Dipl.-Ing. Carsten Weber, Dr.-Ing. Ulrich Kramer, Dipl.-Ing. Rainer Friedfeldt,
Dr.-Ing. Helmut Ruhland, Dr.-Ing. Frank Krämer,
Ford-Werke GmbH, Köln

Entwicklung eines speziell für den Betrieb mit Methan ausgelegten Verbrennungsmotors

Development of a New Combustion Engine Dedicated to Methane Operation

1 Summary

This paper describes the development and optimization of a new combustion engine concept dedicated to the exclusive operation with methane as fuel (natural gas, bio-methane, power-to-gas methane).
The engine is based on the new Ford 1.0L GTDI Ecoboost engine which has been presented to the public end of 2017.
It is designed to withstand very high combustion pressures in order to utilize the thermodynamic properties of methane at its maximum.

The engine delivers a maximum power output of 110 kW and provides a brake mean effective pressure (BMEP) of 30 bar over a wide engine speed range.
In order to achieve this high specific torque output and to minimize the fuel consumption at the same time the engine is equipped with an innovative technology bundle as follows:

- Fully variable mechanical actuation of the intake and exhaust valves to allow load dependent adjustments of the intake valve event length, primarily applied to increase the efficiency under part load operation conditions, and for control of the boosting system.

- Parallel sequential two-stage turbocharger system in order to achieve a high specific engine torque over a wide engine speed range

- Methane direct injection to meet the targeted torque output at low engine speeds and to support superior transient engine performance.

The engine concept and the design of the major engine components are described and first thermodynamic results are presented.

2 Introduction

The modern combustion engine is a highly developed, sophisticated system that transfers chemical energy into mechanical energy following the natural laws of thermodynamics. Currently it is under enormous pressure with regard to emissions and sustainability. The ongoing discussion often neglects the fact that efficiency and emissions are related to the chemical properties of the energy carrier used by the engine. For some fuels, efficiency and emissions are often inversely proportional to each other and cannot be optimized simultaneously.

A very promising alternative to classical liquid fuels is the use of methane as fuel. In addition to natural gas, especially the recent developments in the field of “power to gas” technology are predestined for a wider use in automotive applications.

Due to its favourable hydrogen to carbon ratio the use of methane as fuel immediately reduces the CO_2 emissions by significantly more than twenty percent vs. gasoline applications, just because of its chemical composition. Furthermore the very beneficial combustion behavior can be used to further improve the engine efficiency /2/, /7-9/.

All passenger cars currently in the market are using derivatives of gasoline engines. These engine designs have limitations regarding structural and thermal capability as well as restricted breathing characteristics, caused by the port gas injection, preventing an increase of power output and efficiency.

The introduction of the direct injection system is significantly improving the low end torque of the engine and therefore it enables a higher downsizing capability.

To harvest the full benefit of the high knock resistance of methane, an increase of the geometrical compression ratio is essential. As a consequence, significantly raised peak pressure and increased thermal load lead to much stronger requirements to the engine design.

As the overall target for the development of such a dedicated Methane Turbo Direct Injection (MTDI) engine a twenty percent CO_2 reduction in comparison to the actual best in class compressed natural gas vehicle has been set up, and this has to be demonstrated.

Figure 1 outlines the roadmap of this project. The reference engine with 1.6L displacement, distributed over four cylinders, is replaced by a dedicated 1.0L, three-cylinder engine. This already reduces the CO_2 emissions considerably due to the high degree of downsizing. Instead of a port fuel injection a recently developed direct injection system for methane /7-9/ is applied. In combination with an advanced parallel sequential twin charging system the vehicle performance is kept without any deterioration.

In order to further reduce gas exchange losses, a continuously variable valve lift (CVVL) system is installed to the intake side. A similar actuation system on the exhaust side enables a high accuracy control of the parallel sequential charging system.

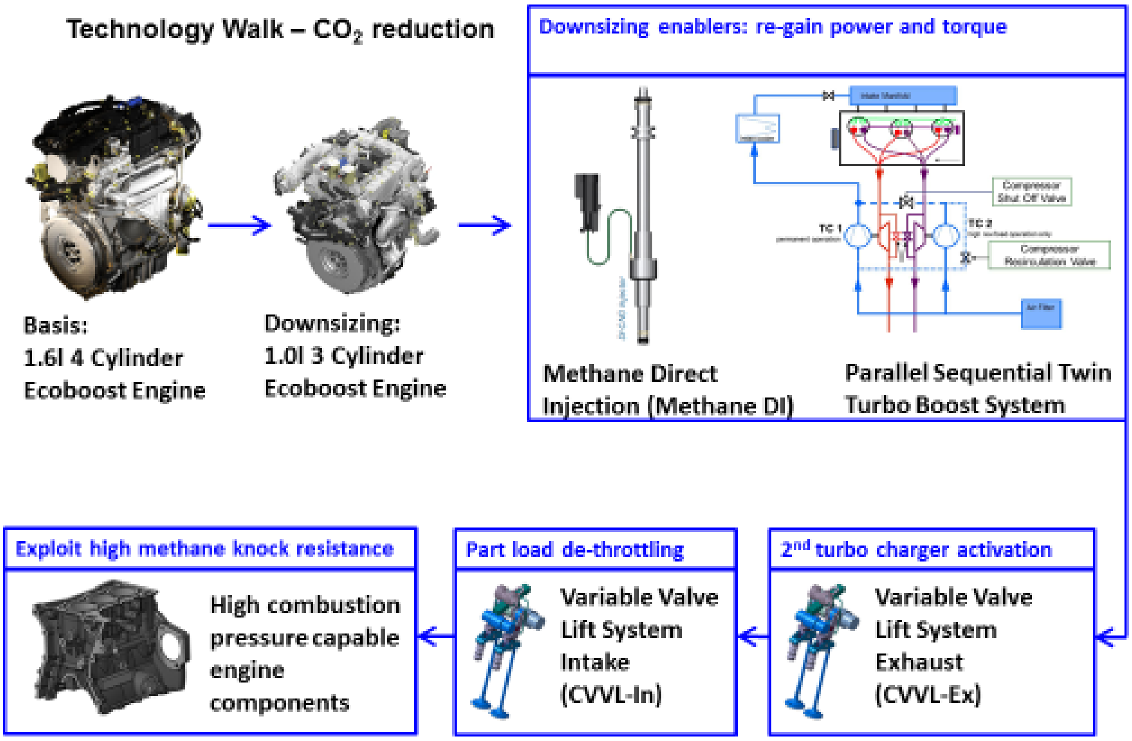

Figure 1: MTDi technology walk for 20% CO2 reduction

The recently presented new Ford 1.0L GTDI Ecoboost engine /1/ is the best foundation with regard to these targets. Nevertheless, especially the high level of combustion pressure requires extensive reinforcements of the entire engine structure to cope with 185 bar peak pressure.
To avoid any negative impact of these reinforcements regarding friction, all related dimensions of the crank train are retained. Moreover the design is protected for the introduction of a variable compression ratio (VCR) system, which is expected to further improve the efficiency by reduction of the maximum combustion pressure.

Table 1 summarizes the main targets for the development of the 1.0L MTDI engine.

Item	MTDi Target
Fuel	Methane
CR	13 :1
Peak Power	110 kW
Low End Torque @ lowest rpm	240 Nm @ 1500 rpm
Maximum Combustion Pressure Capability (avg. / peak)	pmax = 160 / 185 bar
Rated Engine Speed	6000 rpm

Table 1: MTDi engine targets

3 Engine Subsystem Upgrades

Cooling System

The targeted high power output of the engine and the methane combustion itself - with its early center of energy conversion mass - increases the thermal load on the engine components significantly. Furthermore, additional components, such as the bearing housing of the second turbo charger and a gas regulator need sufficient cooling. This requires a redesign and an enforcement of the cooling system as known from the 1.0 L Ford gasoline engine, to guarantee engine durability in the entire operating range. Significant CAE support has been utilized to enhance the coolant flow rate and improve local cooling efficiencies by optimizing the system design without increasing the pressure loss of the entire cooling system and therewith avoiding any negative impact on the engine friction.
The highly sophisticated functionality of the cooling circuit needs an adapted transient response of the system as well.
Beside other features, the plate thermostat is substituted by a wax actuated ball valve. Furthermore, the engine is equipped with an electrically actuated block water jacket valve to allow higher coolant temperatures during part load operation in order to reduce the piston-liner friction and therewith contributes to the overall fuel consumption saving.

Cylinder Head Design

The main dimensions of the MTDi cylinder head like bore spacing, bore diameter and valve sizes are carried over from the production engine. Although the intake duct length from valve to intake plenum is not extended, the split line between head and intake manifold is moved considerably outwards, due to the width of the fully variable valve train module.

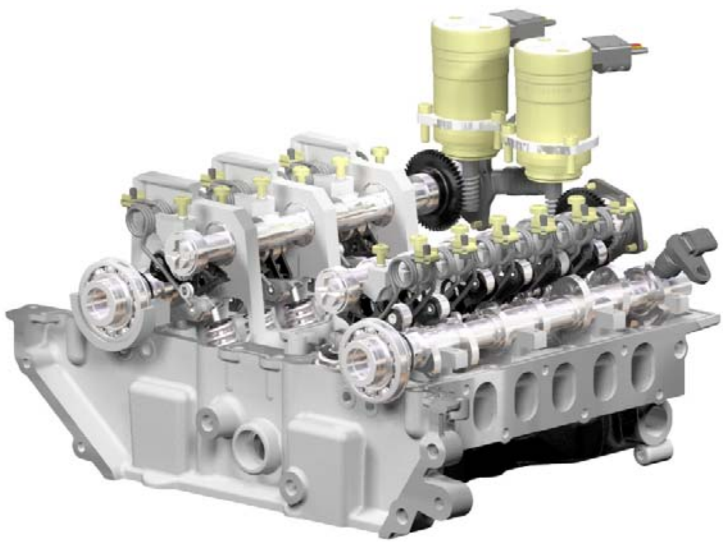

Figure 2: MTDi cylinder head

The all-new MTDi cylinder head depicted in Figure 2 includes two separate integrated exhaust manifolds, connecting to only one of two exhaust valves per cylinder and each

driving a turbocharger. A three-piece water jacket ensures sufficient cooling. Two additional cylinder head bolts per cylinder are required to adapt the engine to the increased combustion pressure level.
Due to the novelty of the three-piece water jacket in combination with the high specific power output of 110 kW/l, a thermal investigation has been required. CFD simulations have been used to optimize the flow velocity distribution in the water jackets (Figure 3 right). Furthermore, a conjugated heat transfer simulation has yielded minimum material temperatures within the cylinder head (Figure 3 left).

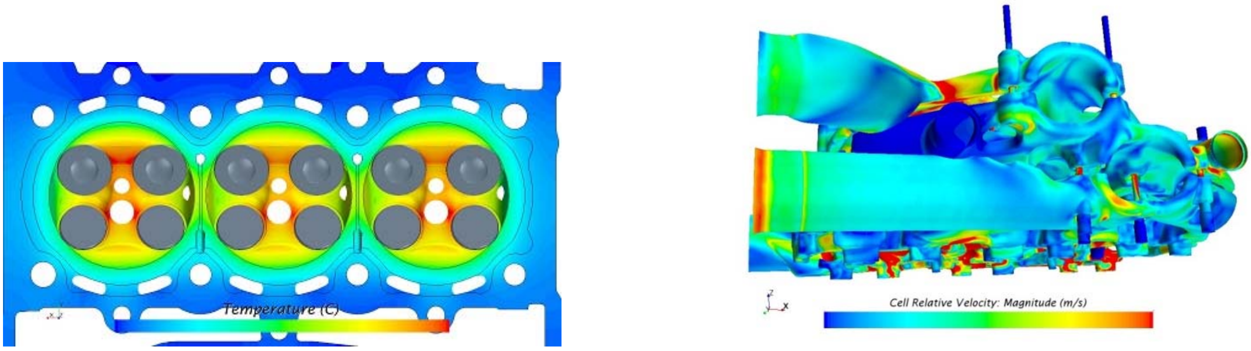

Figure 3: Cylinder head thermal analysis

Exhaust Port and Manifold Optimization

The design of all exhaust ports has been optimized for best mass flow rate by means of a new combination of CFD topology and shape optimization /6/ (Figure 4). With this method increased mass flow rates up to 11% are achieved, with positive effect on exhaust back pressure and hence knock behavior, as well as fuel economy.

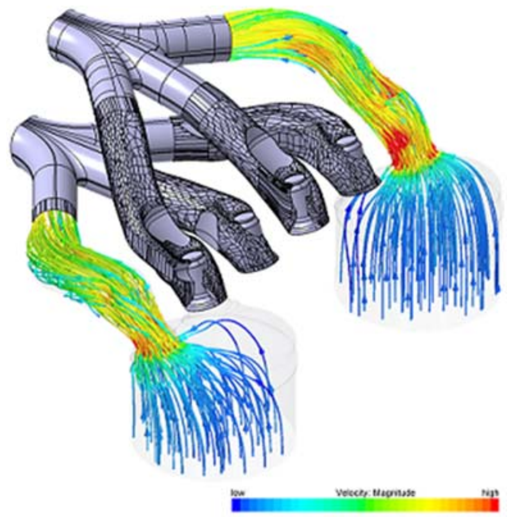

Figure 4: CFD based exhaust port optimization

Piston

The piston requires a dedicated design to cope with the high combustion pressure level and the accompanying high thermal load. Therefore, the piston is gallery cooled and equipped with a ring carrier. The design of the piston top land is the result of a CFD-based mixing optimization under the boundary conditions for a compression ratio of 13 (Figure 5).

Figure 5: 3D-CFD optimized piston crown

Cylinder Block and Crank Train

The main dimensions of the MTDi cylinder block are carried over from the 1.0l Ecoboost production engine architecture. However, in order to cope with the new requirements the block design needs to be reinforced in the area of the bulk heads and the liner top ends. Due to strong CAE support appropriate reinforcement has been achieved without increasing the weight of the cylinder block significantly.

The main dimensions of the new MTDi engine cylinder block are listed in Table 2.

Bore Diameter	71.9 mm
Bore Distance	78 mm
Material	Cast iron

Table 2: "MTDi engine – main cylinder block dimensions"

The block is designed as an open deck deep skirt block with machined inter-bore cooling cuts. A separate oil gallery is applied for the map controlled piston cooling jets serving the gallery cooled pistons. Combined oil return drains and blow-by passages are cast into the block, adjusted to the increased blow-by flow due to the high combustion pressures. Finally, bosses for the additional cylinder head bolts are introduced. A picture of the new cylinder block is shown in Figure 6.

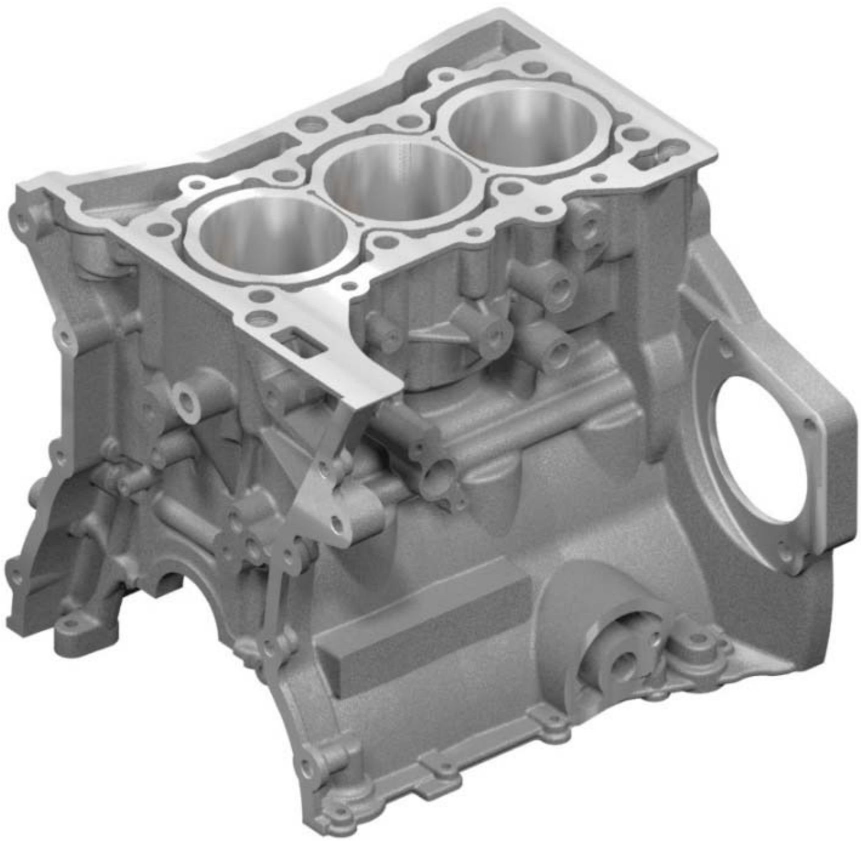

Figure 6: New MTDi engine cylinder block

Figure 7 the critical temperature areas are located between the cylinders on the combustion chamber side of the cylinder liner. The introduction of cuts into the bore bridges brings water close to the hot spots and keeps the material temperatures within acceptable limits.

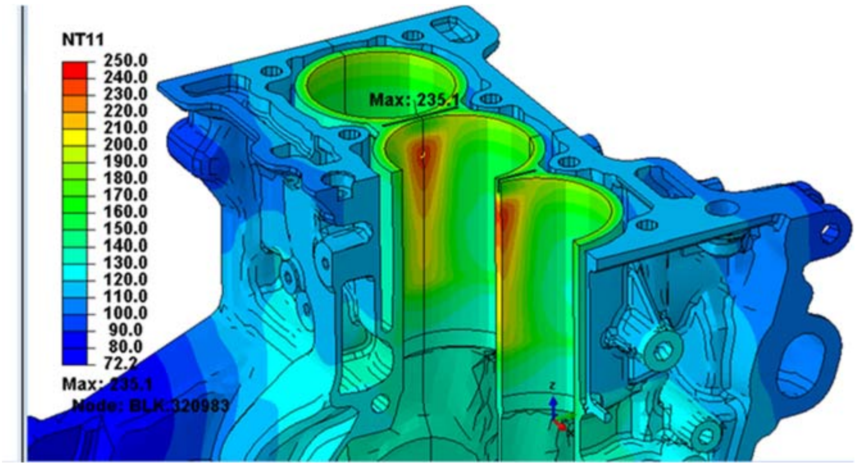

Figure 7: Cylinder block temperature analysis by CAE

To cope with the increased combustion pressures the cast iron crankshaft of the production engine is substituted by a cold forged high tensile steel crankshaft.

VCR System

Right from the beginning the engine has been protected to take a Variable Compression Ratio System. Its function and the working principle is described in /3, 4/. The application of this system to an engine with such limited package space - due to the narrow bore distance on the one hand side and extensively high combustion pressures on the other hand side - is the actual challenge.

Figure 8 shows the VCR system integrated into the bottom end of the engine. First test results will be presented at a later point in time.

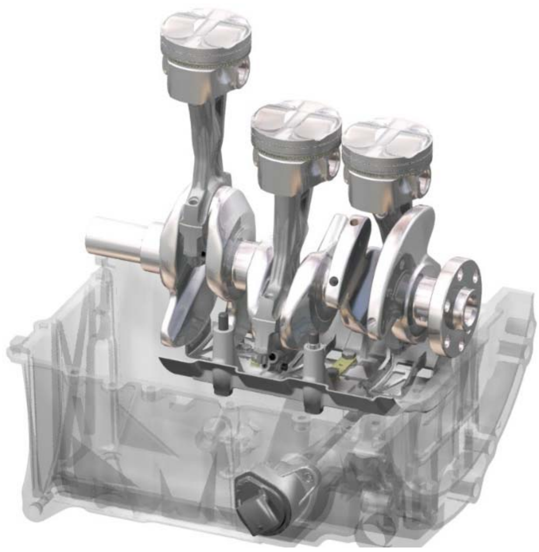

Figure 8: Cam-disk-actuator arrangement in oil pan

Methane Direct Injection System

The engine is equipped with a methane direct injection system that meets the packaging requirements and the performance targets as well /7-9/. Special care is taken to optimize the flow path from the injector inlet to the valve group using CFD.
According to experience a methane injector will be damaged when the combustion meets an open injector; hence this condition must be avoided in any case. For that purpose, safety features are implemented within the control strategy, which predict the latest possible end of injection (EOI).

Twin Boosting System

In order to achieve the high boost pressure level which is required for the high specific low end torque and peak power with methane fuel (240 Nm/l and 110 kW/l), a boosting system comprising two turbo chargers, a recirculation valve and a compressor shut-off valve is applied as shown in Figure 9.

While the 1st turbo charger (TC 1) is permanently operated, the 2nd turbo charger (TC 2) is only activated in case of high speed / high load conditions. TC 2 is activated by means of a new CVVL system, which enables a separate control of each exhaust valve per cylinder. In combination with the split IEM (Integrated Exhaust Manifold), it can be used for activation of the second turbocharger. Therefore, TC 1 is connected to the permanently actuated exhaust valves of the cylinders 1 to 3, while TC 2 is fed by the disengageable secondary exhaust valves of each cylinder.

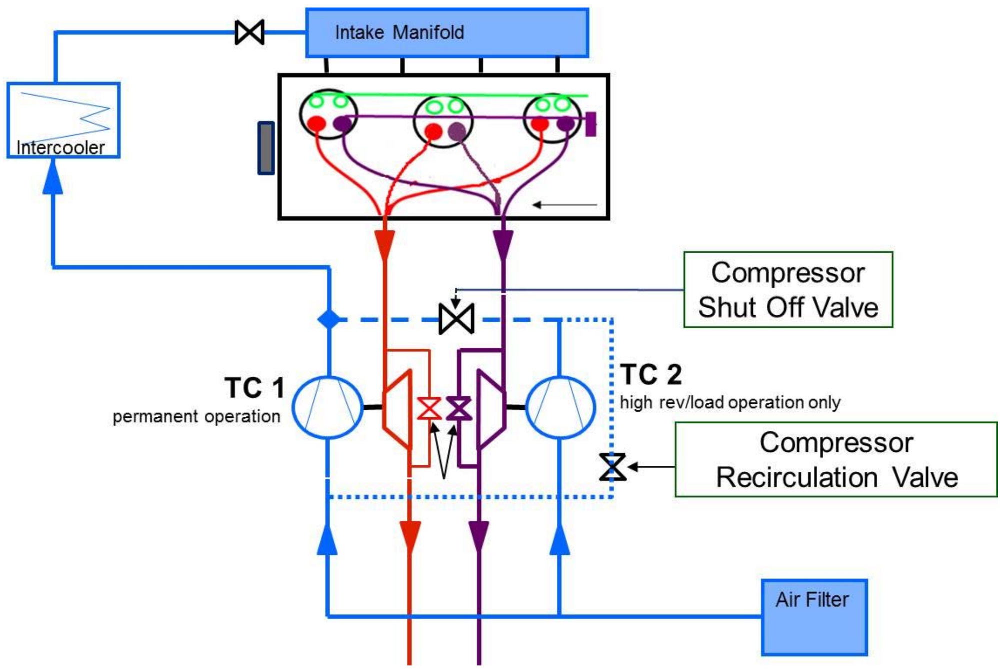

Figure 9: Twin parallel sequential boost scheme

With this configuration TC 1 has been laid out to supply considerable boost pressure already at low engine speeds. At high engine speeds and loads, TC 2 is activated to support TC 1. A smooth transition between one and two turbo charger operation of the engine is a challenge of this system. For this purpose a compressor shut-off valve and a recirculation valve are installed. These valves, in combination with the continuously variable exhaust valve event, enable a calibration of a satisfying transition behavior.
Based on initially assumed boundary conditions, a detailed turbo charger matching has been carried out by means of 1D CAE simulation. With the proposed turbo charger specification the target BMEP of 30 bar is predicted to be achievable in a wide engine speed range from 1,400 to 4,400 rpm according to Figure 10.

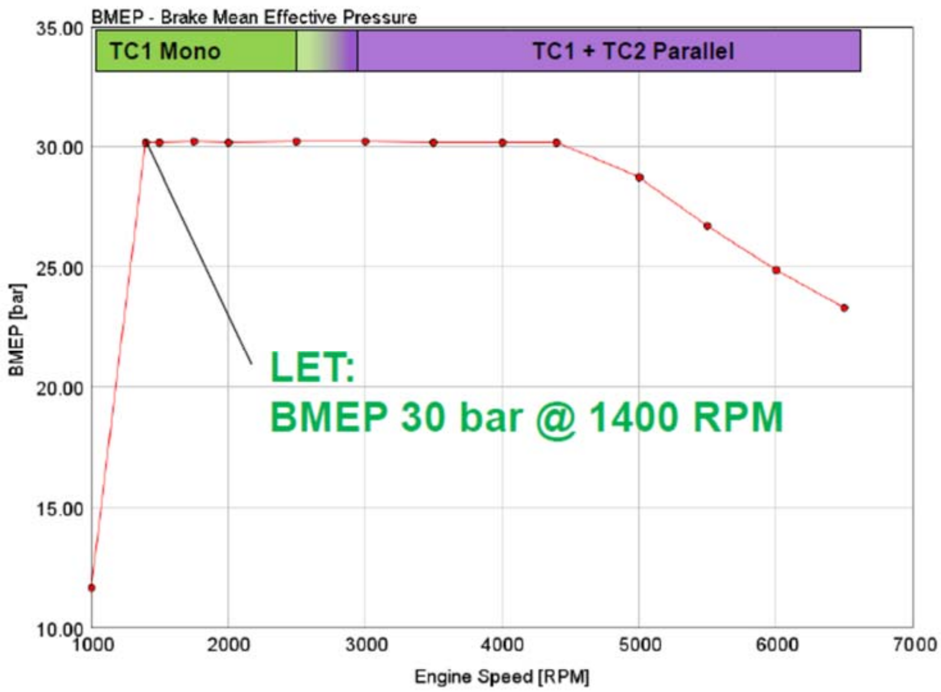

Figure 10: Predicted Full Load BMEP

Due to the specific requirements of the twin boosting system for methane operation, very high pressure ratios and large compressor map widths are required. Since these characteristics are far beyond the requirements of conventional gasoline or diesel turbochargers, dedicated turbine and compressor stages have been developed. Due to the high pressure ratios the temperatures post compressor exceed the temperatures of conventional boosting systems already at low and medium engine speeds. This has to be considered for the material selection of the compressor as well as the downstream piping. The maximum exhaust gas temperature on the turbine side is lower than for gasoline applications, due to the more favorable combustion phasing. It is found in the range of modern diesel applications, which allows the selection of materials known from these applications. The transition from single turbo operation (TC1) to parallel operation (TC1+TC2) has been investigated during the matching process. Based on the selected sizing of the turbochargers, transition occurs at an engine speed between 2,700 rpm and 2,800 rpm.

Figure 11 shows the final turbocharger system, including the electrical waste gates, which are mounted on the compressor housing to prevent excessive heat input into the electronic components.

Figure 11: Turbo Charger Design

Continuously Variable Valve Lift System

Figure 12). This system is applied to both, intake and exhaust camshaft.

On the intake side the CVVL system is used for valve event based load control and part load de-throttling. On the exhaust side it acts for adjusting optimized cam events, depending on speed and load, plus it enables the activation of TC 2 as already described.

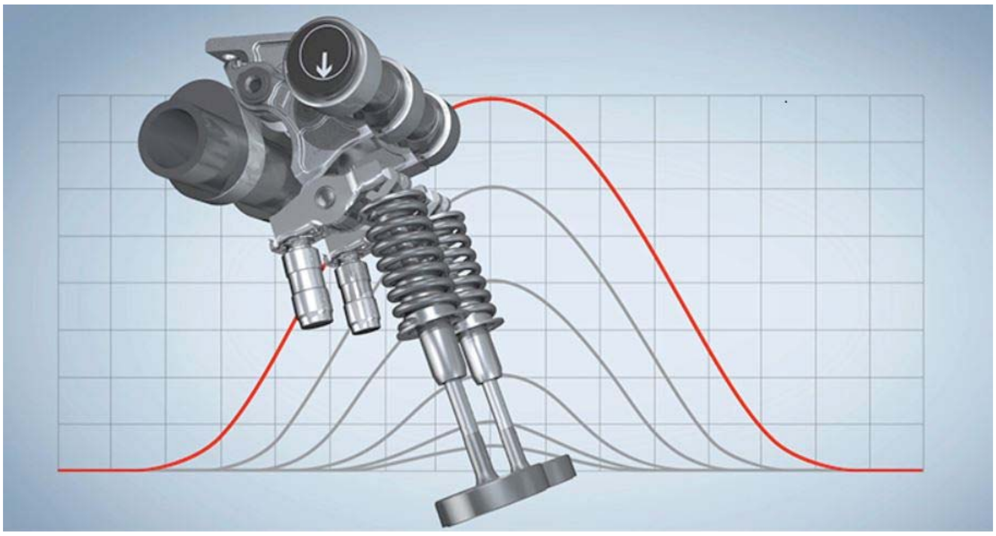

Figure 12: Generic CVVL-system architecture and feasible valve lifts /5/

A detailed description of the working principle of the CVVL- System and its dynamic behavior, as well as the requirements, is given in /5/.

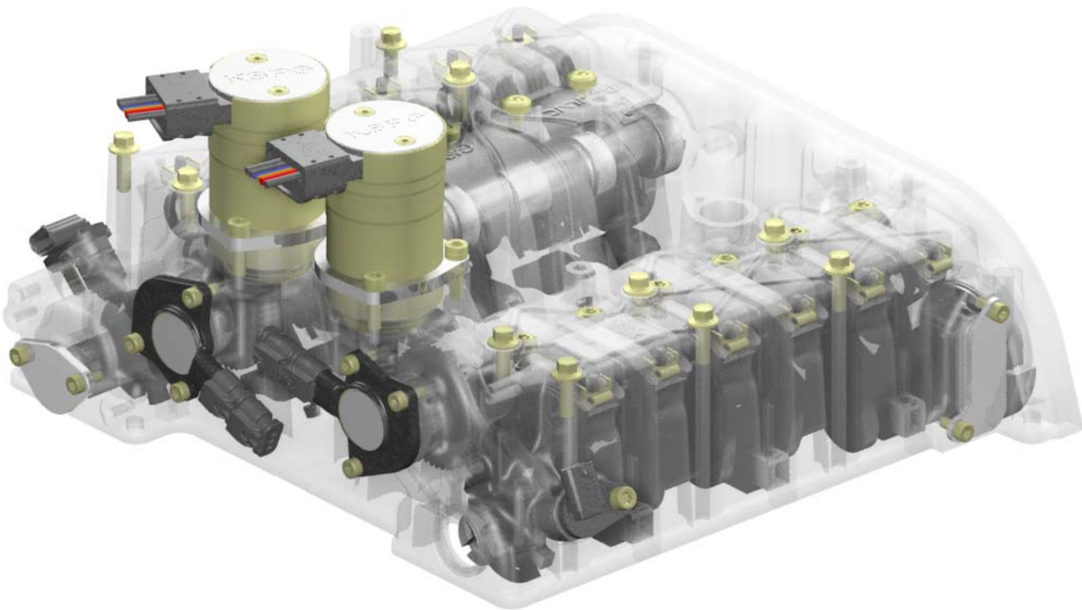

Figure 13: Structural Cover with CVVL-System

All specific valve train components like camshafts, control shafts, circular guide, rocker, rocker spring and actuator for intake and exhaust side are included in this cover, depicted in Figure 13.
The advantages of this architecture are e.g.

- cover with a high structural integrity to take all direct and lateral forces of the valve train, contributing to the desired, high stiffness of the valve train
- no access to head bolts through valve train necessary, thus less compromised valve train layout
- no need for an additional, oil retaining cover

To ensure a high response of the system, the control shaft is equipped with needle bearings with split cages. It can be introduced from one end of the cover without any need for bearing caps. This design additionally supports the cover stiffness.

The applied CVVL-System is a centre biased system. That means, the timing of the maximum valve lift keeps almost unchanged during valve event adjustment. To adjust the valve opening point a fast camshaft phasing system with a wide range of authority is required. Electrically actuated cam phasers with a shifting range of up to 75° crank angle and a shifting speed of up to 600° CA/s are used for this purpose.

Figure 14 shows the visited peak lift for all angular positions of the control shaft of the intake side on the left. In the green area, both intake valves open in parallel and allow a complete load control by EIVC (Early Intake Valve Closing). In the blue area, only one intake valve is engaged with the goal to impel a swirling charge motion. Because of high curvatures the grey area is for transition only.

In the same manner as on the intake side, Figure 14 discloses the peak lift for all angular positions of the exhaust control shaft on the right hand side. In the green area, both exhaust valves are actuated in parallel and both turbines are engaged. Within the blue area, the complete exhaust flow is directed towards the first turbo charger. This operation mode is used to achieve a high low end torque.

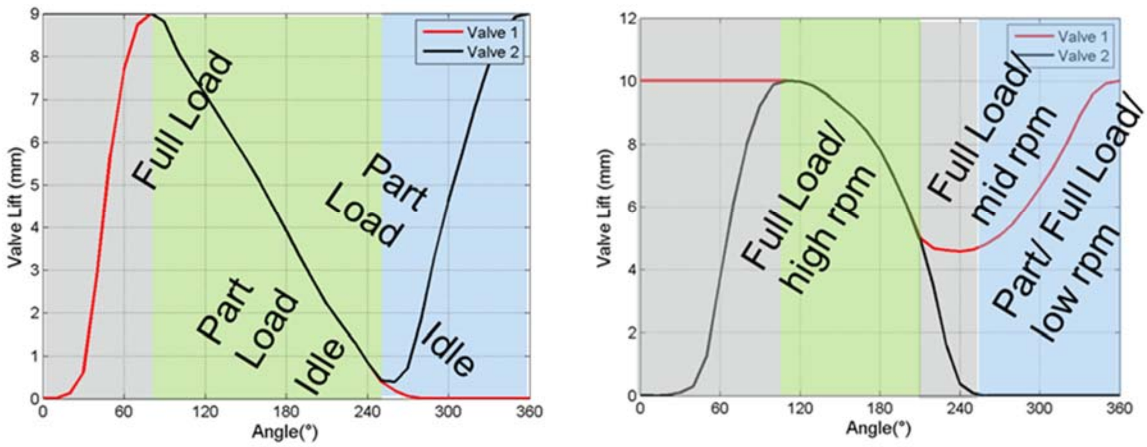

Figure 14: Intake (left) and exhaust (right) valve lift as a function of control shaft angular setting

Figure 15 gives an outside view of this complete new engine, dedicated to a high efficient use of methane, including the previously described technologies.

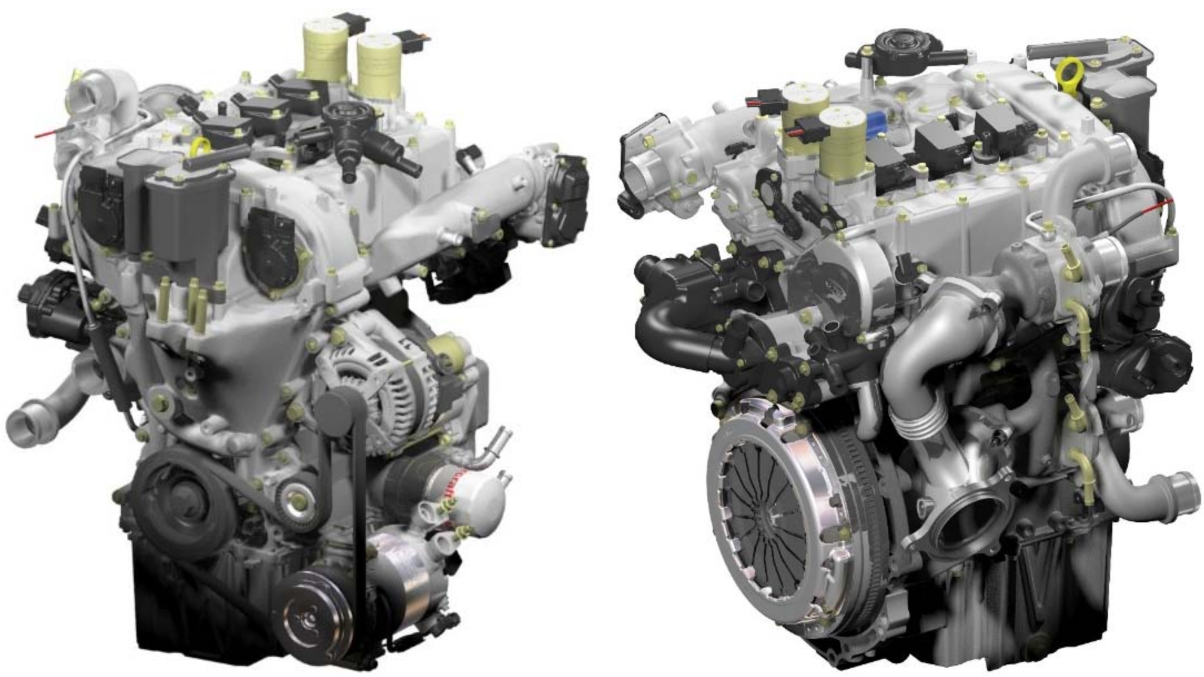

Figure 15: Methane dedicated MTDi engine

4 Combustion System Development and Test Results

The program target to deliver an advanced high power density methane engine requires the design of a demanding combustion system. The key elements of the combustion system - intake ports, combustion chamber, injector, and piston top are designed and optimized for this specific application.

Intake Port and Piston Optimization

The intake ports are redesigned in order to increase the tumble level, with minimum deterioration of the port flow capability. Various port designs with different tumble levels and flow rates have been simulated and an optimum compromise has been selected. One of the main drawbacks of a CVVL system is the fact that for EIVC, the intake-generated charge motion is rapidly dissipating that just a small amount is left for conversion into turbulence at spark timing. The result is comparably slow flame propagation speed. To preserve sufficient charge motion until spark ignition despite EIVC, the general charge motion level needs to be increased compared to the base gasoline production engine. In addition, valve masking is added to the combustion chamber to further increase charge motion at low valve lift operating conditions.

FigureFigure 16 depicts the effect of masking on tumble and turbulence level as a function of crank angle for two different pistons, for masked and unmasked intake valve conditions, again based on 3D-CFD simulation results. The simulation has been carried out at a part load operating condition of 1,500 rpm, 1 bar BMEP with a 2 mm maximum valve lift. The applied valve masking increases the tumble during intake and compression phase and turbulent kinetic energy (TKE) at spark timing.

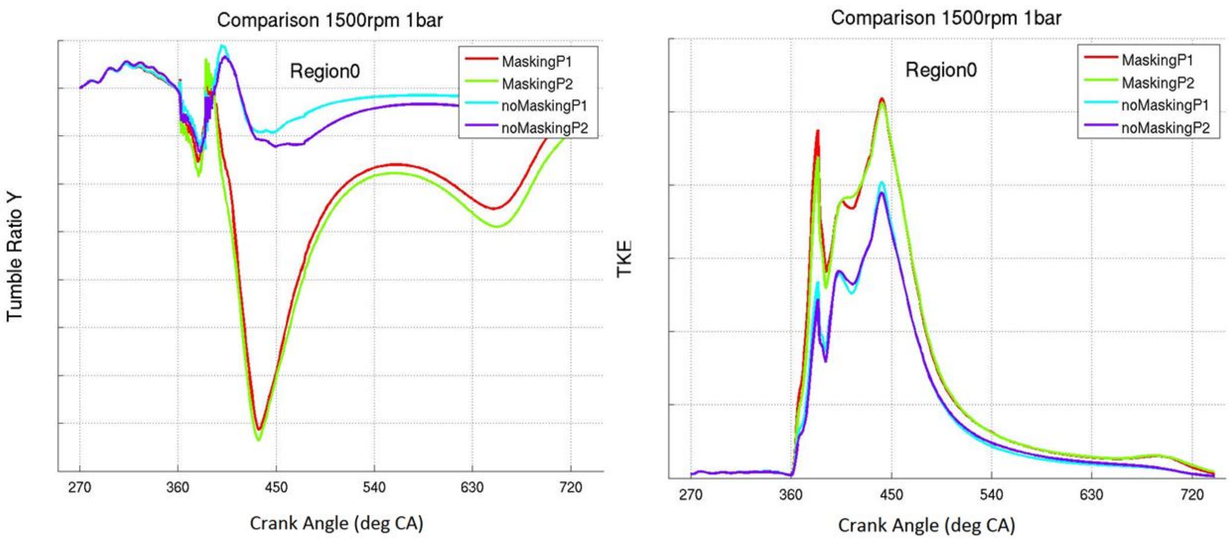

Figure 16: Effect of valve masking on tumble and TKE

The piston crown shape is - to a considerable content - already pre-defined by the need for the high compression ratio in that small displacement engine. Additionally a wide actuation range of the variable valve train system drives the need for large valve cutouts in the pistons. Despite of all these restrictions, the piston crown shape is also optimized for the methane application (Figure 16). Target for this 3D-CFD based optimization has been an optimized mixture homogenization.

Methane Direct Injection Process Optimization

Combustion system optimization for a high performance methane direct injection engine requires a detailed 3D CFD numerical simulation study due to the very complex and distinctive behavior of the injection process. In comparison to a gasoline engine, the DI methane engine does not need to deal with evaporation of the fuel. However, the gas dynamics and momentums are considerably different, since the injected fuel is in gaseous form instead of liquid phase, as it is for gasoline.

As a first step, modelling techniques have been developed and tested to ensure sufficient simulation accuracy. As an example Figure 17 shows a comparison of detailed numerical simulations with LES and various RANS CFD models. Although RANS provided a more detailed representation of the flow field, LES showed a better correlation of the highly dynamic gas jet to measured data.

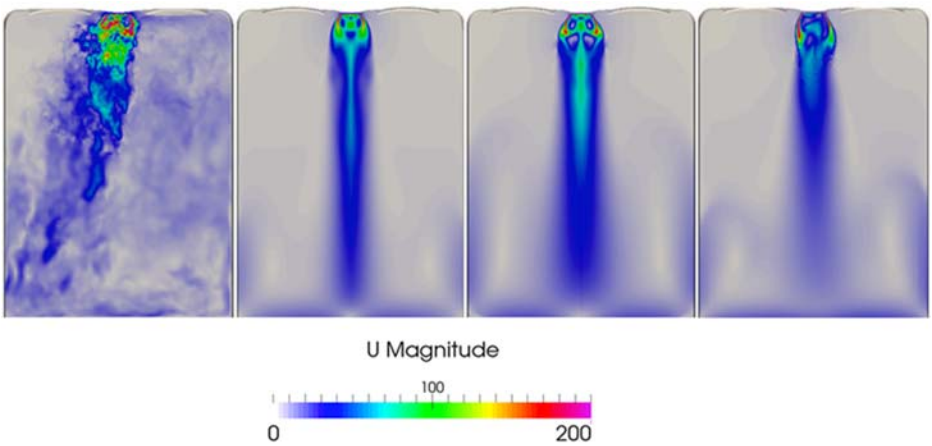

Figure 17: Numerical simulation of the direct injection process

Full Load Results

As already shown in
Table 1, the full load target for this MTDi engine has been set to 110 kW and 240 Nm, respectively 30 bar BMEP, for engine speeds above 1,500 rpm.

Figure 18 shows, that in achieving 120 kW (163 hp), the power target is overachieved by 10 kW. Furthermore, the 240 Nm (30 bar BMEP) torque target at 1,500 rpm is clearly achieved. In this chart, the red lines depict the single turbo operation, while the black lines describe the twin turbo operation.

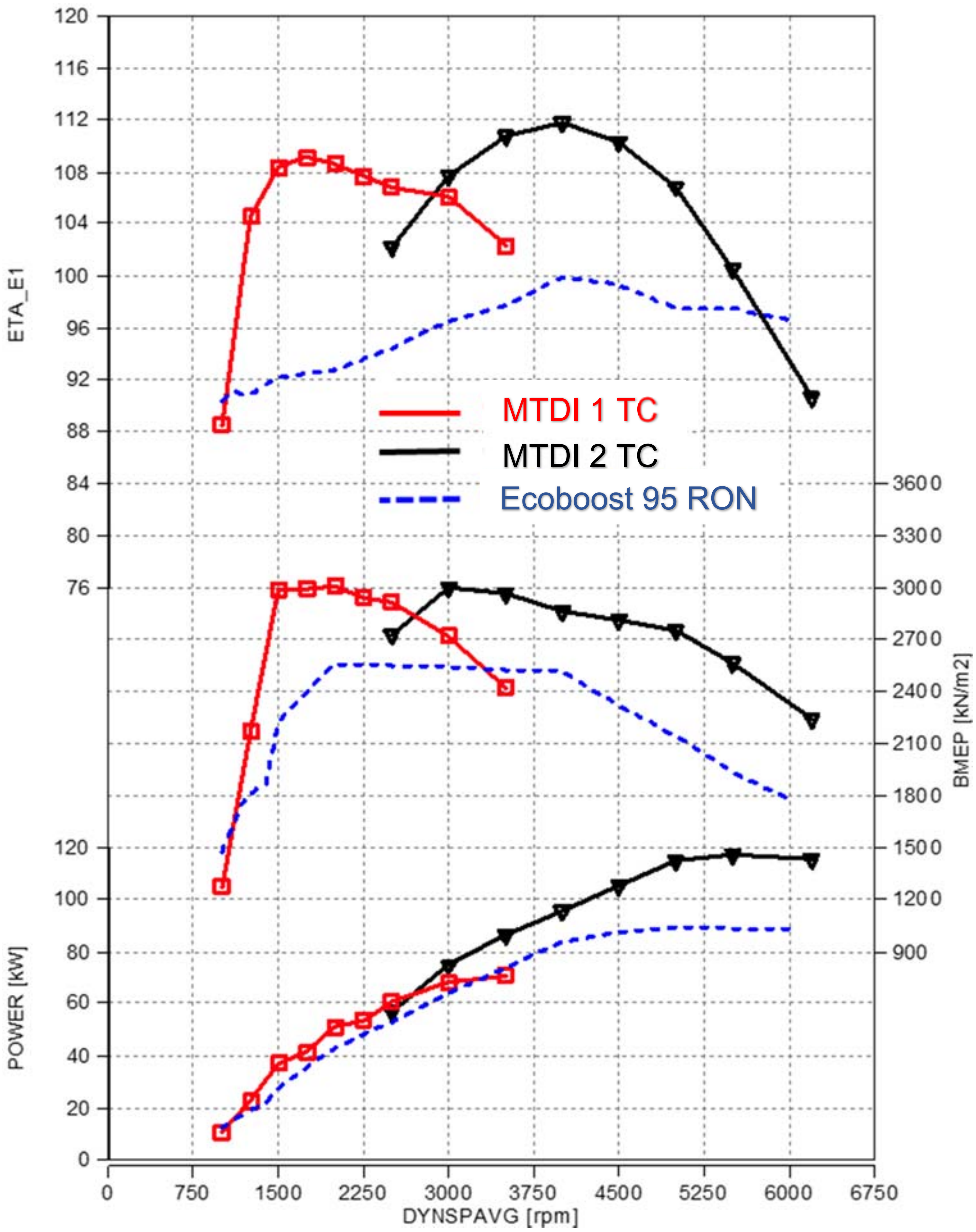

Figure 18: Full load performance MTDi engine

In Figure 18, the engine efficiency is shown, normalized to the peak full load efficiency of the Ecoboost engine. These data demonstrate the enormous potential of this newly developed methane direct injection engine. Peak efficiency improves by 12% compared to the reference. At 1,500rpm, the efficiency even improves by 16% above this reference. Here, the combination of direct injection, high compression ratio and high combustion pressure resistance are revealing their full potential.

Combustion peak pressure and combustion phasing for full load operation, relative to the max combustion pressure of the current gasoline version of this engine, is shown in Figure 19. In the engine speed range below 3,000rpm, the peak pressure limitation is resulting in a slightly retarded combustion phasing. For the high speed range, combustion phasing has to be retarded due to engine knock, rather than peak pressure limitation.

Considering the high compression ratio of 13, the combustion phasing of this engine is still on a remarkable early level.

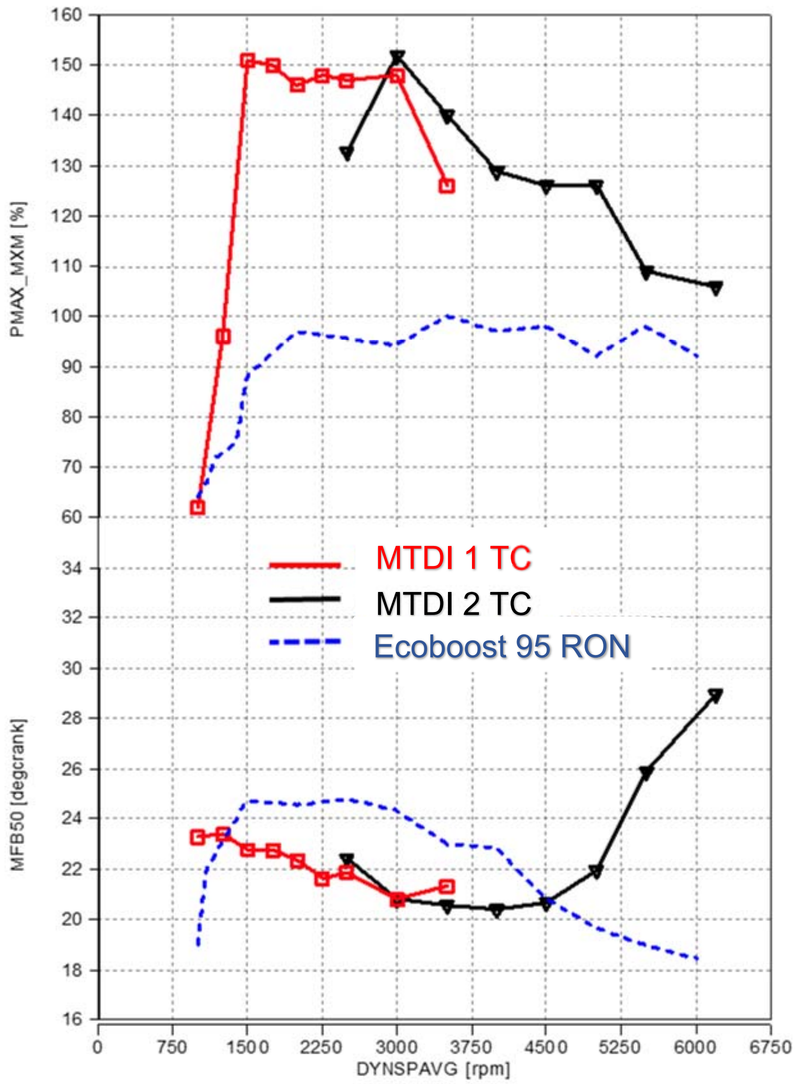

Figure 19: Peak pressure and combustion phasing at full load

Part Load and Mapping Results

The enormous level of adjustable parameters (fully variable valve train on intake and exhaust side, twin turbo setup, etc.) of this engine requires the extensive use of a DoE (Design of Experiments) approach, in order to find the best settings for all parameters. One of the key parameter sets for efficiency optimization are the valve lifts and the associated event lengths of intake and exhaust valve timing.

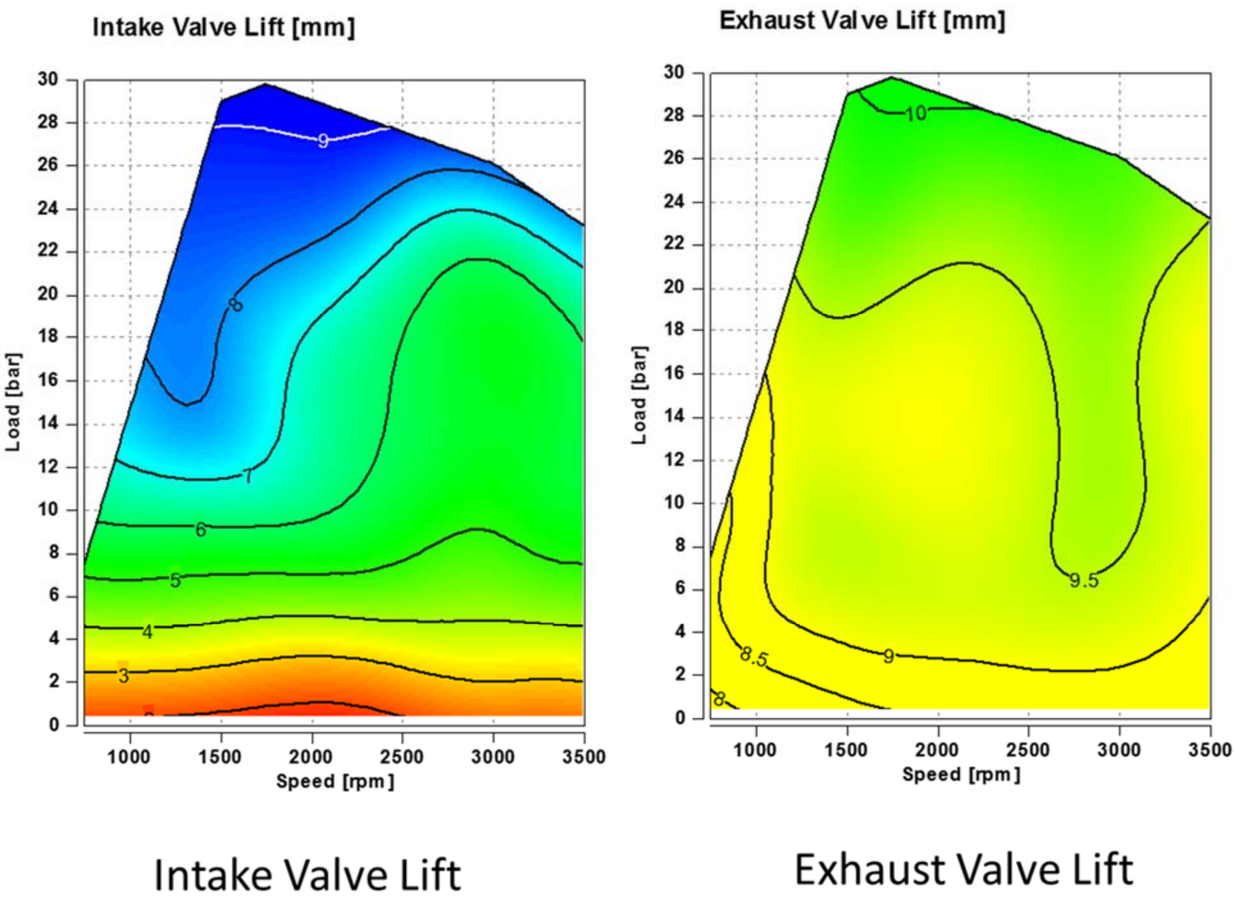

Figure 20: DoE optimized intake and exhaust valve lift

As indicated in Figure 20, a wide range of intake valve lift is utilized for best fuel economy. Although the fully variable valve train could adjust lower valve lifts than the 2 mm shown here, a further reduction of valve lift does not further improve fuel economy. The resulting valve lift curve then leads to a throttling at the intake valve, rather at the throttle body. Hence, no further improvement in fuel economy is possible. At the exhaust side, only small variations of valve lift are used to find the optimum efficiency settings. So, the potential for variable exhaust valve actuation is mainly used for controlling the two turbochargers accordingly.

The resulting engine efficiencies are shown in Figure 21, for single turbo operation in the left chart and for twin turbo operation on the right side. A peak efficiency of 38 % can be observed. However, much more important than this good peak efficiency is the extremely wide area of high efficiency. This is – besides the high compression ratio - mainly a result of the fully variable valve train, which reduces pumping losses at low load significantly.

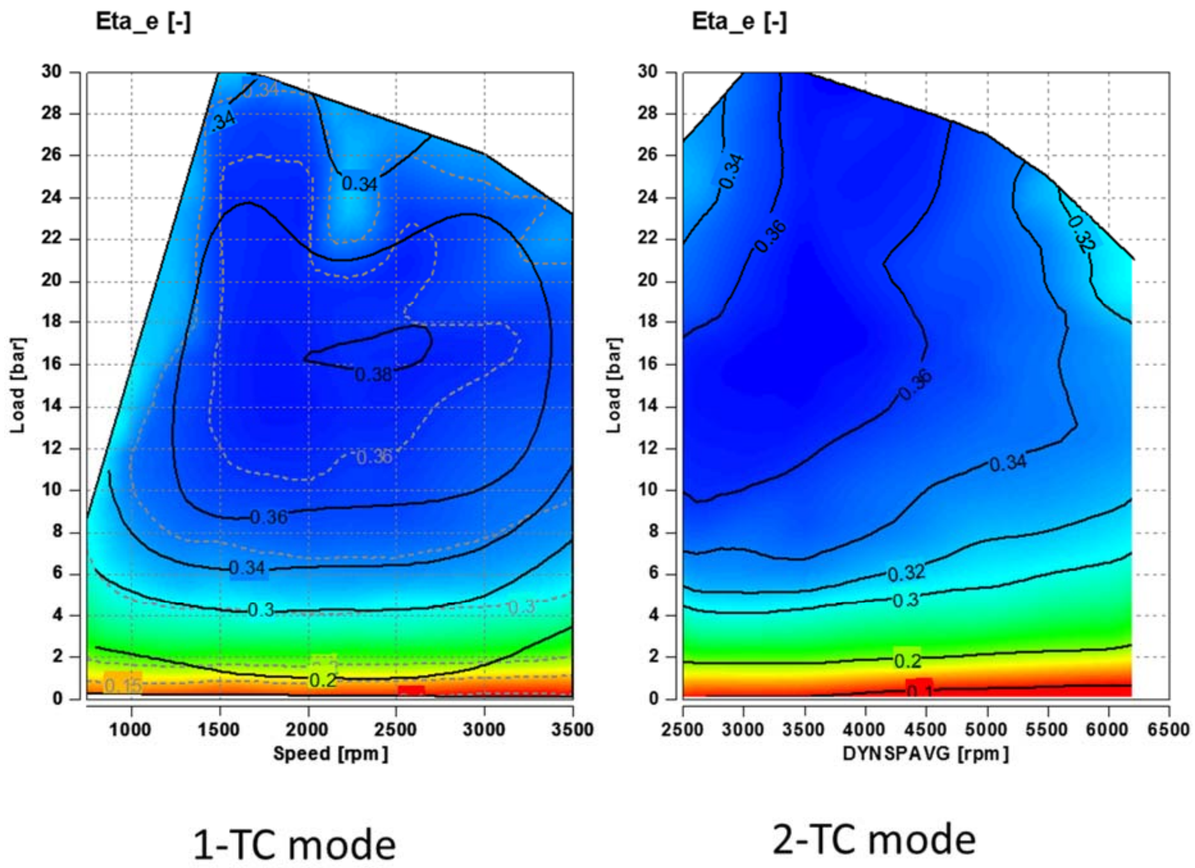

Figure 21: MTDi efficiency for single- and bi-turbo operation

Based on these steady state measurements vehicle cycle simulations have been conducted to verify conformance to NEDC and WLTC targets, based on a mid-class 7-seater van. The NEDC cycle prediction indicates a CO_2 emission of 93 g/km, well below the initial target of 100 CO_2 g/km. The prediction for the higher loaded WLTP cycle shows CO_2 emissions of 120 g/km. The combination of these technologies leads to exceptionally low CO_2 emissions which support the future emission glide path.

5 Acknowledgement

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