Optimization of the valve lift strategy during the acceleration of a diesel engine using WAVE and DOE

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Abstract

This paper presents the use of WAVE 3.4.3 in combination with statistical methods known as design of experiments (DOE). These tools are used to find an optimized valve lift strategy during the acceleration of a turbocharged passenger car diesel engine. For the valve train a future valve timing system is assumed that allows almost completely independent inlet and exhaust valve timing.

It is demonstrated how statistical methods provide substantial aid during the simulation and optimization process when a large field of parameters has to be varied.

Introduction

Electromagnetic and electrohydraulic valve timing systems [1], [2], [3] offer a huge field of possibilities for running a combustion engine. These systems are able to open or close the valves at any desired time. Restrictions only result from the fact that the clearance between valves and piston must be maintained. This means that with the high compression ratios of modern passenger car diesels of ca. 20:1 it is not possible to open valves around TDC. Otherwise collision with the piston would occur. Valve pockets in the piston top could avoid the problem but would be very deep and therefore would have a massive impact on swirl, spray formation and combustion in general.

The aim of this study is to find the potential of such a valve actuation system on a passenger car turbocharged diesel engine under transient conditions. To demonstrate this potential it is necessary to find a valve lift strategy that leads to the best transient engine acceleration behavior. It can easily be seen that the possible opening and closing timings can be varied in a large range. Therefore a classical one-after-the-other-variation would result in hundreds of combinations and simulations. In order to manage this problem statistical methods like DOE can provide substantial help. Using DOE it is possible to reduce the number of simulations necessary and at the same time get a better understanding of the dominant as well as the rather unimportant factors that influence the engine performance.

For this study a system is assumed that is able to open and close valves at any time (apart from TDC) within 36 degrees crankangle which is much quicker than what usual cam shaft driven systems provide. Usually camless systems are crankangle based rather than time based. This leads to almost rectangular valve lift profiles at low engine speeds while near rated power the profiles look more normal. In this study the profiles are kept constant during a simulation and represent an average valve speed that appears during the transient. This makes the simulation much easier and gives nevertheless an idea about the potential of variable valve timing.

Since no proper data is available it is further assumed that the power to drive the valve train is for all systems the same. The goal of all developers of camless systems must of course be to reach a state where the camless valve train runs with less power than the classic cam driven.

WAVE Model

Table 1 shows some data of the engine that was chosen as the baseline. It is the Mercedes Benz OM611 4 cylinder engine [4] that is in production since 1998 featuring e.g. Common Rail Direct Injection, wastegate turbocharger with intercooler, swirl control system and EGR cooler.

Mercedes Benz OM611	
No. of cylinders/valves per cylinder	4 inline/4
Displacement	2151 cc
Bore	88 mm
Stroke	88.4 mm
Rated Power	92 kW at 4200 RPM
Max. Torque	300 Nm from 1800 to 2600 RPM
Compression ratio	19:1

Table 1: OM611 passenger car diesel engine

Figure 1 shows a schematic of the engine's geometry. *Figure 2* shows the corresponding WAVE model including EGR ducts and cooler, swirl control system and wastegate turbocharger. The setup and calibration of the model were carried out for the engine version for the Mercedes Benz C-Class under steady conditions. *Figure 3* shows the matching with measured data for the calibrated model along full load line. The model has been in use and been modified for more than a year prior to this work. It is therefore quite well correlated with the actual engine. Additionally the TC inertia and duct and junction volumes were checked for accuracy in order to properly describe the transient conditions.

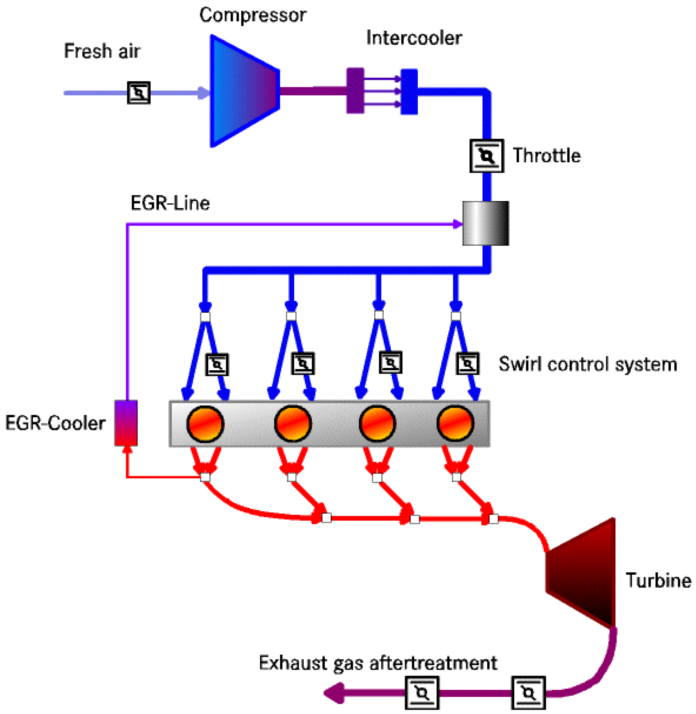


Figure 1: Schematic of the OM611 engine

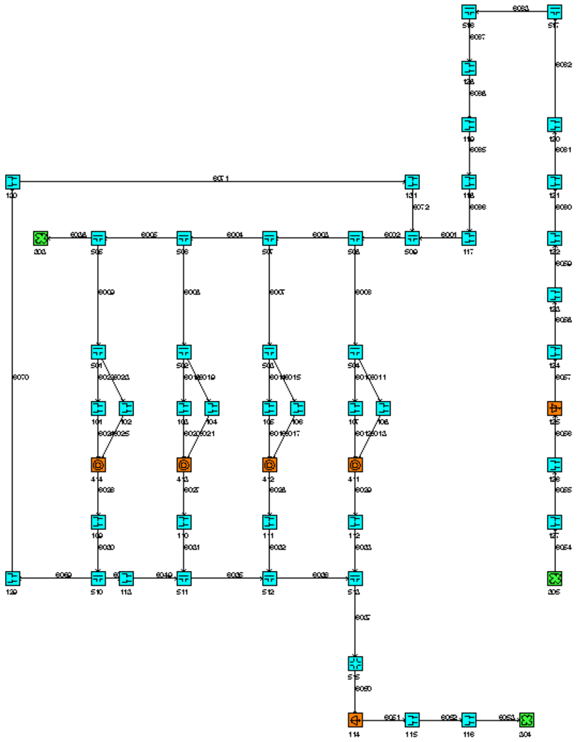


Figure 2: WAVE model of the OM611 engine

Baseline acceleration

The starting point for the transient is part load 2 bar BMEP. The object of this study is a full load acceleration in first gear of a Mercedes Benz C-Class. First about 20 cycles are simulated in order to bring the baseline model to a converged state at 2000 RPM. Then at time t=0s the injected fuel mass is increased until the smoke limit of the engine is reached (equivalence ratio ca. 0.83) and the engine starts accelerating. At the same time all intake ports are opened and the EGR valve is closed. *Figure 4* illustrates the standard valve lift curves and diagrams for engine speed, TC speed, fresh air flow, intake manifold pressure, turbine inlet pressure and engine torque. It can be seen that at t=1.8s the wastegate of the TC starts opening and the boost pressure is limited. Therefore the TC afterwards remains at constant speed or accelerates very slowly. The fresh air flow grows only as a result of the increasing engine speed not the increasing boost pressure.

Design Of Experiments

A. Optimization criteria

As a first step the question must be answered what the aim of the optimization is. Here it seems that the time that passes until the engine reaches a certain speed (e.g. 4000 RPM) should be minimized. But since the valve profiles cannot be altered during a simulation run in WAVE 3.4.3 it is necessary to look at a few more features. It must be expected that in the beginning when the wastegate of the TC is closed other valve timings will be needed compared to when the wastegate opens. To get a better understanding four cases will be investigated:

1. Valve timings for shortest time until wastegate opens.
   This case leads to the valve timings for best TC acceleration.
2. Valve timings for shortest time until the engine reaches 4000 RPM.
   The found timings do not have to be necessarily the most sensible ones for the real engine. As mentioned above best acceleration will probably not be obtained keeping one set of valve timings during the acceleration.
3. Valve timings for fastest change in engine speed before wastegate opens
   This case leads to best car acceleration and does not necessarily mean the same valve timings as case 1. Quick TC acceleration does not mean quick engine acceleration and vice versa.
4. Valve timings for fastest change in engine speed while wastegate opens
   In this case all simulations have reached the maximum boost pressure and the wastegate opens. Further TC acceleration does not take place. This means the found valve timings lead to best engine acceleration through optimized pumping work.

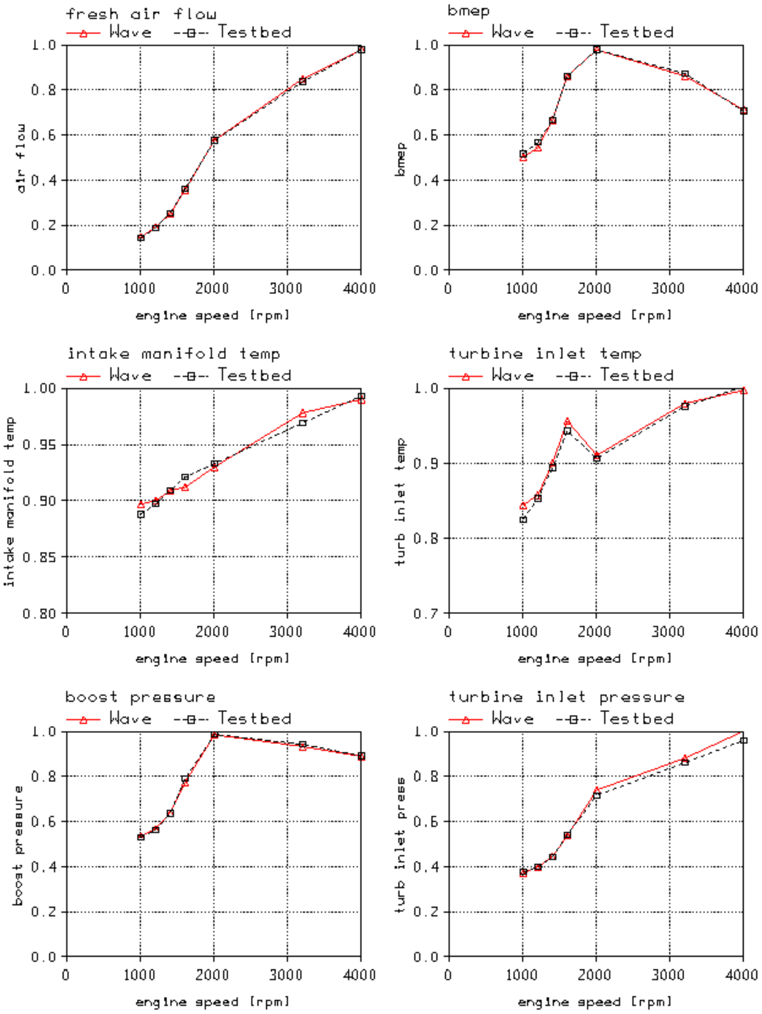


Figure 3: Correlation of the calibrated WAVE model with measured data

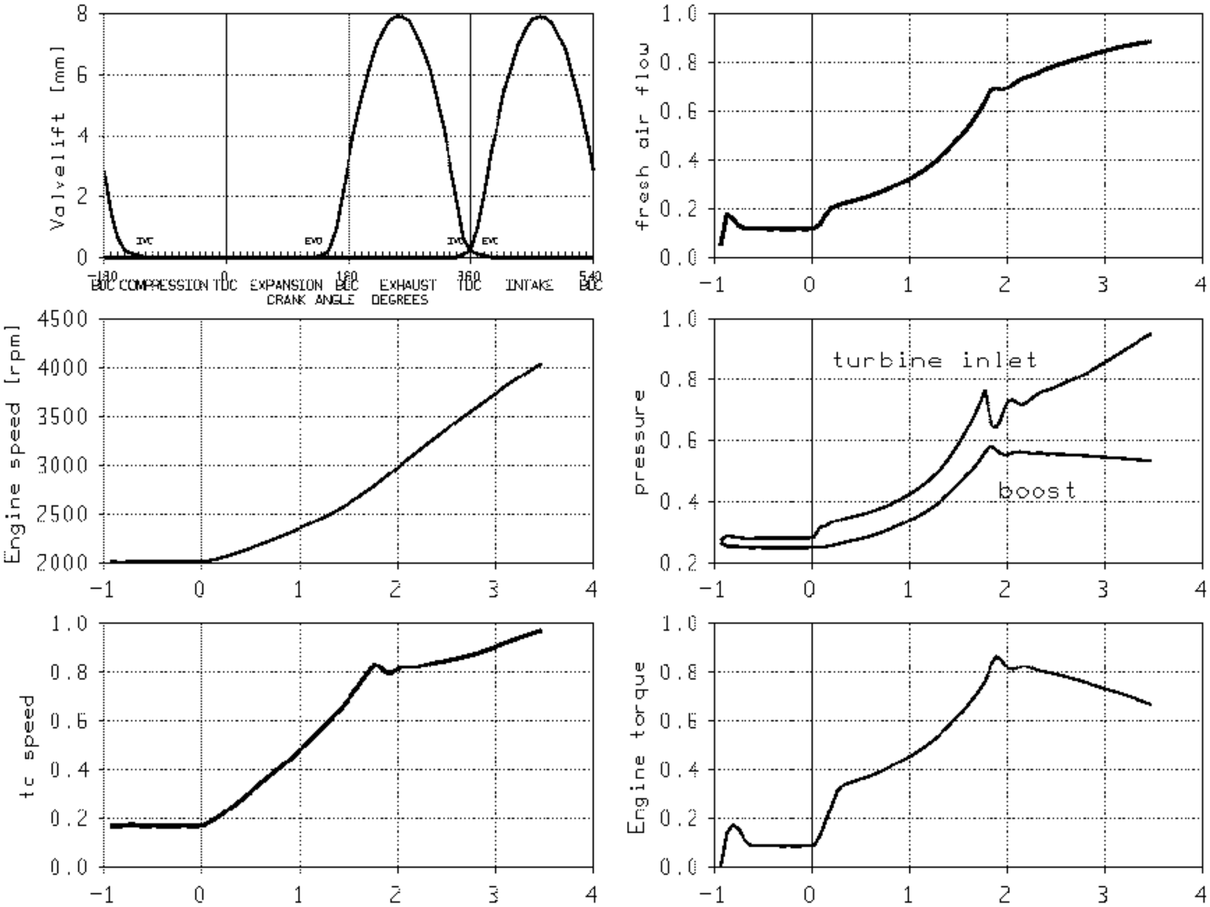


Figure 4: Baseline acceleration

B. Choice of simulation points

The steps of the DOE method are as follows:

1. Using experience a suitable model is chosen that is capable of describing the dependencies between parameters (valve timings) and target (e.g. time for engine acceleration). Additionally the variation range for the parameters must be determined.
2. The combination of parameters for each simulation is chosen. Through systematical variation of several parameters at the same time it is possible during the following analysis to separate the effects of each parameter and the interactions amongst them.
3. After the testing/simulation statistical models are built and compared.
4. Using these models optima can be found and combinations of parameters be determined. To validate the models the combinations should be tested/simulated as well.

Figure 5 illustrates the steps of this statistical method.

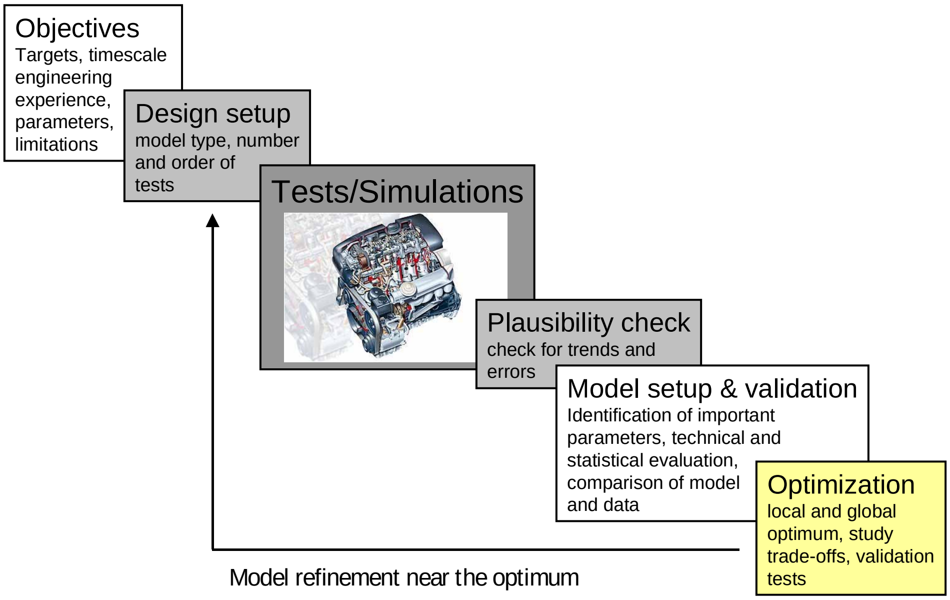


Figure 5: DOE approach

For this case a 2nd order Taylor function was chosen to model the target values as a function of the four parameters. Such functions consist of one constant term, four linear terms, four terms of 2nd order and six 2nd order interaction terms. The number of combinations simulated must be at least 15 in order to solve the equations for these terms.

Here a design for the combinations has been chosen that consists of 25 combinations of parameters. This way it is possible to collect enough data for a proper model setup and at the same time produce all simulation results within a day. In this case there are four parameters that can be altered independently:

1. Inlet valve opening (**IVO**)

2. Inlet valve closing (**IVC**)

3. Exhaust valve opening (**EVO**)

4. Exhaust valve closing (**EVC**)

To determine a useful range for the variation of these four parameters knowledge from steady investigations or engineering experience is very helpful.

For IVO it is known, that it is only possible after TDC. Otherwise collision with the piston might occur. There is also no reason to wait too long after TDC for IVO. In this case the fresh air flow would be reduced and so would be the engine torque.

For IVC a wider range seems interesting. From experience it is known that in steady cases for low engine speeds IVC around BDC=540°CA after combustion TDC is useful. For engine speeds near rated power sometimes IVC up to 80°CA after BDC makes sense because of the dynamic behavior of the oscillating air in the intake manifold.

EVO is for a turbocharged diesel the most interesting parameter under transient conditions. From steady simulations it is known that early EVO leads to high turbine inlet temperature and pressure. In this way especially at low engine speeds (e.g. less than 2000 RPM) the steady TC speed can be increased significantly and in the same way boost pressure and injected fuel mass and in the end engine torque [5]. On the other hand this method leads of course to higher specific fuel consumption: since the exhaust valves are opened long before BDC the exhaust gas leaves the cylinder and reduces the cylinder pressure while the piston is still moving downwards. In this way a more or less significant amount of expansion work is lost and must be compensated by higher boost pressure and increased injected fuel mass. Also too late EVO may produce problems. Especially at high engine speeds there is only little time for pumping the exhaust gas out of the cylinder. If the exhaust valves open too late the cylinder pressure during the upward motion of the piston is too high and reduced BMEP is the result.

For EVC exist similar restrictions as for IVO. Again because of valve piston clearance problems it is not possible to remain open until after TDC. In this case there also seems to be no benefit in closing too early since then the time for pumping out the exhaust gases would be reduced and the amount internal EGR increased.

Table 2 shows the ranges of valve timings that were identified for this study to be of interest. It must be remarked that the statistical models can also describe values outside these ranges, but the further they lie outside the less accurate the model will be.

Parameter	Range of variation [°CA after CTDC]
IVO	360 – 380
IVC	570 – 620
EVO	70 – 180
EVC	340 – 360

Table 2: Interesting valve timing ranges

According to a so called central composite design 25 valve timing combinations were chosen that have to be simulated in transient WAVE runs. It can be seen that all simulations can be performed easily in only one day.

D. Analysis and optimization of valve timings

The data produced by WAVE were evaluated and statistically analyzed on a PC using MINITAB 12 and MATLAB 5.3 with the statistics and optimization toolbox. The models to describe the target value as a Taylor function were set up in MINITAB. The optimization was carried out within MATLAB.

The Taylor functions consist as mentioned above of 15 coefficients, for example

Time=a1 + b1*IVC + b2*IVO + b3*EVC + b4*EVO
+ c1*IVC^2 + c2*IVO^2 + c3*EVC^2 + c4*EVO^2
+ d1*IVO*IVC + d2*IVO*EVO + d3*IVO*EVC
+ d4*IVC*EVO + d5*IVC*EVO + d6*EVC*EVO

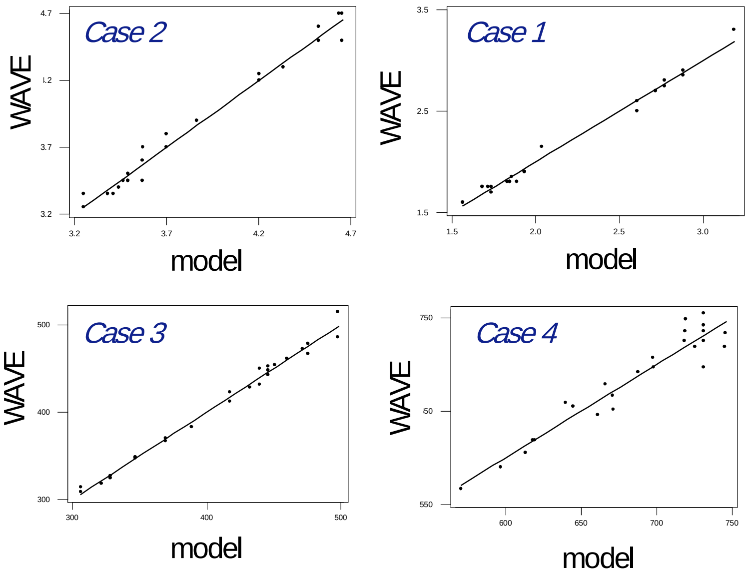
When the parameters are standardized the values of the coefficients (a1, b1, b2 etc.) are an indication for the importance of the corresponding parameter (IVO, IVC, etc.). *Table 3* shows the found coefficients. Where no value is given the coefficient is too small or zero and the parameter is, therefore, statistically unimportant.

Parameter	**Criterion to be modeled**			
	Case 1: Time until wastegate opens	Case 2: Time to reach 4000 RPM	Case 3: Engine acceleration before wastegate opens	Case 4: Engine acceleration after wastegate opens
Constant	3.49	1.93	446	731
IVO				
IVC	0.426	0.463	-53.5	-21.3
EVO	-0.143	0.0502	22.2	35.4
EVC	-0.0578	-0.0741	9.92	
IVO^2				
IVC^2	0.279	0.308	-25.2	-18.1
EVO^2	0.0877	-0.0748	-13.3	-31.8
EVC^2				
IVO*IVC				
IVO*EVO				
IVO*EVC				
IVC*EVO			-3.5	
IVC*EVC				-10.6
EVO*EVC				

Table 3: Coefficients for the Taylor functions

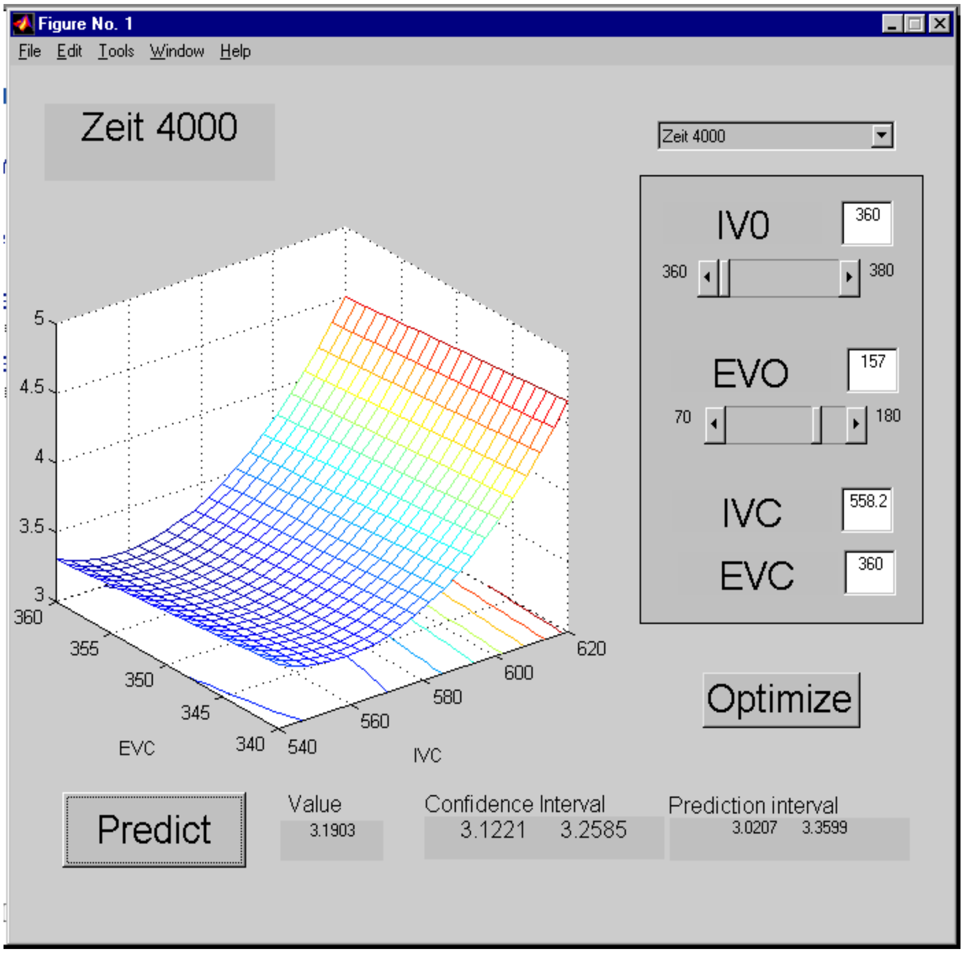
Figure 6 shows plots of the model fit where the 25 simulation results are compared to the values the statistical model produces. It can be seen that for all four criteria the dots fit the lines quite well and the quality of the model is satisfactory.

In order to visualize the models an interactive GUI that has been developed in MATLAB by DAIMLERCHRYSLER is used. *Figure 7* shows a screenshot where a 3D surface illustrates the time the engine needs to reach 4000 RPM as a function of IVC and EVC. On the right side of the screen it is possible to change IVO and EVO by moving sliders. The 3D plot is updated automatically.



The same tool is used to find the best parameters for the four criteria. *Figure 8* shows the valve lift curves that result.

Figure 6: Plots of the model fit



Zeit 4000

Zeit 4000

IV0 360

360 380

EVO 157

70 180

IVC 558.2

EVC 360

Value 3.1903

Confidence Interval 3.1221 3.2585

Prediction interval 3.0207 3.3599

Figure 7: GUI for visualization and optimization

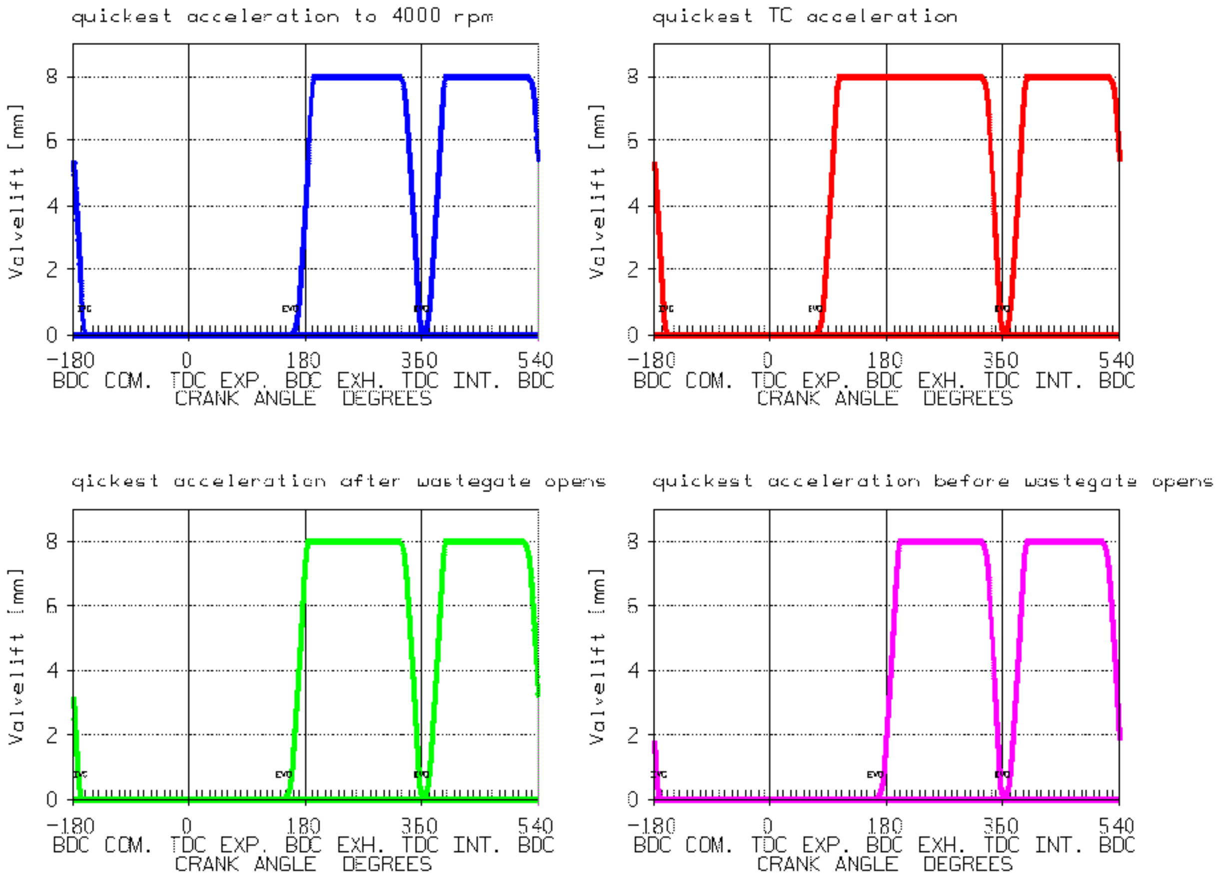


Figure 8: Optimized valve lift curves

Optimized engine acceleration

Figure 9 compares the engine accelerations simulated with the optimized valve lift profiles and the standard acceleration. Illustrated are engine speed, tc speed and fresh air flow, engine torque, boost pressure and pressure at turbine inlet.

The red curve represents the case for the shortest time until the wastegate opens. It can be seen that the early EVO leads to higher temperature and pressure at turbine inlet and therefore results in the desired quick TC response. The negative effect of early EVO is illustrated by the curves for engine torque. Due to the incomplete expansion in the cylinder the piston work is substantially reduced. This cannot be compensated for by the increased boost pressure. The torque curve reaches only the values of the other cases at t=1.5s but does not lead to higher values. After the wastegate has opened it makes as expected no sense to practice early EVO. Since the boost pressure is limited it is useless to offer more enthalpy to the turbine. This leads only to further wastegate opening.

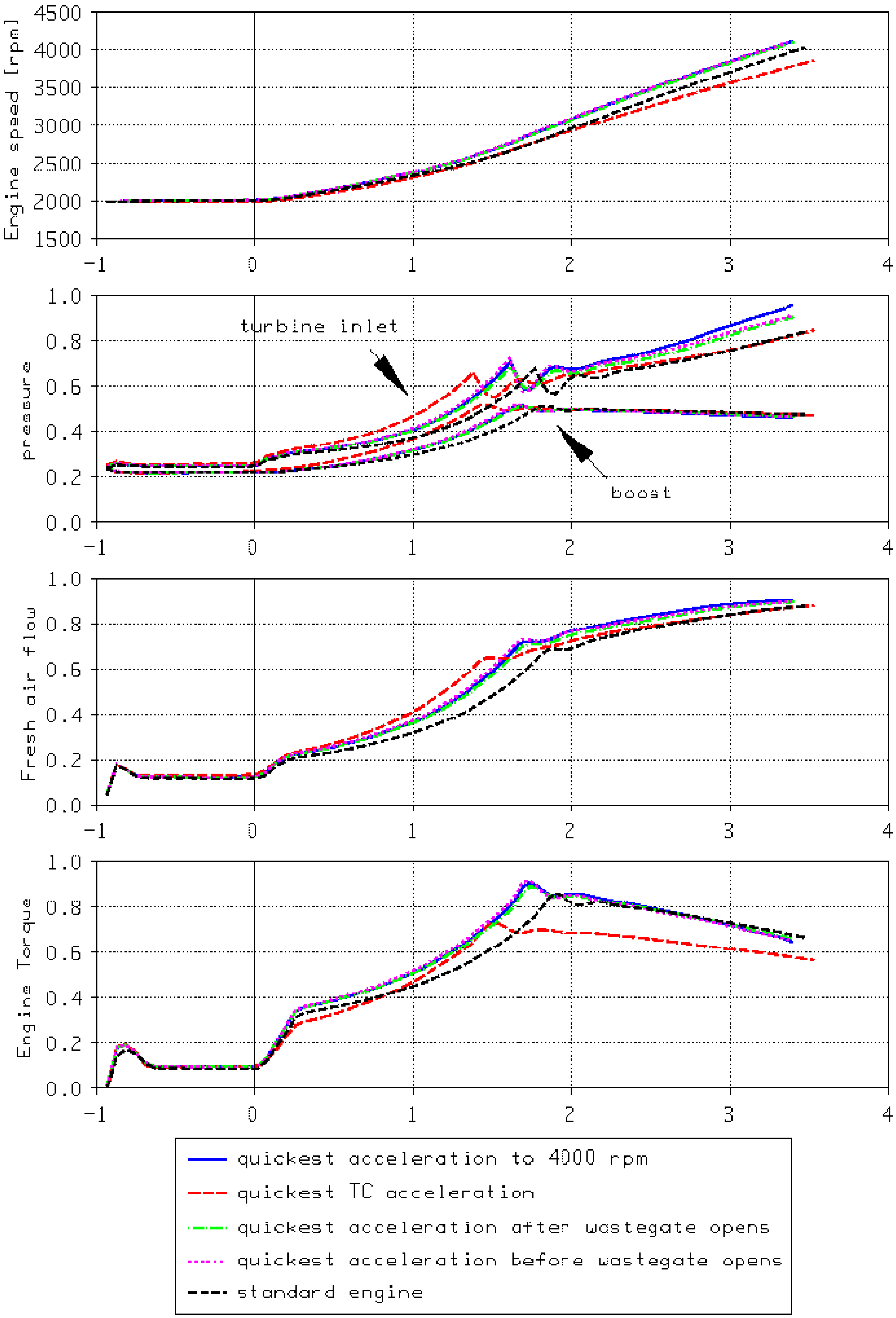


Figure 9: Transient engine behavior, standard and optimized valve timings

In fact the quickest accelerations before and after wastegate opens have both a quite late EVO in common. Due to the fast valve opening compared to camshaft driven systems the exhaust gas leaves the cylinder more easily and the EVO timing can be set quite late. It can be seen (figure 8) that before wastegate opens a slightly later EVO is recommendable. Since this is the timing for the initial seconds of the transient

the engine speed is still low and therefore so is the mass flow. Later after the wastegate opens when the engine speed rises longer opening time is advantageous to pump the exhaust gas out of the cylinder.

The same effect can be seen on the intake side. Before wastegate opens at low engine speed IVC should take place only about 10°CA after BDC. Later when the wastegate has opened a shift to slightly later IVC is recommended.

The simulation results show that only small changes in IVC and EVO are necessary during this transient. Since the inertia of modern TC are quite low the time to reach the boost pressure limitation is short. Therefore it is difficult to make use of a quicker TC response when on the other hand engine torque is lost. Perhaps in a transient that starts at very low TC speed it makes sense to accelerate the TC by early EVO for a certain time and then shift to a normal EVO and make benefit from the higher boost pressure.

Conclusions

The combination of WAVE and DOE provides a very quick method to obtain the potential benefits of a fully variable valve timing system for the acceleration of a turbocharged diesel engine. It is possible to get a deeper understanding for the thermodynamic effects from two approaches: the mathematical/statistical side as well as the engineering side. The use of WAVE on a well known engine model produces the reliable basis that is necessary to predict a transient performance.

The simulations found that the biggest benefits in acceleration result from optimized IVC. The effect of early EVO can clearly be seen in a quicker TC response but does not lead to quicker engine acceleration. The loss of piston work cannot be compensated for by higher boost pressure and increased fuel injection.

References

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