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THE INFLUENCE OF VARIABLE VALVE ACTUATION ON THE PART LOAD FUEL ECONOMY OF A MODERN LIGHT-DUTY DIESEL ENGINE

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ABSTRACT

An investigation has been carried out to identify how variable valve actuation (VVA) can be used to improve the part load fuel economy of a modern light-duty diesel engine. The base engine examined employed a variable geometry turbo-charger (VGT) with air to air inter-cooling, cooled exhaust gas re-circulation (EGR) and common rail fuel injection. The VVA system investigated was of the variable duration type, controlling primarily exhaust valve opening and intake valve closing events.

This paper describes how the fuel economy of the base engine was affected by the interactions of the VGT and EGR systems. It then presents simulation data that shows the methods by which fuel economy can be improved by the use of VVA, explains the phenomena that lead to this improvement and quantifies the extent to which it can be improved: Indicated and brake specific fuel consumption improvements of up to 6% and 19% respectively, were found during the course of this work.

INTRODUCTION

In Europe in recent years there has been substantial growth in the number of light passenger vehicles using high speed light-duty diesel engines. Annual sales of diesel vehicles have doubled in the last ten years and are projected to grow further with estimates of this vehicle sector ranging from 35% to 50% of the total market by 2005. Some countries such as Austria and France, where fuel price differentials and fiscal incentives are large, are already in the region of 50%. Also, the combination of good fuel economy and high levels of torque with improved refinement offered by the larger engines in this classification is leading to them being widely used in the larger car segment.

The increasing stringency of the emissions regulations applied to diesel engines is emphasizing the importance of after-treatment. [1] Catalysts and particulate traps are increasing exhaust back pressure and particulate trap regeneration requires short term enrichment to raise exhaust gas temperatures, both of which are eating into the fuel economy advantage that diesel engines have over conventional gasoline engines.

In addition the introduction of direct injection gasoline engines is further eroding the fuel economy advantage the diesel engine has.

Two practical methods of improving the fuel economy of the diesel engine are available to restore its advantage over gasoline engines:

1. Increasing the specific output of the engine, to allow downsizing, leading to reduced weight and friction.
2. Improving the cycle efficiency of the engine through reducing unwanted losses such as friction and pumping.

Work on increasing the low speed torque of diesel engines, by the application of VVA, has recently been reported [2], [3] and the types of VVA applicable to these engines is discussed in [3]

This work presented in this paper was based on simulating the application of a VVA system of the “variable angular velocity” type to a recently released, European, 4 cylinder light-duty diesel with state of the art systems. A discussion of the VVA system detail can be found in [3] and further data about system variants can be found in [4],[5],[6] and [7] The specification of the engine and its systems can be found in *Appendix 1*

Figure 1 shows the family of valve lift curves used for this piece of work. It can be seen from this that the VVA systems were applied to both intake and exhaust valves and primarily controlled exhaust valve opening, “EVO” and intake valve closing, “IVC.” *Table 1* shows the numerical values for opening and closing at ramp bottom and ramp top.

Besides the discussion of the benefits of VVA on full load torque, the paper by Lancefield *et al* [3] also presents some limited information on the part load effects of VVA. However, the engine modeled in that piece of work had a waste-gated turbine to control boost pressure and a throttle to help control EGR flow (at the expense of increased pumping work and fuel consumption.)

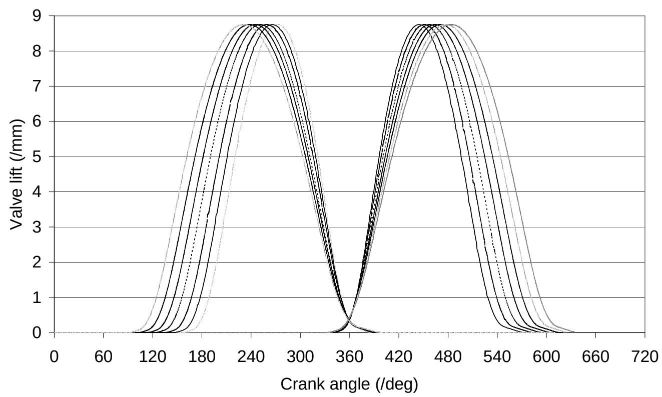


Figure 1 Family of valve lift curves used for the simulation runs. The standard profile is the middle of each group.

VVA setting number	Intake valve				Exhaust valve			
	Opening		Closing		Opening		Closing	
	RB	RT	RB	RT	RB	RT	RB	RT
1	338.1	352.7	577.3	545.6	154.2	170.3	390.9	364.4
2	336.2	351.3	586.7	556.6	143	159.5	392.5	365
3	334.4	350.5	597.2	567.2	131.6	149	394.3	365.7
4	333.6	349.8	608.8	578	120.3	137	395.2	366
5	332	349	619.5	588.9	110.2	127.5	399.1	366.2
6	329.7	348.3	630.6	599.5	99	116.3	400.9	366.5
7	329	347.8	640.9	610.6	88.7	105.3	402.7	366.7

Table 1 Summary of ramp top (RT) and ramp bottom (RB) valve timings generated by the VVA systems in this investigation. Standard is setting 4.

Current state of the art engines are using variable geometry turbines which can combine the functions of the waste-gate and throttle. Opening the nozzles reduces boost, much as a waste-gate does, but it also reduces the exhaust back pressure on the engine. So conversely closing the turbine nozzles can be used to manage the pressure difference across the engine to help control EGR. The work presented here is for an engine with a VGT and examines the effects of the VGT and EGR systems on fuel consumption.

SYSTEM MODELLING

The simulation model was implemented and evaluated using GT-Power[1] and was constructed to include all of the hardware systems that would be present in a real engine, including the control systems and strategies required to produce converged steady state operating conditions. (The evaluation of transient behavior is not part of this paper.)

MODEL SUB-SYSTEMS

EGR system

In part load operation light-duty diesel engines use large quantities (up to 60% by mass) of EGR to control NOx formation. To provide EGR in diesels requires a combination of a conventional, if somewhat larger than normal, EGR valve and, because the engine being investigated ran un-throttled, some means of managing the pressure differential across the engine. Until recently this pressure differential would have been controlled by a throttle, but with the increasing market presence of the VGT, techniques have been developed for using the VGT setting (which controls the inlet guide vane angle and nozzle open areas) to control the pressure differential.[8] For this exercise a closed loop controller on VGT setting position, responsive to desired EGR level, was implemented with a simulation run time input of EGR valve flow area.

The very high levels of EGR used by modern engines make the use of some kind of heat exchanger to lower its temperature very attractive. These are typically either an external shell and tube device or an in-head passage, both of which reject heat to the engine coolant. This feature was modelled by a separate heat exchanger system with specified outlet temperature.

Part load operating priorities and control strategies

Typical, modern, light-duty diesel engines do not need to operate at full load in the vehicle certification emissions drive cycles. As a consequence the emphasis in calibration is on part load fuel economy and emissions. Emissions targets must be met, whilst the fuel consumption of diesels is more than acceptable. Therefore the calibration typically favours emissions over fuel economy where there is a conflict. Thus, fuel injection timing is optimised for the NOx /PM trade-off and EGR is used to reduce the NOx still further. (Increasing EGR levels also affects the operating AFR of the engine which can influence the PM emissions.)

As can be seen from the above discussion a modern light-duty diesel engine has a number of control systems that can affect output, fuel consumption and emissions: Fuel injection timing (and rate shaping/multi-shot), EGR valve and VGT setting.

It is known [9] that current engines use a mass air flow method for calculating and controlling EGR levels. This is based on the assumption that for a given amount of fuel, without EGR the engine will consume a given amount of air at a given VGT setting. Therefore, any short-fall in measured mass air flow is displaced by EGR, giving a notional EGR flow. But both EGR valve opening and VGT setting can influence the amount of EGR provided. So, some form of hierarchy of controls is needed in order that one is subservient to the other. In practice it appears that for fine control the VGT setting is made dominant as this also allows control of boost pressure and thereby AFR, but in high rate transients the behaviour of the EGR control valve must take precedence, or large AFR excursions can result in un-wanted smoke. However, in this work, to produce converged steady state conditions only the EGR rate was controlled via the VGT setting on a time basis all other variables that would be controlled in a real engine were input as constants at simulation run time. This included the VVA system settings

[[1] GT-Power CAE software from Gamma Technologies Westmont Illinois]

Combustion

Initially, the use of the cylinder-condition-responsive Wolfer ignition delay correlation [10] and 3-term-Wiebe function [11] to describe the fuel burning rate were investigated. Unfortunately, due to the high levels of EGR being used these were found to significantly under-predict the time from start of injection to start of combustion (by 3° to 5° with 50% EGR) and not to provide sufficient flexibility to represent the shapes of the fuel burning rate curves adequately. At the other end of the combustion simulation spectrum, because of lack of detailed data it was also not possible to use the sophisticated three dimensional spray burning model present in GT-Power. As a consequence a limited amount of heat release data produced from measured cylinder pressure data was used. The heat release characteristics were held constant for all cases in each simulation run, so no account was taken of variation of ignition delay or fuel burning rate with changing trapped conditions.

Fuelling

One consequence of altering valve timing is that bsfc and bmep can be altered. Therefore, to investigate part load operation at a fixed bmep requires the adjustment of the fuelling to maintain the required bmep. This requires a controller on the fuelling. But, since the model already had adjustable VGT setting and EGR valve open area, each of which can influence bmep, it was considered that there might be multiple solutions for convergence, with bsfc variation among them. As a consequence it was decided that the fuelling amount should be fixed, but with the injection timing and start of combustion aligned with test data. Therefore, in this section of work, the effects of the VVA appear as changes in bmep and bsfc.

SIMULATION RESULTS

In order to fully explain the effects that VVA can have on the part load fuel economy of a diesel engine it is necessary to understand the underlying influences of the other systems that can affect it, in particular the VGT and EGR valve settings:

VGT EFFECTS ON OUTPUT AND FUEL ECONOMY

Initially a set of simulations was run at 2000 rpm, with 10mg/stroke fuelling and standard valve timings, but without EGR, to investigate the underlying trends associated with VGT setting. *Figure 2* (left) shows the effects of VGT setting on output and brake specific fuel consumption, with standard valve timing. It can be seen that maximum bmep and minimum bsfc are achieved with the VGT fully open. This is a consequence of minimised pumping work associated with reduced air flow and exhaust restriction, as demonstrated in *figure 2* (right) which shows the corresponding effects of VGT setting on air flow, AFR and pmep (bdc to bdc).

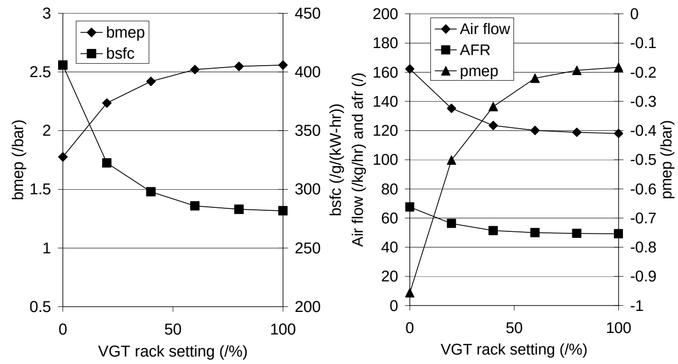


Figure 2 The effects of the VGT setting on bmep (left) and air flow, AFR and pmep (bdc to bdc) (right) with 10mg/stroke fuelling, no EGR and standard valve timings.

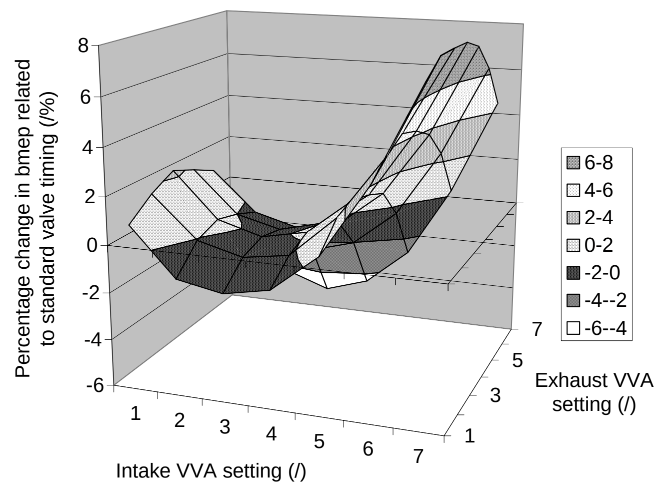


Figure 3 The effects of intake and exhaust VVA on bmep with fully closed VGT. (2000 rpm 10mg/stroke fuelling.)

INTERACTIONS BETWEEN VGT AND VVA

A conclusion from the above is that minimising air flow and reducing the restriction offered by the exhaust system are key to good part load fuel economy. To see how VVA could assist with this another set of simulations was run, again with zero EGR, fuelling at 10mg/stroke and several settings of the VGT, each exploring the full ranges of both intake and exhaust VVA. *Figure 3* shows the effect of VVA on bmep with the VGT fully closed. (For brevity the VVA settings are shown as indices, the reader is referred to *table 1* for further detail.) The results are relative to the output with standard valve timings. *Figure 4* shows the corresponding change in bsfc. It can be seen from *figures 3* and *4* that the greatest improvements in bmep and bsfc occur with the latest IVC (It is worth noting that advancing IVC also increases bmep and improves bsfc, but not as much as retarding, within this range of timings.) The data shows, as illustrated by *figure 3,* that exhaust VVA setting 3 is the other optimum. The improvements available with changing intake valve closing timing are larger than those with exhaust valve opening timing and improvements in bsfc of the order of 8% are available at this fuelling level with no EGR and the VGT fully closed.

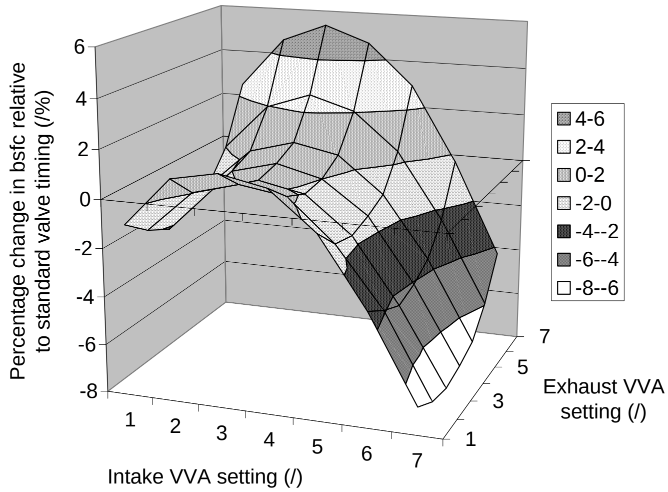


Figure 4 The effects of intake and exhaust VVA on bsfc with fully closed VGT. (2000 rpm 10mg/stroke fuelling.)

Simulations carried out at other, more open VGT settings, revealed that the optimum valve timings did not change. Within the ranges of the VVA system evaluated best fuel economy and corresponding bmep occurred at intake VVA setting 7 (retarded some 33º from standard) and exhaust VVA setting 3 (retarded some 11º from standard.)

It can be seen from *figure 5* that the benefits that can be generated by the use of VVA are strongly affected by the VGT: At 10mg/stroke fuelling, with closed VGT setting they can be as high as 8% improvement in bsfc, but at 60% open VGT and beyond they only amount to 1% or less. This underlying behaviour needs to be kept in mind for the next section.

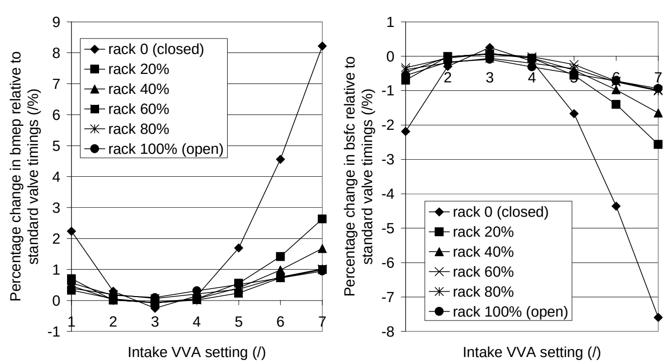


Figure 5 Effects of intake VVA setting on bmep (left) and bsfc (right) at a range of VGT settings. Exhaust VVA setting 3.

INTERACTIONS BETWEEN VGT AND EGR SYSTEMS

It might be thought, from the above, that minimising fuel consumption at part load operation is only a matter of fully opening the VGT to open the turbine inlet guide vanes to minimize boost and exhaust back pressures. However, as mentioned earlier, the VGT setting is used to control the pressure differential across the EGR valve to help manage the EGR flow. Therefore this approach to minimising fuel consumption may not be possible when using EGR. To investigate this a series of simulations runs, again at 2000 rpm, but with two levels of fuelling, 5mg and 10mg/stroke, each at three levels of EGR, 10%, 30% and 50%, were carried out. Cylinder pressure data, which was translated into heat release curves, was available for the 0%, 30% and 50% EGR runs, but as no data was available for the 10% EGR setting it was assumed that the combustion characteristics were the same as for the zero EGR case.

The EGR flow is governed, not just by the pressure difference across the engine, but also by the size of the EGR valve orifice. It was established, by simulation, that for 50% EGR it was necessary for the EGR valve to be fully open (effective orifice diameter 14mm) if the VGT setting were to be anything other than fully closed. But at the 30% and 10% settings a range of combinations of VGT and EGR valve settings could be used to provide the necessary levels of EGR. Therefore another “driver” was needed to govern the hierarchy of controls. In this case a major problem is the closing response time of the EGR valve when a positive load transient occurs, so to some extent there is a need to minimise the EGR valve opening. However, no data was available to define the EGR valve settings at 10% and 30% so for the 10% operating point an EGR valve setting that required a relatively closed VGT setting was selected (6mm diameter orifice) and for the 30% an EGR valve setting that allowed a relatively open VGT setting (12mm diameter orifice) was used.

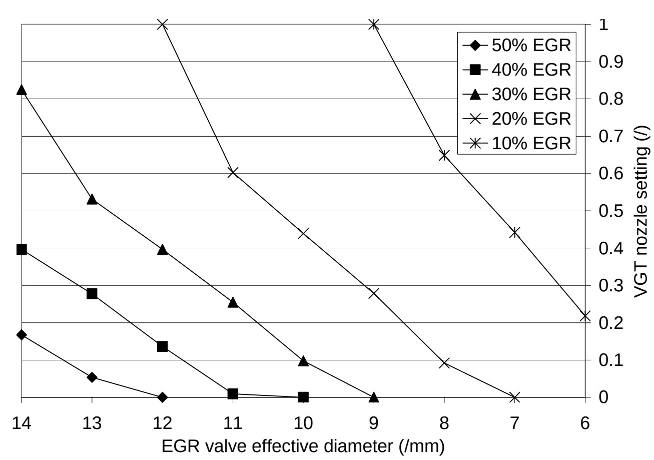


Figure 6 VGT settings required to generate specified EGR levels for a range of effective EGR valve diameters. Standard valve timings.

Figure 6 shows the range of authority that VGT setting has over EGR levels at a range of effective orifice diameters. It can be seen that for 50% EGR an effective diameter of greater than 12mm is required and for 30% EGR an effective diameter of greater than 9mm is required if the VGT setting is to be anything other than fully closed. It should be noted that at values lower than these insufficient EGR results. For the 10% EGR case it can be seen that sufficient EGR could be provided by effective diameters of less than 6 mm, but VGT settings of less than 0.2 would result. It can also be seen that for effective diameters of greater than 9mm too much EGR is introduced even with the VGT fully open.

Figure 7 shows the VGT settings resulting from the 6 combinations of fuelling and EGR level tested at the optimised operating conditions of exhaust VVA setting 3 and intake VVA setting 7. It may seem counter-intuitive that the higher fuelling rate cases require a more closed VGT setting. This is explained by the fact that the turbo-charger produces more boost, with higher fuelling, so a more closed VGT is needed to restore the necessary pressure differential across the engine and EGR valve.

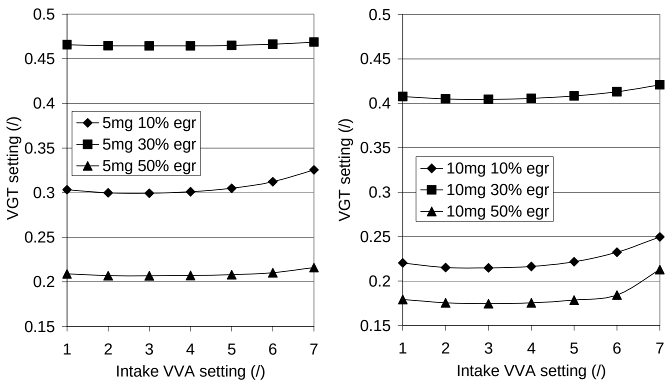


Figure 7 The VGT settings taken for 5mg/stroke (left) and 10mg/stroke (right), each at 3 EGR levels. Exhaust VVA setting 3.

DETAILED EFFECTS OF VVA

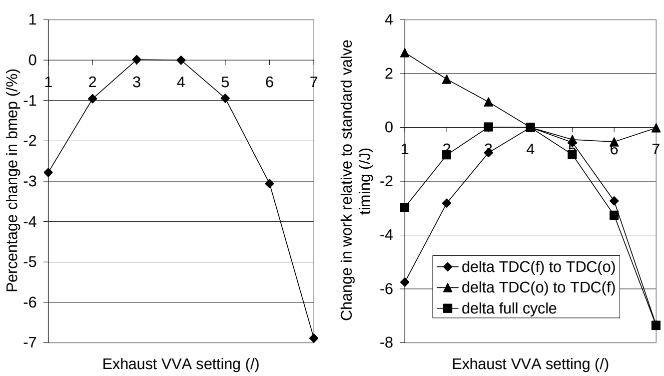


Figure 8 The effects of exhaust VVA on bmep (left) and the work done on the piston in various parts of the cycle (right) 10mg/stroke fuelling,10%EGR and standard intake valve timing.

For the 10mg/stroke fuelling and 10% EGR case, *figure 8,* left, shows the optimum exhaust setting to be 3 , but it is only marginally higher than 4, indicating that the optimum is likely to be between 3 and 4. However, as the numerically greater setting 3 was used for the remainder of the investigation. It can also be seen from *figure 8*, right, that both advancing and retarding EVO, relative to standard, reduces the amount of work done during the expansion and exhaust strokes, (TDC(f) to TDC(o)). Earlier EVO reduces expansion work more than it reduces exhaust pumping work and later EVO increases exhaust pumping work more than it increases expansion work. However, the effects of EVO on the work done in the intake and compression strokes, (TDC(o) to TDC(f)), through the influence of reduced mass flow, offer the ability to marginally improve the work output with EVO retarded by approximately 11º.

Examination of the behaviour of the system with exhaust VVA setting 3 and varying the intake valve timing over the full range shows that output and bsfc can both be improved. Consideration of the work distribution in the cycle and inspection of the changes that take place in the work in the various strokes highlights the reasons for this. (Note for *figures 9* through to *12* the basis for reference is the standard intake valve timing, setting 4, but with exhaust VVA setting 3.)

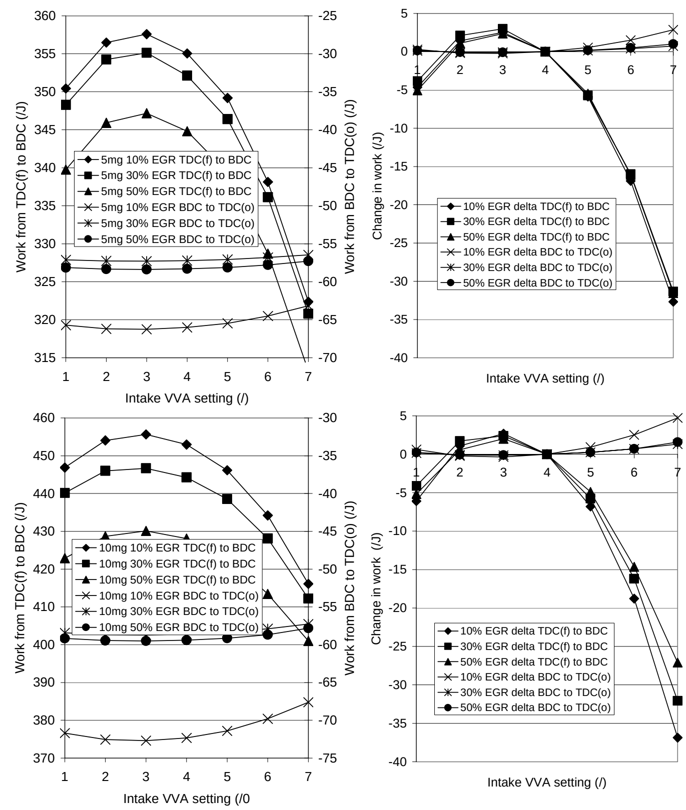
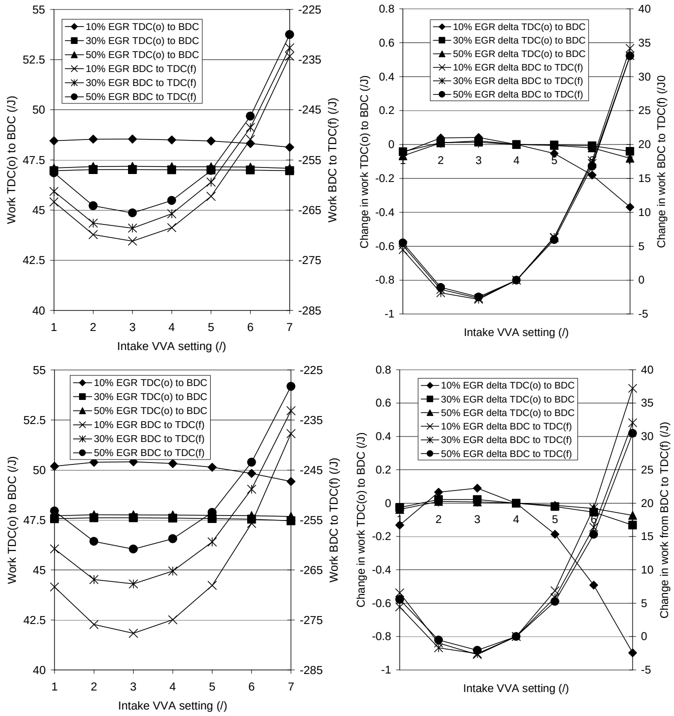


Figure 9 Work done on the piston between TDC(f) and BDC and BDC and TDC(o), (left) and changes in work relative to intake VVA setting 4 (std) (right) for 5mg/stroke (top) and 10mg/stroke (bottom.)

Figure 9 shows the work done from TDC(f) to BDC and BDC to TDC(o), in absolute terms (left) and as changes from the standard intake valve timing value, (right) for 5mg/stroke, (top) and 10mg/stroke, (bottom). By summing the work in these two strokes, it can be seen that for intake VVA settings 2 and 3 small improvements in work output can be achieved. These are of the order of 3 joules. (Note for this engine 50 joule change is equivalent to 1 bar mep.) For all other intake VVA settings the total output from these two strokes is worse than for standard intake timing as there is a more



significant reduction in expansion work.

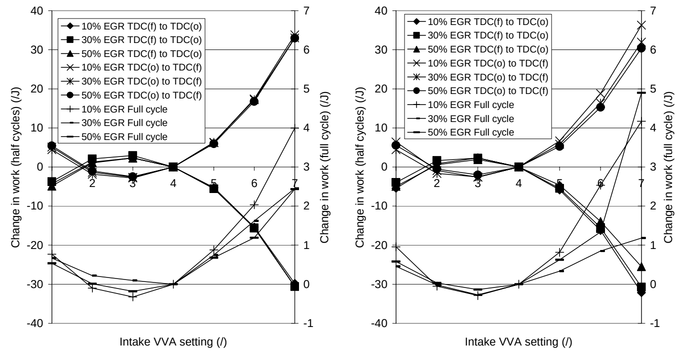
Figure 10 Work done on the piston between TDC(o) and BDC and BDC and TDC(f), (left) and changes in work relative to intake VVA setting 4 (std) (right) for 5mg/stroke (top) and 10mg/stroke (bottom.)

Figure 10, shows the data sets for the same operating conditions as in *figure 9*, but with work values for the induction and compression strokes. It can be seen that intake valve timing only has a small effect on the (+ve) work in the intake stroke, (the changes in nominal value are caused by changes in boost pressure.) However, it has a significant effect on the (-ve) work done in the compression stroke, being able to substantially reduce the work required to compress the trapped mass, by retarding IVC.

This reduction in compression work occurs because from BDC to IVC the pressure operative in the work calculation, ∫pdv, is largely that of the intake manifold rather than that of the compressed gas which it would be after IVC. Also retarded IVC reduces the trapped mass, which in turn further reduces this work integral. The above also indicates that early intake valve closing, (late IVC mirrored about BDC) should have the same effect, but the valve dynamics implications of this make such early IVC difficult to achieve with acceptable valve lift and a cam operated valve train.

From the above it can be seen that there are significant changes in expansion and compression work and small changes in induction work and exhaust pumping levels. *Figure 11* shows how these combine in the two half cycles and over the full cycle. It can be seen that the net results of the latest intake valve timings are small single digit changes in work output. However, these need to be considered in the context of the overall work output of the cycle.

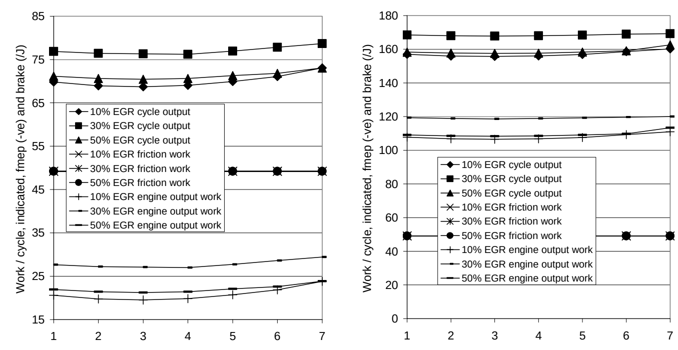
Figure 11 Changes in work done in the two half cycles from TDC(f) to TDC(o) and from TDC(o) to TDC(f) and



the full cycle, relative to intake VVA setting 4 (std) for 5mg/stroke (left) and 10mg/stroke (right) fuelling.

Figure 12 shows the full cycle work, the fmep converted to work/cycle and the resulting work available as output. In this context it can be seen that the changes in output from the intake valve timing variations are more significant, when considering brake output. In the 5mg case the change of approximately 5 joule per cycle represents approximately 25% of the resultant output, and in the 10mg/stroke case the change is of the order of 4% although the actual change in work is greater.

Figure 12 Indicated, friction and resulting output work for 5mg/stroke (left) and 10mg/stroke fuelling (right.)



(Please note that for *figures 13* and *14*, the basis for comparison is the operation of the engine with standard intake and exhaust valve timings, and so the data represent the overall benefits available from intake and exhaust VVA.)

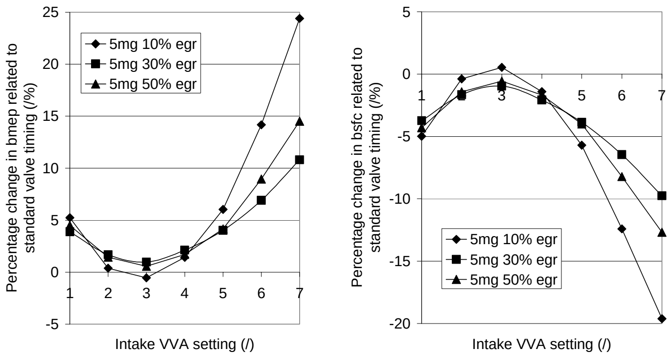


Figure 13 Changes in bmep (left) and bsfc (right) for changing intake VVA, with exhaust VVA setting 3 at 5mg/stroke fuelling and three levels of EGR.

It can be seen from *figures 13* and *14* that significant increases in bmep and reductions in bsfc are possible with optimised (11° retarded) exhaust and optimised (33° retarded) intake valve timings at the operating conditions investigated, with improvements in bsfc of up to 19% and in bmep of up to 24% at 5mg/stroke fuelling. At the higher fuelling rate of 10mg/stroke these reduce to 5.5% and 4.5% respectively. However, care needs to be taken in the interpretation of bsfc values at very light loads, where a significant proportion of the fuel is used simply to overcome friction as small changes in output as calculated here with fixed fuelling can produce large changes in bsfc.

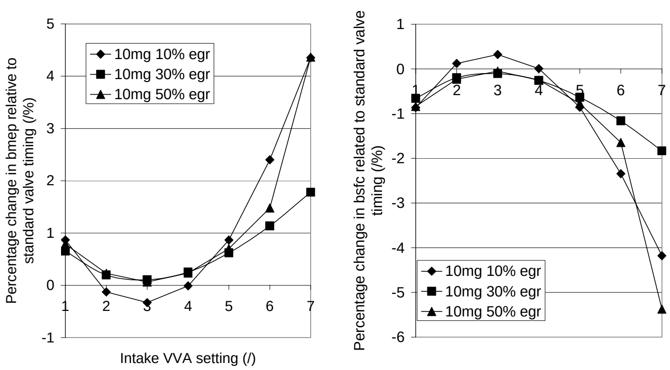


Figure 14 Changes in bmep (left) and bsfc (right) for changing intake VVA, with exhaust VVA setting 3 at 10mg/stroke fuelling and three levels of EGR.

As was mentioned earlier, because of a lack of data about the engine calibration, the EGR valve openings for the 10% and 30% EGR settings were somewhat arbitrarily chosen, with a valve opening necessitating a relatively closed VGT setting for 10% EGR and one allowing a relatively open VGT setting for 30% EGR. From *figures 13* and *14* in combination with *figure 7*, showing VGT settings, it can be seen that the more closed the VGT setting, the greater the improvement in bmep and bsfc offered by the use of VVA.

There are two factors causing this:

1. The more closed the VGT setting the greater the exhaust pumping work, *figure 9*, with attendant higher bsfc.

2. The more closed the VGT setting the greater the boost pressure and the greater the benefits of late intake valve closing in reducing compression work.

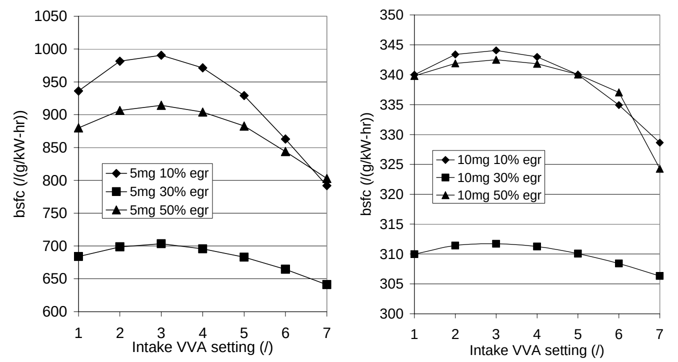


Figure 15 bsfc values for 5mg/stroke (left) and 10mg/stroke fuelling (right) at 2000 rpm and three EGR rates.

The results presented above deal with changes in bsfc resulting from changes in engine brake output caused by the application of VVA, with fixed fuelling. With very light engine load the proportion of the fuel consumed to overcome friction is high and consequently small improvements in brake output can result in large improvements in bsfc. An alternative view of these results can be had through isfc, where the effects of friction are included. *Figure 16* shows the changes in isfc that correspond to the changes in bsfc in *figures 13* and *14*. Despite the isfc improvements being smaller, they are still significant if they can be turned into real world fuel economy.

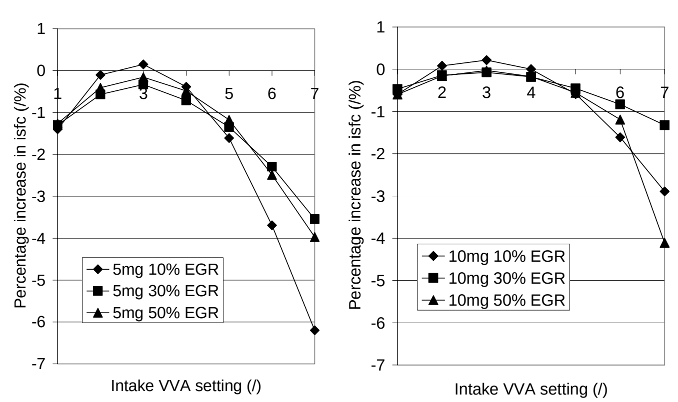


Figure 16 Percentage change in isfc values with optimized exhaust VVA and varying intake VVA, for 5mg/stroke (left) and 10mg/stroke fuelling (right) at 2000 rpm and three EGR rates.

Finally, a short discussion of the implications of calibration trade-offs to the part load fuel economy benefits of VVA: from the above, the fuel economy penalties of running with high levels of EGR, a smaller than necessary EGR valve orifice or small VGT nozzle openings are apparent. Effectively, the use of EGR is mandatory as it is the most cost effective method of controlling NOx, but the amount of EGR required is largely dictated by the engine characteristic and the applicable legislation. However, the combination of EGR valve and VGT settings used to implement it are a matter of calibration.

At present, typically it is the EGR valve system that is the slowest to respond to changing operating conditions on a diesel. During positive (tip-in) load transients it is critical that the air-fuel ratio is controlled to avoid smoke. Therefore, there is a requirement that the EGR flow is reduced as quickly as possible to divert more mass flow to the turbo-charger in order that its speed rises rapidly to provide the air needed to increase engine output - all without the AFR falling below the smoke threshold. Consequently a calibration for best transient torque rise will require a calibration where the EGR valve is more closed than would be the case for minimum fuel consumption, in order that it can be closed more quickly.

The use of variable geometry turbines has promoted the use of larger compressors which have larger inertias. In some cases, this has led to a deterioration of the tip-in performance of the engines, focusing more attention on the calibration for good transient behavior and less on fuel economy.

Thus it can be seen that as the calibration of the VGT and EGR systems is likely to favor best transient behavior the use of VVA to improve light load fuel economy offers greater benefits.

CONSIDERATIONS IN INTERPRETATION OF THE RESULTS

It must be remembered that this piece of work has been carried out using simulation, albeit with a well correlated model. The three areas of modeling that are of particular significance in the interpretation of the results of this piece of work are:

1. As was stated earlier, in the discussion of the combustion model, fixed fuel burning rate characteristics were used for each operating condition. These were derived from measured cylinder pressure data, but did not respond to changes in trapped conditions in terms of pressures and temperatures at start of injection. The use of VVA altered the trapped mass significantly.

2. At the light loads under investigation both the turbine and compressor were operating in regions of their maps that were outside the areas covered by the turbo-charger manufacturer's data, therefore, their predicted operation relied on extrapolated data.

3. The simulations were carried out with fixed fuelling and the perturbations resulting from the application of VVA appeared as changes in output. The changes in bsfc are associated with these changes in bmep. At light loads engine output is sensitive to engine friction as a large proportion of the fuel is used to overcome friction. Inspection of isfc changes may provide alternative insight into the trends.

CONCLUSIONS

1. This piece of work sought to identify and investigate the effects of VVA that can improve diesel engine part load fuel economy. This has been achieved, but the true significance of these effects can only be quantified by experiment.

2. On the engine investigated, light part-load fuel economy can be minimally improved by retarding exhaust valve opening by up to 11º from standard.

3. It was found that outside the range from standard EVO to 11º retarded EVO, changing EVO reduced output because:

    - Advancing EVO from standard reduced expansion work more than it reduced exhaust pumping.

    - Retarding EVO further than 11º increased exhaust pumping work more than it increased expansion work.

4. On the engine investigated, light part-load bsfc can be improved by up to 19% by retarding intake valve closing by 33º from standard.

5. On the engine investigated, light part-load bsfc can be improved by up to 5% by advancing intake valve closing by 33º from standard.

6. On the engine investigated, light part-load isfc can be improved by up to 6% by retarding intake valve closing by 33º from standard.

7. On the engine investigated, light part-load isfc can be improved by up to 1.3% by advancing intake valve closing by 33º from standard.

8. These improvements are produced by minimizing the mass flow through the engine, which reduces compression work and exhaust pumping work.

9. The greater improvements associated with lighter loads highlight the potential for significant fuel economy benefits during the standard drive cycles, where large percentages of the cycle time are spent at light load and idle.

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APPENDIX 1 - BASE ENGINE SPECIFICATION

4 cylinder in-line, 4 valves per cylinder twin overhead camshaft

Overall:

- Swept volume 1998 cm^3.
- Compression ratio 19:1
- Bore 86mm, stroke 86mm.
- Standard output 130PS at 4000 rpm.

Systems:

- Cooled EGR.
- VGT
- Air to air intercooler
- “Common Rail” injection

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