Transient torque rise of a modern light duty diesel engine with variable valve actuation

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ABSTRACT

Simulation has been used to investigate how variable duration VVA can influence the transient torque rise of a modern light-duty diesel engine. The base engine examined employed a variable geometry turbo-charger with air to air inter-cooling, cooled EGR and common rail fuel injection. The VVA system investigated was variable duration, applied to both intake and exhaust valves, controlling primarily exhaust opening and intake closing events.

This paper describes the assumptions and methods used to deal with transient combustion, the interaction of the VGT and EGR sub-systems, the VVA, VGT and the EGR controllers, and contrasts the behaviour of the standard engine with that using VVA by presenting result to explain the phenomena that lead to the differences in torque rise profile.

INTRODUCTION

Increasingly, engine performance prediction, through computer simulation, is being used as a tool in the "Rapid Prototyping" toolbox of the engine developer. This trend has followed the realisation that as hardware development on an iterative, test, modify, retest, basis has become increasingly expensive, so computing power, that makes accurate computer simulation realistic on acceptable time scales, has become cheaper.

The increasing availability of cost effective computing power has allowed the development of simulation models that include more and more sophisticated physics, to represent the processes taking place inside the engine and its associated systems, and effective numerical methods for producing solutions to normally intractable equations. This has led to modern engine simulation models now being widely used. The data produced by these models is regularly used to predict operating trends, and is now starting to be accepted as producing good quantitative results without significant "tuning" of the models. This is particularly so in terms of full-load operation, but increasingly accurate predictions of pumping work, heat transfer and cylinder residuals can also be produced leading to indications of part load fuel economy trends.

Much has been written about the methods used by the simulation codes, with the work by Benson [1] on the "Method of Characteristics" perhaps the most widely referenced, although other methods of solving the unsteady flow equations such as the "Lax-Wendroff" method [2] [3] and particularly "Finite Volume" methods [4] [5] are in use today. Other areas that have attracted a lot of interest are combustion [6],[7],[8] and heat transfer [9],[10],[11]. The culmination of this great body of work is a considerable number of simulation codes, with commercial, "in house" and academic roots.

Fundamental to the commercially available codes are ease of use, flexibility in the range of engines and engine sub-system models that can be constructed and reliable well-correlated solvers based on high levels of physics. The level of development of these attributes is now such that the limitations of use are largely those of the imagination, flexibility and understanding of the user.

Many of the phenomena dealt with by these codes are fundamentally unsteady: wave action in pipes, heat transfer and combustion for example, which all happen cyclically, making the codes suitable for application to unsteady engine operation. This, in combination with the increasing refinement and reliability of the codes has therefore led to their increasing use for the analysis of transient engine operation, particularly in the development of analytical calibration methods, [12], turbo-charging configurations, [13], [14] and control systems and strategies., [15], [16]

This paper discusses the use of a commercially available code, GT-Power[1] , to investigate the transient torque rise of a turbo-charged light duty diesel engine with variable valve actuation "VVA". It describes the models used, the control systems needed to instigate and control the transient operation of the engine sub-systems, such as the variable geometry turbine "VGT" and exhaust gas re-circulation systems "EGR", contrasts the behavior with and without VVA, investigates sensitivity to control speed and seeks to explain the phenomena contributing to the differences.

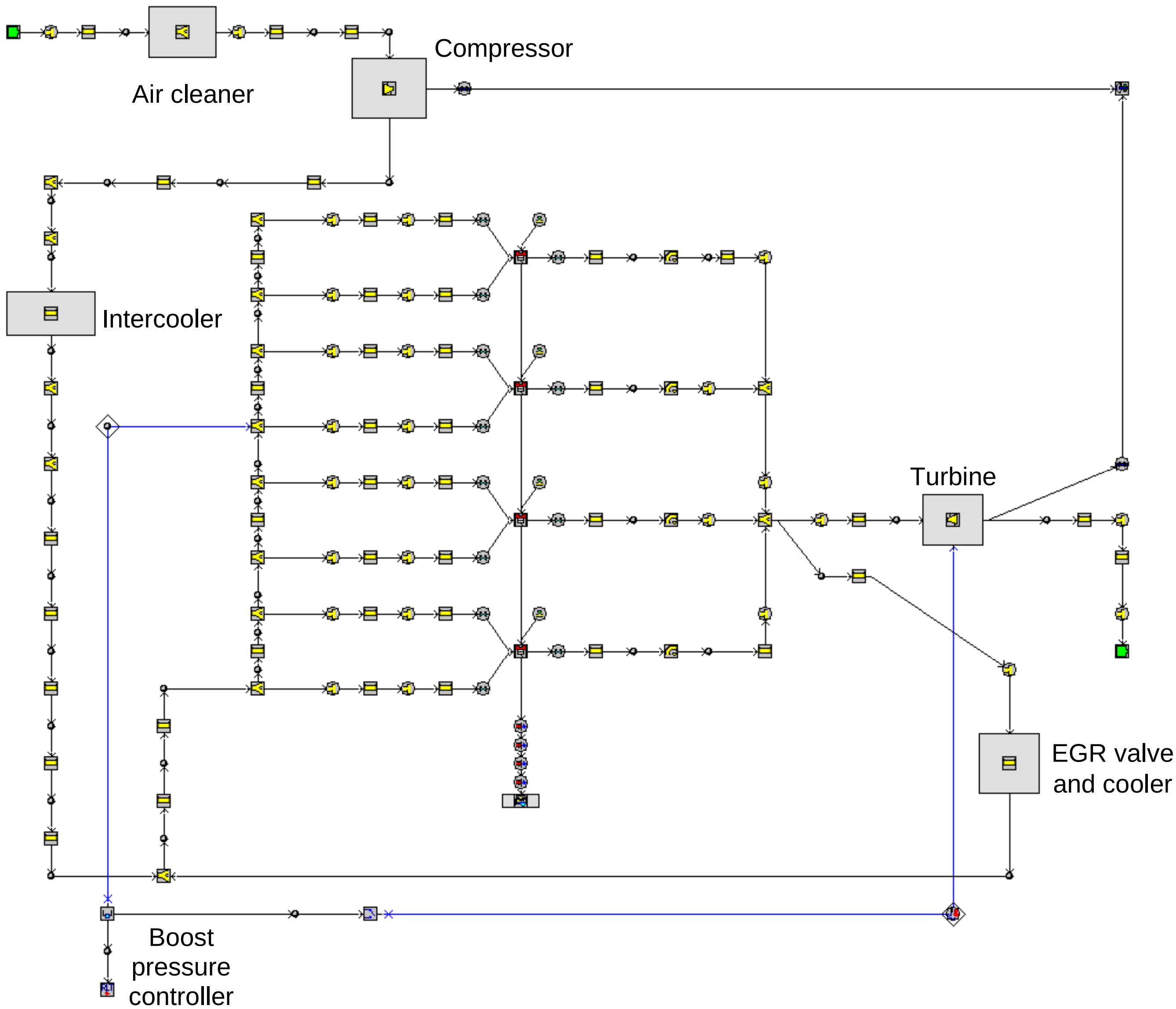


Figure 1 Base model of standard configuration for steady state operation. Major sub-systems highlighted

MODEL DEVELOPMENT

THE STEADY STATE MODELS

<u>The main engine model</u>

The main engine model was constructed to represent the engine systems as completely as possible by including detailed geometry, materials, surface finishes and local ambient conditions for heat transfer etc. A detailed discussion of the data and techniques used for this exercise is outside the scope of this paper, but *figure 1* shows the overall scheme of the model, with sub-systems important to this piece of work highlighted.

Sub-systems that required care in implementation because of special features were: the air filter, EGR mixer, separate ports and discharge coefficients for the two intake valves (swirl and high flow), compressor, intercooler, the EGR valve and EGR cooler, exhaust manifold and turbine.

Simplifying assumptions were made in the treatment of the exhaust valves and the exhaust system after the turbine: The engine has two exhaust valves per cylinder, but since they discharge into a single port, it was decided to treat them effectively as one larger valve with twice the discharge coefficient. The exhaust system after the turbine was simplified to two short pipes and 2 orifices that were configured to provide a good representation of the downstream pressures seen by the turbine with the complete exhaust system [17]

Steady state full-load combustion was simulated by a combination of the Wolfer ignition delay correlation [18] and a 3 term Wiebe function. [19], [20] These were implemented as a user routine (see later discussion of user routines in the section on VVA.) This method was chosen as the simplified built in models did not have the correct forms and insufficient data was available to use the fully phenomenological combustion models. Cylinder heat transfer was modeled using the Woschni correlation. [21]

The use of a variable geometry turbine offers the opportunity to use larger compressors than would be possible with a waste-gate controlled turbine. Also it was found that VVA alters the compressor match, requiring the compressor maps to be extended significantly outside the range of the manufacturer's data, into the region of soft stall and to higher speeds and pressure ratios.

The built in models of turbines allow simple implementation of a VGT, but again some extrapolation of the manufacturer's data, was required, particularly at the more closed nozzle settings. Four maps were used to represent the behavior of this device.

The EGR valve was represented as a controlled orifice and the cooler as a bundle of tubes surrounded by engine coolant at constant temperature..

For steady state full-load operation the VGT was boost pressure target controlled and no EGR was used.

For part load operation a combination of VGT setting (to control pressure differential across the engine) and EGR valve open area was used to achieve the desired EGR flow.

Fuelling was primarily AFR controlled and measured fuel injector needle lift and timing were used as the basis for the injection characteristics.

The complete model was cross checked against a combination of measured data and trusted simulation data, where no measured data was available. It was found to produce results that were typically less than 3% different from the steady state base-line data across the full-load torque curve and it was therefore considered a suitable basis for investigation of other non-standard areas of engine operation:

<u>The VVA system model</u>

The use of simulation to investigate the potential for increasing the low speed torque of diesel engines, by the application of VVA, has recently been reported [22], [23] and the types of VVA applicable to these engines is discussed in [22]

This investigation of transient behaviour was based on the simulation of a VVA system of the "variable angular velocity" type as applied to a recently released, European, 4 cylinder light-duty diesel with state of the art systems. A discussion of the details of the VVA system of this type can be found in [22] and further data about other similar systems can be found in [24],[25] and [26] The specification of the engine and its systems can be found in *Appendix 1*

Figure 2 shows the families 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.

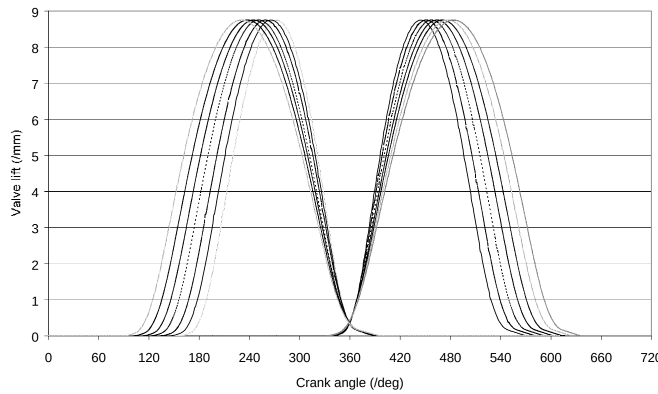


Figure 2 Family of valve lift curves used for the simulation runs. The standard profile is the middle of each group.

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.

Since any VVA system of this type has characteristics that are unique to its geometry there were no standard models available to represent them. But GT-Power allows the use of "User Routines," pieces of code that can be user generated and compiled to form part of a ".dll" file used at run time. GT-Power provides the "hooks" to allow bi-directional data transfer between the main model and the user's subroutine. A user routine representing the detailed behavior of the VVA system was written in Fortran, to provide crank angle based valve lifts to the main simulation model at any crank angle and VVA setting passed from the main model to the subroutine.

<u>Modifications for transient operation</u>

The major changes to the model fell into two areas:

1. The provision of a combustion correlation that could deal with the trends of ignition delay and fuel burning rate with varying engine load, EGR levels and boost conditions

2. The provision of either time based controller characteristics or actual controllers for the various sub-systems that respond to the load transient.

The ways in which these were implemented are discussed in the next two sections. Besides these areas of significant work it was a matter of routine to provide the model with a new set of initial pipe wall temperatures that represented the converged steady state conditions at the start of the transient.

Combustion

Unlike the full-load steady-state operating points used to test the accuracy of the model, it was found not to be possible to adequately model the ignition delay and fuel burning rate curves, that were derived from measured part load cylinder pressure data, by using the Wolfer correlation and 3-term-Wiebe function. Therefore ignition delays and burn rate curves derived from measured data were used to deal with the steady part load operation. This method was extended to provide a "map" of combustion characteristics (including provision for the ignition delays) that covered the fuelling – EGR operating space.

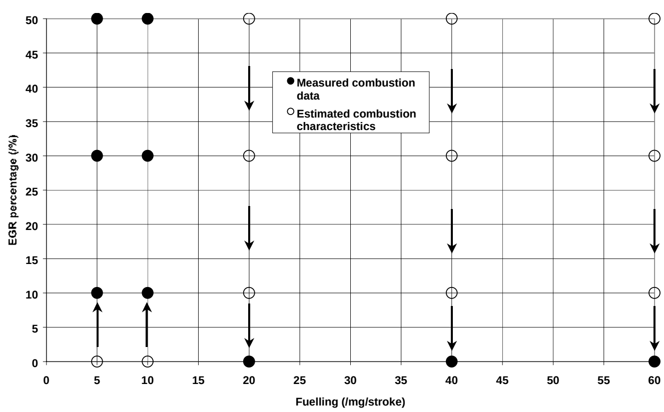


Figure 3 combustion characteristic map in the fuelling-EGR space. (Arrows point ***to*** the data used when measured data was not available)

Figure 3 shows the way in which the map was populated, by indicating the actual test fuelling/EGR percentage conditions leading to the ignition delays and burn rate curves. However, as might be expected, measured pressure data was not available for all combinations of fuelling and EGR level needed to populate the map. This was particularly true at the higher levels of fuelling. After some consideration it was assumed that there would be little or no EGR at the higher fuelling levels, as the air flow required to use these amounts of fuel, with the prevailing air fuel ratio, would take sufficient time for the EGR valve to have shut. Therefore, the zero EGR burn curves have been propagated upwards across the maps to the higher EGR levels at the high fuelling values. The validity of this assumption will be discussed later. This map was installed into the GT-Power model, where interpolation between the curves for intermediate values of both fuelling and EGR level was carried out automatically. (No extrapolation was allowed, if the operating point went off the edge of the map the value at the edge of the map was used)

Controllers

Current state of the art diesel engines use variable geometry turbines which can combine the functions of a waste-gate and throttle, which would have been used in earlier engines. Opening the VGT 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 increase the pressure difference across the engine to increase EGR, a function that would previously have been carried out by a throttle. Modern EGR valves are position controlled, rather than on/off, and as a consequence the EGR valve and the VGT both need relatively sophisticated controllers and control strategies.

The relative priorities and speeds of response of the various sub-system controllers is a major constraint on calibration of light duty diesel engines as there is a trade-off between fuel economy and transient drivability through the interaction of the VGT rack and EGR valve orifice settings. [27] Fuelling systems can respond on a stroke by stroke basis and will, within measurement system accuracy, follow a prescribed air-fuel ratio trajectory very closely. Therefore, the limitations on transient operation are imposed by the air/EGR management. When using AFR fuelling control (as is common these days) transient torque rise is a function of how quickly the air flow can be increased, this is initially a problem of rapidly reducing the amount of EGR to increase air flow and then becomes a matter of achieving a target boost level, through management of the turbine and, if present VVA, as quickly as possible to maximise volumetric efficiency throughout the transient.

From the foregoing it can be seen that five controllers were needed for this piece of work:

1. Fuelling/AFR
2. Intake VVA
3. Exhaust VVA
4. VGT rack (2 modes)
5. EGR valve

In order to discuss the operation of these controllers it is necessary to describe the sequence of events during the transient: In a real driving situation the transient from light load cruise to full load would be initiated by the driver pressing the accelerator pedal. In a modern diesel engine management system this would trigger a change of operating mode from minimised emissions and near minimized fuel consumption to maximised output. This would be achieved by lowering the target AFR and providing adequate fuelling to achieve it, closing the EGR valve and closing the VGT rack to maximise boost.

To replicate this sequence of events in the model time based series were used to represent the changing inputs, with simultaneous step changes in AFR and available fuel representing the combination of driver input and engine management mode change. Up to the start of the particular transient investigated the EGR valve is fully open and the VGT rack position is controlled to manage the amount of EGR. At the time of the step change, in fuelling and AFR, the controller for the VGT rack is switched to a boost pressure target from an EGR target and a time based series representing the EGR valve closing response is applied. (The strategy governing the operation of the VVA controllers is discussed later)

GT-Power provides a number of ways in which controllers can be imposed on system actuators, and in addition to the time based controller response series a PID controller was implemented to control the VGT rack position. *Table 2* summarizes the controller characteristics:

Variable controlled	Controller type	Comments
Fuelling / AFR	Time based series	Step changes (during one cycle)
Intake VVA	Time based series	Taken from measured controller response data
Exhaust VVA	Time based series	Taken from measured controller response data
Variable Geometry Turbine rack	PID – switches between EGR target and boost pressure target at the start of the transient	Tuned for response to match baseline model characteristics for speed, overshoot and stability
EGR valve orifice diameter	Time based series	Taken from simulation results of detailed study of complete EGR valve control system

Table 2 Summary of controller characteristics

Figure 4 shows the time based controller characteristics used during the standard valve timing run. The transient is set to start at t=4.8 seconds. The air-fuel ratio switches from 47:1 to 19:1, which is representative of very light cruising load to smoke limited full load. The available fuel switches from 5mg/stroke to 65mg/stroke, but the actual amount of fuel used is set by the AFR limit once the transient starts. As already mentioned the VGT controller switches from EGR control to boost control at the start of the transient and this results in the target boost level changing from approximately 1 bar absolute to 2.44 bar absolute pressure. Inspection of *figure 4* also shows that there is a delay between the initiation of the transient and the EGR valve beginning to close. This time series was chosen to be representative of typical EGR valve response and includes the detailed behaviour of the valve, the controller and the vacuum actuation system and it is the vacuum actuation system that introduces the delay. [15] In *figure 3* it will also be noted that the time series for the controllers finish at t=5.85. This is because the last of the time series has stabilised and from then on GT-Power uses the last value in the series.

The behaviour of the PID controller implemented to control the VGT setting for boost pressure control can be seen later in *figure 6*, where the transient VGT setting is shown.

As mentioned above, closing the nozzles of a VGT increases the exhaust back pressure, thereby increasing pumping work and fuel consumption. [27] It can also be reasoned that the smaller the EGR valve orifice the greater the pressure differential required to provide a given level of EGR, and again the greater the fuel consumption. A smaller EGR valve opening is best for transient response, because it can be closed more quickly. But, this results in the need for a more closed VGT to provide the necessary EGR flow – increased fuel consumption again resulting. Thus it can be seen that good transient operation is in conflict with best part load fuel economy.

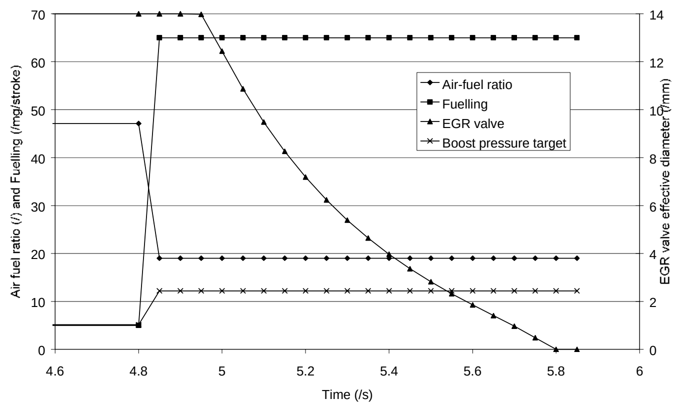


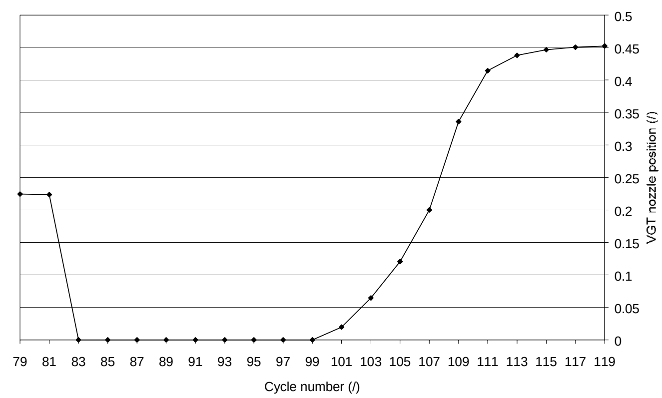
Figure 4 Time based controller characteristics used with standard valve timing (The first 4.8 seconds is used to stabilize the low-load operating condition)

The transient and discussion of the results with standard valve timings

Once the behavior of the control systems had been adequately modeled, simulation was carried out to investigate the transient torque rise of the engine at a constant 2000 rpm from 5mg/stroke fixed fuelling with 50% EGR to full load, zero EGR, with a target boost level and AFR limit. This was selected as being representative of the start of an overtaking manoeuvre in a high gear.

N.B. In all of the following figures the data presented is converged data for the end of a given cycle. The start of the transient is the end of cycle 80 at t = 4.8 s.

Figure 5 shows the VGT behaviour throughout the transient. At the desired level of EGR, 50%, at 5mg/stroke fuelling the VGT rack setting is approximately 0.22, which generates the pressure differential to cause the desired EGR flow rate (The EGR valve is fully open). At the switching time, the end of cycle 80, the priority becomes maximising boost and the error term in the PI controller becomes large causing the VGT nozzles to be rapidly closed. (The time taken in the above plot is consistent with the slew rate of an electrical actuator) As the error in boost pressure reduces so the controller opens the nozzles, finally stabilising at 0.45 for the target boost level of 2.44 bar absolute. *Figure 6* shows the engine torque predicted during the transient. It will be seen from this that there is a small torque peak shortly after the transient begins. This is a consequence of the difficulty of reducing EGR and increasing boost pressure quickly and, whilst it might be thought that this is positive, fluctuations in torque of this nature are not



good for the transmission and there is an emissions implication for a real application:

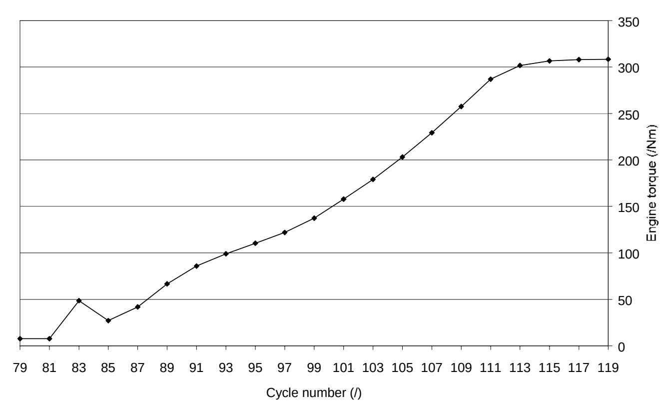
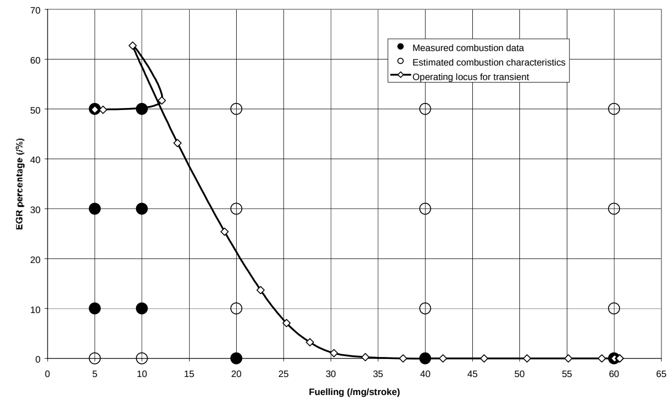


Figure 5 VGT nozzle setting as a function of engine cycle number with standard valve timings

Figure 6 Transient torque rise with standard valve timings

Figure 7 shows the EGR-fuelling operating locus on the combustion characteristic map, initially shown in *figure 3*. It can be seen from this that during the transient, the model moves into operating regimes (combinations of fuelling and EGR level) that are not well populated with measured combustion data. In the region above 50%



EGR the model simply uses the 50% EGR condition, for fuelling levels between 10 and 20 mg/stroke interpolated data provides some of the effects of the EGR, but probably not all, and for fuelling levels between 20 and 40 mg/stroke the data used does not include the effects of EGR, despite the locus showing that some is needed. It is not possible to comment on how important these excursions from measured data are, but it can be seen that for the majority of the transient the major effects of EGR are captured.

Figure 7 Transient EGR-fuelling operating locus with standard valve timing (Cycle 79 is at left, 50% EGR and 5mg/stroke fuelling)

Figure 7 also shows that the torque spike occurring around cycle 83 is associated with a combination of increased fuelling and increased EGR (from starting conditions). *Figure 8*, shows that a consequence of the increase in EGR is a dip in airflow. It also shows the explanation for the rise in EGR and reduction in air flow: It can be seen that, in comparison with the EGR valve control the VGT control is fast, so the VGT starts to close, increasing the pressure driving the EGR across the engine, before the EGR valve starts to close to reduce the EGR level. However, by cycle 86 the EGR valve is closed sufficiently to reduce EGR flow and the air flow starts to increase again. Despite this dip in airflow, because of the implementation of the AFR controller the amount of fuel injected into each cylinder is correct and the AFR remains correct. Thus if the dip in airflow could be avoided an almost step change in torque at the start of the transient and a smooth rise from there on might be expected . This is discussed again later

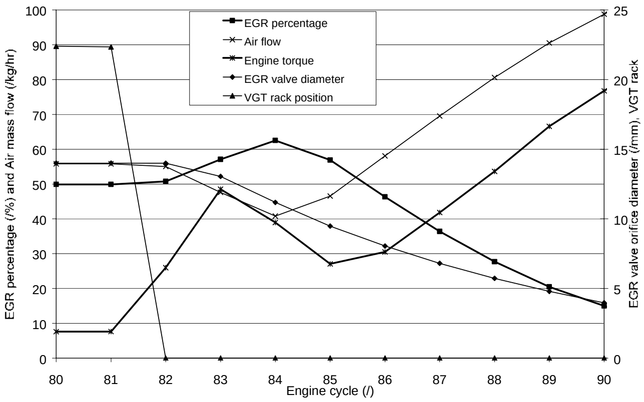


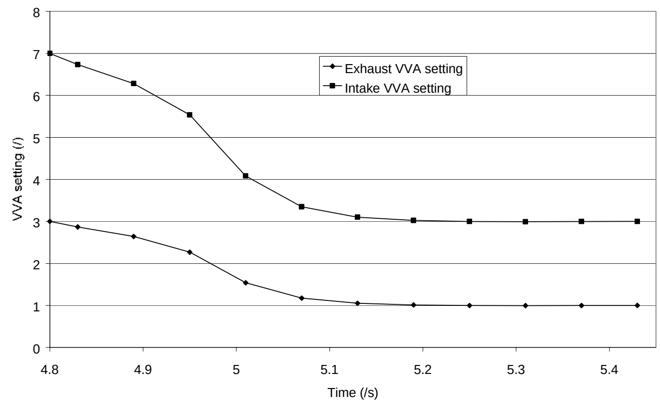
Figure 8 Response of the EGR orifice diameter and percentage, VGT nozzle position, air flow and torque with standard valve timing in the region of the torque spike.

It should be pointed out that the model uses a "smoke limiting" AFR controller that has access to the air mass trapped on a cylinder by cylinder basis to adjust the fuel injected for each cylinder. Thus there are no delays and accurate AFR control is possible with the simulation model. But, in a real system, the air mass flow data would be measured for the whole engine and would be historical, i.e. delayed, thus falling air flow would tend to result in a reduction in AFR below the target, possibly leading to smoke, and increasing air mass flow would result in an AFR above the target, leading to slower torque rise than would be achievable with more accurate fuel control. Intelligent predictive control taking into account rates of change of air flow and correcting for delays in measurement might be able to compensate for this.

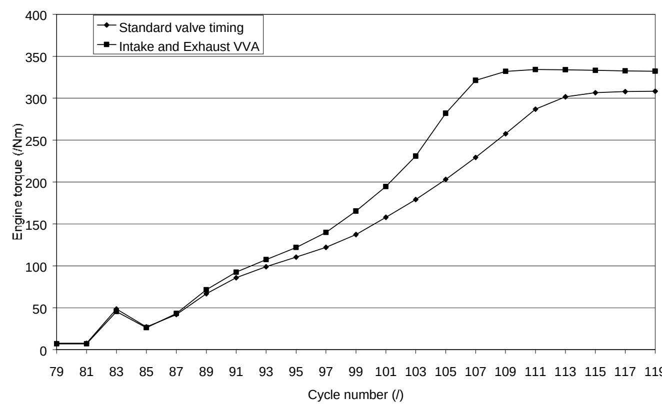
Overcoming this shortcoming of the control system used in the simulation model, which produces this torque spike is practical in a real system, where the response of the VGT controller could be delayed to allow the EGR valve to partially close before the VGT setting reduced, but this would have taken considerable effort to implement in the simulation model. Therefore, since it was considered that the early torque rise and fall was unlikely to mask any underlying differences between the behaviour with and without VVA it was decided that this controller strategy would be retained for the comparison in torque rise rate.

COMPARISON OF STANDARD VALVE TIMING AND VVA PERFORMANCE

The only difference between the models with standard valve timing and with VVA was the addition of controllers for both intake and exhaust VVA systems. The difficulty with these was that the actuators used in real VVA hardware systems are subject to the reversing loads imposed by the valve train events and as such are highly dynamic, often moving to their target positions in a



stop-start manner in small high frequency steps. Within the scope of this piece of work is was considered that it was not possible to replicate this type of behaviour for



the controllers and a higher level approach was needed. The controller characteristics used were therefore based on measured data, sampled at a low speed, such that the high frequency stop start behaviour was filtered out. This data was measured from a hydraulically actuated VVA system on a test bench and is represented as time series in the model.

A separate piece of work, [27] showed that at 2000 rpm, 5mg/stroke fuelling and 50% EGR, the initial condition for the transient, for minimum fuel consumption the intake valve closing setting would need to be 7 and the exhaust valve opening setting 3 and in the full load investigation it was shown that for maximum output, at 2000 rpm full load, these settings would need to be 3 and 1 respectively. Thus the intake VVA controller needed to change the setting from 7 to 3 and the exhaust VVA controller from 3 to 1. *Figure 9* shows these time series characteristics.

Figure 9 Time series of the VVA controllers. t=4.8s (end of cycle 80) is the start of the transient.

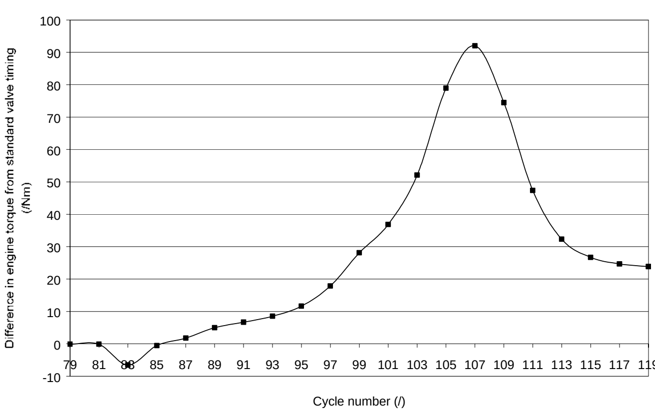
Figure 10 Transient torque rise with and without VVA

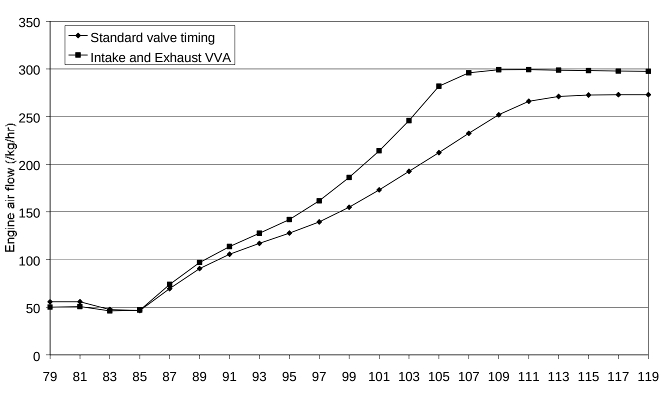
Figure 11 Difference between the torque levels with and without VVA

Figure 10 shows the transient torque rise characteristics for the model with fixed, standard, valve timing and with VVA operating according to the strategy outlined above. It can be seen that whilst the initial characteristics in the region of the torque spike, discussed earlier, show the two curves to be very similar, beyond cycle 87 the model with VVA has a notably higher rate of torque rise. *Figure 11* shows the difference between the two torque levels, a positive difference indicating greater torque with VVA. It can be seen that initially the torque output is marginally higher with VVA, as the fuelling is fixed at 5mg/ stroke in both cases, but the VVA reduces pumping work. But this situation reverses as soon as the AFR controlled fuelling starts at the end of cycle 80 as with the VVA air flow is lower. The situation again reverses after cycle 85, beyond which the engine model with VVA predicts greater torque for the remainder of the transient. The reason for this behaviour is shown by *figure 12* which shows that after cycle 85 there is greater air flow through the engine with VVA and because the fuelling is AFR controlled a greater output results.

Figure 12 Air flow with and without VVA

The factors that lead to this greater airflow are greater volumetric efficiency, greater power from the turbine (partly from increased efficiency and partly from increased mass flow) and increased compressor efficiency, all of which result from the use of VVA. *Figure 13* shows the turbine and compressor cycle averaged





efficiencies for the transient and it can be seen that from cycle 87 onwards the efficiencies of both are higher with VVA than standard valve timing until the transient is over and the models have stabilised at maximum torque. (It should be noted that with VVA the stabilised torque output is higher as the volumetric efficiency of the engine is higher)

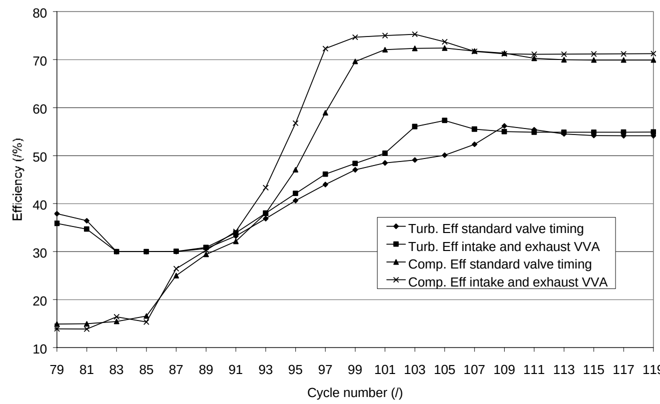


Figure 13 Turbine and compressor cycle averaged efficiencies with and without VVA

Figure 14 shows the effect of the VVA on the volumetric efficiency. It also shows the time based characteristics of the VVA controllers as well as the standard valve timing settings. From this it can be seen that the VVA systems are already at their target positions by the end of the dip in the volumetric efficiency caused by the EGR level excursion (cycle 85) It should be noted that on the basis of the work carried out in the full load investigation that the exhaust VVA system starts the transient, and remains, at settings that are superior to the standard, while for the intake VVA system the transient starting condition is worse than standard for volumetric efficiency. However, since they have both reach their final positions before the air flow actually starts to rise the disadvantageous intake starting point appears not to be a problem, but slower VVA control systems might reduce the benefits.

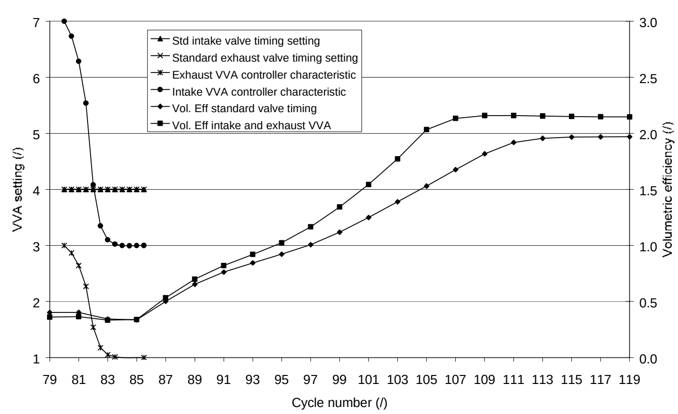


Figure 14 Volumetric efficiency and valve timing settings

Figure 15 shows the effects of the VVA on available torque for accelerating the turbo-charger. It is not clear if the 200+% greater available torque around cycle 83 is significant overall, but the availability of 15% to 25% greater torque to speed up the device, from cycle 87 until the boost pressure is achieved, (cycle 107) is significant and demonstrates another reason why the improved transient operation is achieved.

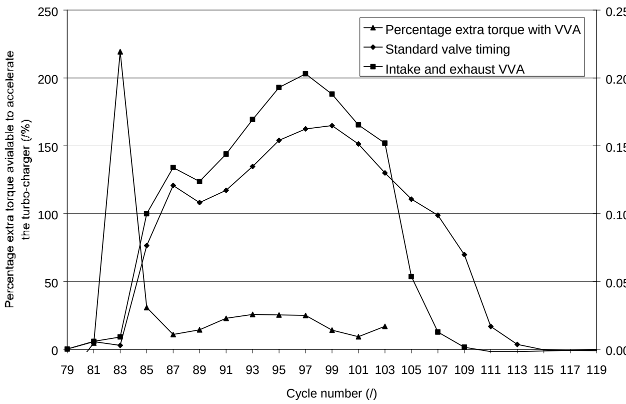
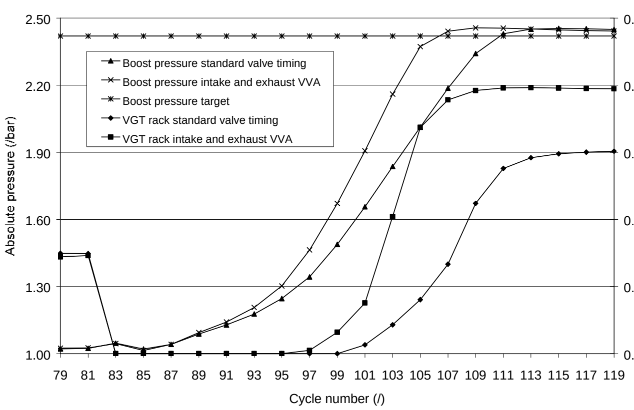


Figure 15 Mean torque available to accelerate the turbo-charger and percentage greater torque available with VVA

The fact that there is still torque to accelerate the turbo-charger at cycle 107, despite boost pressure being achieved is symptomatic of overshoot of the boost pressure controller. This can be seen in *figure 16*, which shows the VGT nozzle settings, the boost pressure target and actual boost pressure characteristics for the transients, the later demonstrating the overshoot. (It is interesting to note that despite greater airflow and output the model with VVA has a less restrictive VGT setting at the full load operating point because the volumetric



efficiency of the engine is predicted to be higher.)

Figure 16 VGT settings and boost pressures during the transient with and without VVA

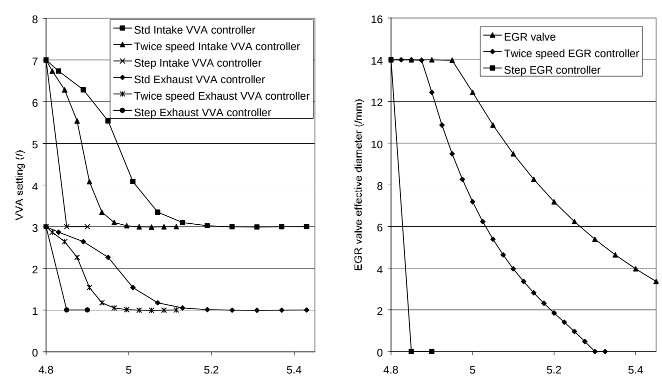
Thus it can be seen that VVA offers faster transient torque rise capability by maximising volumetric efficiency and thereby mass flow through the engine. This increased mass flow and the improved interaction between the turbine, the flow through it produce greater turbine net torque to accelerate the turbo-charger, leading to a faster rise in boost pressure. However, with the controller characteristics investigated here the application of VVA has no impact on the torque spike and air flow dip early in the transient.

Analysis of the above indicates that slowing the initial response of the VGT rack controller could remove the EGR spike and consequent air flow dip. However, it seems likely that this will be at the expense of slower and therefore less accurate control of EGR and less rapid transient response. Therefore consideration of the effects of a faster EGR valve and VVA controllers might provide alternatives that do not cause reductions in other aspects of the engine's performance:

SENSITIVITY TO CONTROLLER SPEED

An investigation was carried out into the effects of faster VVA and EGR controllers. For each of them a controller that was twice as fast, i.e. took half the time to complete the movement and a step response controller i.e. one that changed instantaneously were implemented. *Figure 17* shows the time characteristics of these controllers.

Figure 18 shows the effects of the faster VVA controllers. It can be seen that the controller with the step characteristic offers only very small improvements over the twice speed and standard controllers in the early cycles. This is because it offers the possibility to optimise the intake VVA setting earlier in the transient. However, the advantages of the faster controller are so small as to offer little incentive to implement faster, and necessarily more expensive, controllers for the VVA systems. (It is worth noting that direct valve actuation



systems are capable of implementing these step changes.)

Figure 17 Time series characteristics of the "twice speed" and step controllers applied to the VVA (left) and EGR systems (right)

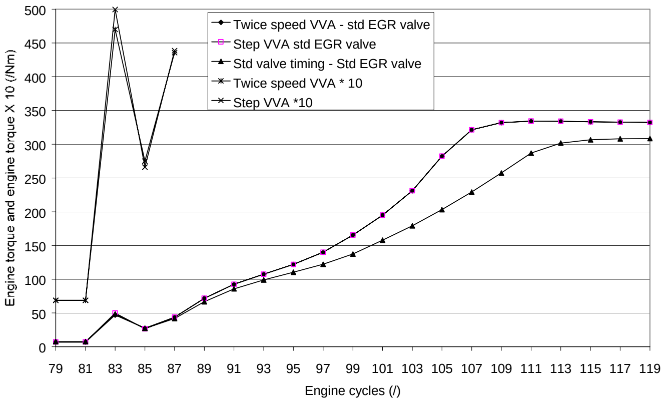


Figure 18 The effects of faster VVA controllers on torque rise

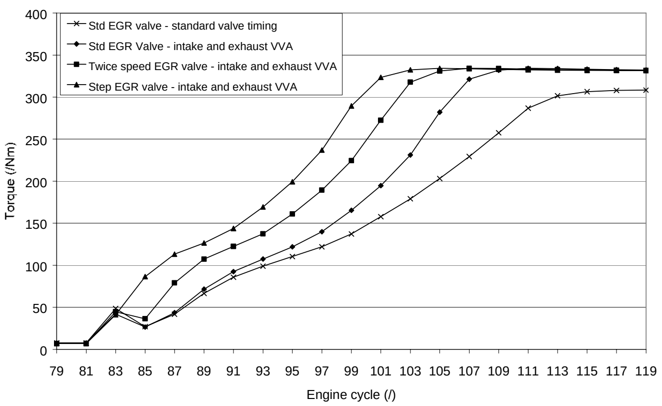


Figure 19 The effects of faster EGR controllers and VVA on torque rise

From *figure 19*, it can be seen that EGR valve speed is very influential in the rate of torque rise. With a step characteristic EGR valve and VVA approximately 100% more torque is available 10 cycles (0.6s) after the start of the transient when compared to standard and maximum torque is achieved 10 cycles earlier. From the above it is possible to comment on the validity of the assumptions made in the implementations of the controllers: it can be seen that faster VVA controllers offer little advantage, and provided they are faster than the EGR valve, making them slower is unlikely to affect the results of this type of work significantly. But, faster EGR valve control does have a significant affect on torque rise, therefore in investigations such as this it is important to have representative data for the response of the EGR valve.

CONCLUSIONS

1. It has been demonstrated that when a variable geometry turbine is used to modulate engine pressure differential to control the flow of EGR, if the VGT actuation is substantially faster than EGR valve actuation, the interaction between the two systems can lead to short term increases in EGR levels at the beginning of “tip-in” transients.

2. It has been demonstrated that the use of variable duration VVA systems applied to both intake and exhaust valves can lead to improved transient torque rise. For the transient from 5mg/stroke fuelling and 50% EGR to full load at 2000 rpm the improvements shown by simulation were:

   10 cycles into the transient X Nm or Y% more torque is available

   100 cycles into the transient X Nm or Y% more torque is available

   Full load torque is achieved after ZZ cycles, with VVA, aa cycles or b seconds sooner than without.

3. The reasons for these improvements are:

   The VVA increases the engine’s volumetric efficiency which in turn increases engine mass flow.

   The efficiencies of the turbine and compressor are increased

   Etc?

4. An investigation into the sensitivity of the transient operation to controller speeds demonstrated that:

   Faster VVA controllers have little effect

   A faster EGR valve controller has a significant effect on torque rise rate and can overcome the initial EGR excursion referred to in 1, above.

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APPENDIX 1 - BASE ENGINE SPECIFICATION

4 cylinder in-line, 4 valves per cylinder twin overhead camshaft with inward acting rockers.

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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