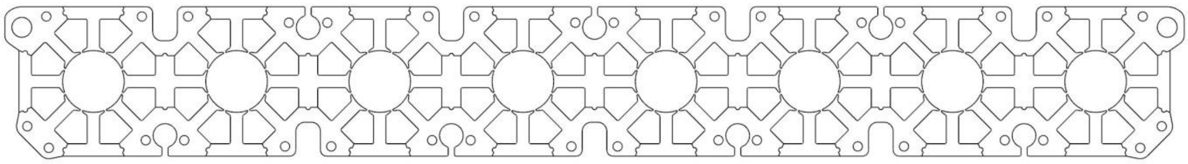

Fig. 4: Stator geometry

The single rocker assembly carries two roller followers one of which runs on each cam, thus trapping the rocker to follow the cam profile regardless of the direction of the inertia forces. The clearance between the rollers and the cam profiles needs to be adjusted during the build process but not thereafter. The rocker rotates about a fixed pivot and is connected to the drop link which then transmits the motion. It is necessary to adjust the backlash between the cam profiles and, on the current hardware, this is done by mounting one of the roller followers on an eccentric as can be seen in Figure 6. This arrangement will be revised in the next design level, it is a "one time only" adjustment completed during the module build process.

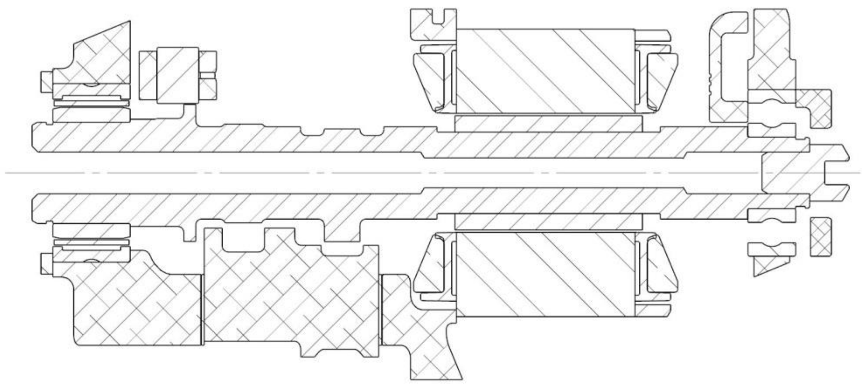

Fig. 5: Section through the mechanism on the rotor centre line

The drop link, shown in Figure 7, incorporates a pre-loaded spring arrangement which is designed both to provide the prescribed level of seating load when the valve is closed and to provide compliance so that the mechanism cannot “go solid” and jam for any reason including thermal expansion of the various components. The spring pre-load is set when the drop link is assembled, the seat load is set once the unit is assembled onto the cylinder head – this is achieved by adjusting the position of the fixed abutment at the pivot point of the finger follower in such a way that the deceleration cam effectively tries to “overclose” the valve and in so doing stretches the drop link, compressing the spring. Note that once the valve lifts from the seat the spring is “out of circuit” and imposes no load into the valve train.

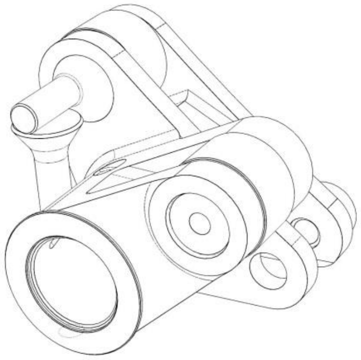

Fig. 6: Isometric view of current rocker assembly

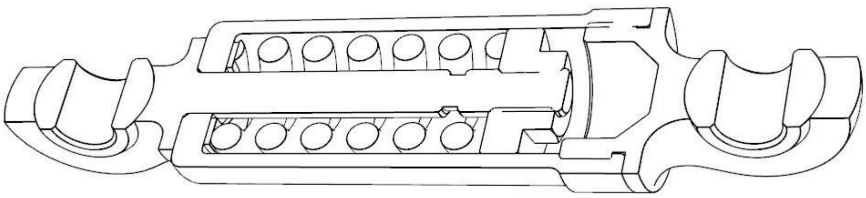

Fig. 7: Sectioned view of drop link

The two piece finger follower assembly is unique to the IVA system and provides control of the valve motion in both directions. A simplified schematic diagram of the operating principle is in Figure 8.

The finger follower is formed by the 'upper' and 'lower' parts, each of which has a part spherical contact surface of the same diameter towards the rear of the part. On nominal tolerances, these two surfaces form a contact 'sphere', which is retained from above by the assembly adjuster, and below by the HLA. The effective centre of this sphere is the pivot point about which the finger follower rotates.

Since the assembly adjuster position is fixed, the HLA takes up the clearance at the rear of the finger, clamping the two halves together, which closes the jaws until they are in contact with the valve and collet assembly. Clearly, the HLA will compensate for thermal expansion in this part of the mechanism.

At the other end of the finger follower, the top surface of the valve tip and lower surface of the collet holder combine to form a cylindrical contact surface of the same nominal diameter as the rear contact sphere. Thus, as the finger follower is rotated around the rear pivot point in response to a force from the drop link, the cylindrical surfaces of the valve and collet assembly slide over the flat surfaces on the jaws of the two finger follower halves, and the valve is moved along its axis.

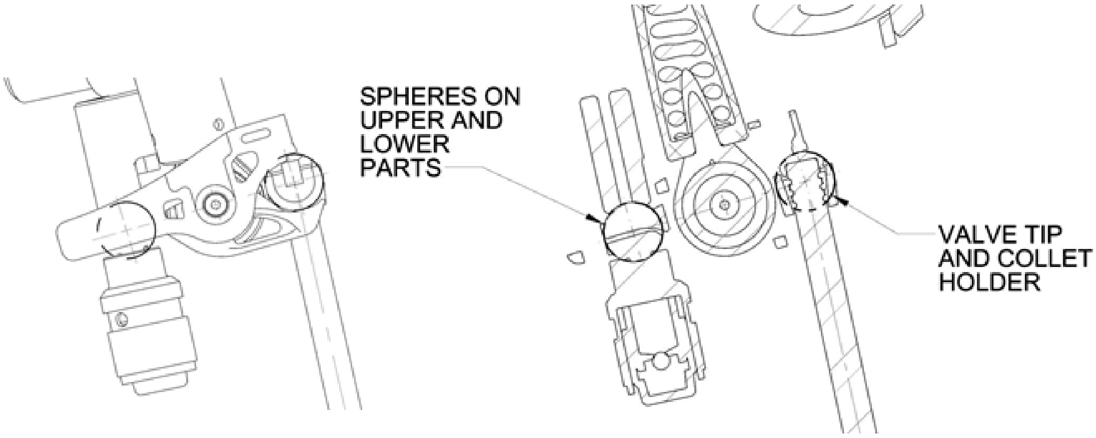

Fig. 8: Finger follower and valve retainer

2.3 Electronics and control

The electronic architecture employed for the prototype units consists of a local valve control unit for each actuator. A supervisory controller is also employed to manage and co-ordinate the required events for all the valves. The supervisor transmits event commands to and receives state reports from every valve individually. The valve control units then determine and control the target trajectories for both angular position and angular velocity of the actuator rotors. A diagram showing this architecture is shown in Figure 9.

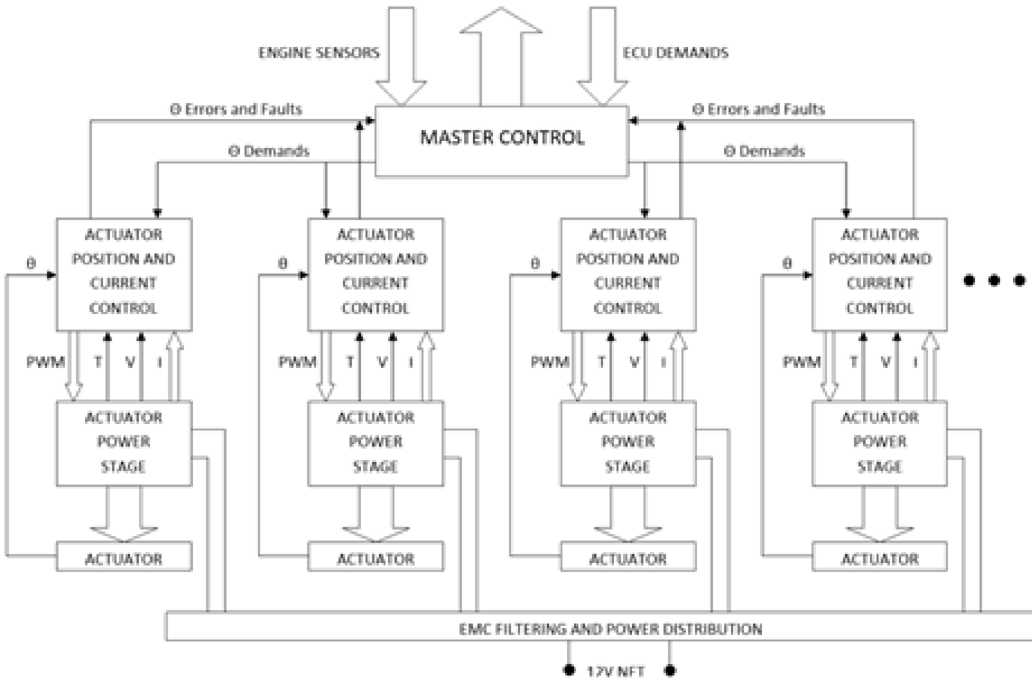

Fig. 9: Schematic layout of the electronics

Event profiles are constructed from key parameters as defined in Table 1: IVO, IVC, MOP and lift, more complex event types can be commanded with a second set of parameters and this results in an array of possible valve event profiles.

A profile generator function defines a real time rotor trajectory in terms of actuator and cam position, velocity and acceleration (torque). The plan is defined in such a way that allows the system to compensate for changes in engine state during each cycle. It also allows the commanded event to change from cycle to cycle. This capability is maintained even under conditions of very high rates of change of engine speed.

The actuator control architecture is an adaptation of a classical closed-loop servo motor control function. The main control loops are combined with information from a computationally efficient feed-forward model of the system running the planned cam trajectory. This minimises controller phase lag which in turn improves the precision of the valve behaviour and reduces electrical power consumption.

The development electronics control board contains a micro-processor for each valve and is connected to a gate-driver board containing the gate drivers which control a series of Full-H-bridges. As each actuator currently has 4 phase windings, a total of 4 independent H-Bridges are required per valve. Each H-Bridge includes a current measurement shunt for closed loop current control.

A separate power board contains the power MOSFETS for driving individual actuator phases and these boards along with the main DC-link capacitors are thermally sunk onto the electronics housing to enable efficient cooling. The power stages have been designed to target minimal conduction and switching losses at typical operating conditions include protection against overcurrent, under voltage, over temperature as well as diagnostics.

There is no direct feedback of the valve position required: the rotor position is the control parameter used in the valve control feedback loop. Non contacting absolute rotary encoders determine the rotor position for each actuator. Indirect measurement of the valve position facilitates precise and consistent control of the valve events and timing.

Note that, for a production implementation, considerable rationalisation of the electronics architecture will be possible. For example, using two processors per valve gave considerable flexibility in the exploratory stages of the programme but larger, more capable processors would be used for production and the number of valves controlled per processor would be a cost based decision. It is also planned to radically simplify the power stage architecture to reduce cost.

2.4 Base engine modifications

Note that the cylinder head design has been modified only as required to mount the IVA mechanism in place of the production arrangement. The combustion chamber, ports (and piston crown) are unchanged, as is head height. The inlet valve is modified only in overall length, which is shorter because the valve spring is not required

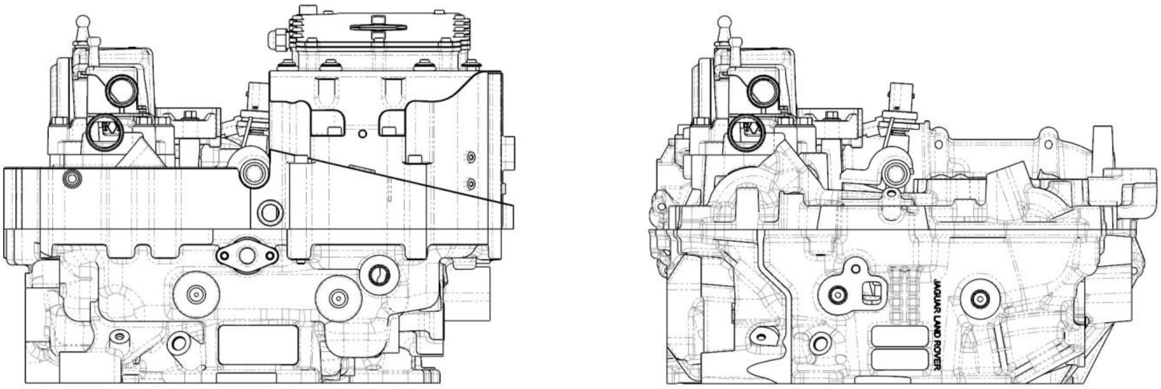

Fig. 10: . Front View of Engine Package; Inlet only IVA left, standard right

2.5 Package

The IVA cylinder head package for the inlet only design is shown in Figure 10. The valve control electronics slope upwards towards the rear on this North-South installation - but this is to accommodate large diameter, off-the-shelf connectors. Later iterations will be more compact. In the case of the forthcoming 16 valve arrangement it will be necessary to re-package the crankcase breather separator but this can be easily re-located into the space previously occupied by the cam phasers. This is possible because the entire timing drive system can, of course, be deleted providing an overall reduction in engine length at the crank of approximately 30mm. The minimum clearance between the IVA arrangement and the bonnet in the donor vehicle, a Jaguar XE, is approximately 20mm and the under-bonnet installation is shown in Figure 11.

Fig. 11: Photograph of IVA engine installed in Jaguar XE

The next level of hardware – both mechanical and electronic will be significantly simpler and smaller and it should be borne in mind that there is considerable flexibility in the electronics packaging – some of the electronics can be moved from one row of valves to the other for east-west installations for instance.

3 Operating modes

There are a number of different operating modes for IVA depending on the event targeted. At the simplest level the actuator will always park on the cam base circle but this is not always the case.

For a basic full lift event an angular trajectory is planned for the IVA rotor such that, at the target valve opening crank position, the rotor arrives not only at the angular position corresponding to the start of lift but also does so at the required angular velocity. A specific angular velocity profile is then followed until the crank reaches the position corresponding to valve closing. Once again, the rotor is synchronised such that both its angular position and velocity are as required at that point. The rotor can then complete its rotation and park ready for the next event. Clearly, the average angular velocity of the rotor as the lift profile is traversed defines the valve period in crank degrees. Note also that any changes in crank speed are monitored and compensated for in the trajectory plan in real time.

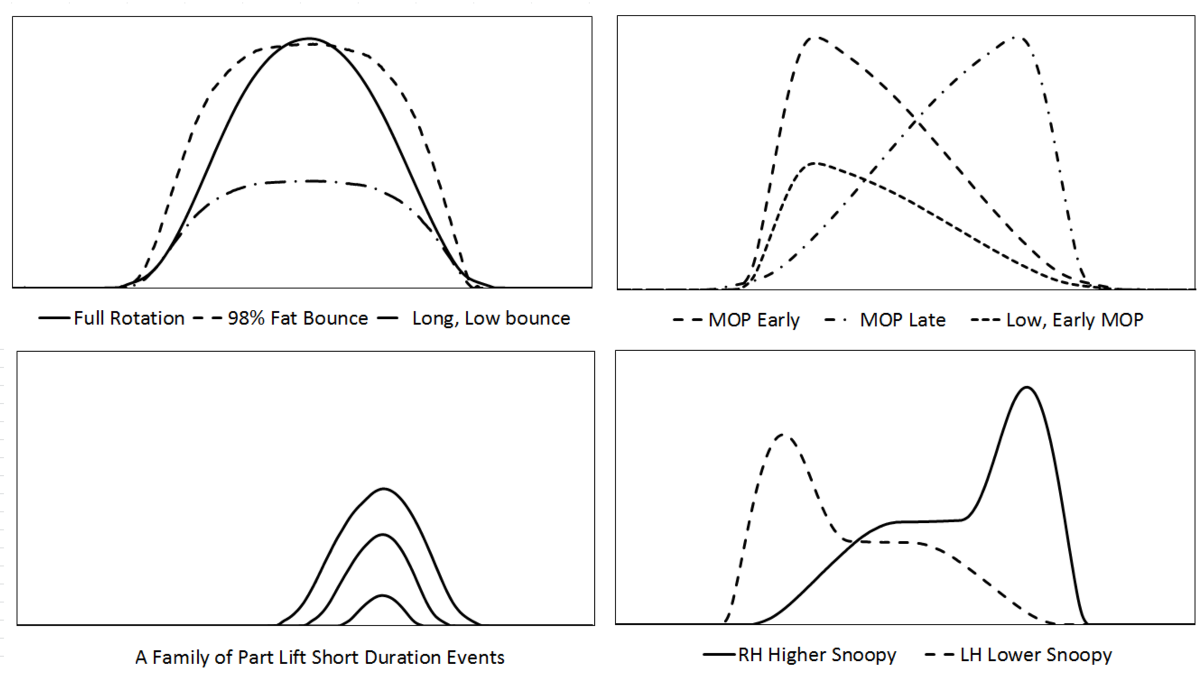

Fig. 12: IVA valve lift event shapes available at present

Part lift or “bounce” events are achieved rather differently, the rotor starts from a parked position and accelerates through a planned angular trajectory initially as for the full rotation event. However, the trajectory includes a reversal of rotation direction occurring at the planned lift before the rotor is returned to its original parking position.

Complex event shapes can be attained by adjusting the rotor velocity profile during the valve lift period. Examples of the shapes possible are shown in Figure 12. Because the synchronisation of the rotor motion relative to the crankshaft is purely electronic multiple events during one cycle can be demanded, missed events for single valve operation or cylinder de-activation are easily achieved. It is also possible to go from a low, short period bounce event to a full lift event from one cycle to the next giving a transient capability far faster than current fuelling systems can respond. It is this cycle to cycle control that allows transient torque control by trapped mass rather than ignition retard to be a realistic proposition – and a significant CO2 opportunity in its own right.

4 Development

4.1 Electrical Power Consumption

Minimising IVA electrical power consumption was a significant target for early development activities. Several factors come into play: the high acceleration rates required, actuator performance, resistive losses, friction, system mass and the event planning used for the control system. Simulation was employed as the mechanical system was designed in order to reduce the mechanical work implied by the valve event itself. During physical testing the mechanical system is essentially fixed and the focus was moved to the control system including the target profile generation algorithms. IVA events are inherently highly dynamic and a focus on minimising accelerations whilst achieving the desired event was key. Of course accuracy (and indeed repeatability) of events relative to the target event is also critical – both for engine function and for the minimisation of power consumption.

Measures have been developed for event quality and which reflect the accuracy of the start and finish timing plus the time-area integral so that power consumption improvements can be monitored against a common standard for accuracy and repeatability. Figure 13 shows a statistical analysis for 300 consecutive events.

The power demand required by the IVA system is a critical parameter and significant headway has been made with each generation of hardware and software. Figure 14 shows plots of sample measured power consumptions for IVA taken from the dynamometer testing. This is shown both as electrical demand and the approximate equivalent crank power to drive the alternator assuming 50% alternator efficiency. The data for full load testing was all completed using both inlet valves using a 98% lift bounce event. This was done because the shape of such an event offers a larger time area integral than a full rotation event which has a substantially lower electrical power demand. The curve shape can be seen in Figure 13. However, the next generation software will allow a similar lift profile to be achieved whilst still using a full rotation event. This will result is a significant reduction in power demand at higher speeds. Furthermore, the next generation of actuator has been considerably improved and the power demand will be reduced by between 40% (shown) and 60%. For comparison purposes, lines of constant FMEP at 0.05 and 0.2 bar are also shown to represent a state of the art and a more typical valve train parasitic loss – this shows that, even today, the full alternator loss is equal to or less than the best conventional valve train losses up to 3000 RPM. The next generation will maintain this to over 4000 RPM.

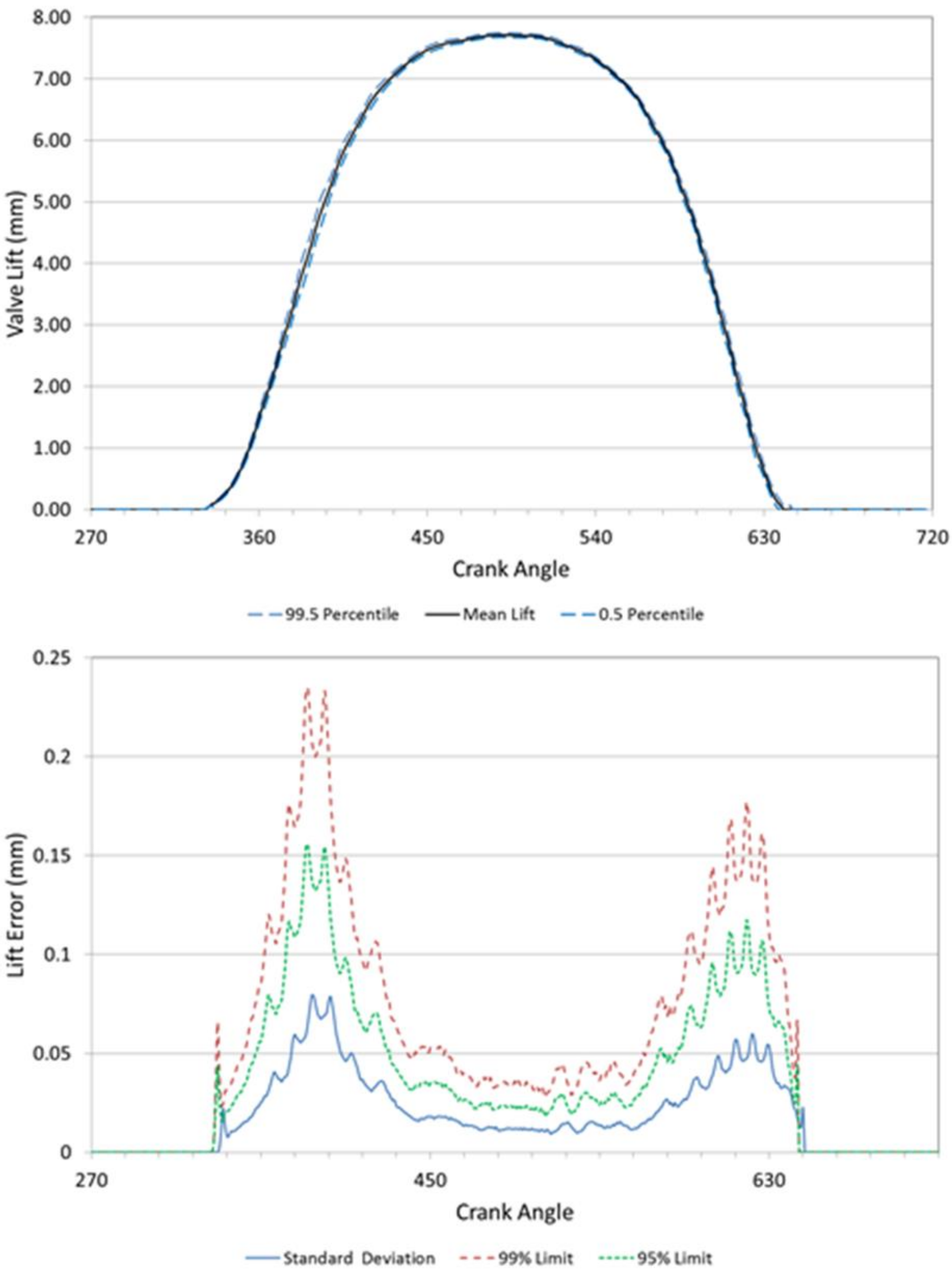

Fig. 13: Statistical Analysis of 300 Consecutive "High Bounce" Events (from fired engine testing)

The part load power demand is reflected in the right hand part of Figure 14 and, once again we can see that the current results – even if we allow for alternator efficiency are comparable with a "best in class" conventional arrangement. The base engine was equipped with a very advanced variable system which does not achieve best in class FMEP in this respect. "Smart charging" opportunities may allow the results to be exploited to greater effect than is immediately obvious.

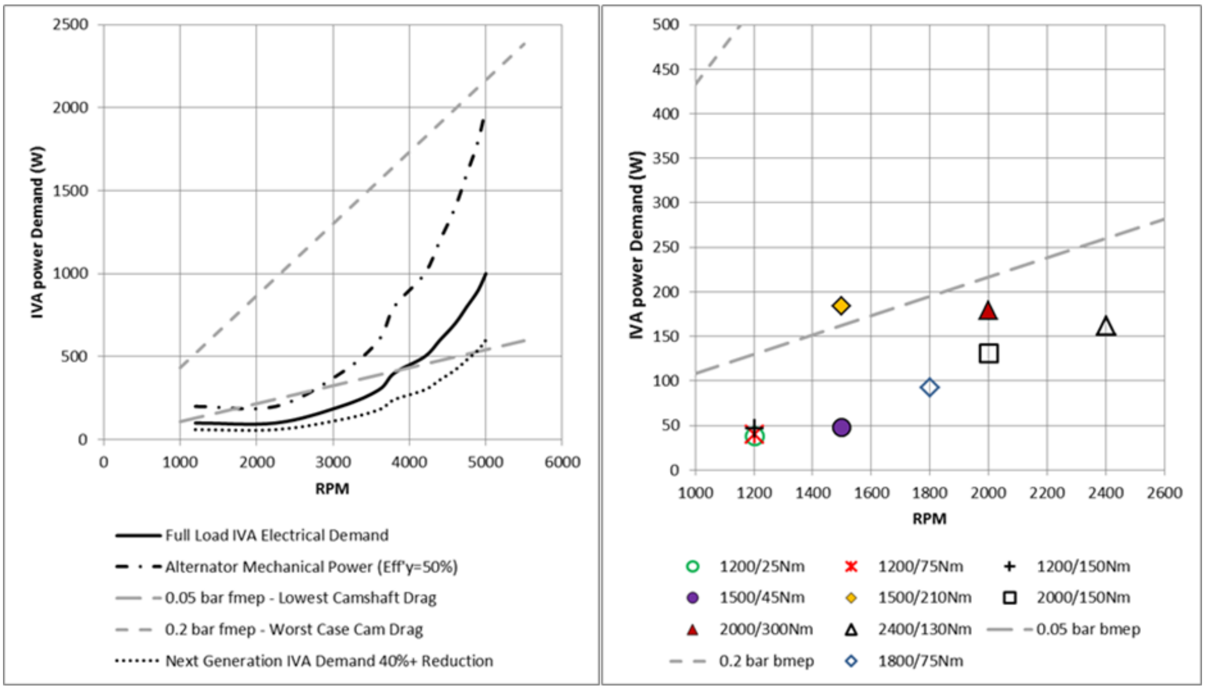

Fig. 14: IVA Power Demand at Full Engine Load and at the Minimap Points

4.2 Rig test

To develop the electrical machine a motoring dynamometer rig has been employed. This has enabled the refinement of the actuator design itself, steady state characterisation and control development. The rig has been used for a mechanical characterisation, establishing the angle by angle torque required to turn the mechanism. This detailed understanding of frictional loads helps to drive the development of low frictional elements and the calibration of the simulation models. The rig is able to simulate the thermal environment in which the IVA system operates.

As the IVA actuator differs from a standard motor in that it does not rotate at constant velocity, but each actuation being highly dynamic a 'commutation rig' was developed. The commutation rig contains the actuator and a large inertia. This allows the development of the best commutation strategy for a desired level of acceleration. The focus here is on both power consumption and dynamic response.

Multi-purpose rigs have been developed which allow for mechanical and control software development. These rigs allow the behaviour of the actuator in a hot environment to be understood and developed.

One point that became apparent as soon as rig testing began was the astonishingly low noise levels associated with IVA operation. Whilst no specific anechoic chamber work has yet been undertaken the subjective results in comparison with a conventional cam rig are dramatic. Whereas the conventional camshaft rig operating a single valve requires ear defenders at all times, it is perfectly possible to have a mobile phone conversation whilst standing right next to the IVA rig with only a perspex splashguard cover at the equivalent of an engine speed of 3000 RPM and with all 8 valves running.

4.3 Durability testing

The high level of analysis and simulation of the system gave confidence that the IVA system would have a high level of durability. In order to gain further confidence before starting fired engine testing, a 500 hour valve train durability test was performed. This test cycle had to be adapted from the standard Jaguar Land Rover valve train test cycle as IVA runs a range of events from slow and fast partial lift events to slow and fast full lift events. This testing showed that, with a few very minor improvements, the system had sufficient durability to move to fired engine testing with confidence and over 1000 hours have been completed on the fired engine with remarkably few IVA issues.

5 Performance and economy development

The Ingenium engine is an in-line 4-cylinder unit of 83mm bore and 92.3mm stroke. The combustion system is based on a 4-valve per cylinder architecture with direct fuel injection in a symmetric layout; CR is 10.5:1. The base valvetrain has variable intake and exhaust cam timing; the inlet valves feature an electro-hydraulic fully variable valve lift system. The boosting system has a twin scroll turbo-charger and the engine is designed for SULEV 30/EU6c/CN 6. Performance is rated at 184kW at 5500rpm with maximum torque of 365Nm between 1300 and 4500rpm. All test work was carried out using 98RON ULG.

5.1 Full load performance and knock sensitivity

At full load it was confirmed that the same full load characteristics, at the same levels of boost, were achieved over the engine speed range as the standard engine. Comparison curves are shown in Figure 15.

Optimising IVA under part load WOT conditions showed that improvements in efficiency were achievable by reducing the residual mass fraction and knock sensitivity in comparison to optimised base engine performance. An example at 1500 RPM, 13 bar BMEP is shown in Figure 16: at the same Knock Intensity BSFC was reduced by 2.4%

5.2 Part load performance

Given the degrees of freedom that IVA offers, a Design of Experiments approach was taken to model the BSFC response to IVA control parameters. The parameters for each intake valve were defined as shown in Figure 17 and defined in Table 1 which also shows the range of adjustment used in the DoE. Clearly, only a part of the range of options available from IVA has been included in order to reduce the size of the DoE.

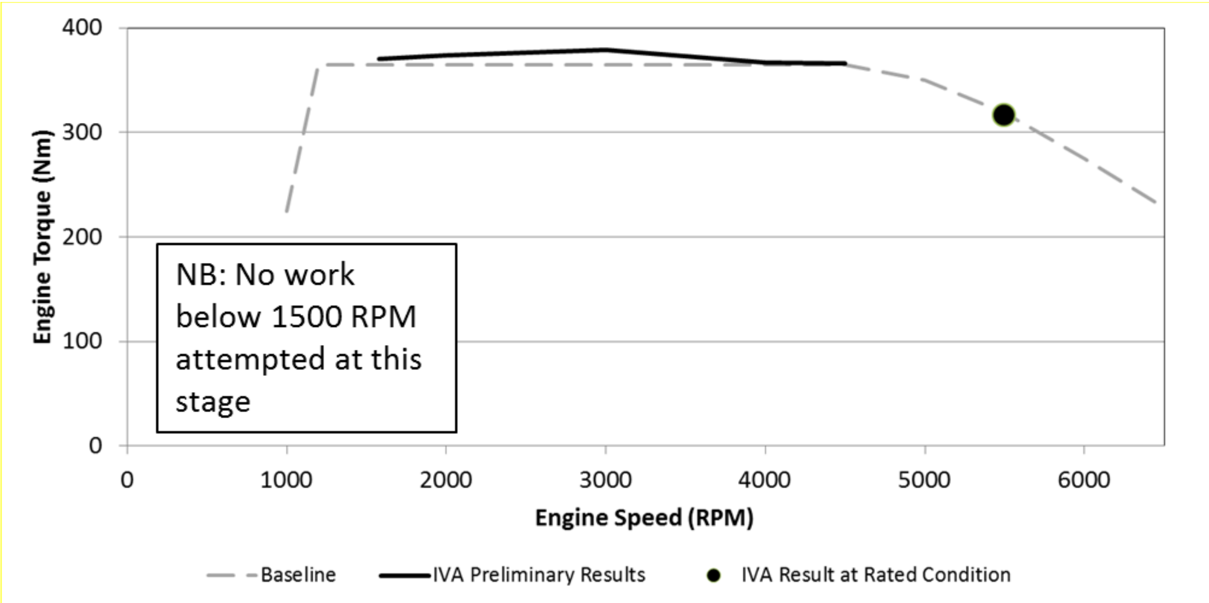

Fig. 15: Comparison of Full Load Performance over the engine speed range

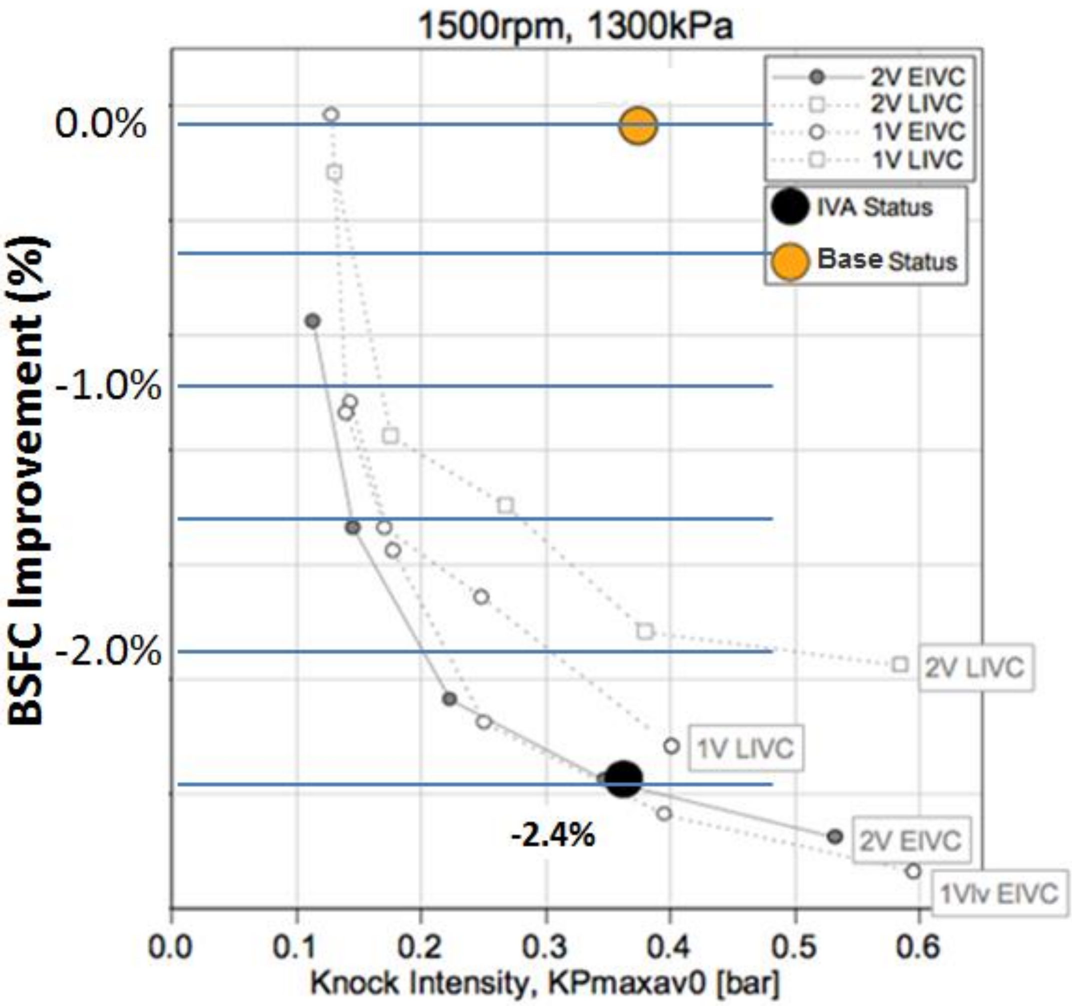

Fig. 16: BSFC improvements at comparable knock intensity

The DOE model was mathematically optimised for lowest BSFC and then those settings were confirmed by running that condition on the engine dynamometer. The validation tests consisted of a BMEP sweep at all sites below 50Nm, as small inaccuracies in the BMEP had a strong effect on BSFC.

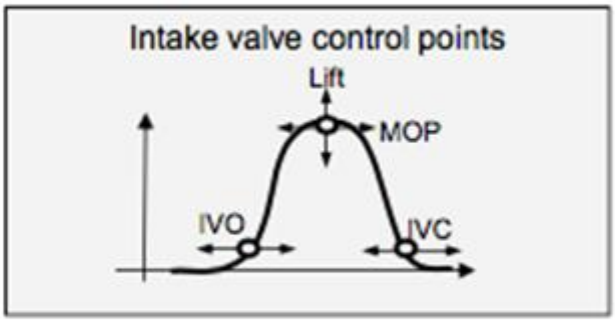

Fig. 17: IVA Control Parameters

IVA Parameter	Description	Minimum	Maximum
IVO	Inlet Valve Opening [deg ATDC]	-95	28
IVC	Inlet Valve Closing [deg crank]		
Period	Inlet valve period [deg crank]	68	236
Lift	Inlet valve Lift [% 0f 7.8 mm]	17	98
MOP	Inlet Valve Maximum Opening Point [%]	22	80
	Exhaust Cam Phasing [deg crank]	-25	-50
	Start of Injection [deg ATDC]	245	380

Tab. 1: IVA parameter definitions and DoE Variables

The engine was tested over a wide range of speed:load points. The IVA equipped engine demonstrated BSFC reductions in comparison with the base engine of between 0.8 and 7.5%. One of the test points was the internationally recognised World Wide Mapping Point at 1500 rpm and 2.62bar BMEP. The DOE experiment was carried out and the data reduced to show that single inlet valve operation had a significant advantage over 2-valve operation.

The DOE data for the single valve operation is presented in the following paragraphs. The single inlet valve DOE included some 120 test points covering the ranges shown in Table 1. The DOE model is compared with the test data in Figure 18, the correlation coefficient, R2, was 88%.

When comparing the IVA BSFC and emissions with the base engine there are three elements to consider, pumping losses, friction losses and combustion. For the purposes of brevity, we are only reporting detail on the 1500 RPM, 2.62 bar BMEP results but the underlying story is similar across all the points tested - in short: IVA offers benefit in all three areas

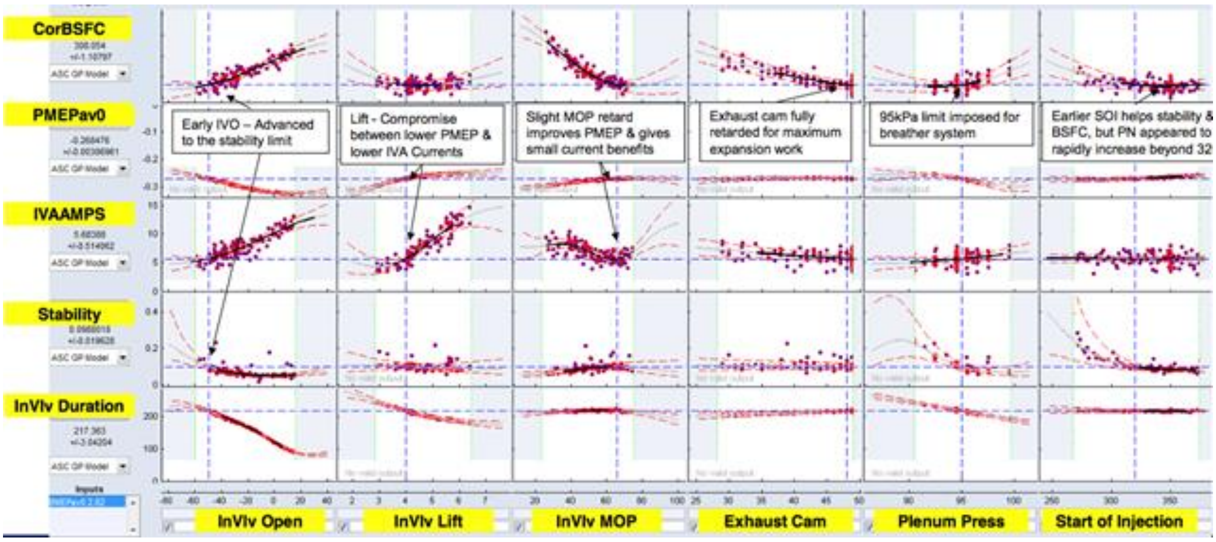

Fig. 18: 1500rpm/2.62bar DOE model

5.3 FMEP and PMEP vs BSFC

FMEP and PMEP are plotted in Figure 19. The data shows that IVA is able to improve pumping losses significantly. FMEP varied from -0.77 bar to a minimum of -0.60 bar and PMEP from 0.27 bar to 0.41 bar. Minimum BSFC coincided with low pumping losses and demonstrated a 5.3% reduction from base engine levels.

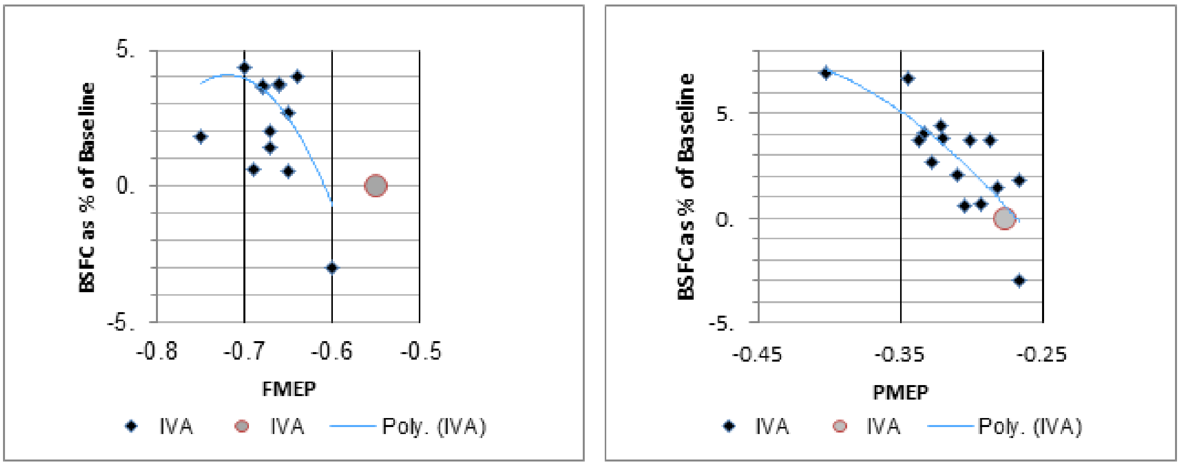

Fig. 19: Friction and Pumping Loss Characteristics

5.4 IVA calibration for minimum BSFC

It is clear from the above that IVA is able to control or affect many engine operating parameters from pumping and friction losses to in-cylinder air motion, gas exchange and ultimately combustion. The plots shown in Figure 20 demonstrate a result from this process where combustion and air motion parameters are compared with BSNOx exhaust emission and the result shows that minimum BSFC can be achieved with very low NOx levels and long burn periods. This operating condition also coincides with high levels of predicted swirl. These conditions can be reached with high levels of dilution through internal EGR. As Figure 21 shows IVO is early and allows for a large exhaust valve overlap, which facilitates the internal exhaust gas flow in to the cylinder. High dilution levels require high levels of TKE and air motion in order to reach stable combustion as demonstrated in this calibration, where STD of IMEP<0.1.

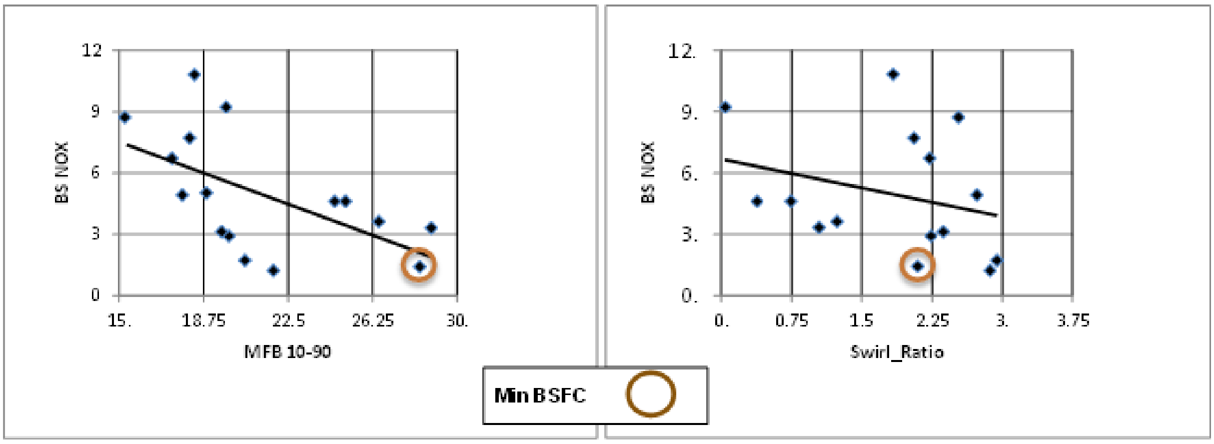

Fig. 20: Minimum BSFC and NOx emissions.

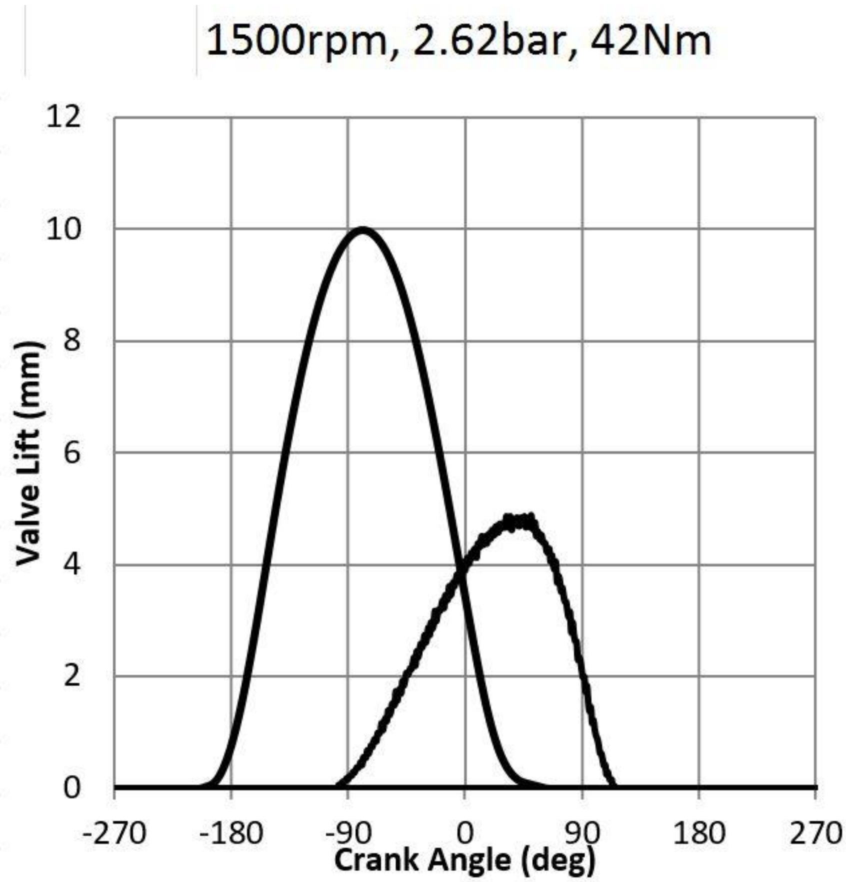

Fig. 21: Optimised IVA event for the results shown in Figure 18

5.5 IVA Part Load Results Summary

Figure 22 shows the optimised valve profiles for the mini map sites tested, and Table 2 shows the BSFC improvement from the baseline engine compared with the IVA result, it can be seen that the application of IVA has a positive effect and reduces engine BSFC at all sites tested.

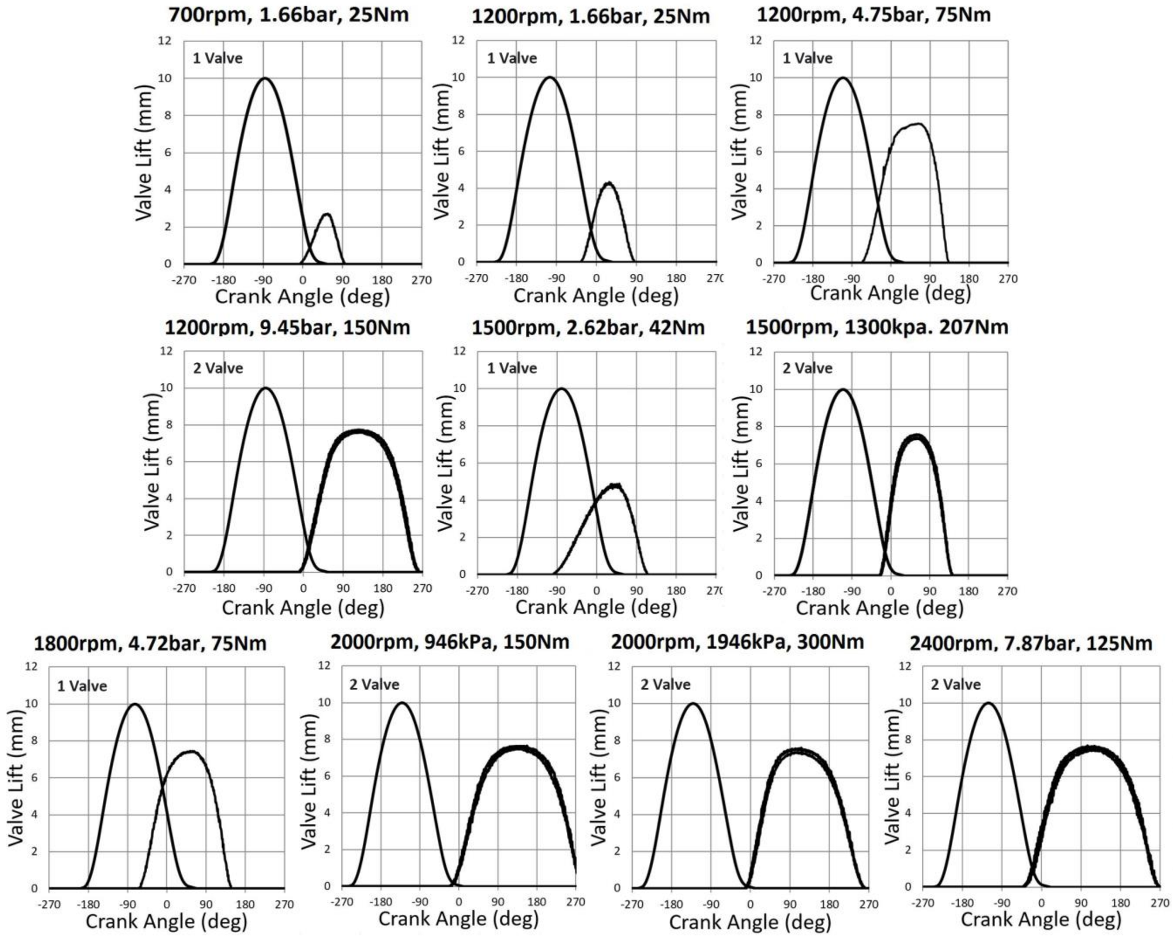

Fig. 22: Optimised IVA Valve Events for the minimap points

Speed Points	Torque Points, Nm	BMEP, Bar	BSFC Improvement
700	25	1.575	6.4%
1200	25	1.575	7.5%
1200	75	4.725	3.1%
1200	150	9.450	2.8%
1500	42	2.621	2.1%
1500	207	13.015	2.4%
1800	75	4.725	2.5%
2000	150	9.450	1.2%
2000	300	18.899	3.1%
2400	125	7.875	0.8%

Tab. 2: The detailed BSFC results for the minimap points selected

6 Value

The projected production cost of the hardware based on the next iteration design is well understood and has been generated with the aid of both JLR and a Tier 1 supplier. The “value” diagram shown in Figure 23 is based on the forecast volume production costs and the fuel consumption benefits based purely on the steady state results obtained to date using inlet valve IVA only on a base engine already equipped with the most advanced variable valve system currently available. We can be very confident that those fuel consumption advantages will increase with the next generation hardware, with the application of IVA to address transient operation and as combustion chambers and porting are modified to take advantage of the new opportunities presented. For example, it has been clearly demonstrated that a chamber optimised for tumble responds well to the use of swirl at low loads and low to medium speeds – could the use of differential inlet port and valve sizes enhance that effect at the lowest loads and speeds whilst also extending its usefulness over a greater proportion of the load:speed map?

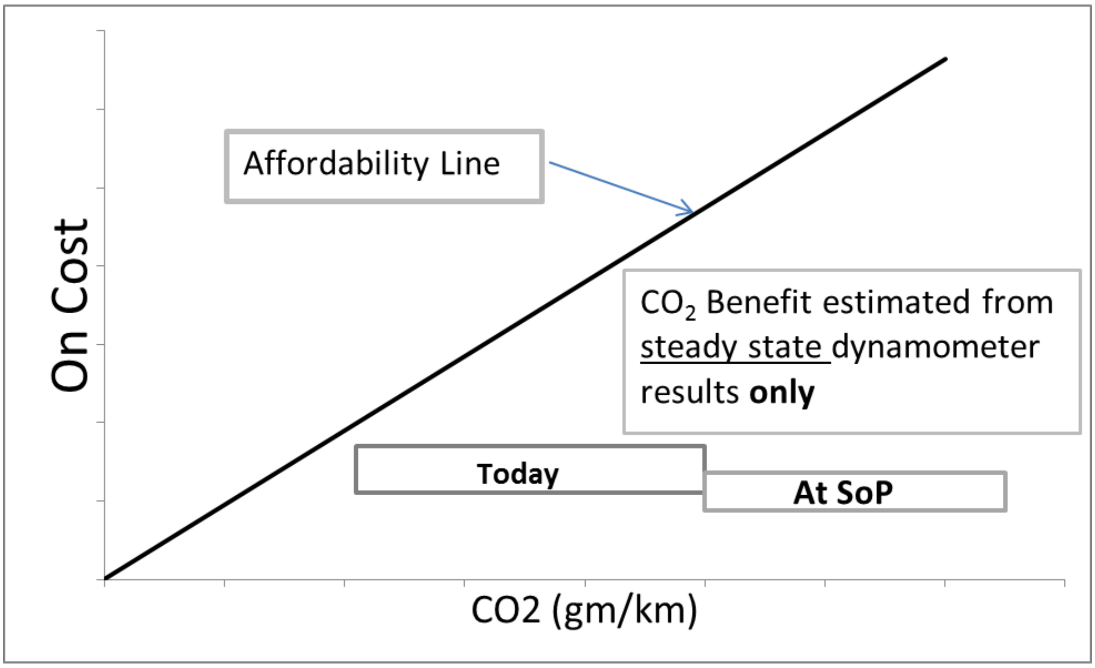

Fig. 23: Diagrammatic representation of CO2 cost:benefit status and forecast

7 Conclusions and Next Steps

The hardware reported in this paper represents the culmination of a phase of work begun in 2014 and has successfully demonstrated that the fundamental concept works – we can run a real engine over the full load speed range using IVA on the inlet valves. We have shown that the additional event control capability is genuinely valuable in terms of engine operation and we have built a car as a demonstrator. We have also learned a huge amount both about how to optimise the actuator design and the fact that, despite greater capability than any alternative system, the current hardware is not fast enough to allow us to reach the best fuel consumption results over a significant part of the speed-load range – the engine wants more! In fact, it is fair to say that, in order to extract maximum value from the IVA opportunity other engine systems including the combustion system will have to be improved or re-optimised. The work to date does not scratch the surface of the potential. The next level of hardware will provide more – the “Mark 4” actuators are currently being procured and will feature a minimum full rotation event duration of 5.5 ms rather than 7ms, rated speed will be 6750 rather than 6000 RPM, power demand for a given event will be 40-60% less than today’s hardware. Furthermore, the new design is significantly more compact than that used to produce the results reported in this paper – as can be seen in the comparison shown in Figure 24.

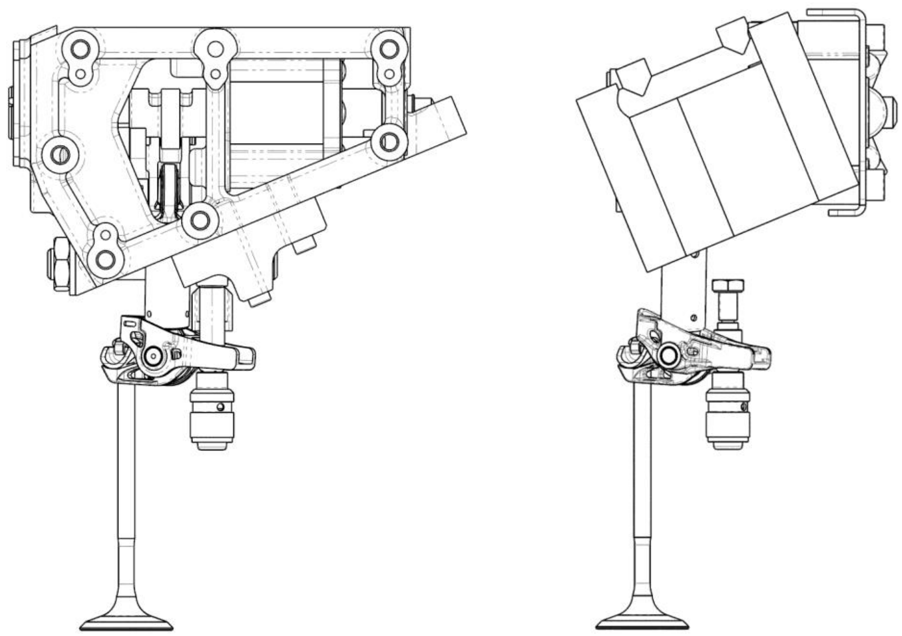

Fig. 24: Comparison of the current hardware package and the next iteration (on the right)

There will also be an exhaust valve capable version and a 16 valve engine on the dynamometer. The new hardware has been designed with cost and production feasibility very high on the list of objectives. Nevertheless, the opportunities for integrating IVA with other engine and vehicle systems should not be overlooked. Some of these are obvious, for example, in the case of inlet plus exhaust IVA redesign of the turbo/wastegate plumbing. However, it may well be possible to reduce cold cranking torque to the point where, for some stop-start architectures, a separate starter motor maybe deleted entirely. However, CO_2 emissions reduction is the main objective and we anticipate very significant further progress in this regard as we progress to the next hardware and software iterations.

8 Acknowledgements

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9 Nomenclature

- ATDC - After Top Dead Centre
- BTDC - Before Top Dead Centre
- BMEP - Brake Mean Effective Pressure
- BSNOx - Brake Specific NOx
- CA - Crank Angle
- CO - Carbon Monoxide
- CoV IMEP - Coefficient of Variation of IMEP
- IVA - Intelligent Valve Actuation
- DOE - Design of Experiments
- ECU - Engine Control Unit
- EIVC - Early Inlet Valve Closing
- FMEP -Friction Mean Effective Pressure
- IMEP - Indicated Mean Effective Pressure
- IVA – Intelligent Valve Actuation
- IVC - Intake Valve Closing Angle
- IVO - Intake Valve Opening Angle
- JLR - Jaguar Land Rover
- LIVC - Late Inlet Valve Closing
- MBT - Maximum Brake Torque
- MOP - Maximum valve Opening Point
- NMEP - Net Mean Effective Pressure
- PMEP - Pumping Mean Effective Presure
- TKE - Turbulent Kinetic Energy
- VCT - Variable Camshaft Timing
- VCU - Valve Control Unit
- WOT - Wide Open Throttle
- WWMP - World Wide Mapping Point