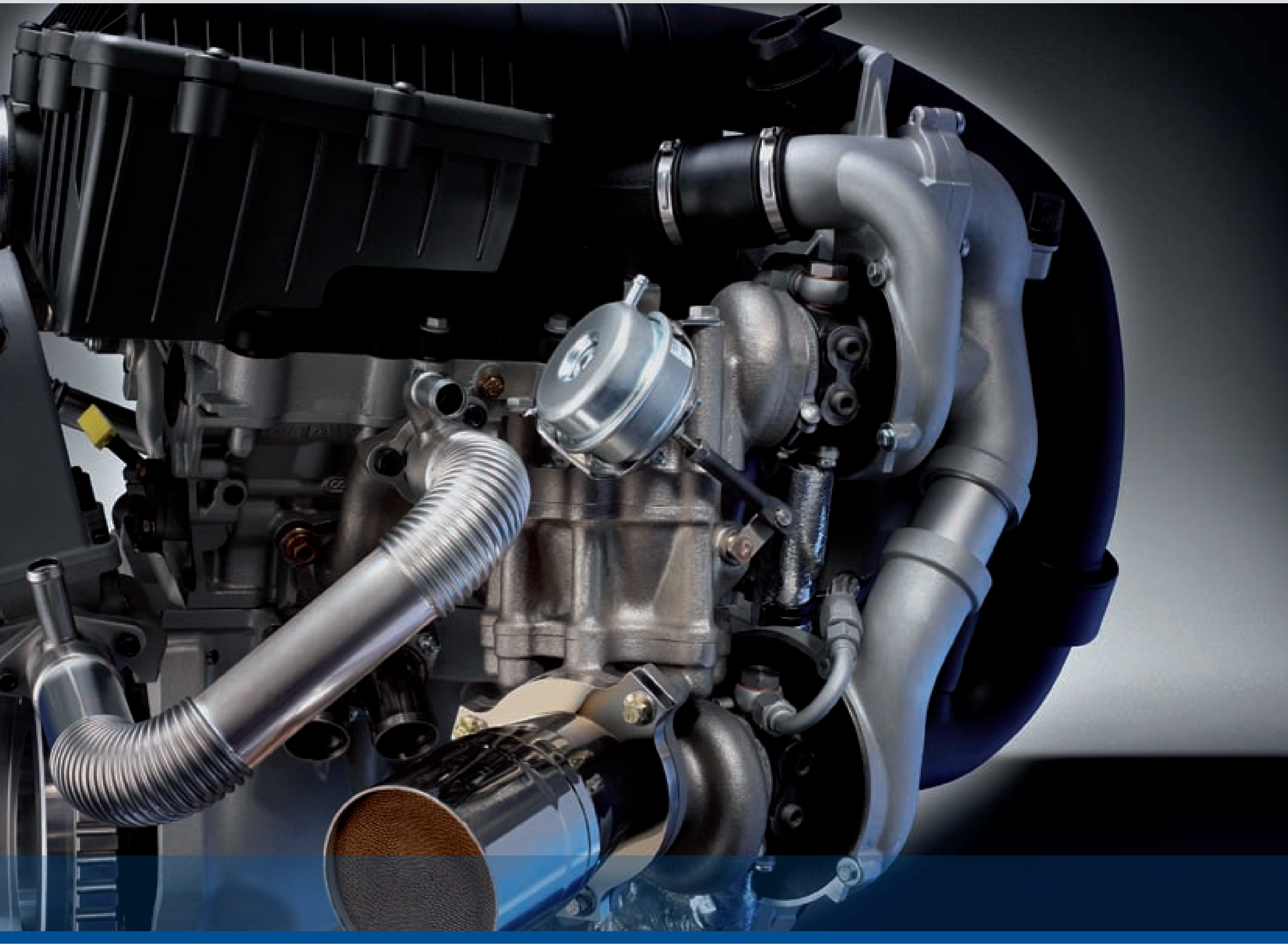

The Mahle Downsized Engine as Technology Demonstrator

Concept, Layout and Design

Downsizing – the combination of reduced engine displacement and pressure charging – is an important option, especially for gasoline engines, to minimise fuel consumption and CO_2 emissions while maintaining good driveability. To demonstrate the possibilities and core technologies of an advanced and innovative downsizing concept, Mahle is currently developing its own engine. This engine with a displacement of 1.2 litres will achieve values for torque and power output, which will enable a 50 % downsized engine, offering a potential for the reduction of fuel consumption and CO_2 emissions up to 30% in the New European Driving Cycle (NEDC).

1 Introduction

The reduction of CO_2 emissions combined with increasingly stringent legislation for HC, CO and NO_x is the most important challenge for designers of future engines and vehicles.

As shown in **Figure 1**, it will be difficult to meet the proposed target of 130 g/km for CO_2 emissions in the NEDC with vehicles powered by naturally aspirated gasoline engines with a displacement above 1.4 l. Limiting the displacement to 1.4 l in conjunction with natural aspiration would also limit torque and power output to approximately 160 Nm or 80 kW. Depending on vehicle size and weight vehicle dynamics and driveability would therefore deteriorate.

To overcome this trade-off between CO_2 emissions, fuel consumption and driving dynamics, downsizing in combination with turbo charging is a promising and already established option for gasoline engines. Concepts presented in the past showed for comparable performance with a naturally aspirated engine a displacement reduction of about 25%, resulting in a fuel economy advantage up to 15%. More current developments (based on GDI) show a downsizing potential of more than 35% with a fuel consumption benefit of more than 20%. The challenges for more advanced downsizing and the key for customer acceptance are the steady state and transient low speed torque characteristics.

2 Objectives

The opportunity of an innovative downsizing concept drove the development of an engine as technology demonstrator. Main topics for the development are the pressure charging system, the gasoline direct injection and variable valve timing as the core technologies of this downsizing concept. The thermodynamic and mechanical properties (friction and weight) of the engine were also optimised as a prerequisite for a good efficiency. The engine has been designed not only as a technology demonstrator but also as production capable, to enable developments and experiences gained to be as close as possible to future needs.

The engine was designed as a three-cylinder in-line configuration displacing V_H = 1.2 l with both a single turbocharged and a twin turbocharged concept. The objectives for WOT characteristic and fuel consumption figures are shown in **Table 1**. These figures for torque and power output are outlined for a vehicle application with a kerb weight of m ≈ 1,600 kg to show a comparable driving performance to a naturally aspirated engine with a displacement of V_H = 2.4 l. This would equate to a downsizing of 50 %, resulting in a re-

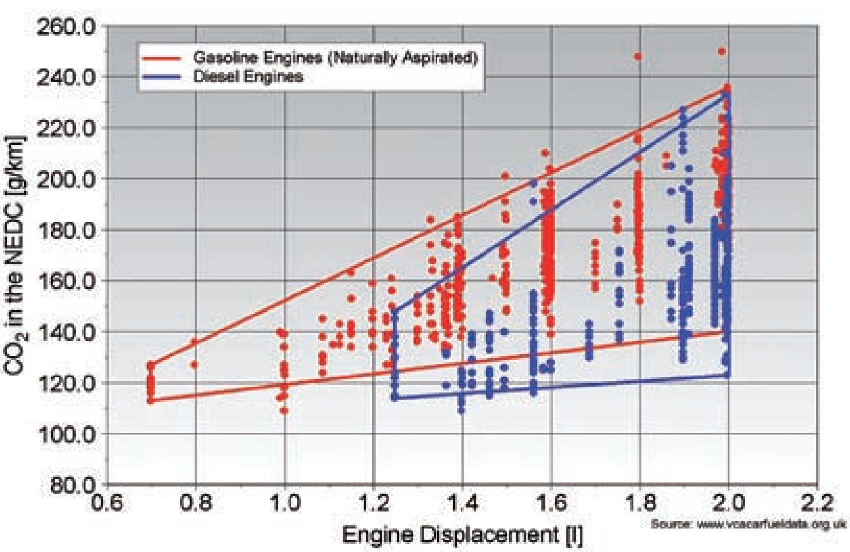

Figure 1: CO_2 emissions of gasoline and diesel engines, depending on engine displacement

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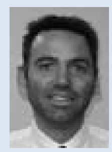

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Table 1: Performance and fuel consumption data objectives for the downsized 1.2 l engine

Performance at Full Load		
Single turbo:		
Torque		
Maximum at n = 2,500 - 3,000 rpm	M_d = 210 Nm	(p_{me} = 22 bar)
Power Output		
Maximum at n = 6,000 rpm	P_{max} = 90 kW	(P_{max} / V_H = 75 kW / l)
Twin turbo:		
Torque		
at n = 1,000 rpm	M_d = 153 Nm	(p_{me} = 16 bar)
Maximum at n = 2,500 - 3,000 rpm	M_d = 286 Nm	(p_{me} = 30 bar)
Power Output		
Maximum at n = 6,500 rpm	P_{max} = 144 kW	(P_{max} / V_H = 120 kW / l)
Fuel Consumption at Part Load		
Optimum	b_e = 235 g / kWh	
at n = 2000 rpm / p_{me} = 4 bar	b_e < 295 g / kWh	

duction of fuel consumption and CO_2 emissions in the NEDC of approx. 30 % compared to a vehicle application with a typical 2.4 l cylinder engine. The engine is designed for λ1- operation over the total engine map and for conformity with the EU 5 emission regulation. The most important engine design data are shown in **Table 2**.

3 Base Engine

3.1 Cylinder Head

Within the combustion chamber, the positioning of injectors and spark plugs and the configuration of intake and exhaust ports are core components for the thermodynamic and combustion systems of the engine. The main design topic was the combustion chamber layout in conjunction with optimum cooling conditions. **Figure 2** shows the combustion chamber, designed with four valves per cylinder and a cylinder bore of 83 mm for a compact geometry with a small surface area. For the GDI system a central spray- guided design (for homogeneous operation at λ 1) is used, the M10 spark plug is positioned slightly off cylinder centre towards the exhaust side. This combustion chamber configuration enables a compression ratio of ε = 9.75 which is relatively high for pressure charged engines.

The combustion chamber configuration required several iterations to optimise the cooling in the cylinder head to keep the thermal loading as low as possible in order to minimise injector tip temperature and knock sensitivity. This was achieved by using a cross flow cooling arrangement together with an optimised water jacket in the critical areas. Cylinder head and cam ladder are made from aluminium. The cylinder head is made from the A 356 alloy using the Coscast process. This process allows small tolerances in the casting and therefore the reduction of wall thicknesses combined with a complex geometry as a prerequisite for low weight and good cooling. The inlet ports are fully machined and designed for the future option of high and medium tumble ports.

The cylinder head and the core for the complex water jacket geometry are shown in **Figure 3**. The design of the cylinder head, valve size and port geometry was supported by CFD and GT Power simulation.

3.2 Cylinder Block and Crankcase

Design objectives for cylinder block and crankcase were: (1) a low component weight combined with high stiffness and low bore distortion, (2) a good control of cooling and thermal man-

Table 2: Main design data of the downsized 1.2 l engine

Main Data, Cylinder Block and Cranktrain			
Cylinderblock (material)		**A 356**	**(COSCAST)**
Bore		83,0	mm
Stroke		73,9	mm
Stroke-Bore ratio		0,89	
Cylinder displacement		0,400	l
Cylinder number and arrangement		3	in line
Displacement		1.200	l
Bore spacing		91	mm
Conrod length		123	mm
Block height		189,5	mm
Compression ratio		9,75	
Firing order		1 - 3 - 2	
Cylinder Head and Valvetrain			
Cylinderhead (material)		**A 356**	**(COSCAST)**
4 valves/cylinder			
DOHC architecture with roller finger followers and dual independant cam phasing			
Valve head diameter	Intake / Exhaust	31.4 / 25.5	mm
Valve stem diameter	Intake / Exhaust	6 / 6	mm
Maximum valve lift	Intake / Exhaust	11 / 11	mm
Valve angle	Intake / Exhaust	21.5 / 20.0	°

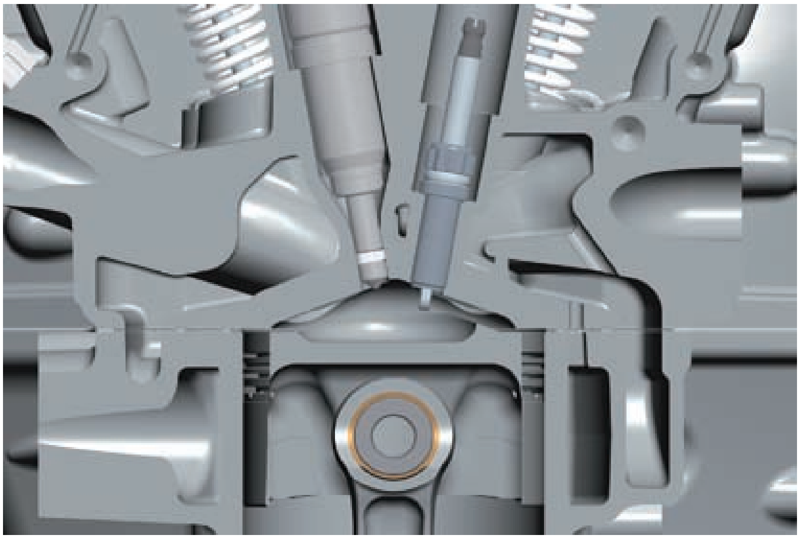

Figure 2: Combustion chamber geometry

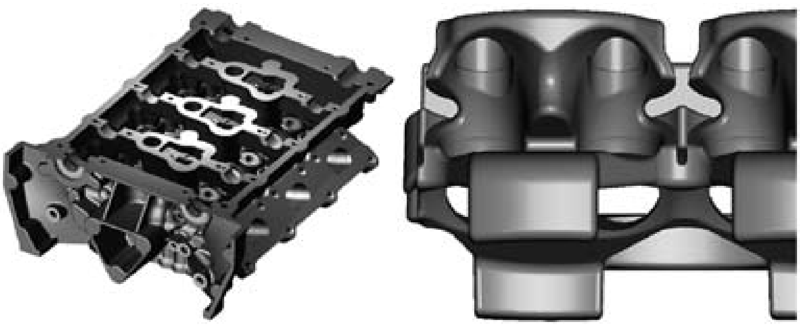

Figure 3: Cylinder head and water jacket core

agement. The Coscast process (Aluminium alloy A 356) is also used for the cylinder block. The cylinder block, **Figure 4**, is designed as „closed deck" for high stiffness with a Mahle Nikasil-coated cylinder bore for good heat transfer.

The bolting of cylinder head, cylinder block and bedplate is designed as a through bolting system to minimise bore distortion and to reduce the component weights, Figure 4.

The piston cooling feeds via separate (non-filtered) oil channels directly from the oil pump control gallery. The bedplate design features integrated baffles and avoids the use of cast-in steel inserts. A multi-layer steel gasket is used for the cylinder head gasket.

3.3 Crank and Valve Train

Focus of the development was a design for expected peak cylinder pressures of 140 bar, low friction and low wear.

The crank train is designed with

- Forged Mahle light-weight pistons [1],
- A Mahle ring package with two compression rings (one barrel faced ring, one Napier ring and a three-piece oil-scraper ring,
- A DLC-coated piston pin,
- Forged Mahle steel conrod [2], and
- A steel crankshaft.

Big end conrod bearings are thermally sprayed for increased load capability. The bearing of the crankshaft is designed with thick wall main bearings (4 mm) to reduce the effects of temperature on the bearing clearance. The cam drive uses an 8 mm pitch low noise Iwistar chain.

The valve train was designed for low weight, low moving mass, low wear and low friction with a DOHC architecture and dual independent cam phasing (intake and exhaust). The lightweight assembled camshafts (steel cam lobes) actuate the valves via roller-finger followers for low friction. For intake and exhaust, new light-weight valves [1, 3] are used, which enable reduced friction (due to lower valve spring forces) and lower wear between valve and valve seat insert (due to reduced closing impulse). With large sodium-filled cavities, the Mahle lightweight valves also offer a significantly increased potential for heat rejection and thermal resistance, compared to conventional valves. This is advantageous for supercharged GDI engines with lean burn operation.

Figure 5 shows the complete arrangement of crank train, cam drive, valve train and balancer system.

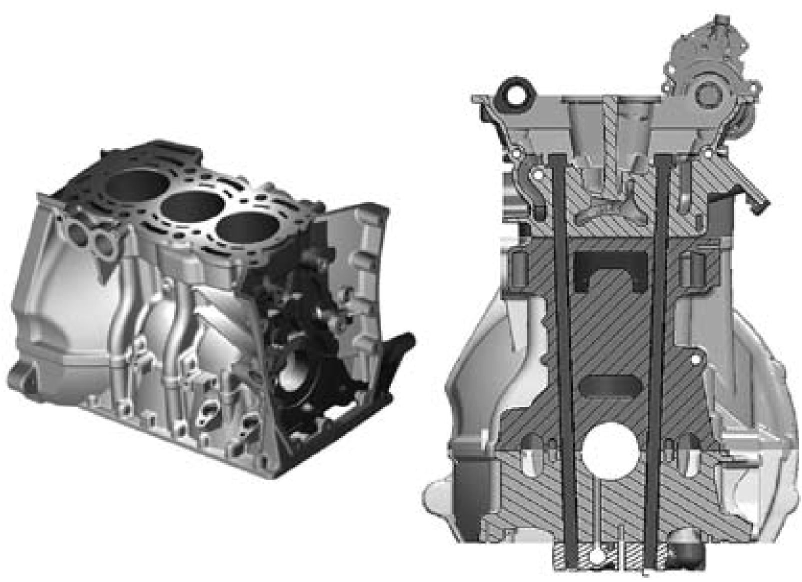

Figure 4: Cylinder block and bolting with cylinder head

Figure 5: Complete drive train with cam drive, valve train and mass balancing

The assembled camshaft, produced using the Mahle process, also integrates an innovative device for initial oil mist pre-separation in the breather system, **Figure 6**, resulting in enhanced oil-separation [4].

Cylinder head and valve train are protected for a future application of a system for variable (switchable) valve lift [5].

A three-cylinder in-line engine has free rotating moments of 1st and 2nd order which, if left unbalanced, result in a worse smoothness than a four-cylinder in-line engine. To improve mass balancing without significantly increasing weight, packaging and friction, a system was chosen with two small masses, housed in plastic gears driven from the crankshaft.

3.4 Cooling and Oil Circuit

Efficient cooling control was the main objective for the design of the cooling system, as a prerequisite for optimising combustion and friction behaviour. The cooling system uses separate cooling passages for cylinder head and block to allow coolant flow optimisation. Water flow is distributed via an electric water pump and electrically controlled diverter valves and uses an integrated water temperature sensor in the cylinder head gasket for control feedback. Low friction, low weight and reduced package were the focus for the design of the oil circuit and the lubrication system. A separate gallery supplies the unfiltered oil for piston cooling without pressure loss for the lubrication system. The control valves for the two cam phasers have a separate feed from the lubrication of the crankshaft and camshaft bearings. A plastic oil-filter module, integrated aluminium oil cooler and an oil pump housing with Coscast aluminium castings are used for weight reduction, **Figure 7**.

4 Charging, Intake and EGR System

Two different pressure charging concepts with single and sequential twin stage turbocharging were selected for this engine. The focus of the development was the twin stage system, which was designed for the maximum torque and power output figures shown in Ta-

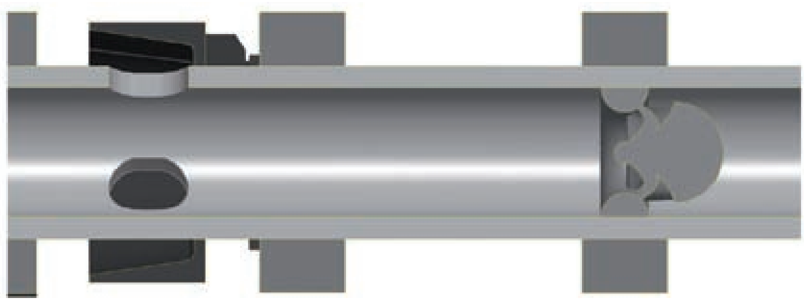

Figure 6: Assembled Mahle camshaft with integrated oil mist pre-separator

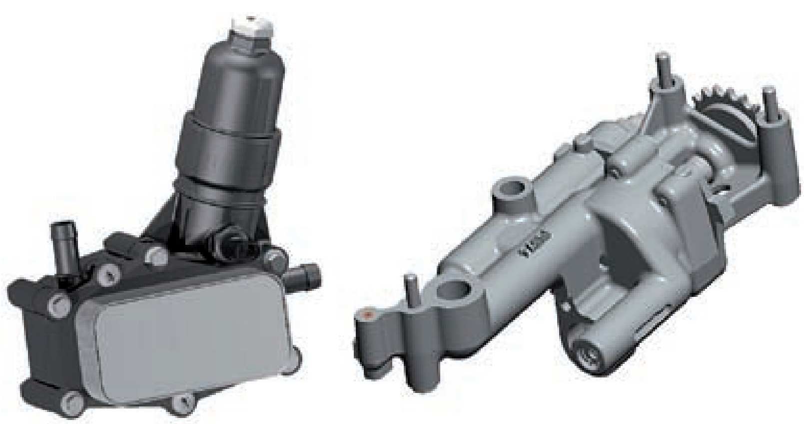

Figure 7: Mahle oil filter module and oil pump housing

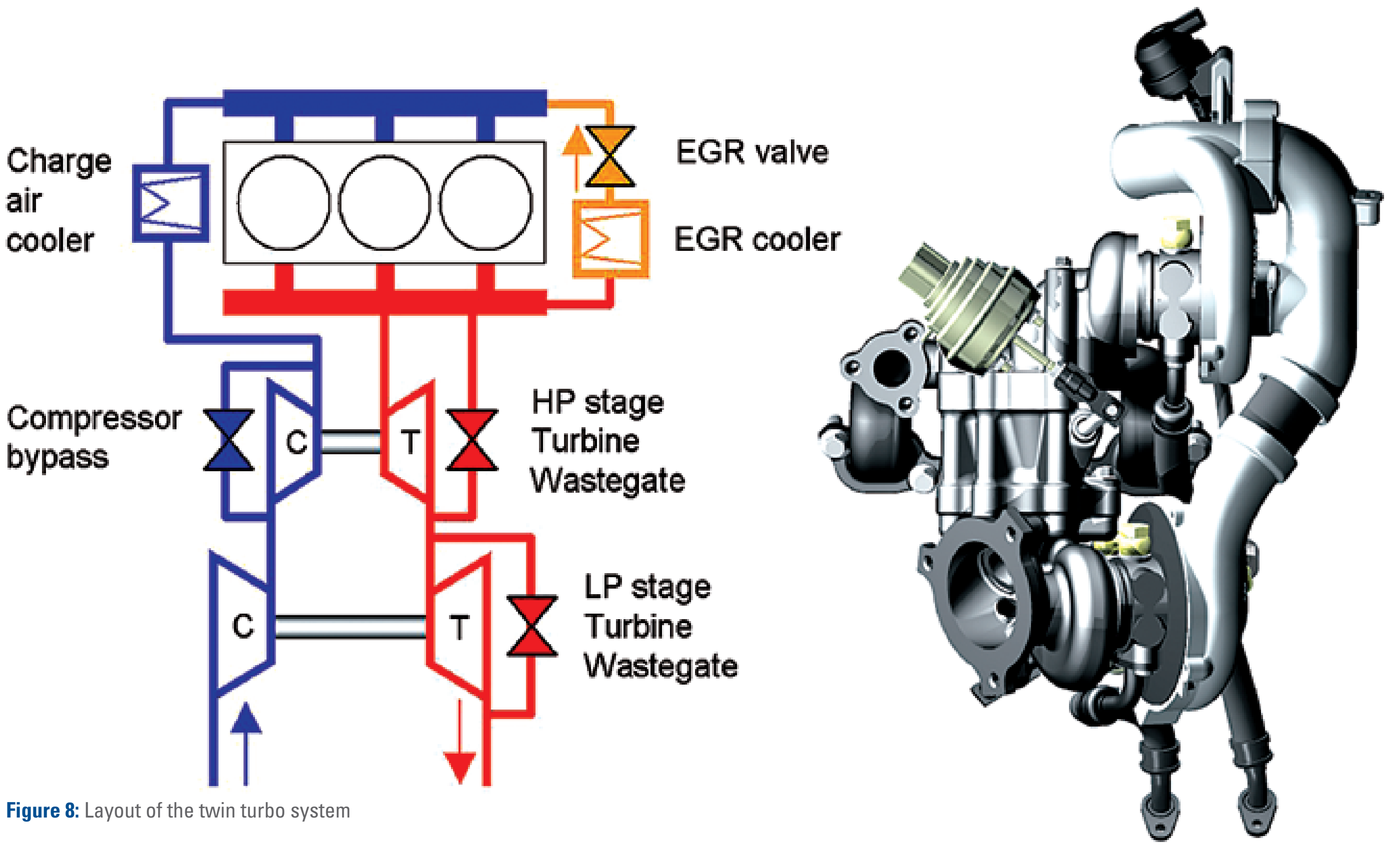

Figure 8: Layout of the twin turbo system

ble 1 as well as for an optimum transient response. Both configurations of turbochargers, control and by-pass valves and integration of charge-air cooling and EGR system are shown in **Figure 8**. The compressor maps of the chosen twin turbochargers are shown in **Figure 9**.

The high-pressure stage with the small turbocharger is effective at low engine speed, while the low-pressure stage with the large turbocharger is rotating without delivering a significant part of the charge pressure. The wastegate valves of both turbochargers are closed to achieve a maximum pressure and flow. The small turbocharger attains its maximum speed at WOT and an engine speed of approximately 2,500 rpm, when the wastegate of its turbine and the bypass of its compressor are opened. The large turbocharger now supplies the entire boost pressure. The maximum boost pressure achieved is 2.8 bar. The turbine housing of the high pressure stage is integrated into the cast steel exhaust manifold. All components are designed for a maximum exhaust temperature of 1050 °C. The single turbo concept uses a turbocharger with variable turbine geometry (VTG), based on a standard specification for Diesel engines. The system is combined with a water-cooled aluminium exhaust manifold and cooled exhaust gas recirculation (EGR) in the range of high load and engine speed to reduce the exhaust temperature down to an acceptable level of 850 °C. A complete charge cycle simulation supported the design of both pressure charging concepts with GT Power.

Low weight, high compactness, low flow losses and good transient behaviour were the objectives for the complete intake system. Therefore the engine uses a highly integrated system with a maximum use of plastic components for much of the air path: air filtration, mass air-flow meter, throttle and

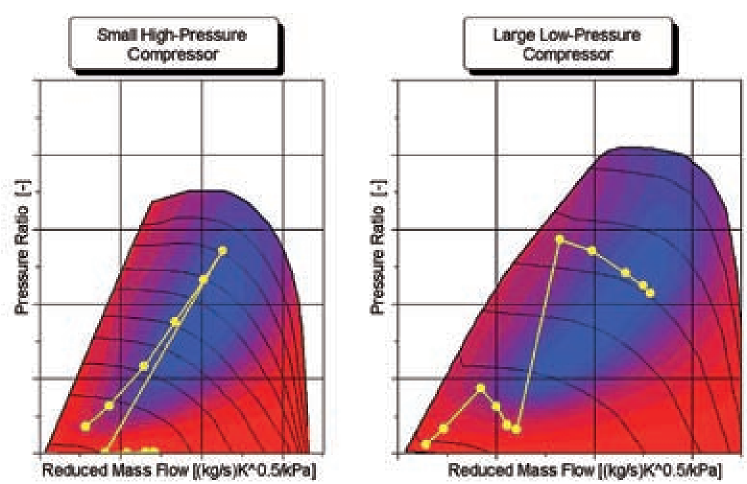

Figure 9: Compressor maps with operating points at WOT

intake manifold. Both charge-air concepts include an aluminium charge-air cooler, **Figure 10**. A triple throttle assembly optimises transient response and flow losses. The low volume air cleaner assembly is equipped with additional NVH measures, a Helmholtz resonator behind the air cleaner and a gap resonator before the turbocharger.

Figure 11 shows the EGR system which is used not only for the part load range, as is customary, but also for high loads and engine speeds in order to reduce the requirement for over-fuelling and to improve knocking sensitivity [7]. The exhaust gas is taken from the exhaust manifold before the turbocharger. The EGR flows through the cooler assembly, where the cooler uses a bypass system to maintain a constant gas temperature. The EGR flow is lead to the intake manifold, where it is fed through a barrel type EGR valve. Separate entry ports are used for each cylinder for minimum residual EGR gas. The control valve is integrated into the intake manifold. The EGR system is designed for EGR rates up to 15 % at high engine load and speed. An novel, fast-switching air pulse valve is integrated into the single turbo system to avoid backflow and to enable the high EGR rates over a greater load and speed range.

Figure 10: Integrated intake system for two charging concepts with charge-air cooling

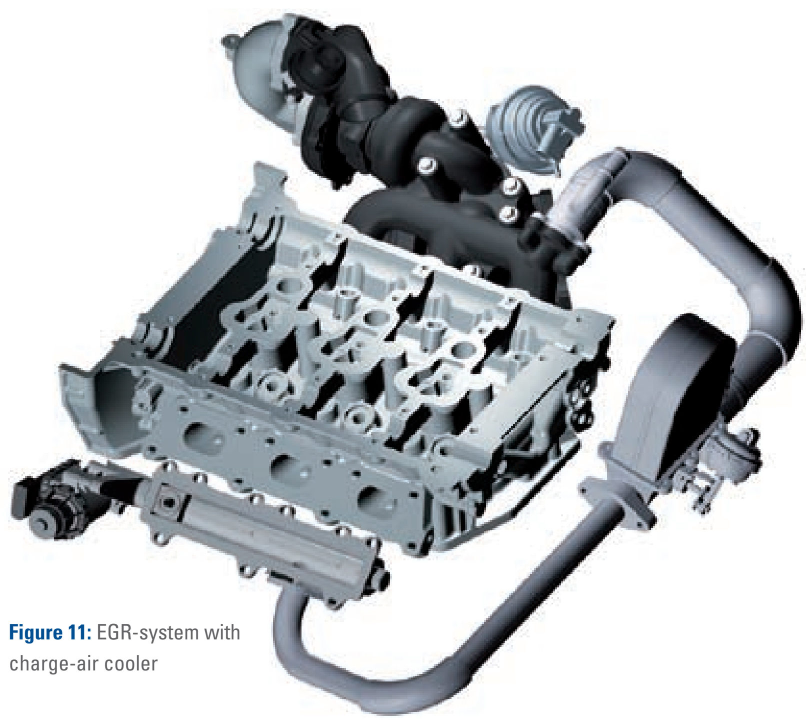

Figure 11: EGR-system with charge-air cooler

5 Ancillaries

The reduction of the driving losses for the ancillaries [8] was the main motive for the design of the ancillary machines and drive. In addition to the previously described optimisation of the oil circuit and lubrication system, the use of an electrically driven water pump was designed. To enable an additional potential for fuel consumption reduction a combined starter-generator, as used in mild hybrid vehicles, is used instead of a standard generator.

6 Summary

Downsizing is an important option for minimising fuel consumption and CO_2 emissions, especially for gasoline engines, in combination with good driving performance.

For this scenario a downsizing demonstrator engine with a displacement of 1.2 l was developed which has a potential for fuel and CO_2 reduction of 30 % in the NEDC. The focus of the concept design was the pressure charging (single and sequential twin turbo), gasoline direct injection and variable valve timing. The development work is continuing with the focus on the thermodynamic and mechanical development of the engine. The results will be reported later in a further MTZ publication.