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Optimized Driving Cycle Oriented Control for a Highly Turbocharged Gas Engine

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Abstract

The article is focused on a 1-D drive dynamic simulation of a highly turbocharged gas engine. A mono fuel CNG engine has been developed as a downsized replacement of the diesel engine for a medium size van. The basic engine parameters optimization is provided in a steady state operation and a control adjustment is applied to a dynamic vehicle model for a transient response improvement in highly dynamic operation modes of the WLTC (world light duty test cycle), selected for investigation. Vehicle simulation model with optimized control system is used for driving cycle fuel consumption and CO_2 emissions predictions compared with the basic engine settings.

Introduction

Increasing number of cars operated specially in and in close vicinity of the city negatively influences living environment. Implementation of emission limits applied to combustion engines significantly improves air quality, however, the number of vehicles increases and air quality, especially PM emissions, get worse. Despite the higher complexity of engine design, the engine's reliability was maintained at the same level. With the discussion about the planet global warming there was a new goal to achieve - reduce CO_2 emissions. Amount of CO_2 in engine exhaust gas is dependent on carbon fraction in fuel, which is oxidized during combustion process, and on a total fuel consumption. Car manufacturers were given a task and challenge to decrease specific CO_2 emissions to certain fleet average values. Research and development centers look for the new ways, how to increase total efficiency of cars by optimizing every subsystem in the drivetrain and also vehicle aerodynamics.

Engine downsizing was chosen as an effective approach of engine total efficiency improvement [1]. The utility vehicles and vans are, so far, relatively not influenced by downsizing in contrary to passenger cars. It was observed from previous research experience [2], [3] that engines, optimized in stationary operation with high level of boosting, do not offer sufficient acceleration response in transient modes. Therefore, acceleration control correction is implemented in the ECU.

Since CO_2 production is a product of complete oxidation of carbon contained in fuel, the second option is to use a different type of fuel, with lower carbon content than conventional fossil fuels. The fuel that, to some extent, fulfils this requirement is natural gas [4], [5], and the real world emission studies confirm that [6], [7], [8], [9], [10].

Natural gas is currently the most practical fuel, however, the ultimate solution from latter reasons is carbonless fuel, e.g. hydrogen. The usage of natural gas in engine, with the same brake efficiency as in a diesel engine, reduces CO_2 emission by 25% [11].

One part of the European international project GasOn [12] addresses reduction of the van CO_2 production to the level of a middle class passenger car. Both principles mentioned above were applied to the base vehicle, which was a nine seat van, powered by a diesel engine. The high performance engine reached limits of today downsizing and adaptation of gaseous fuel was applied in the engine design.

A base 4 cylinder diesel engine of 1.6 l displacement was redesigned for a port fuel injection of compressed natural gas (CNG) and the intake ports configuration for high level of cylinder charge turbulence and good mixture homogenization with a new shape of the combustion chamber with reduced compression ratio to 13.5:1. Engine was equipped with a variable valve-train timing of intake and exhaust valves, intake cam profile switching and a low pressure EGR system. Engine boosting device was a single stage turbocharger with variable geometry turbine. Emission system is based on a relatively simple three way catalyst and a stoichiometric operation within the full range of complete operation map.

VEHICLE SIMULATION

At early phase of the project, a simulation model of the vehicle, including the engine, was developed. The goal was to support the design and a calibration of the engine and the entire vehicle. GT-Suite simulation software was used for vehicle simulation. 1-D engine simulation as a basic function of GT Suite [13] can be extended to complex simulation models, including various systems, such as cooling system, lubrication system, control system etc. Transient nature of 1-D simulation also allows simulation of a complete car drivetrain and prediction of the vehicle fuel consumption in artificial and real world drive cycles.

Definition of Engine Full Load Parameters

A full load curve was set to cover the needs for a modern vehicle drivetrain. The limits were selected based on results of previous project, validated by the single cylinder experiments [15]:

- Engine torque 180 Nm (BMEP 14 bar) at 1000 rpm
- Peak engine torque 380 Nm (BMEP 30 bar) at 2000 rpm
- Maximum power 125 kW (78 kW/l) at 5500 rpm.

High demand for low speed torque and relatively high maximum engine power set quite challenging task for the single stage turbocharger. Nevertheless, low technical complexity of the boosting system was required.

Specification of engine design brought limits and constraints to the engine optimization. A robust cranktrain of the original diesel engine is able to work under significantly higher peak combustion pressures than it is typical for downsized gasoline engines. This operation would not be achievable without knock resistant fuel such as natural gas, with octane number around 120 depending on CNG composition. Operational constraints given by engine and turbocharger manufacturers were:

- Compressor outlet temperature T_2
- Turbine inlet temperature T_3
- Turbocharger speed $T\text{-}RPM_{MAX}$
- Peak cylinder pressure p_{MAX}
- Compressor surge limit was set according to the steady state compressor map.

The cam design was derived from existing cam profiles of similar engine. The main task was to find appropriate turbocharger size to meet the target full load curve. Small turbine can help to achieve the target BMEP in low and medium engine speeds. On the other hand, a larger turbine gives a sufficient flow reserve in the full power mode, but it cannot give a sufficient level of the boost pressure in low engine speed. Lowering the engine speed for the maximum power eliminates this compromise and the gas engine characteristics approach the characteristics of a diesel engine. Transition of the speed of the maximum engine power output towards lower values provides reduction of the BSFC due to a positive effect of downspeeding reflected in lower cycle frequency. Therefore, the speed of 3500 rpm was selected as speed for maximum power. Equal speed range of the gas and the diesel engine provides the ability to implement a new engine with original transmission gear ratios.

Vehicle Simulation Model

Vehicle simulation model for highly boosted engines requires an engine model capable of prediction of engine and turbocharger dynamics. Simple map-based models are not capable of the prediction of turbo-lag phenomenon during vehicle acceleration and therefore, the fuel consumption prediction can serve only as a first approximation. In contrary, the 1-D engine model takes into account the influence of different control strategies.

Engine 1-D Simulation Model

1-D simulation models are extremely capable because of their width in system description, including especially engine control system. On the other hand, the computational cost increases together with simulation complexity. The level of complexity of original 3-D model (Figure 1) has to be adapted to desired result accuracy. The physical part of a calibrated engine model is usually simplified for vehicle simulations, since only transient behavior of engine itself is critical in case of drive cycle simulation. Intake and exhaust manifolds (Figure 2) are simplified and transformed to 0-D objects (Figure 3). However, the detailed physics (e.g. model response as heat transfer, flow resistance etc.) can be in a form of multidimensional lookup tables. Simplified models, or so called Fast Running Models (FRM) [13], still very well predict transient behavior. Their computation speed is typically 3-5 times slower than real time, while engine parameters prediction is at acceptable level, especially at high engine load – Figure 4, Figure 5 and Figure 6. In our case, the simplification level of flow volumes of pipes were down from 242 for 1-D model to 56 for FRM. Compared to complete 1-D models, the FRMs are approximately 10 times faster and more suitable for multiparameter optimizations.

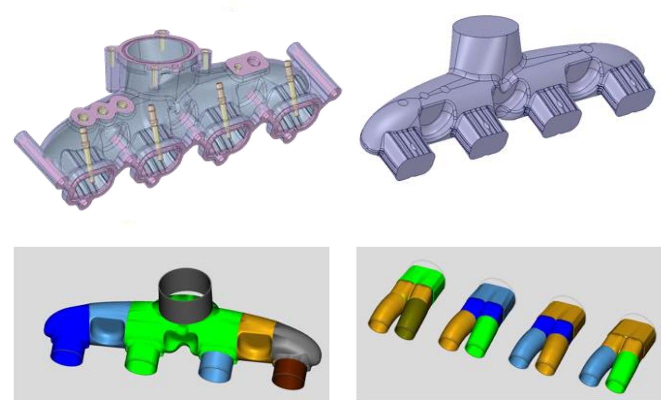

Figure 1 Transformation of 3-D CAD data to 1-D thermodynamic model.

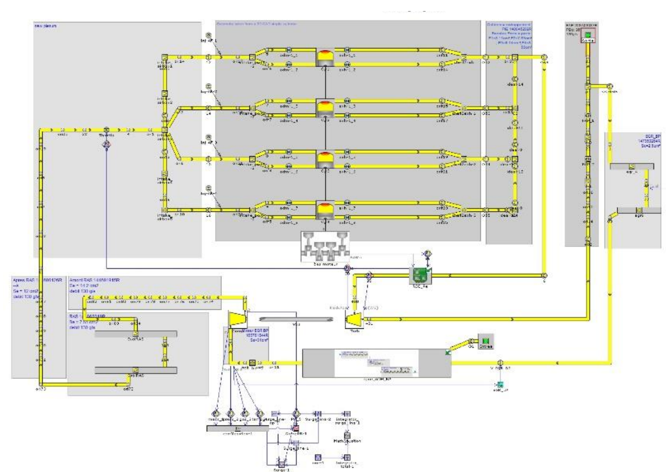

Figure 2 Detailed 1-D engine model modelled in GT-Suite.

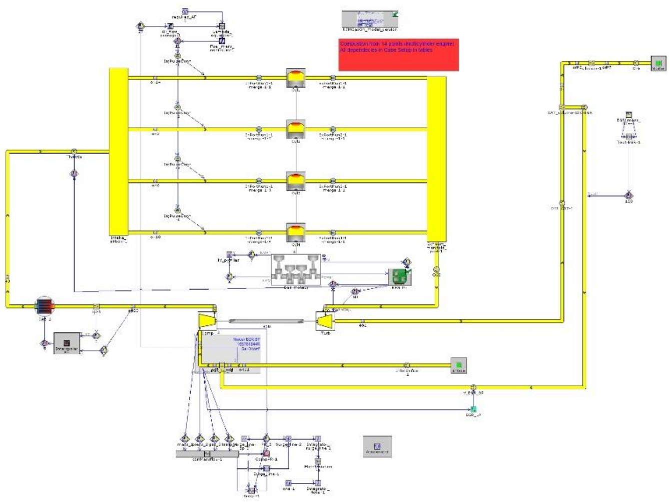

Figure 3 Fast running engine model modelled in GT-Suite.

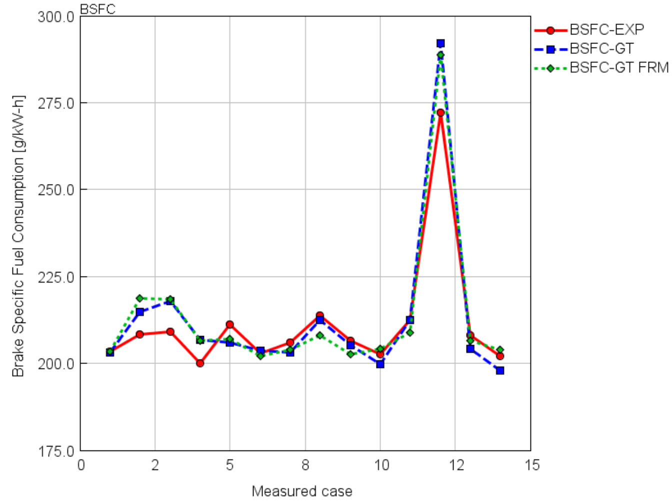

Figure 5 Comparison of Brake Specific Fuel Consumption between experiments (red), fully 1-D model (blue) and the FRM (green).

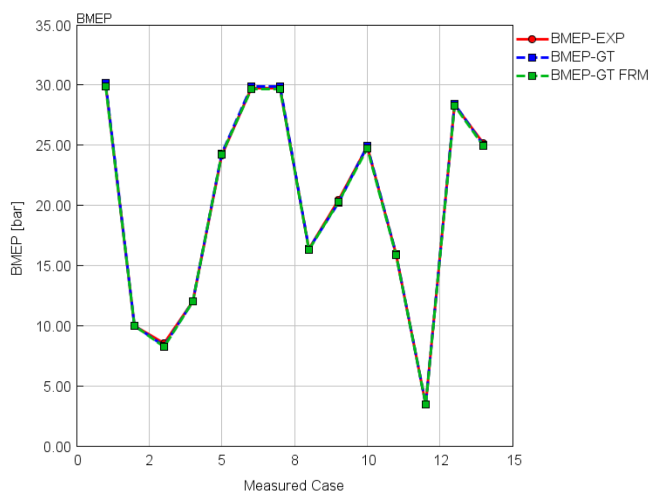

Figure 4 Comparison of Brake Mean Effective Pressure between experiments (red), fully 1-D model (blue) and the FRM (green).

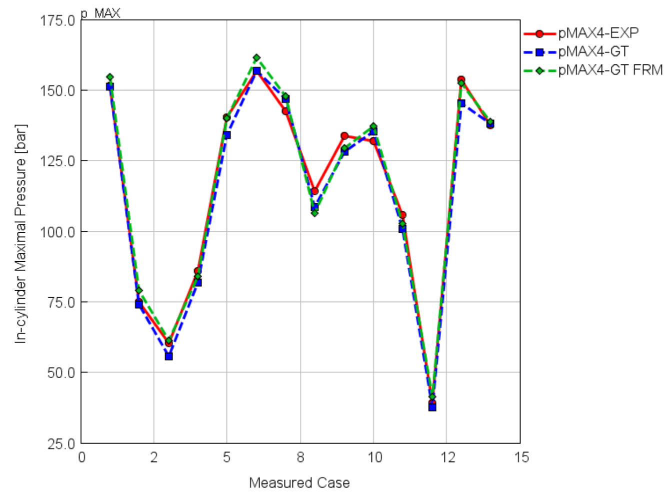

Figure 6 Comparison of maximal in-cylinder pressure between experiments (red), fully 1-D model (blue) and the FRM (green).

Port fuel injection of natural gas was implemented to the engine model. Fuel was defined as a mixture of species according to natural gas composition. The software provides chemical equilibrium calculation for combustion and CO_2 emission in exhaust gases is properly predicted.

A large optimization of engine setting in static operation condition performed with help of FRM. Boost pressure level, valve timing (VVT), combustion timing and EGR percentage were optimized for the whole engine operation map using genetic optimization algorithm. Switchable intake cam profiles enables change of short profile with the valve lift opening of 154 crank angle degrees (defined at 0.7 mm lift) to long profile corresponding to the opening angle of 187 crank angle degrees. VVT optimization was performed for both cam profiles in the whole engine operation map (Figure 7 and Figure 8). BSFC difference of engine with short and long intake cam profile (Figure 9) indicates region where it is beneficial to use certain intake cam profile (Figure 10). Steady state optimization often leads to Millerization of valve-train, as was confirmed in previous work [3]. Early intake valve closing for Millerization is supported by short intake cam profile (Figure 10). Optimized parameters were set into the look-up tables dependent on engine load and engine speed, in similar way as is done in a real engine control unit (ECU).

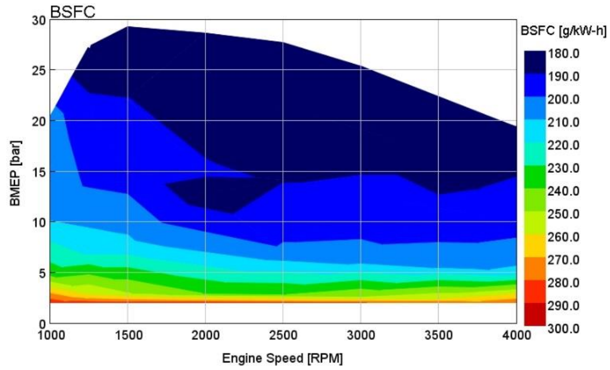

Figure 7 Brake specific fuel consumption for optimized engine used with short (154 CA) intake cam profile.

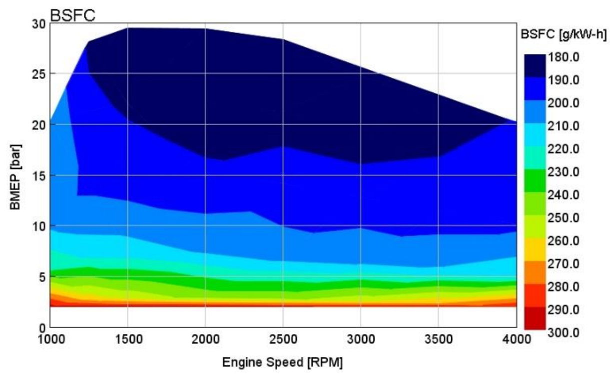

Figure 8 Brake specific fuel consumption for optimized engine used with long (187 CA) intake cam profile.

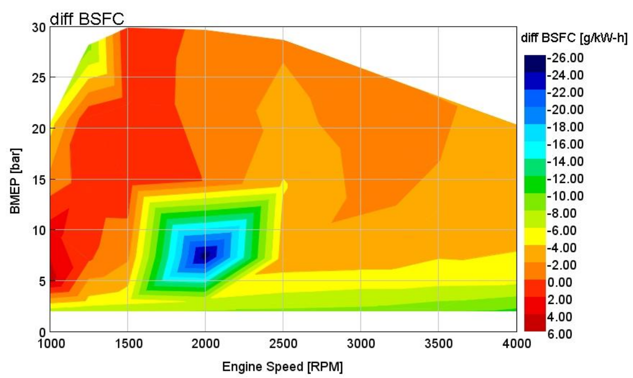

Figure 9 Difference of long cam profile BSFC map subtracted from short cam profile BSFC map.

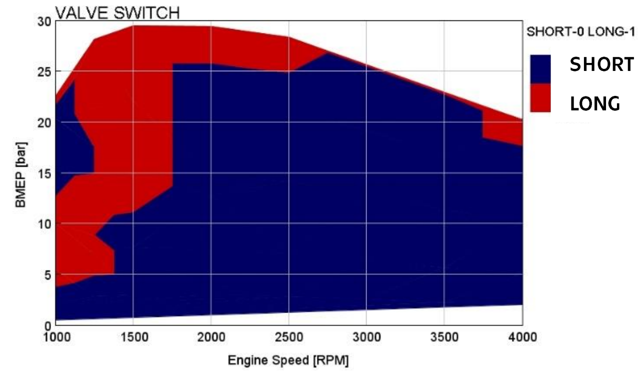

Figure 10 Intake cam profile switch operation map used for engine control.

Basic engine control system consists of throttle controller and controller of turbine geometry position (VGT). Typical controllers operation consists of two phases, where the engine is throttled in low engine load, while VGT is open for minimal engine pumping losses. In high engine load, the boost pressure is controlled by VGT with fully open throttle. This configuration (Figure 11) is convenient from the fuel consumption point of view. Nevertheless, open VGT for low engine load results in low turbocharger speed and engine transient operation is influenced by strong turbo-lag. Transient operation capability can be improved by keeping turbocharger speed high at low engine load with closed VGT (Figure 12). Closed VGT causes higher turbine upstream pressure and higher engine pumping work together with increased BSFC. Application of this control modification should be only when turbo-lag influences the vehicle acceleration ability in homologation drive cycles.

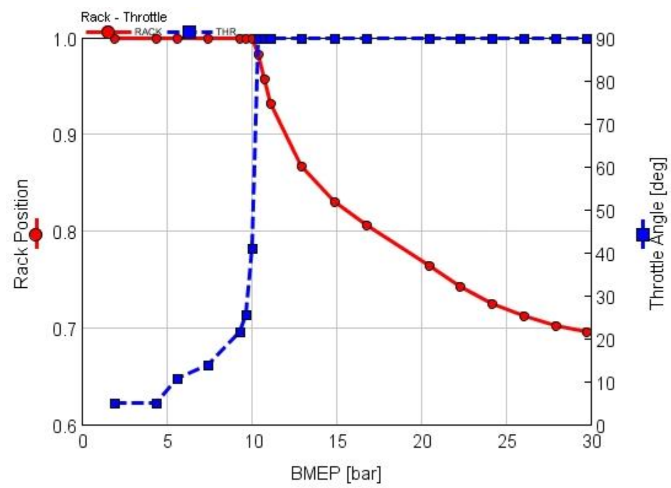

Figure 11 Throttle and VGT controllers behavior with increasing engine load.

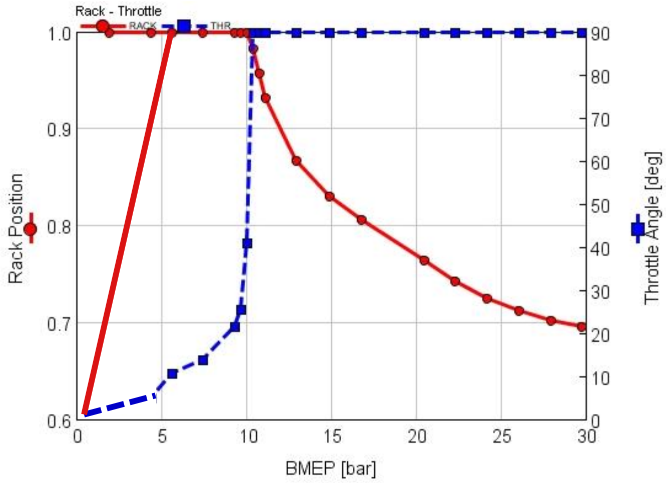

Figure 12 Modified throttle and VGT controller behavior with increasing engine load for increased turbocharger speed in low engine load.

Vehicle Simulation Model

Figure 13 shows a GT-Suite vehicle simulation model that implements a FRM engine model (highlighted in red), a transmission and vehicle object (green) and an electric starter with the battery (in orange) into one simulation model. Engine powertrain includes all inertia masses and frictional losses. Vehicle model is defined by important parameters for longitudinal vehicle dynamics description, such as vehicle mass, frontal area, drag coefficient, wheel characteristics and transmission gear ratios and inertias.

Similarly to the real vehicle, the control system supplements the physical part of the model (complete drivetrain). The control system consists of the two main parts: the driver (blue) and the engine control unit. The driver is represented by a set of PID controllers and shifting algorithms for a manual transmission. Optional start/stop system was taken into account as a measure to reduce driving fuel consumption. The electric system of the vehicle was not modeled in detail. The battery state of charge was not monitored.

Vehicle drive cycle is prescribed as a vehicle speed profile, which is the main driver model input. Two drive cycle profiles were simulated - an NEDC and a new WLTC homologation test. Both drive cycles are described as a speed profile in time. NEDC transmission shifting is also described as a sequence of changes in time. WLTC transmission shifting strategy is part of directive [14] and shifting points have to be generated before drive cycle procedure, based on the engine full load torque curve and vehicle characteristics as a vehicle size, vehicle mass and transmission gear ratios. The shifting strategy was derived using an MS Excel based in-house developed software routines [16].

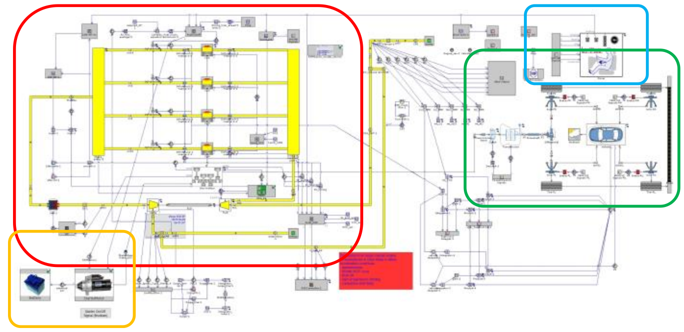

Figure 13 Vehicle simulation model: RED area– FRM engine model, GREEN area– vehicle model, BLUE area– driver model, ORANGE area – engine starter model.

Engine control unit was simulated by a set of control algorithms for:

- Calculation of fuel amount based on cycle averaged intake air mass flow rate, sensed upstream the compressor and EGR mixer. Full time lambda = 1 control was required. Injected fuel quantity was equal for all four cylinders.
- Fuel cut-off during engine deceleration and engine over speed limit.
- Engine idle speed was controlled by the throttle position.
- Valve timing and intake cam profile switching in dependence on engine speed and actual boost pressure (Figure 10).
- Engine power controller was based on a combination of the throttle and VGT (Figure 11). Demanded boost pressure results from the map, based on engine speed and accelerator pedal position (Figure 14) results in requested engine power.
- EGR mass flow rate control system based on an engine speed and boost pressure.
- Optional start/stop function activated, when the vehicle was stopped and idle controller was active.

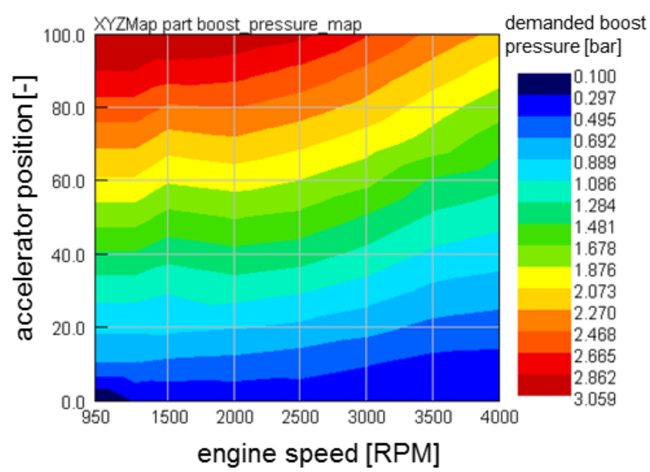

Figure 14 Transformation map between accelerator position and demanded engine power by map of required boost pressure for throttle and VGT controllers.

Basic engine control was derived from the results of static optimizations for complete range of engine speed and load. It is necessary that a corrections of control system need to be made for engine transient operation. NEDC drive cycle is not very dynamic and no correction was necessary to pass the vehicle speed profile of the test cycle. More dynamic WLTC simulation brought out acceleration issues especially in three parts, where required acceleration was the highest. High performance turbocharged engines suffer with a time lag of boost pressure during acceleration, especially in low engine speed as mentioned in introduction section.

Acceleration control system corrections were implemented into the vehicle simulation model. At this stage, a simple correction algorithm was used. If the actual boost pressure is below target value by 0.35 bar (target value corresponds to the required engine power), the following corrections are applied to maximize turbocharger turbine power output by:

- EGR valve was fully closed.
- Combustion phasing was retarded by 15°CA to expansion stroke.
- Valve timings and intake cam profile – values for maximal volumetric flow rate and temperature upstream turbine was set to short intake valve lift profile, intake valve opening at 360°CA after firing TDC and exhaust valve lift opening at 130°CA after firing TDC (both timings defined at 0.7 mm valve lift).

The comparison of drive cycle results without and with acceleration control system corrections is displayed in Figure 15 and Figure 16. The figures show 40 seconds period with a transition from the low speed to the high acceleration mode. The WLTC gear shifting is prescribed in a fixed time and there is no dependency on real engine speed. Due to turbocharger inertia and its insufficient speed response, well known as turbo lag, the engine is not capable of following preset speed profile (Figure 15). Transmission shifting directive also allows engaging of the clutch when the engine speed is near the idle speed. Since the transient vehicle performance is sensitive to the clutch control, this clutching was implemented into the control system as well. The clutching increases engine speed helps to increase turbocharger speed at the beginning of acceleration phase. Therefore, the acceleration control strategy, focused on maximal engine volumetric efficiency together with already mentioned clutch operation, can minimize the turbo lag as can be seen from a turbo speed line in Figure 16, and enables successful acceleration from the low engine speed.

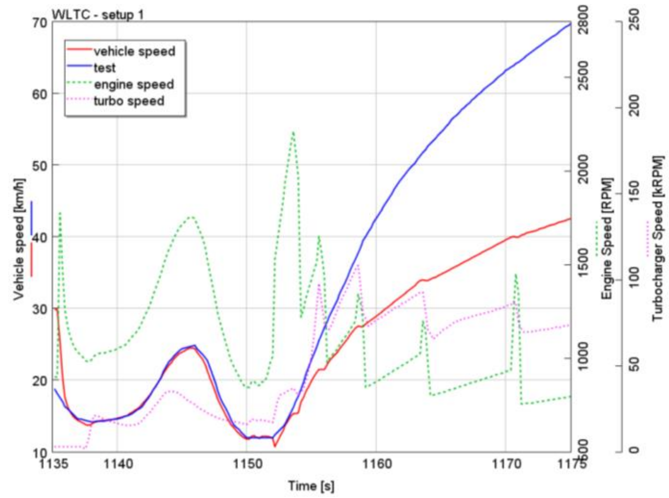

Figure 15 Vehicle speed in WLTC acceleration phase from 1135s without acceleration correction. Red line is simulated vehicle speed, Blue line is drive cycle speed, Green dashed line is engine speed, Magenta dotted line is turbocharger speed.

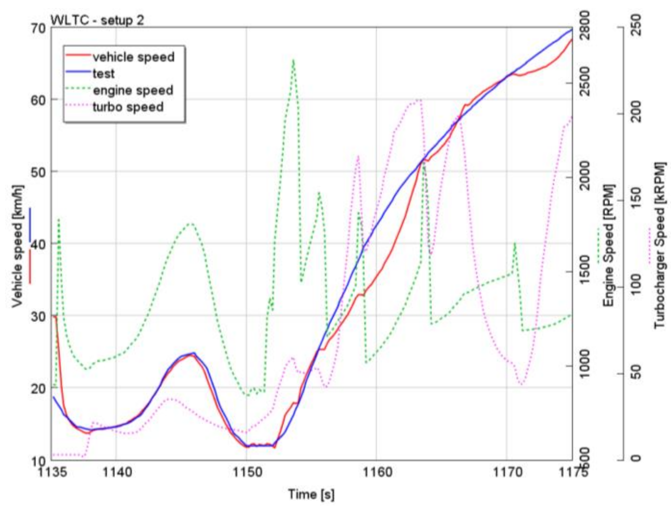

Figure 16 Vehicle speed in WLTC acceleration phase from 1135s with acceleration correction. Red line is simulated vehicle speed, Blue line is drive cycle speed, Green dashed line is engine speed, Magenta dotted line is turbocharger speed.

Drive Cycle Fuel Consumption Prediction

Drive cycle simulation was performed for both NEDC and WLTC test cycles. The simulation model does not take into account the engine and the catalyst warm up. Therefore, resulting vehicle fuel consumption might be a slightly optimistic in comparison to the real world.

Several simulation cases were calculated to find the benefit of implemented fuel saving strategies. The first function is a start/stop option. The second function is an application of optimized valvetrain timing. Non optimized valvetrain timing data was based on the test bench data measured in a laboratory of industrial partner. Optimized valvetrain data was obtained by a simulation. Engine optimization provided improvement of the fuel consumption. On the other hand, the dynamic capability was reduced (mainly because of so called "Millerization" and higher EGR rates). Already mentioned dynamic control correction was able to improve transient response during test cycle and maintain acceleration capability unchanged.

Drive Cycle Simulation Results

Fully dynamic vehicle model provides a detailed information on all vehicle system performances in the test cycle and the vehicle fuel consumption. Vehicle and engine speed during WLTC drive cycle start is displayed in Figure 17. In Figure 18 is shown results of highly dynamic part of WLTC drive cycle.

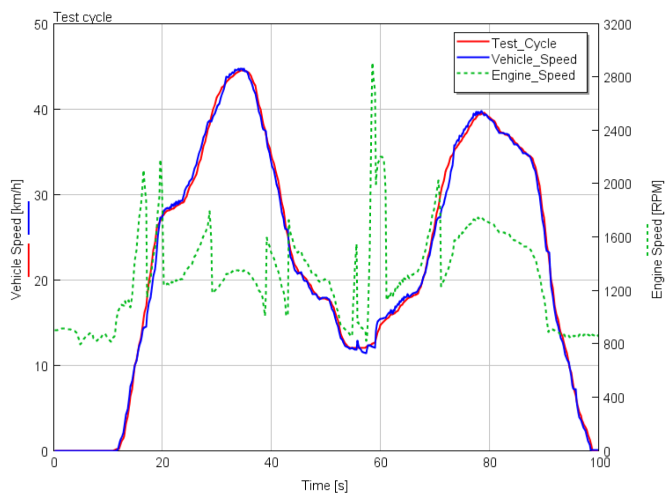

Figure 17 Result of vehicle simulation in low speed phase of WLTC. Red line is simulated vehicle speed, Blue line is drive cycle speed, Green line is engine speed.

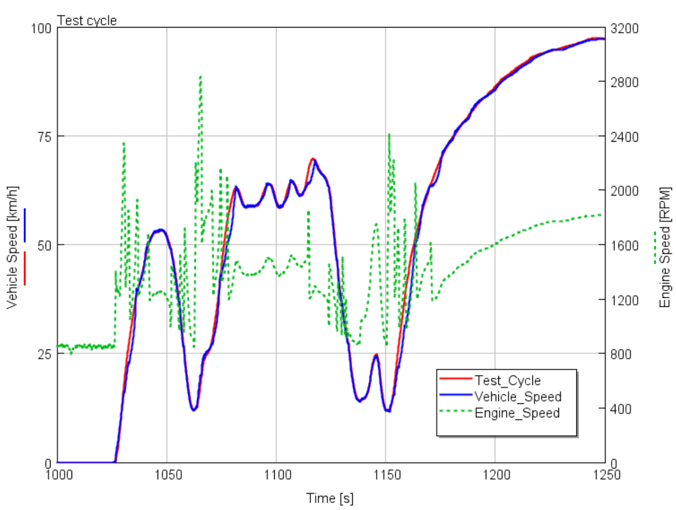

Figure 18 Result of vehicle simulation in high speed phase of WLTC. Red line is simulated vehicle speed, Blue line is drive cycle speed, Green line is engine speed.

Actual engine boost pressure in the first 100 s of WLTC drive cycle (Figure 17) according to demanded boost pressure is displayed in Figure 19. Throttle and VGT controller operation is displayed in Figure 20. Activation of acceleration control correction can be observed by sudden increase of both controllers reaction speed by gain switching of PID controllers, when boost pressure target was not reached. The PID settings were adjusted iteratively to get no significant oscillations.

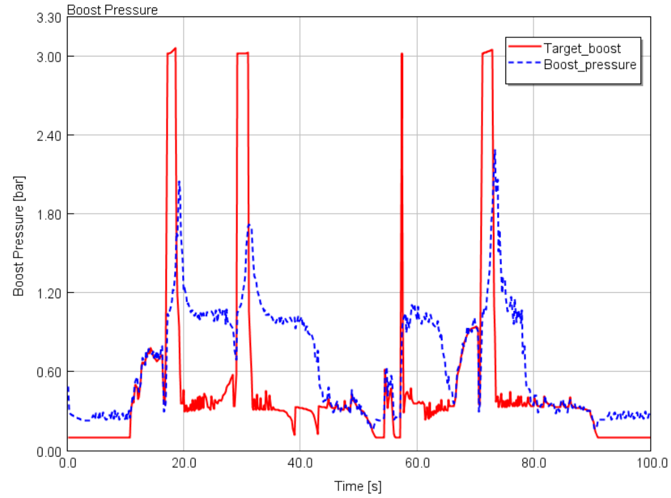

Figure 19 Demanded and actual boost pressure in low speed phase of WLTC test cycle simulation. Red line is target value and Blue dashed line is resulted boost pressure.

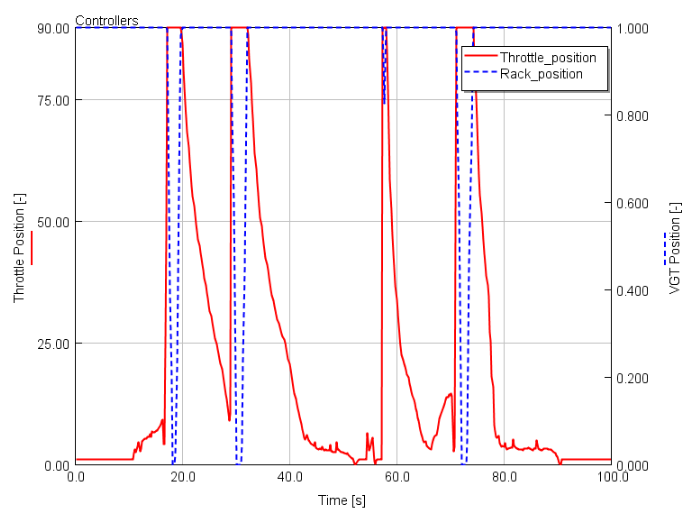

Figure 20 Throttle and turbine VGT controllers performance in low speed phase of WLTC test cycle simulation. Red line is throttle position and Blue dashed line is VGT position.

Table 1 Final specific CO_2 emissions [g/km] in simulated drive cycles for different variant of engine control strategies

Drive cycle	without start/stop	with start/stop	with start/stop + optimized cam timing
NEDC – CO_2 reduction	24.2 %	27.2 %	33.7 %
WLTC – CO_2 reduction	23.3 %	24.7 %	29.7 %

CO_2 emissions of the CNG-fueled vehicle related to the diesel-fueled vehicle in NEDC and WLTC driving cycles is summarized in Table 1. Optimized engine control system improved CO_2 emissions by almost 30 % in WLTC. Additional CO_2 emissions decrease can be achieved by application of the start/stop function. The start/stop function which saves fuel during engine idling is more suitable for NEDC where idling phase is more frequent. CO_2 emissions in WLTC can be improved by another 1.4%.

Summary and Conclusions

This article presents results of the vehicle simulations in drive cycles, which were performed during the complex project focused on the development of high performance gas engine. This work extends the engine development phase, where turbocharger design was defined for required engine full load curve.

Physical depth of simulation model includes thermodynamic engine model, longitudinal vehicle dynamic model with control system. Therefore, the simulation model can describe engine behavior, which affect real vehicle operation in dynamic drive cycle. Prediction capability of such complex models allows control system algorithms development in advance to the real engine optimization at the test bed. Calibrated engine simulation model transformed to fast running model is very well suitable for complex optimization tasks and for long-drive cycle simulations.

Acceleration capability of turbocharged engines optimized in stationary conditions for the best possible fuel consumptions are limited. Engine acceleration control setup has to be used for improvement of the engine dynamic behavior. Increased engine volumetric efficiency, together with exhaust gas enthalpy increase during acceleration phase, lead to reduction of turbo lag and shorter transient response time.

The vehicle simulation of drive cycles showed a potential of CO_2 emissions reduction by usage of the CNG as a fuel in a downsized engine for a medium size van. Optimized engine control system, together with start/stop function, predicts CO_2 specific emission decrese by almost 30 percent lower than the base diesel fueled vehicle. Resulted CO_2 specific emission is at a level for small passenger cars.

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Definitions/Abbreviations

BMEP	brake mean effective pressure
BSFC	brake specific fuel consumption
CA	crank angle
CNG	compressed natural gas
FRM	fast running model
NEDC	new European driving cycle
PID	proportional integral derivative controller
PM	particulate matter
TDC	top death center
VGT	variable geometry turbine
WLTC	Worldwide harmonized light duty vehicle test cycle