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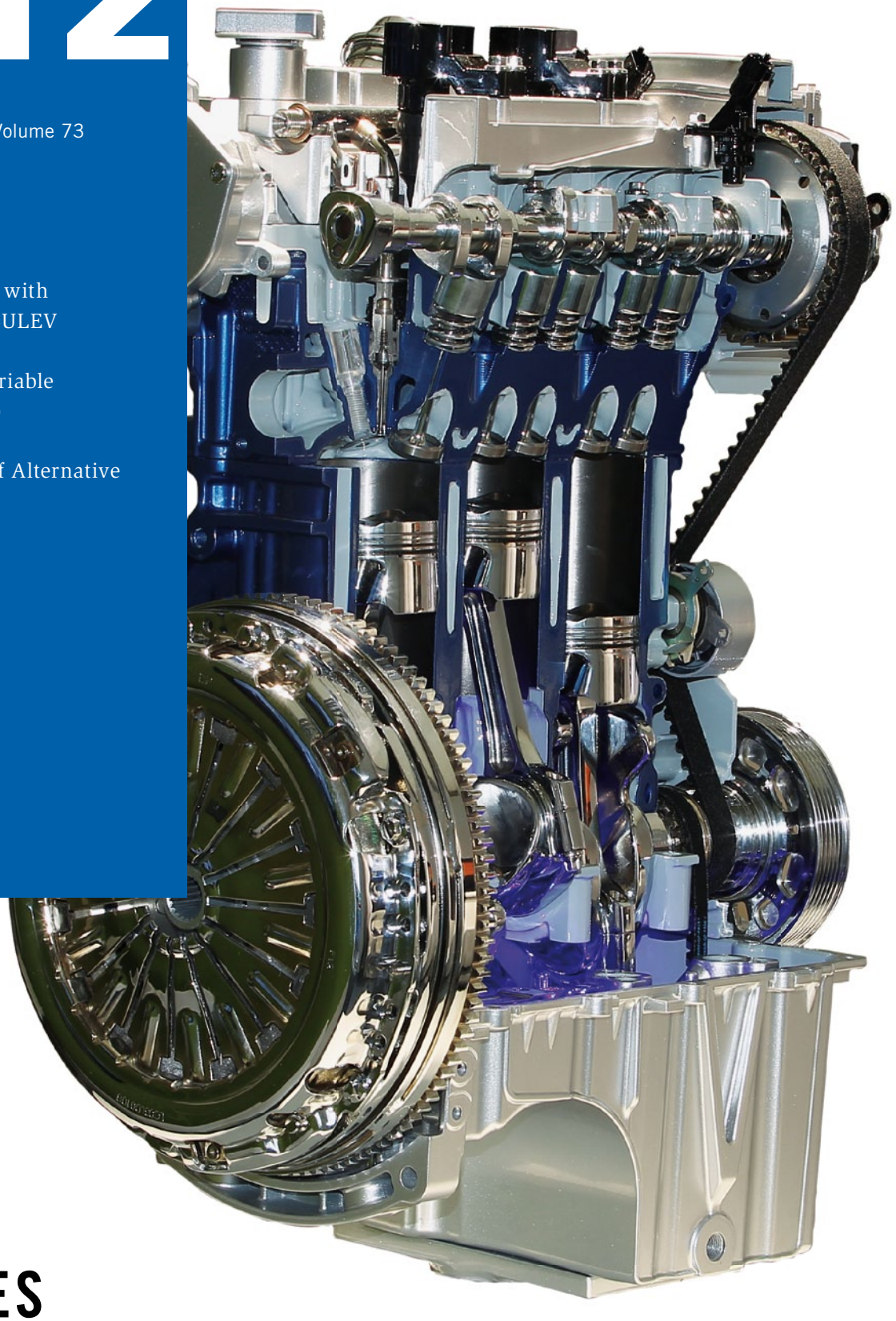
MTZ WORLDWIDE

05 May 2012 | Volume 73

GASOLINE ENGINE with Turbocharger for SULEV

POTENTIALS of Variable Compression Ratio

LASER IGNITION of Alternative Liquid Fuels

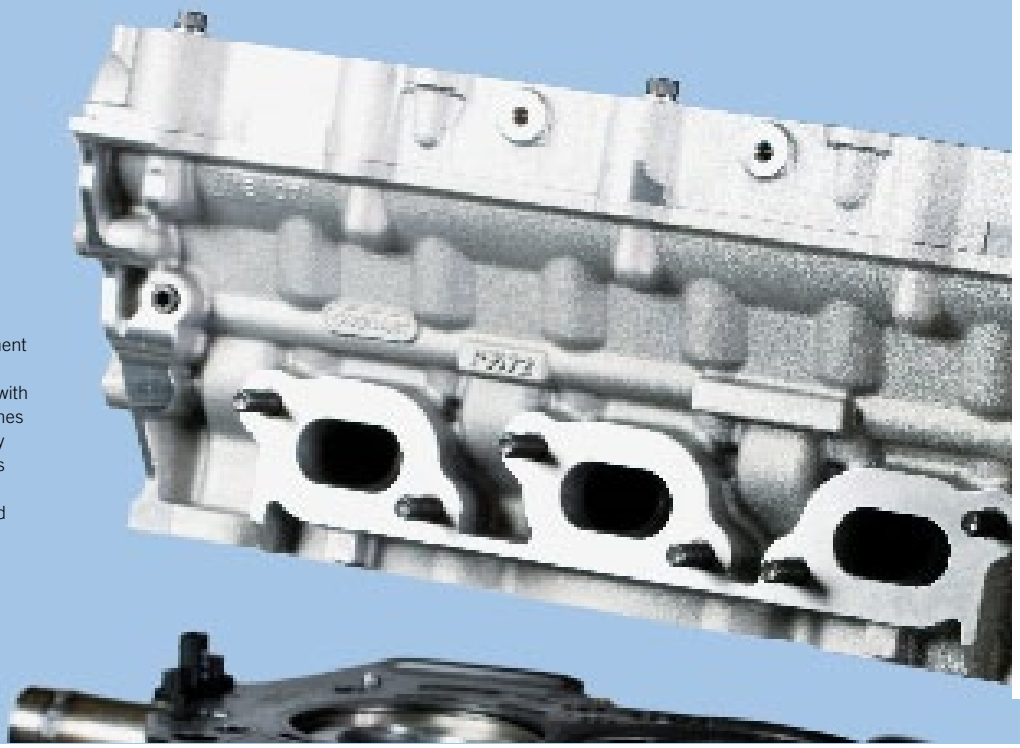


ENGINES
WITH SMALL DISPLACEMENT

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COVER STORY

SMALL-DISPLACEMENT ENGINES



4, 12 | An increasing number of small-displacement two- or three-cylinder engines will come onto the market in the near future. They range from those with a high specific power output used as vehicle engines for downsizing applications to low-cost, technically simple drive units that function as range extenders for electric vehicles. MTZ presents two different concepts of small-displacement engines from Ford and from KSPG and FEV.

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COVER FIGURE © Ford
FIGURE ABOVE © Mahle

PROMOTING AND DEMANDING

Dear Reader,

As part of its National Development Programme for Electric Mobility, the German government intends to provide considerable funding for research and development activities in this field over the next few years. The sum of up to two billion euros is under discussion. Of course, promoting the development of electric vehicles makes sense, whether it is to enable vehicles to have zero emissions locally, or even – if the energy comes from regenerative sources – to be carbon-neutral. It is also correct that there is some catching up to be done in research and development on electric traction, and this is the aim of this latest government funding.

Nevertheless, one should not ignore the fact that the market share of fully electric vehicles is expected to remain negligibly small in the foreseeable future. In all likelihood, they will account for a maximum of just 5 percent of newly registered vehicles in 2020. Or to put it another way: 95 percent of vehicles sold will still be powered by a diesel or gasoline engine. Engine manufacturers and their suppliers are making great efforts to further reduce CO_2 emissions, in some cases through their own research and in others through joint projects, for example under the umbrella of the Research Association of Combustion Engines (FVV). But as far as internal combustion engines are concerned, the government seems to rely solely on legislation that makes demands, as well as on market forces. In contrast to electric mobility, no research funds have been allocated in this area.

As has been demonstrated, the internal combustion engine still has considerable potential for improvement. According to a projection by Bosch, engine optimisation alone can reduce CO_2 emissions by up to 30 percent, and when combined with modifications to the vehicle itself the figure will be 40 percent, with hybridisation offering a further 10 percent reduction. These potential savings should be implemented as quickly as possible in fully developed products, also for the benefit of our environment. The government's funding policy, however, ignores one of the best levers for achieving a fast and comprehensive reduction in CO_2 emissions. What is your opinion on the subject? You can have your say by sending an email to Richard.Backhaus@rb-communications.de or by contributing to the blog at www.ATZonline.de.
I look forward to hearing from you.

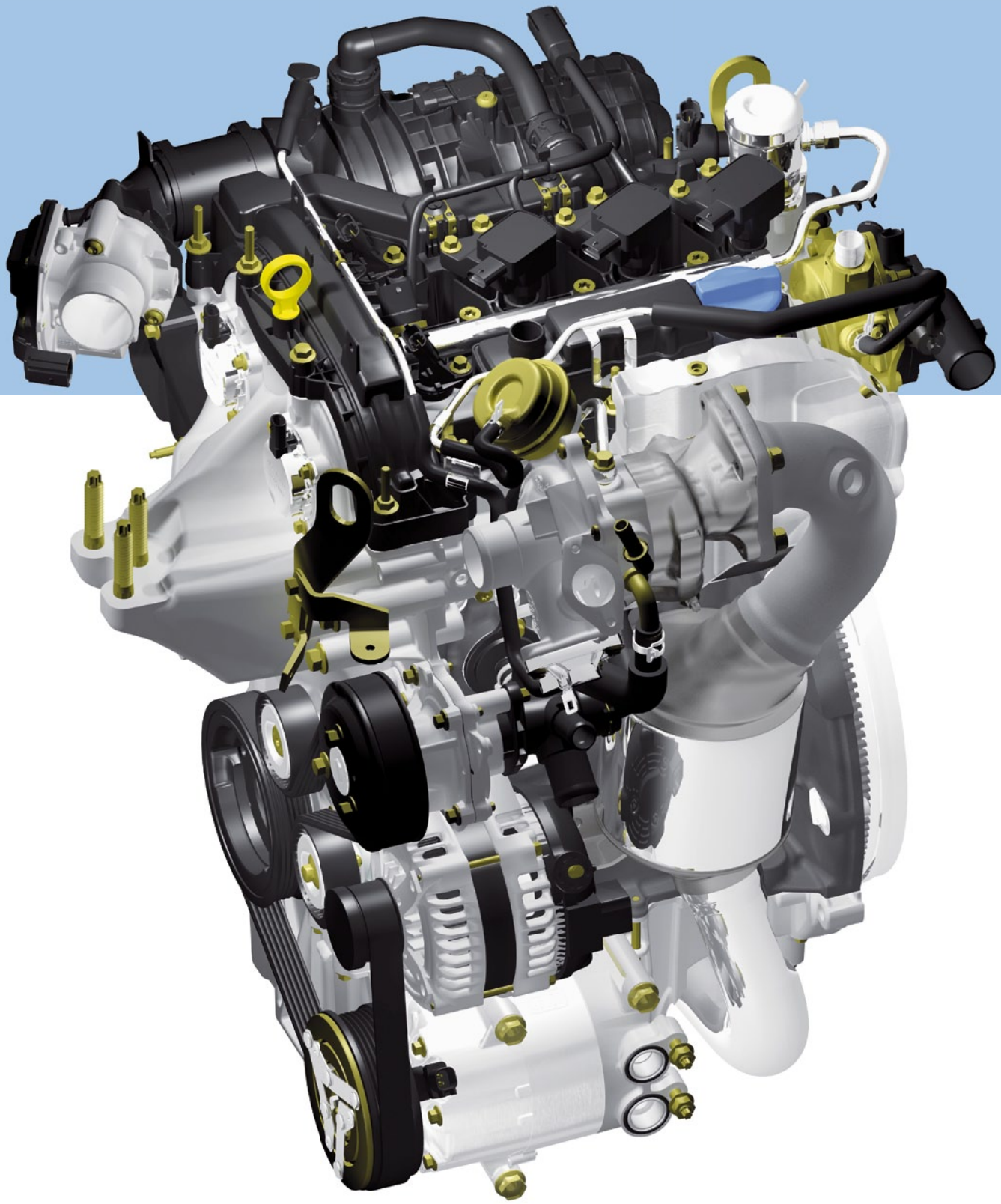
Best regards,



RICHARD BACKHAUS,
Vice-Editor in Chief MTZ
Wiesbaden, 15 March 2012



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Ford has launched a new 1.0-l Eco Boost gasoline direct injection turbocharged three-cylinder engine as part of the Eco Boost engine family. The design is all new, and will be produced in high volumes in several plants for a global market. The engine combines excellent fuel economy with high power and torque at an affordable cost.

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THREE-CYLINDER GASOLINE ENGINE WITH DIRECT INJECTION

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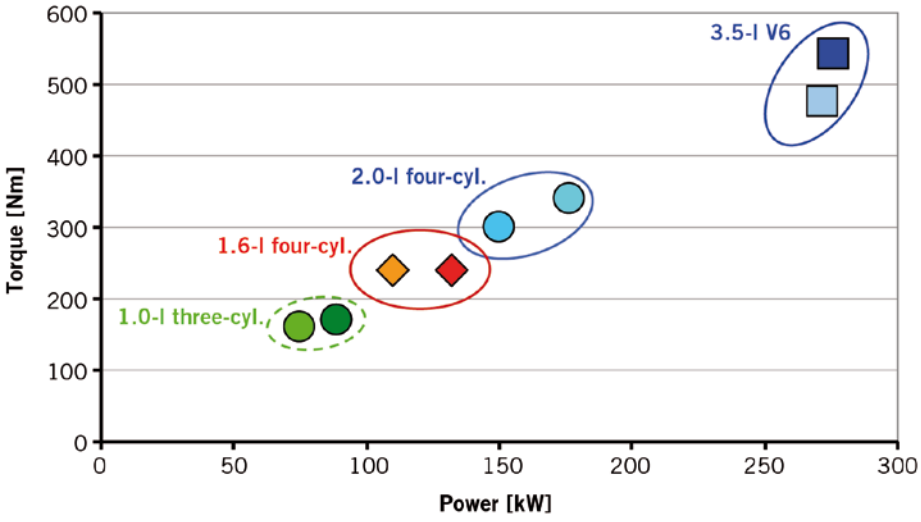


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DOWNSIZING ENGINE FAMILY

The 1.0-l Eco Boost is the newest and technologically most advanced member of the Ford Eco Boost engine family. It was developed with the close cooperation of Ford sites in Cologne, Aachen, Dunton and Dagenham. Currently the engine is being launched in the new Ford Focus, and there will be further vehicle applications later this year. Eco Boost technology is a key element of Ford Motor Company's global blueprint for sustainability [1]. This family of direct injection turbocharged gasoline engines is replacing higher displacement naturally aspirated engines in a very wide range of vehicles in markets around the world. It will give the customer significantly improved fuel economy and emissions without compromising performance feel and power. Within the Eco Boost portfolio the 1.0-l engine covers the torque and power range up to 170 Nm (200 Nm overboost) and 92 kW, ❶. It will replace the 1.25-l PFI, 1.4-l PFI and the 1.6-l TiVCT PFI engines on the medium term.

While the three-cylinder engine beats the 1.6-l engine in terms of performance feel due to its significantly higher torque



❶ 1.0-l engine as part of the Ford Eco Boost engine family

from 1500 to 4500 rpm, ❷, it has significantly better fuel economy than the 1.25-l PFI. In addition to downsizing and downspeeding, this very significant improvement is achieved by a comprehensive set of actions to reduce friction and pumping losses.

ENGINE TARGETS AND CONCEPT DEFINITION

Market studies identified for this decade two segments that could be satisfied with a common base:

: European volume segment around 74 to 92 kW
: small car segments around 48 and 59 kW in Europe but also in rapidly expanding Asian and South American markets, with naturally aspirated 1.0-l engines.

Initial engineering studies established that fuel economy and cost targets could best be met with a three-cylinder engine. Further studies then showed that a three-cylinder engine derived from existing architectures would not achieve the full functional potential and be more expensive than a dedicated small architecture. An all new small engine architecture was therefore developed with a 1.0-l in-line three-cylinder configuration as the basis. The challenge was to design an engine architecture that has excellent fuel economy using latest technologies in the turbo DI version, while offering an efficient but very low cost PFI engine for growth markets.

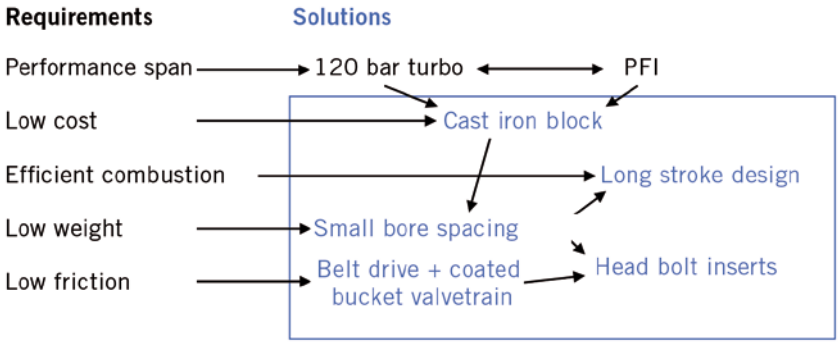
One key element to achieving these conflicting goals is a cast iron block, with an unsurpassed combination of strength and low cost. To keep the weight competitive the engine block has to be designed as compact as possible. This requirement is largely met through the small displacement and three cylinders, and further supported by a long stroke configuration, which also improves combustion efficiency and knock resistance. Even in the highly charged turbo version (24 bar BMEP), the cast iron block enables cooling slots in the bore bridge, which is only 6.1 mm wide.

Before this solution could be implemented, a compatible solution had to be found for the top end. Here the first challenge was to find an effective, low cost and compact solution for the valve train. By means of detailed friction breakdown studies and benchmarking, it was established that while the bucket tappet valve train in itself has higher friction than roller finger followers, this is only part of the losses. The primary drive also makes up a large part of the valve train friction, and here the belt-drive engines were typically much better. Hence a bucket tappet valve train was chosen, with a lifetime belt-in-oil drive. Through intensive development and implementation of polished tappets this valve train achieves friction levels similar to roller finger follower concepts, but at significantly lower cost. The small dimensions of the cylinder head created a packaging conflict between the tappets and the cylinder head bolts, which was solved by threaded steel inserts in the head, ❸.

Having settled the basic configuration of the engine, another major question was the configuration of intake, exhaust and accessory systems. Studies showed that keeping the accessories, turbocharger and catalyst on the same side of the engine offers significant advantages in crash performance and package. Putting them all on the front side provided easier thermal management as well as

❷ Main dimensions, material selection and performance metrics of the new engine

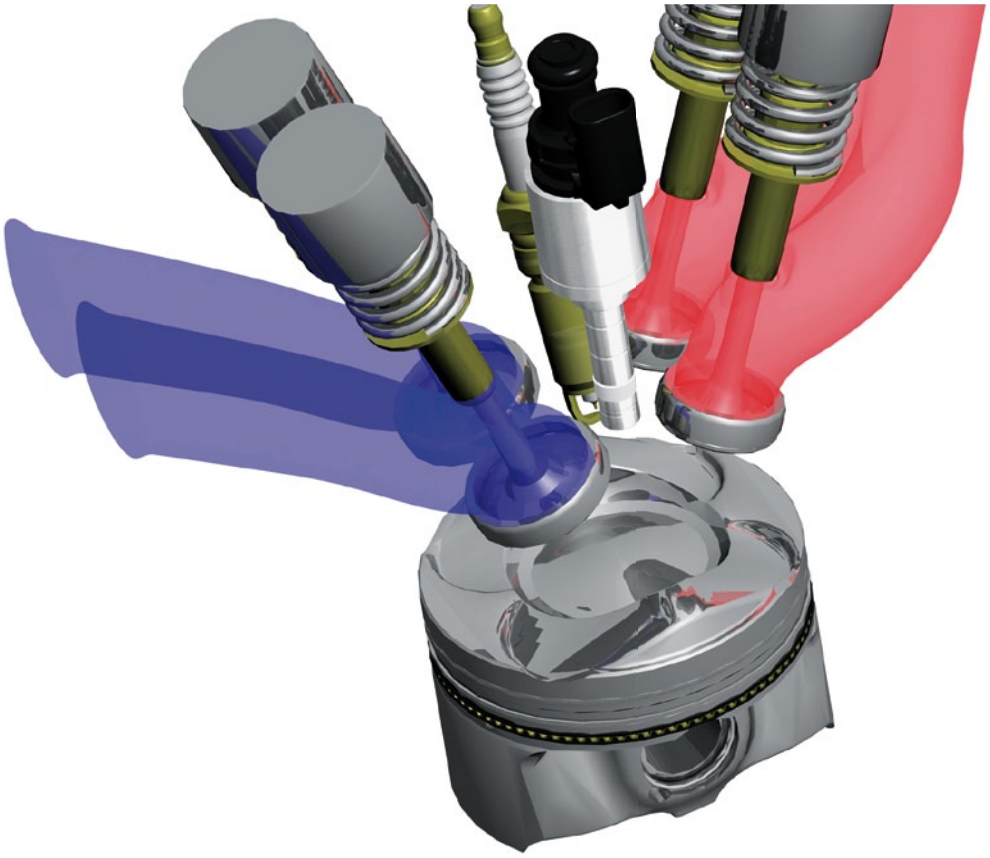
DISPLACEMENT [cm³]	999
VALVE TRAIN	DOHC
VALVES/CYLINDER	4/3
BORE x STROKE [mm x mm]	71.9 × 82.0
BORE/STROKE RATIO	0.88
BORE SPACING [mm]	78
COMPRESSION RATIO	10:1
MAX. POWER [kW at rpm]	92 at 6000 (95 RON)
MAX. TORQUE [Nm at rpm]	170 at 1500 to 4500 (200 Nm overboost)
FUEL SYSTEM	DI
IGNITION SYSTEM	Individual COP with integrated driver
FUEL GRADE (RON)	91 to 98 RON and E85
CYLINDER HEAD	Aluminium
CYLINDER BLOCK	Cast iron
CRANKSHAFT	Cast iron
CAMSHAFT	Cast iron
CONNCETION ROD	Sinter forged
VALVE ANGLE [°]	Int 19.4 and exh 21.5
VALVE DIAMETER [mm]	Int 27.1 and exh 23.1
INTAKE VALVE [mm]	8.26
EXHAUST VALVE LIFT [mm]	8.26



❸ Requirements driving the concept choices

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❹ Combustion system



commonality with the diesel engines used in these vehicles. The thermal challenge of having the alternator close to the exhaust was substantially eased by the integrated exhaust manifold, which radiates significantly less heat versus a conventional manifold.

COMBUSTION SYSTEM

The new 1.0-l is the second of Ford's Eco Boost engines to be featured with the next generation DI combustion system based on central injection. This system was adapted to the smaller bore and the long stroke geometry. It enables outstanding mixture preparation and maximum flexibility in terms of operational strategies. The longitudinal arrangement of injector and spark plug in one plane is beneficial with respect to valve sizes and cooling especially for this very compact engine with 71.9 mm bore. In contrast to the 1.6-l which has an M14 spark plug, the 1.0-l has an M12 to further optimise valve sizes and position, while centring the spark location in the combustion chamber.

The spray pattern of the solenoid driven multi-hole injector has been optimised by CFD in order to suit the smaller bore and the charge motion with minimised wall wetting. The intake ports are designed as straight high tumble ports with optimised flow coefficients, providing the base prerequisite for best possible mixture formation even under the most difficult conditions of high injected fuel quantities during boosted operation. The stratification capability of this combustion system can be used for flexible split injection strategies in homogenous mode and also for future fully stratified part load operation. This is supported by a shallow bowl in the piston crown which was carefully developed to ensure effective overall homogenous operation, ❹.

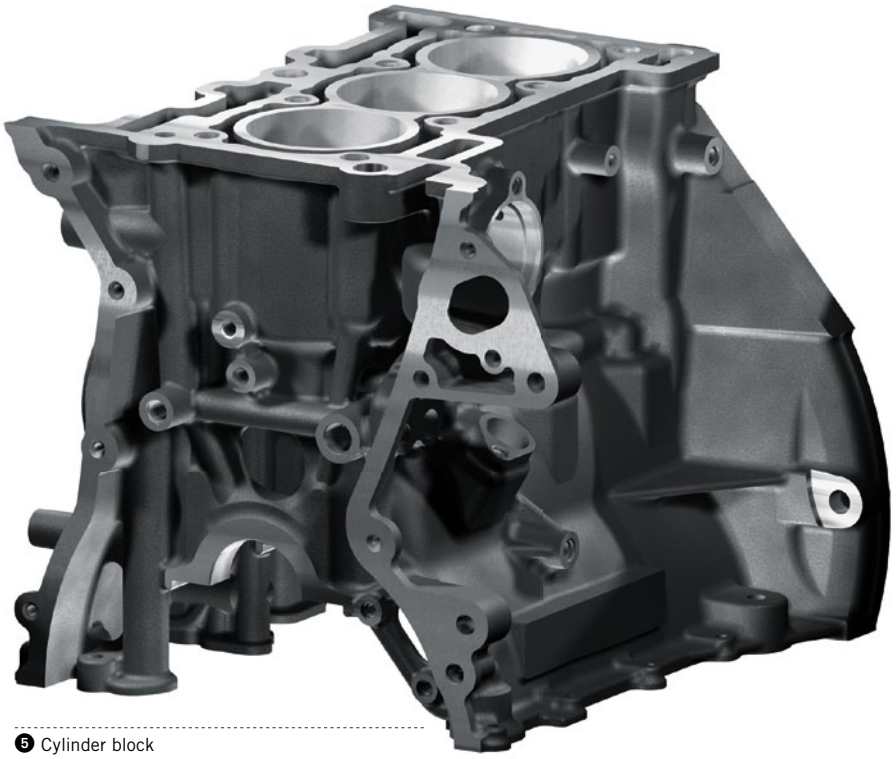
CYLINDER BLOCK

The cylinder block has consequently been designed for downsizing not only by means of displacement reduction and high BMEP but also with regard to the outer dimensions. An overall block length of 286.2 mm was achieved, ❺. This is a prerequisite for future small vehicles and electrification strategies. To enable this size a long stroke concept in combination with a bore spacing of 78 mm and a bore bridge of 6.1 mm was chosen. This compact design was enabled by using grey cast iron. The small size of the block minimised the weight disadvantage versus aluminium but delivered advantages which could not have been achieved by using aluminium, e.g. faster warm-up and lower oil flow of the main bearings. In addition the lower cost supports low cost variants of the engine in emerging markets.

In order to reduce bore distortion and knock sensitivity an open deck design was chosen. To further reduce the temperature in critical areas cooling slots have been incorporated into the bore bridges. A second thermostat has been integrated into the block which can fully isolate the block water jacket from the rest of the cooling system. In combination with the cast iron material, this enables a much faster warm up than any previous Ford engine.

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The long skirt design offers high stiffness to the block structure and delivers, together with the short length of the engine, a powertrain bending behaviour which usually can only be achieved with much more sophisticated and expensive bedplate designs.



❺ Cylinder block

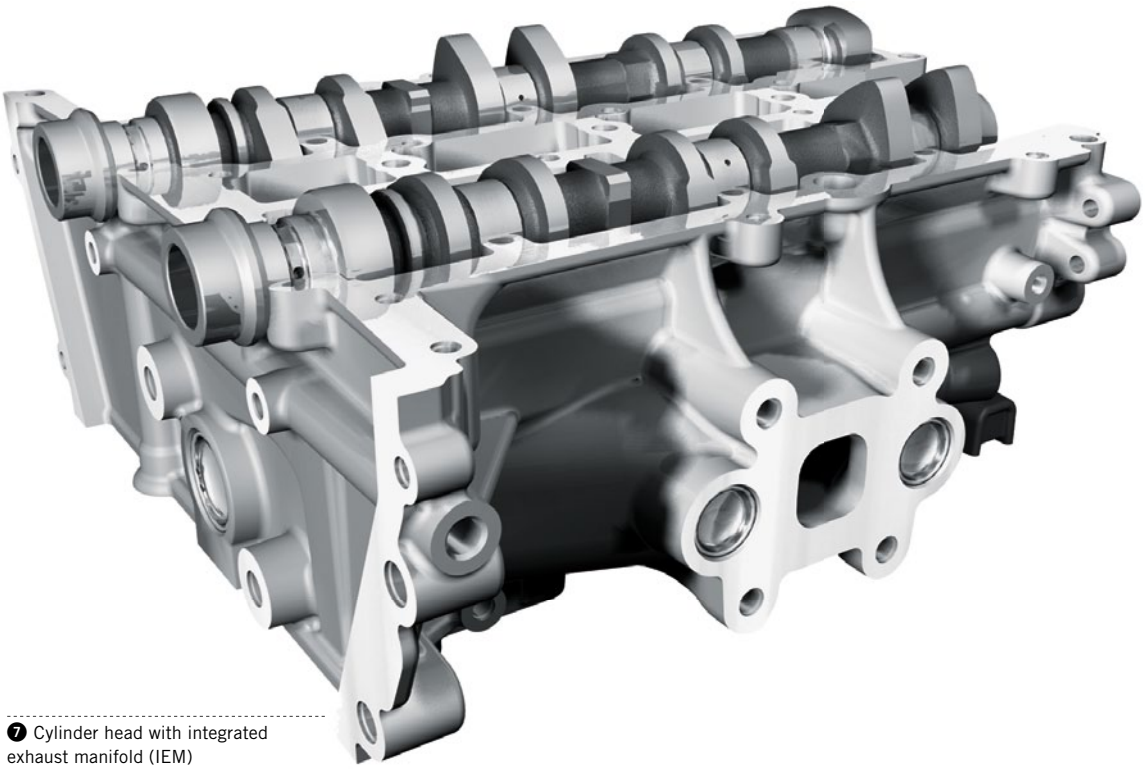
CRANKTRAIN

Due to the short length of this engine, in combination with the long stroke concept and the high specific output, it was a challenge to design a crankshaft that fulfils all needs in terms of strength without compromising parasitic losses by large bearings and the necessity of a balance shaft. Intensive usage of CAE helped to fulfil these conflicting targets and resulted in a cast iron crankshaft with main bearing diameters of 44 mm and conrod bearing diameters of 40 mm, ❻, which can usually be found on naturally aspirated engines with comparable dimensions. This together with a crank offset of 8 mm delivered a substantial contribution to fulfil the fuel economy targets. Originally the engine had been designed with a balance shaft drive, which was used in the first prototypes. Early on in the project however, it was established that the ambitious NVH targets can be met without a balance shaft, which helped to further reduce weight, cost and friction. The enablers for this are the short length of the crankshaft in combination with an intelligent arrangement of counterweights.



❻ Cranktrain

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❼ Cylinder head with integrated exhaust manifold (IEM)

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INTEGRATED EXHAUST MANIFOLD

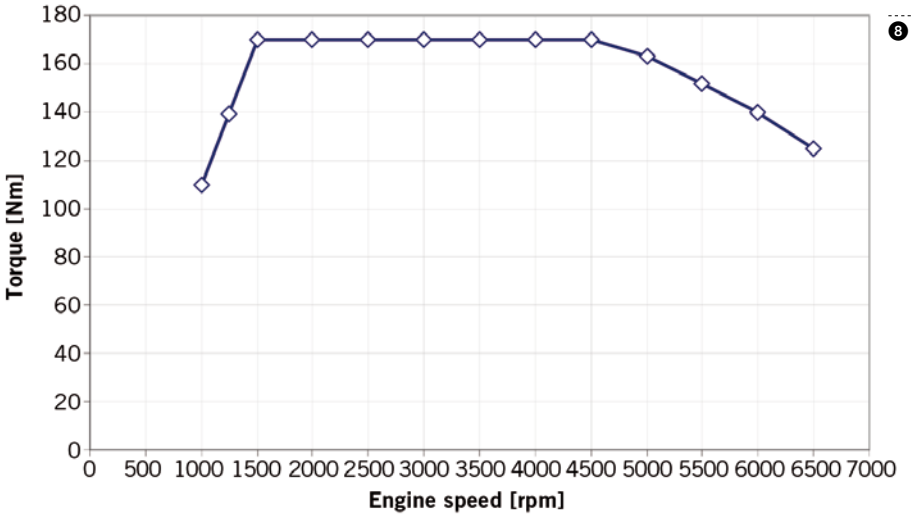
The 1.0-l Eco Boost is the first of Ford's engines in Europe which goes into production with an integrated exhaust manifold (IEM) [3]. A three-cylinder engine is predestined for this technology due to the favourable pulse separation. In contrast to four-cylinder engines, the manifold volume in three-cylinder engines can be kept small without risking backflow in neighbouring cylinders. This helped to keep the cylinder head very compact and realise the full potential of an IEM, ❼. Key benefits of the IEM are improved fuel economy and complexity at an affordable price. The fuel economy benefit in the NEDC is caused by two different effects. One of these is the much faster warm up of the engine resulting in parasitic loss reduction. The other is a faster catalyst warm up so that fuel consuming catalyst heating measures can significantly be reduced. This effect is caused by the smaller surface of the exhaust duct between the valves and the catalyst compared to a conventional manifold. Not relevant for the NEDC but especially important for customer satisfaction is the real world fuel economy. Here the IEM delivers the largest fuel economy benefit as it enables almost full range lambda 1 capability without enrichment for component protection. A further major benefit of the IEM is a significant reduction in cost as no expensive separate manifold with high Nickel content including its gasket, bolts and heat shielding is required.

THERMAL MANAGEMENT

The main function of the split cooling system used in the engine is to accelerate warm-up, and hence improve NEDC fuel economy with quicker friction reduction. The coolant flow from the pump is channeled to the split cooling thermostat, which is mounted in the cylinder block. In cold running conditions, i.e. coolant temperature below 75 °C, this thermostat is closed shutting off flow to the block water jacket and intake side of the head. In this condition all coolant flow is sent to the exhaust side of the head, i.e. through the exhaust valve bridges and integrated exhaust manifold, and to the turbocharger. Once the split cooling thermostat opens at 75 °C, flow is opened to the block water jacket and inlet side of the head. Then the whole system acts in a conventional manner, being controlled by the main 92 °C thermostat on the outlet of the head.

Early on in the project, CAE analysis showed that the high specific performance of the engine made cooling of the bore bridges necessary. Hence an innovative inter-bore cooling slot was developed. The machined slot is 1.2 mm wide leaving 2.45 mm each side for sealing. The coolant is fed into the slot from the cylinder block via a recess in the flame face of the head, and is returned to the exhaust side of the cylinder head. With this design, the coolant pressure delta is always sufficient to ensure adequate flow in the thermally highly loaded bore bridge.

The thermal management of the cylinder head with its integrated exhaust manifold is a critical factor for durability. 3D transient conjugate heat transfer method was used to optimise the water jacket geometry and balance the flow between the hottest areas of the IEM and the exhaust valve bridges. The challenge was to keep the flow rate in all areas between 0.5 and 6 m/s, in order to prevent film boiling at low flow rates and material erosion at high flow rates, while maintaining the required heat rejection performance.



❽ Torque curve

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LUBRICATION SYSTEM

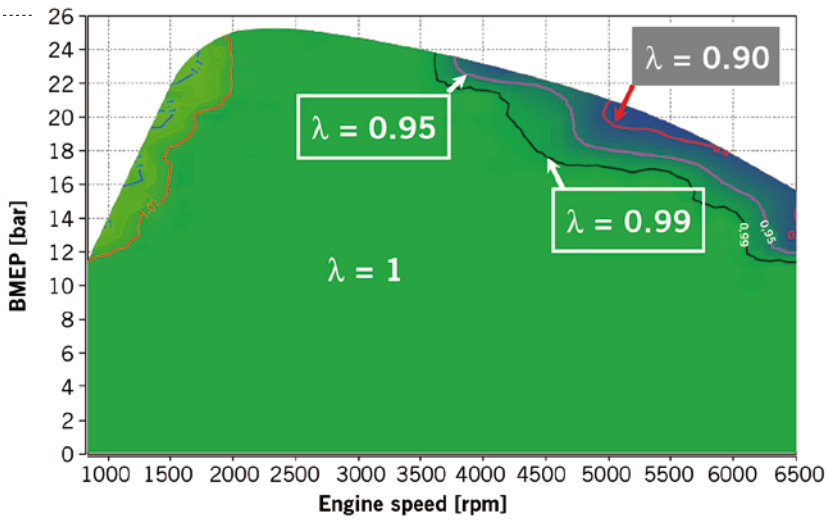
The oil pump is driven by the crank shaft via a 10mm wide belt-in-oil. The belt-in-oil runs without a tensioner, which offers friction and cost advantages as well as excellent NVH performance. The switchable volume flow variable vane oil pump can be run at two target pressure set points of 2 and 4 bar. The oil pressure is actively controlled for various running conditions via a solenoid according to a map in the engine control module. The piston cooling jet flow is controlled individually by a ball valve in each jet. The opening pressure of 2.5 bar is set to provide full flow with the 4 bar "high" oil pump set point, meaning the jets can be switched via the oil pump set pressure by the engine control system. Main oil gallery feedback ensures the flow output of the pump is automatically matched to the engine requirements for any condition.

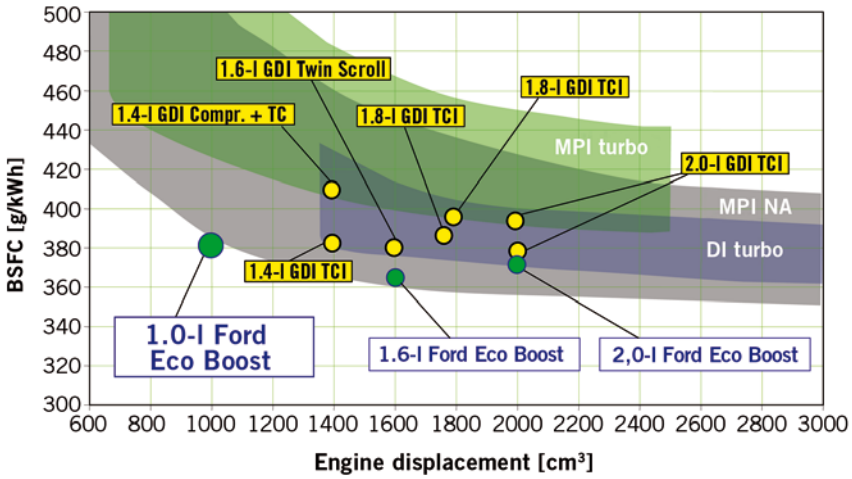
ENGINE OPERATION AND FUEL ECONOMY

The 1.0-l Eco Boost engine uses scavenging strategies at low speeds and high loads to improve low end torque and responsiveness. This leads to very good driveability even in heavier vehicles. The torque kneepoint is reached at 1500 rpm, ❽, and time to get from the naturally aspirated torque to 90 % of maximum torque at 1500 rpm is just 1.5 s. In low part load the engine runs high amounts of internal EGR. Here the cam control is applying a late/late strategy, so that valve overlap is always after gas exchange TDC at throttled part load. Due to this strategy residual gas is not getting into the intake ports, which helps to reduce intake port deposit formation. In order to reduce the occurrence of megaknocks, which charged engines tend to show, a control strategy was developed to detect pre-ignition and to implement countermeasures. Although the engine hardware is robust to a high number of events, the phenomenon has to be prevented in the interest of customer satisfaction.

The engine achieves stochiometric operation almost across the whole operating range, only in a small area near

❾ Lambda map





❿ Break specific fuel consumption at 2000 rpm and 2 bar BMEP (source of scatter band: AVL 2010)

maximum power, some very moderate enrichment is required to achieve the power, ❾. The combination of the excellent combustion and the comprehensive set of design actions for reduction of friction and pumping losses enables the engine to deliver class-leading fuel economy over the whole operating range. ❿ shows the specific fuel consumption at 2000 rpm/2 bar bmep in the competitive environment. With 382 g/kWh, the new 1.0-l engine is clearly below the low end of the scatter band – even for naturally aspirated PFI engines, which makes it the new DI turbo benchmark in the low displacement segment. Accordingly the 92 kW variant in the Focus could be homologated with 114 g CO_2 emissions/km, the 74 kW version with just 109 g/km. In this vehicle class, this fuel economy level could – up to now – only be achieved with diesel engines or expensive hybrids.

SUMMARY

The 1.0-l Eco Boost Ford is a no-compromise design for fuel economy, while also delivering good performance feel and the fun-to-drive element. This was achieved at a very affordable cost, which makes the engine attractive for high volume usage in the price-sensitive small and medium vehicle segments, as well as providing the basis for future applications in emerging markets. With its very small dimensions the engine is suitable for future vehicle platforms with regard to safety requirements, design and electrification strategies.

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KSPG RANGE EXTENDER
A NEW PATHFINDER TO ELECTROMOBILITY

As a modular component of a serial hybrid propulsion concept, the range extender has the technical potential to compensate for the well-known shortcomings of electric vehicles, serving as a bridge technology and paving the way forward for small and very small cars. Customer acceptance here depends heavily on NVH comfort, i.e. noise, vibration and harshness. For this reason, and for the first time, the quiet and low-vibration range extender now being presented by KSPG and FEV features a new roll moment compensation, recently developed by FEV.

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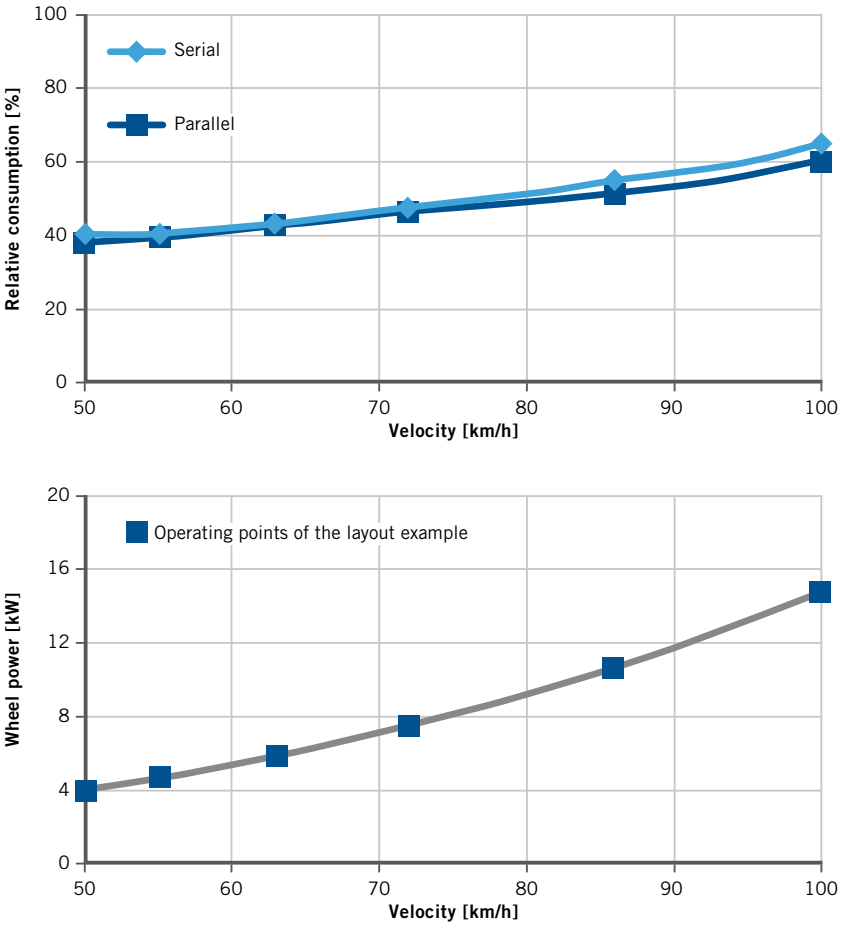


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OPERATIONAL PARAMETERS

In the concept for a plug-in range extender vehicle, an electric traction motor is the primary means of forward propulsion, coupled with a relatively small combustion engine in case of need. The range extender thus constitutes a suitable bridge technology for compensating for the range, heating and air conditioning limitations of battery-powered electric vehicles. This technology is aligned with government policies calling for a rapid shift to electromobility. In the case of serial hybrid propulsion, there is no mechanical connection between the propulsion system and the combustion engine, which is solely used for generating electricity. In order to avoid the progressive loss of efficiency resulting from multiple energy conversions, concepts featuring an optional mechanical driveshaft are also under discussion. Owing to the free shifting of load points, however, a direct comparison reveals that the disadvantages with regard to the efficiency chain are largely offset, and it is only at higher speeds that the mechanical driveshaft offers a significant advantage, ❶. The range extender, ❷, a modular component of a serial hybrid propulsion concept [1, 2] developed by KSPG AG in cooperation with FEV GmbH, therefore focuses on the new segment of compact city cars.

However, despite its intrinsic advantages, the range extender remains controversial in this segment of the vehicle market. These vehicles are designed for urban mobility, making zero emissions the prime consideration. In addition, with small cars it is harder to justify the need for propulsion redundancy. Depending on the electricity mix, in most countries zero-emission travel is a purely local phenomenon. Despite the repeatedly trotted-out daily kilometre statistics, there is no denying the fear caused by current range limitations. The same is true of customer expectations regarding heating and air-conditioning, which have yet to be met. Evidently,



❶ Comparison of fuel consumption at constant speed and charge sustaining with serial and parallel hybrid drive

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increasing the energy density of the electrical energy storage system by a factor of two to three for a series application will not be possible in the foreseeable future, nor will a comparable reduction in costs [3]. Moreover, when account is taken of currently strong legislative backing for the range extender with respect to CO_2 certification in the New European Driving Cycle (NEDC), ❸, the range extender becomes an increasingly attractive option.

❷ Range extender basic module without add-on components

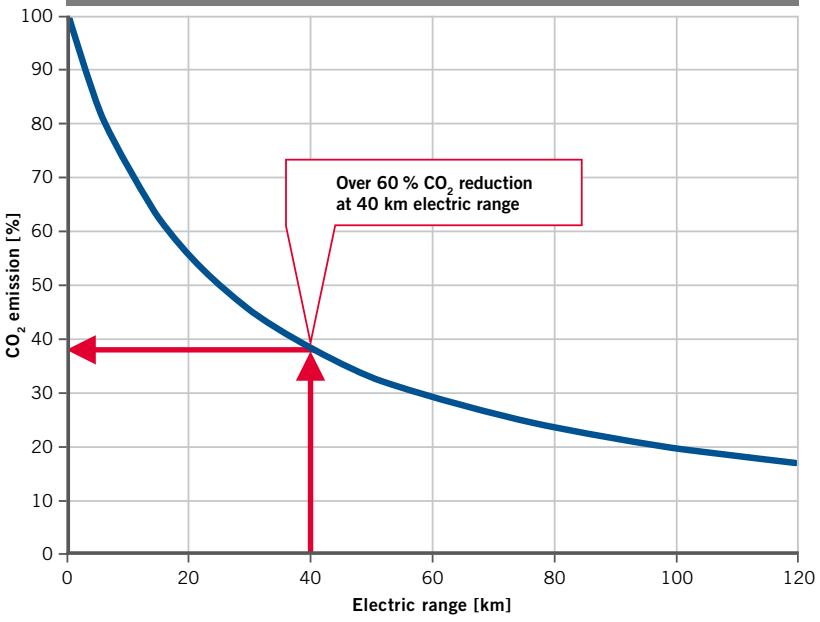


For an automotive supplier, there are multiple motivations for making a commitment. To say that carmakers would predominantly opt to use their own combustion engines (core competence, exploitation of capacity, etc.) is certainly an accurate assessment of current market potential. On the other hand, hardly any of them have small two-cylinder engines in their inventories. When new developments become unavoidable, automotive suppliers see themselves as partners of the automotive industry rather than its competitors. Sourcing a range extender externally, for example, could prove to be a sensible option when automakers do not want to invest in small or medium-sized production lines in-house. Producing on behalf of a carmaker would also make it possible to meet the needs of niche vehicle suppliers at an affordable price. Drawing on the combined expertise of the KSPG Group, the range extender could potentially point the way to a new field of business.

❸ Calculation of CO_2 emissions in accordance with ECE R101 (top) and influence of electric range in the NEDC on CO_2 emissions (bottom)

$$CO_{2,50Cmax} = 0 \text{ If one entire NEDC can be driven electrically}$$

$$CO_2\,[g/km] = \frac{\text{Electric range [km]} \times CO_{2,SOCmax}\,[g/km] + 25\text{ km}\; CO_{2,SOCmin}\,[g/km]}{\text{Electric range [km]} + 25\text{ km}}$$

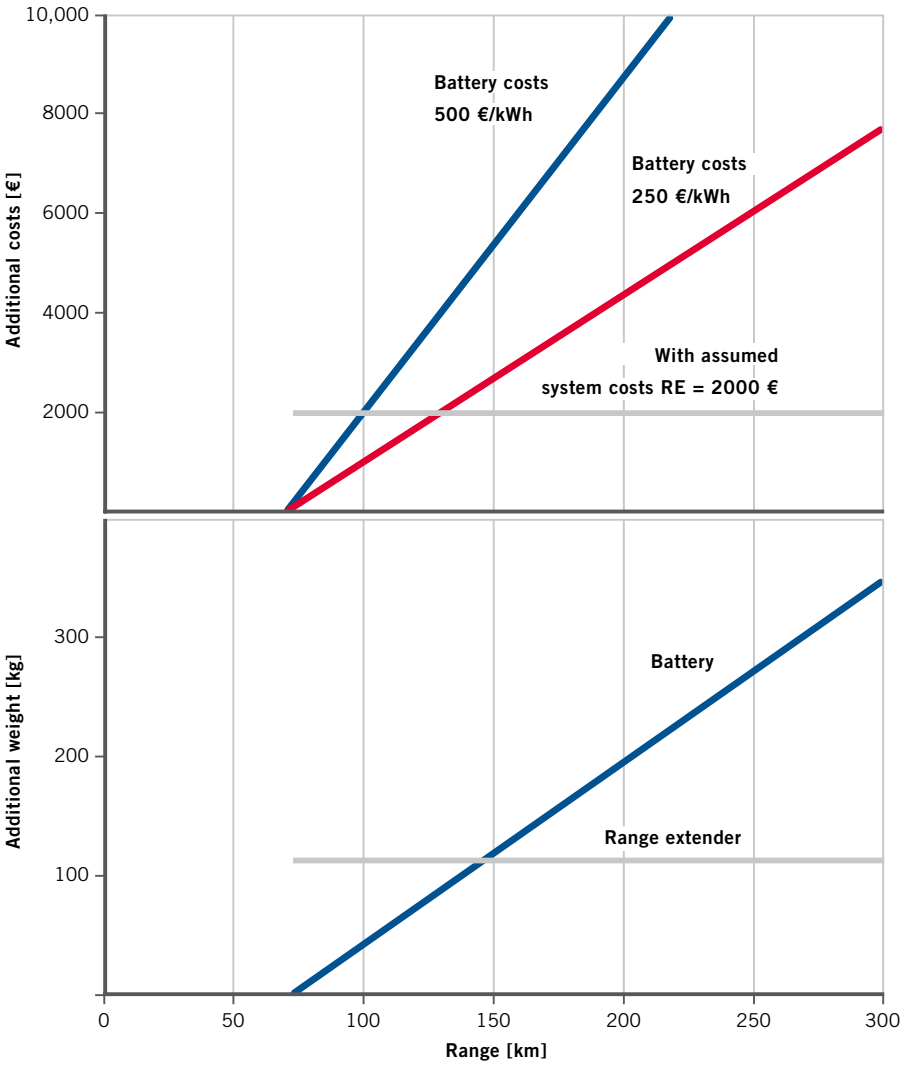


REQUIREMENTS IMPOSED ON THE RANGE EXTENDER

Automotive suppliers attach great importance to a compact modular design that simultaneously minimises the number of interfaces to the vehicle while limiting development and application costs, thus resulting in positive economies of scale. This dictates the use of a purely serial hybrid drive without a mechanical driveshaft. A compact unit installed in a space-saving manner should offer maximum possible freedom with regard to an alternative vehicle package and styling. The range extender presented by KSPG und FEV at IAA 2011 represents just such an attractive vehicle package designed for under-the-floor installation in the rear section of the vehicle, Cover Figure.

The performance level of the internal combustion engine and the two generators arises from the need for adequate climbing power, the aim being to prevent conflicts with vehicles with more powerful engines, thus assuring a stress-free driving experience with a minimum battery charge (SOC_{min}). A speed of 100 km/h on an incline of 3 % and a solid 80 km/h on 5.3 % incline (extremely steep for a Ger-

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❹ Sample cost and weight comparison between a vehicle with a range extender (REEV) and a battery-powered electric vehicle (BEV) (70 km electric range assumed for REEV)

COMBUSTION ENGINE	
Engine type	Two-cylinder four-stroke SI-engine in 90° V configuration
Displacement	799 cm³
Bore x stroke	80 mm × 79.5 mm
Max. power	30 kW at 4500 rpm
Brake mean effective pressure	10.3 bar at 4500 rpm
Max. torque	66 Nm at 4500 rpm
Valve train	OHV, two valves per cylinder
Fuel injection system	Port fuel injection
GENERATOR	
Engine type	Permanent magnet synchronous machine Four pairs of poles in series connection
Max. electric power	2 × 15 kW
Drive	Gear transmission ratio 1:2
MODULE	
Cooling	Combined cooling circuit for inverter, generator and combustion engine
Emission class	Euro 6
Weight (as in ❷)	62 kg
Length × width × height	665 × 550 × 355 mm

❺ Technical data of the range extender module

man autobahn) – based on design data for an assumed micro car – requires wheel power of 25.8 kW or, depending on the degree of efficiency, 30 kW output from the internal combustion engine.

The fact that the energy capacity of the expensive lithium-ion battery can be dimensioned on a very small scale also speaks for the range extender. Together, the battery and range extender must be less expensive than the comparatively large battery for purely electric operation. Moreover, there should also be weight savings. By way of example, and making certain assumptions, ❹ shows the point at which a range extender becomes less expensive and/or results in lower vehicle weight. Besides conceptual cornerstones calling for the most compact design possible and quiet electric driving – i.e. a low engine noise level (supported by downspeeding) and low engine vibration (NVH) – the top priority is to minimise the cost of manufacturing the motor-generator set.

TECHNICAL CONCEPT

The engine concept is specially tailored to the requirements of mini cars. The most important concept characteristics have already been described in detail [2]. One conceptual cornerstone is certainly the V2 design with a 90° V-angle. The two cylinder heads necessary for this stand opposed to the model-related advantages of low insertion height with a vertically installed crankshaft and, as the basis for meeting the extremely stringent NVH criteria, mass force compensation of the first order solely through the counterweights at the crankshaft and only minor moments of inertia of the first order and the mass effects of a higher order. Further NVH optimisation will rely on the first use of FEVcom, the patented roll moment compensation feature developed by the engineering partner FEV, which will be discussed in detail in the section after next. The engine block of the combustion engine, the generator casing and, as an option, the converter, are integrated together in the same aluminium housing and included in the same cooling circuit. The technical data for the range extender unit are summarised in ❺.

ENGINEERING AND DESIGN

Integrating the generators and the internal combustion engine in one housing on

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one hand enables an extremely compact package and systematic implementation of modular thinking, ❻. On the other hand, this also results in a set of entirely new engineering challenges that require a comprehensive approach to development. In one part of the shared housing the regular internal combustion engine parameters pertain, with high temperatures, oil mist and combustion gases. In the part of the housing, which contains the electric components, high electric voltages of several hundred volts have to be safely managed, while stringent requirements with respect to electromagnetic compatibility also have to be met. The generators must be kept at a much lower temperature than the internal combustion engine, and the entry of combustion gases must be prevented. Thus the two spaces must be permanently isolated from each other, even though that coolant has to be able to pass from one to the other and the mechanical output transmitted with minimal friction.



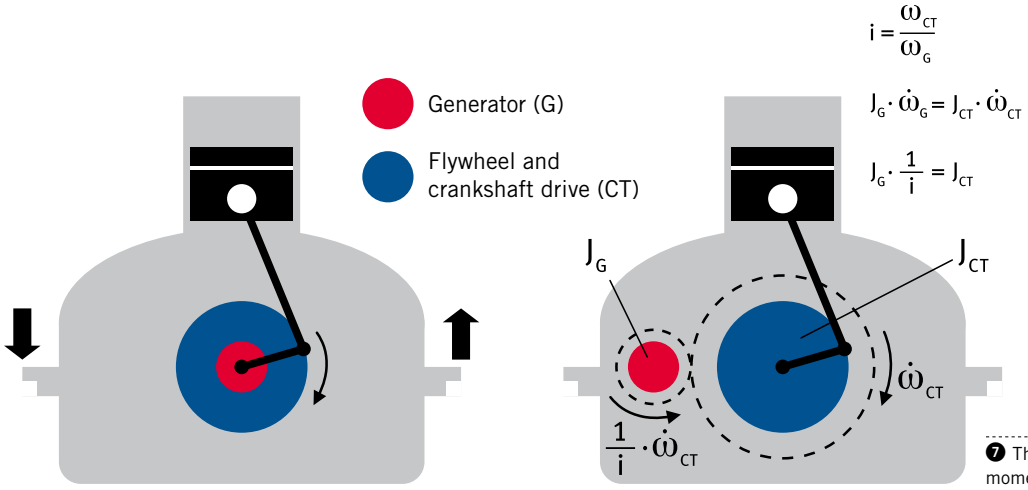
❻ Common aluminium housing for the combustion engine and generators

In this concept, the stators of the generators are pressed into the carrier housing, which is completely rinsed with coolant water. Despite the comparatively high temperature level, this ensures a safe dissipation of electrical power losses with simultaneous low loss of pressure. A controllable electric coolant pump governs the coolant volume flow based on demand.

The permanent magnet synchronous generators are connected electrically in series, thus reducing the connection between the two machines to three connector cables, with the converter requiring only one three-phase connection as an interface. Since both of the three-phase machines are mechanically synchronised through gearing, one resolver suffices for position evaluation. Owing to the fact that all of the electrical components are contained in a single housing, it is possible to use copper railings for the electrical contacts, which minimises the number of feedthroughs as well as providing a reliable shield against disruptive electromagnetic influences.

NOISE EXCITATION

Purely electric driving is characterised by very low noise emissions. A range extender is supposed to increase a vehicle's range without detracting from the experience of quiet driving. Therefore, one of the foremost development goals is low noise excitation [4, 5], which primarily derives from the gas and mass forces of the internal combustion engine. These lead to rigid body motion module and deformations of the structure, which then excite the chassis as air- and structure-borne sound. Optimisation of the internal combustion engine (combustion process, reduction of clearances, selection of materials and stiffnesses, housing optimisation) can minimise structure deforma-



❼ The patented FEVcom roll moment compensation system

tion. Rigid body motion of the module arises from free mass forces as well as gas and mass force-induced torque fluctuations, and can primarily be influenced through selection of the engine form (the number and arrangement of cylinders, the ignition sequence, rotating and oscillating masses, mass balance system) [6].

The torque fluctuations arising from the gas and mass forces cause an acceleration of the rotating inertial mass of the crankshaft and the generator rotor, and produce reactive moments that lead to a rolling motion of the range extender around its crankshaft axis. In conventional engine designs, this roll moment must be supported at the engine bearings and thus causes vibrations of the vehicle structure. Particularly in a low-noise electrically powered vehicle, the resulting structure-borne sound leads to an annoyingly high level of airborne sound emission in the passenger compartment.

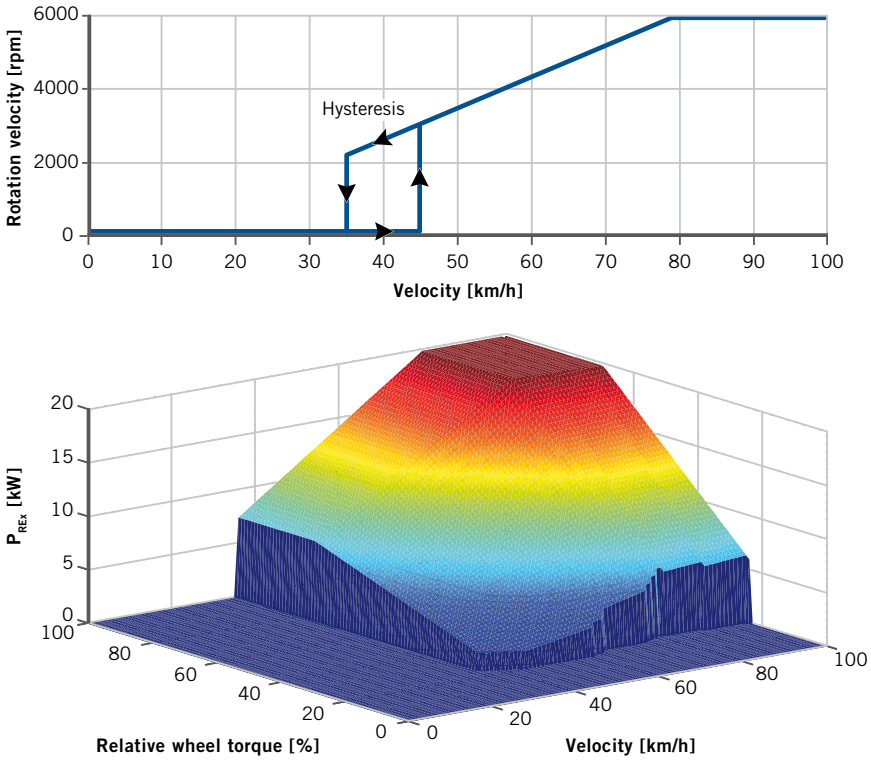
FEV has thus developed the system FEVcom (full engine vibration compensation) that effectively eliminates roll moments in the range extender. It operates on the principle of a rigid gear drive between the crankshaft and the rotor of the generator. A special feature of this concept is that – owing to the advantageous generator package in connection with the V2 design – the required mass inertia is distributed equally between the two gear driven rotors of the generators. The gear reverses the direction of rotation between the combustion engine and the generators, and the rotary inertias from the internal combustion engine and generators are harmonised (inertia of the rotors multiplied with the reciprocal value of the gear ratio corresponds to that of the cranktrain of the internal combustion engine). As a result, the reactive moments are supported directly within the housing, and have no effects on the bearings, ❼ [2].

OPERATING STRATEGY

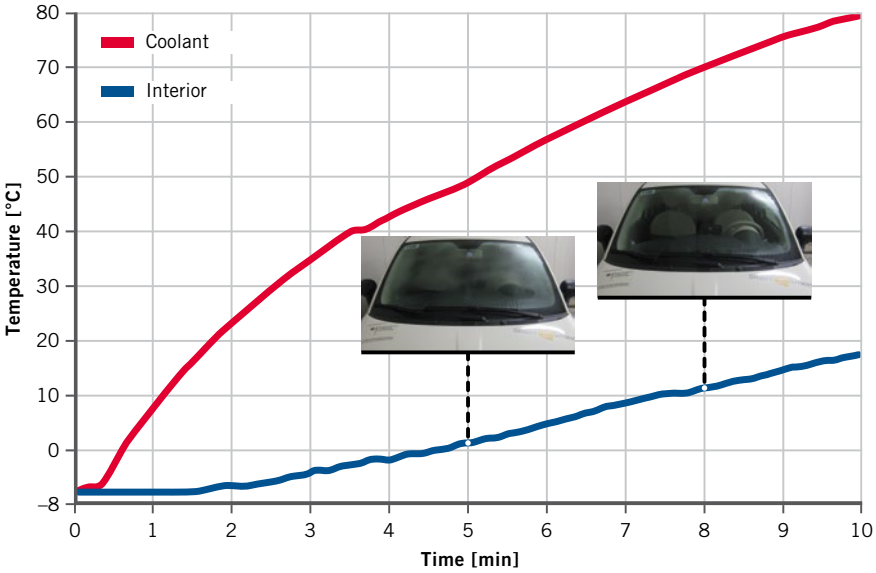
The objective of acoustically optimising the range extender to the degree that it only "hums like a refrigerator compressor" is desirable but essentially unobtainable without an unreasonable level of technical effort. Thus the operating strategy of the range extender, which possesses various degrees of freedom, must be selected so as to exploit effectively the existing velocity-dependent masking due to rolling and wind-induced noise of the electric vehicle. Various operating strategies have already been examined by FEV in a range extender electric vehicle with a Wankel engine. It quickly became clear that one- and two-point strategies would be disadvantageous from both the acoustic and efficiency point of view. A performance tracking mode, which tracks the required output, exploits the existing masking in optimal fashion, leading to greater driver acceptance in that the noise emissions are directly linked to the performance requirement. From the energy standpoint, taking into account the limited efficiency of battery charge and discharge, it also makes sense to conduct the energy generated directly to the electric drive rather than temporarily storing it in the battery.

The operating strategy developed by FEV during the course of numerous practical tests, depends on the battery condition, the speed of driving and the driver's desired wheel torque (accelerator pedal position), ❽ [7]. In line with the vehicle's design as a range extender-equipped electric vehicle, it will drive in purely electrically powered mode until a defined battery state-of-charge has been reached. The range extender switches on only once the battery state-of-charge falls below a defined level. A velocity-dependent switch-on threshold has been implemented here in order to assure adequate masking by the driving noises. At driving speeds above this switch-on threshold, the rotational velocity of the range extender is coupled to the driving speed, the torque to the driver's desired wheel torque. A hysteresis range around the switch-on threshold prevents constant switching on and off. Recuperation takes place during coasting down and braking, during which the range extender produces no output. Owing to the speed-dependent rotational velocity and the accelerator-dependent load control, a dynamic range extender noise can be generated that positively supports the driving impression of an electric vehicle, and approximates the typical background noise of a conventional vehicle.

Furthermore, the range extender can provide very efficient support for climate control of the passenger compartment by using the heat losses in winter (or at lower outside temperatures) to heat the



❽ Operating strategy of the FEV range extender with Wankel engine [7]



❾ Quick defrosting of windows and heating of passenger compartment solely by using rejected heat by the range extender

interior of the vehicle, ❾. Noise-optimised operation of the range extender with lower output (defrost mode) quickly warms the interior, and by producing electricity reduces the strain on the battery, thus increasing the vehicle's range.

SUMMARY AND OUTLOOK

The current state of the art in battery technology means that compact and micro cars are still the only vehicles that can be powered electrically. The high cost of batteries, short ranges and poor heating and air conditioning performance evidently stand in the way of successful marketing. The current driving habits of large segments of the population extend far beyond short-distance trips and low average daily driving. Quicker expansion of the electric car market will require efficient support measures in order to achieve the necessary degree of customer acceptance. The range extender, as a modular component of a serial hybrid propulsion concept, offers above all the technical potential to pave the way for the electric car during a period of transition whose duration is difficult to forecast. Given a goal-oriented design, it is able to compensate for the well-known shortcomings of battery-operated propulsion while simultaneously reducing the costs of driving and effectively limiting CO_2 emissions – without detracting from the experience of quiet driving. Jointly developed by KSPG and FEV, the range extender is characterised by special noise-attenuating features. Here, FEV's roll moment compensation system for minimising noise and vibration excitations in the vehicle structure will be used for the first time. With regard to the acoustic perception of the range extender, the special significance of the operating strategy must be emphasised.

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THE NEW BMW TURBOCHARGED SULEV 2.0-L FOUR-CYLINDER GASOLINE ENGINE

BMW's first turbocharged SULEV gasoline engine will be making its debut this year in the new BMW 328i PZEV. This not only means that the BMW Group is complying with the requirements of the world's most stringent exhaust emission legislation, but also underlines the fact that the term premium is being defined by the many facets of sustainability to an ever increasing extent. Twin Power Turbo technology enables dynamic performance and achievement of low fuel consumption while adhering to the most stringent exhaust emission limits.



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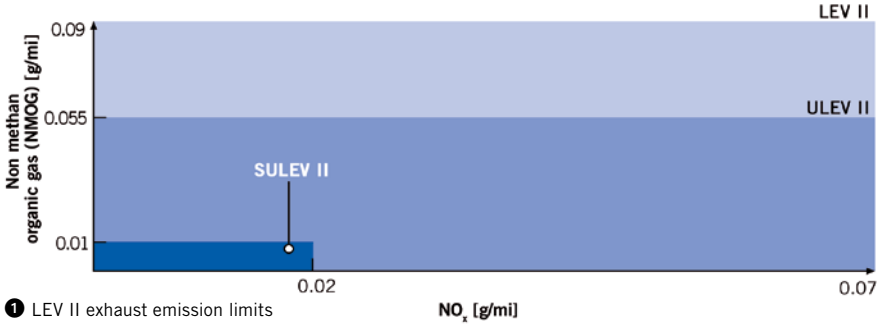
CHALLENGES IN THE DESIGN OF TURBOCHARGED SULEV GASOLINE ENGINES

The California Air Resources Board (CARB) has defined SULEV (Super Ultra Low Emission Vehicle), the world's most stringent exhaust emission standard, with its LEV II exhaust emission regulation, ❶, whereby the CARB exhaust emission regulations not only apply to California, but to other US American states as well. Compliance with the SULEV limits must be assured for 15 years and 150,000 miles before a vehicle can be approved as a Partial Zero Emission Vehicle (PZEV). Furthermore, the vehicle must not produce any fuel evaporative emissions and must adhere to the requirements for level 2 on-board diagnostics. Scheduled for implementation in model year 2015, the LEV III regulation will tighten the exhaust emission limits even further. The current and planned statutory regulations relating to vehicle fleet contaminant emission limits are generating a need for a continuously growing number of vehicles that comply with the SULEV limits. This means that SULEVs will no longer be niche models in the future, but rather the standard in the USA.

The cold start and subsequent catalyst warm-up phase hold the key to meeting the SULEV limits. The majority of all air contaminants are emitted during the first approximately 30 s of the exhaust emission test. This is due to the higher emission level of a cold engine and to the low level or lack of conversion on the part of the catalytic converter before the light-off temperature has been reached. The purpose of the catalyst heating mode at the beginning of the exhaust emission test is to use the exhaust gas to heat the catalytic converter up to light-off temperature as quickly as possible with a minimum of emissions and excellent smooth running. Turbocharged engines constitute a particular challenge in this respect as the exhaust-gas turbocharger acts as an additional heat sink en route to the catalytic converter, thereby delaying light-off accordingly. This article gives an account of the way in which BMW has solved the conflict of objectives between minimum exhaust emissions and other customer demands, such as fun to drive and fuel consumption, by means of innovative solutions.

COMBUSTION TECHNOLOGY AS ENABLER OF SUPER ULTRA LOW EMISSION CONCEPTS

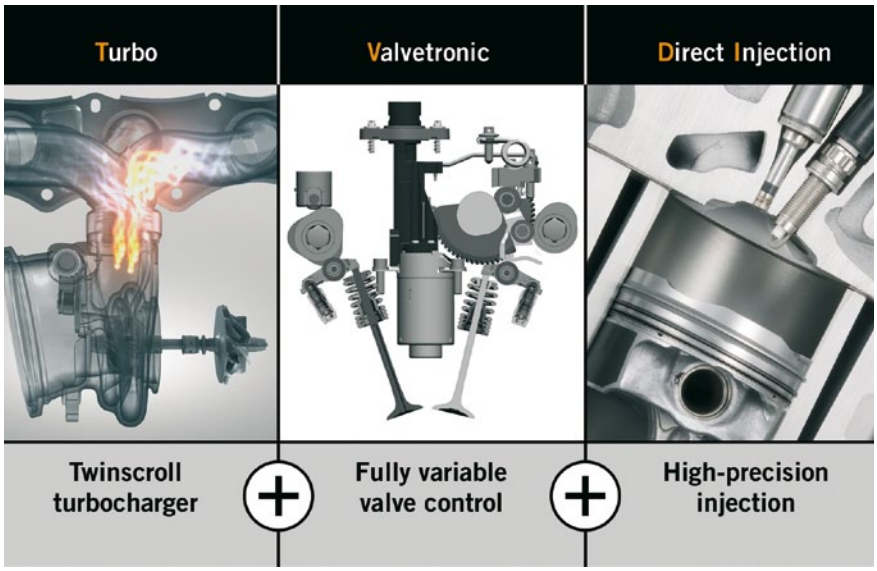
Twin Power Turbo technology [1] is being used to achieve the EfficientDynamics objectives – lower fuel consumption and more driving pleasure – in current and future BMW gasoline and diesel engines. This technology takes the form of the TVDI combus-



❶ LEV II exhaust emission limits

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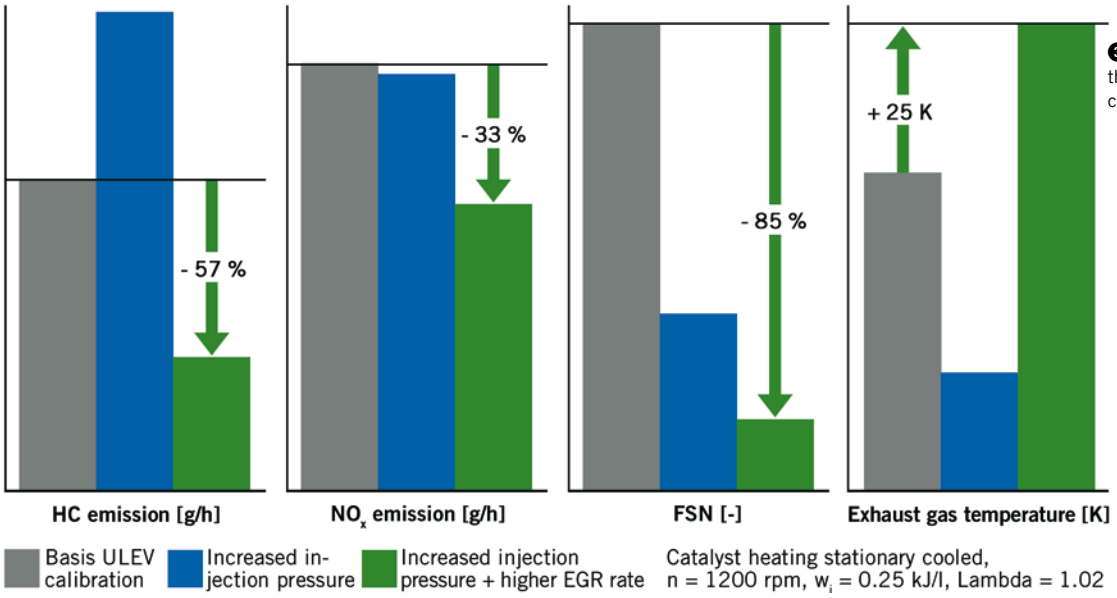
tion concept (combining turbocharging with Valvetronic and direct injection) in a gasoline engine, ❷. Fitted in its current six-cylinder TVDI gasoline engine [2], BMW has been consistently rolling this combustion concept out throughout the BMW Group's entire range of engines since 2009. The BMW 2.0-l four-cylinder gasoline engine [3] presented in 2011 is serving as the basis for the new turbocharged SULEV engine. The TVDI combustion concept enables the achievement of low fuel consumption along with the dynamic driving performance that is typical of BMW all over the world, irrespective of the locally available fuel quality. Furthermore, the degree of freedom offered by TVDI creates favourable prerequisites for the realisation of super ultra low emission concepts.



❷ TVDI combustion concept

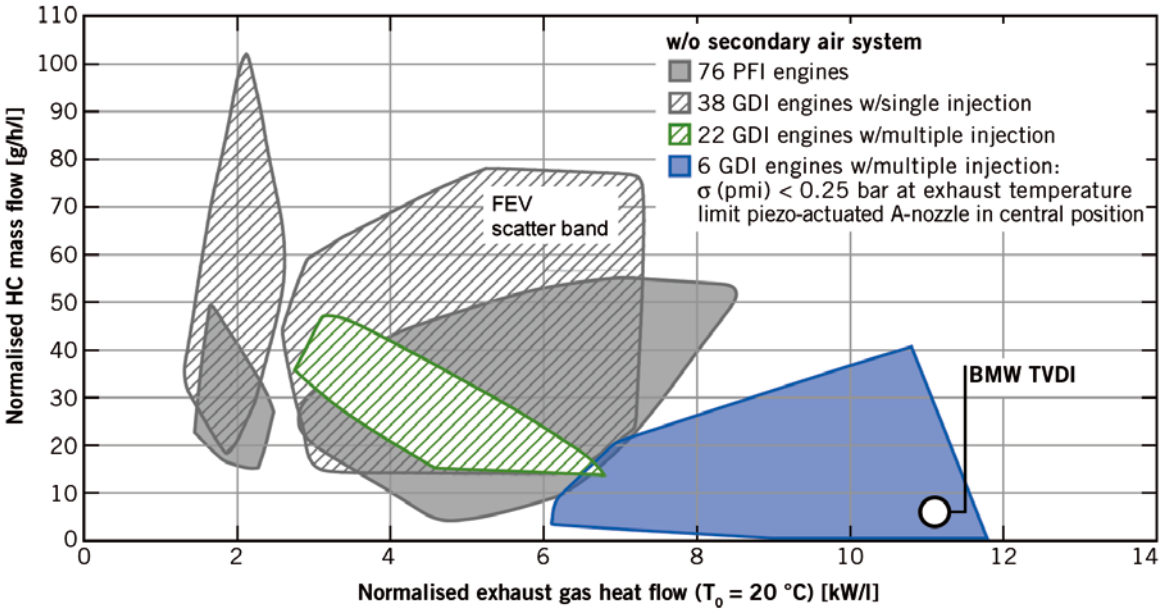
Because of its flexibility in terms of the number and duration of injection pulses per working cycle, the direct injection system with its centrally positioned injector is capable of a much more retarded ignition than can be achieved in an engine with a manifold injection system and it therefore accelerates the catalyst warm-up process considerably. A lean catalyst heating mode is used with double injection. The first injection takes place during the intake phase and contains the main injection volume. It serves to generate a homogeneous, lean basic mixture. The second, lower-volume injection is precipitated as the ignition injection just after ignition and it accelerates or stabilises the combustion process by increasing the turbulence in the already ignited basic mixture, enabling the application of a highly retarded ignition. This lean burn mode enables follow-up reactions with the excess oxygen in the exhaust gas, causing the temperature of the exhaust gas to rise and reducing the CO and HC emissions.

Based on the current ULEV application for catalyst heating operation, further functional improvements can be achieved by increasing the injection pressure from 120 to 200 bar and optimising the charge cycle. ❸ shows the emissions measured during catalyst heating mode with stationary cooling and the temperature of the exhaust gas upstream of the catalytic converter. The increase in injection pressure improves the atomisation and distribution of the fuel and therefore results in a considerable reduction in particle emissions. Becoming stronger as the injection pressure increases, the spray penetrates more deeply, however, so that a larger amount of fuel is sprayed onto the wall of the cylinder, thereby increasing the HC emissions. The variability of the valve train in the TVDI combustion concept can be used to compensate the rise in HC emissions resulting from the higher injection pres-



❸ Optimisation of the application in catalyst heating mode

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❹ Catalyst heating mode benchmark

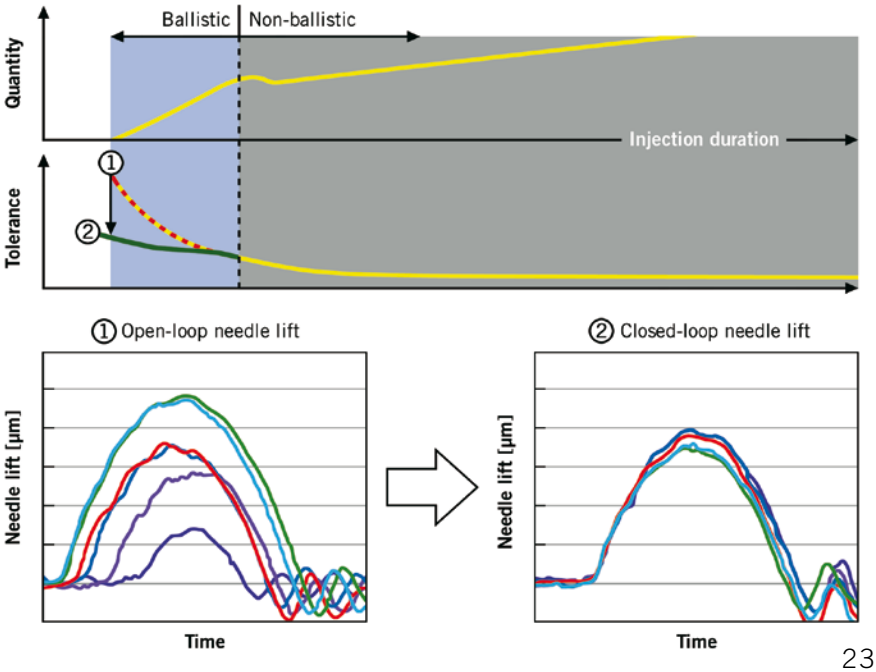
sure by increasing the volume of residual exhaust gas in the combustion chamber. The high residual exhaust gas content in the combustion chamber causes the charge temperature to rise, exerting a positive effect on the atomisation and distribution of the fuel and mixture preparation and ultimately leading to a reduction in HC and particle emissions. The improved NO_x value is the result of slower combustion at a lower peak temperature. The retarded position of the combustion centre of gravity leads to a desired increase in the temperature of the exhaust gas. Therefore, while keeping engine running smoothness constant, both engine-out emissions and catalyst light-off are improved.

❹ shows the catalyst heating mode in the TVDI combustion concept by comparison with competitive concepts. The HC mass flow is applied over the enthalpy flow of the exhaust gas – normalised according to the respective displacement in each case. The purpose of optimising the catalyst heating mode is to achieve maximum enthalpy flow while reducing HC emissions to a minimum at the same time. It is evident that the TVDI combustion concept with centrally positioned multi-hole injector is capable of achieving results within the range of the best piezoelectric DI combustion systems. The catalyst can be heated with up to 22 kW by means of the exhaust gas from the new 2.0-l four-cylinder TVDI gasoline engine.

CONTROLLED VALVE OPERATION AND IGNITION INJECTION OPTIMISATION POTENTIAL

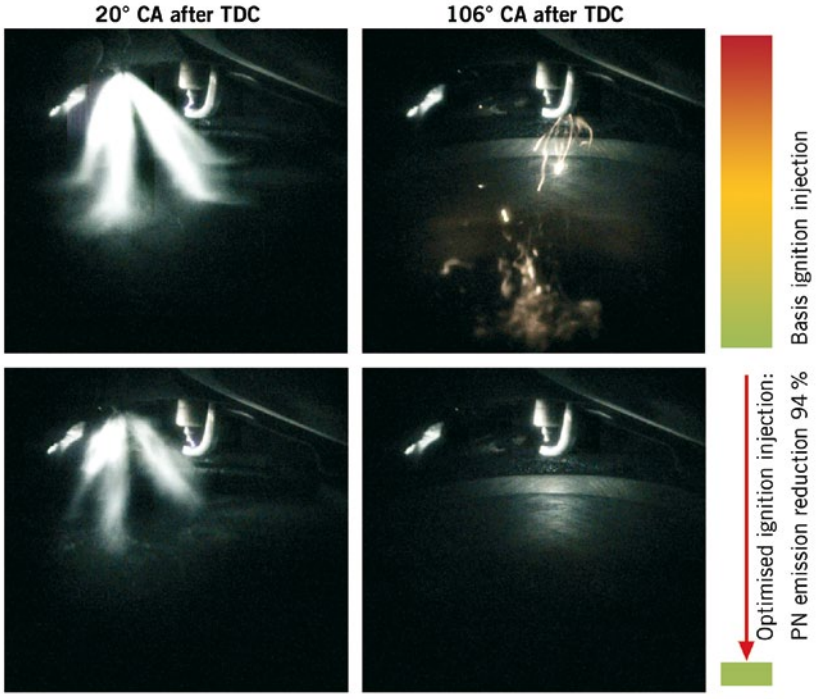
A function that monitors the injector needle opening duration and, with it, the injection volume within the ballistic section of the multi-hole injector characteristic by means of control and adaptation algorithms, ❺ (schematic view), is used in order to ensure that the minimum quantity required for the ignition injection remains stable throughout the life cycle of the vehicle. Characteristic features are identified in the patterns of the injector's electric drive signals for this and these are used to determine the opening time of the injector needle and correct it by adapting the drive circuit to ensure that injection of the minimum quantity required for TVDI catalyst heating mode is assured with a very high level of precision throughout the entire life cycle of the vehicle.

❺ Controlled valve operation ignition injection (schematic view)



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Optimisation of the quantity of fuel precipitated in the ignition injection is crucial to the achievement of low particle emissions in catalyst heating mode – and in the context of the Euro 6 legislation. The controlled valve operation described above offers advantages in this respect as well, as the ignition injection can be produced in a very reproducible manner by extremely small injection volumes. The upper sequence in ❻ shows an ignition injection that has not been optimised in terms of injection volume and subsequent combustion with highly diffusive components. It shows the relatively deep penetration of the spray, which causes fuel to be sprayed onto the piston and leads to very sooty combustion. The optimised, significantly shortened ignition injection prevents fuel being sprayed onto the piston. The diffusion flame produced from the rich zones in the middle of the combustion chamber is much less pronounced. The comparison shows a massive reduction in particle emissions where the ignition injection volume has been optimised. Further improvements to mixture homogenisation, which contribute towards effectively reducing emissions, are achieved by selectively optimising the injector's spray pattern.

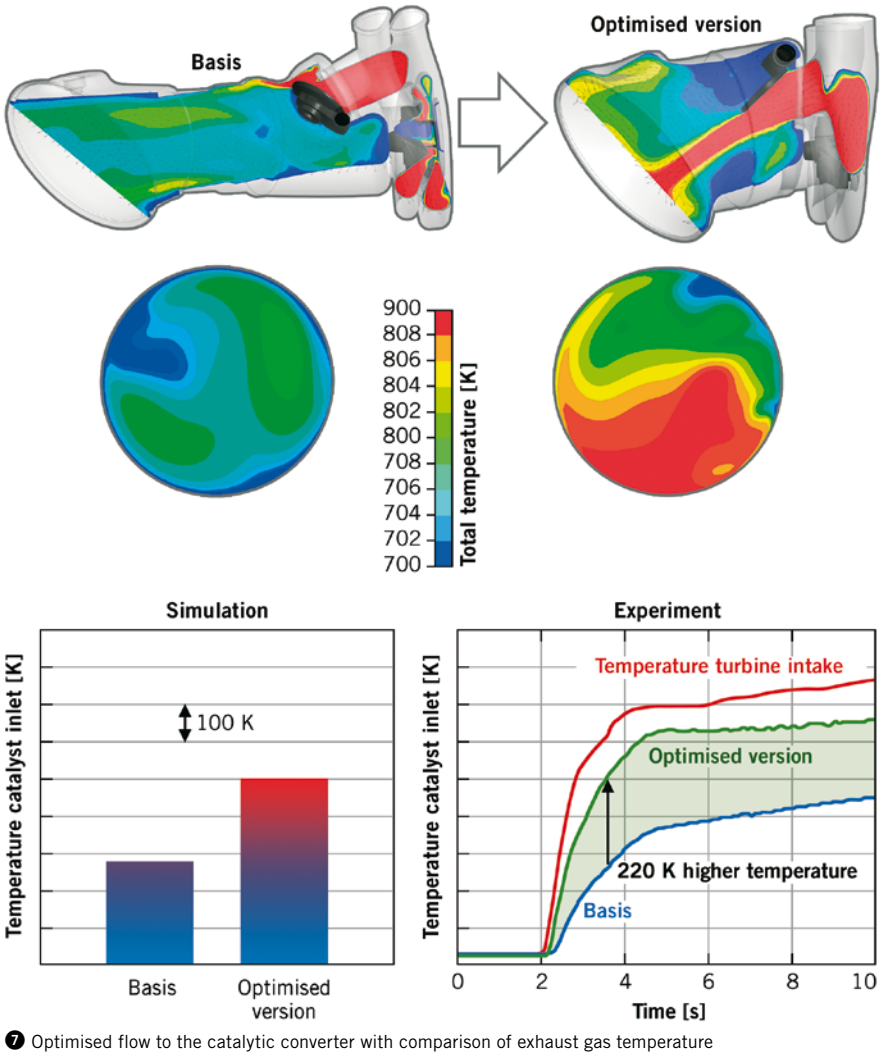


❻ Influence of ignition injection volume on particle emissions in catalyst heating mode

MINIMISING EXHAUST GAS HEAT LOSSES IN CATALYST HEATING MODE

Apart from combustion concept optimisation, another important prerequisite for compliance with the SULEV values is the minimisation of the heat loss by the exhaust gas en route to the catalytic converter. As mentioned above, the turbocharger constitutes a special challenge in its capacity as an additional heat sink upstream of the catalytic converter as it causes a considerable delay in lighting off. ❼ clearly shows a temperature drop of 300 K in the exhaust gas passing through the turbocharger. These massive heat losses are primarily due to the transfer of heat from the exhaust gas to the large surfaces of the turbine scrolls and wheel.

The amount of heat loss by the exhaust gas during catalyst heating can be reduced if as much exhaust gas as possible is routed via the wastegate directly, bypassing the turbine. The exhaust gas should also flow through the wastegate ports into the catalytic

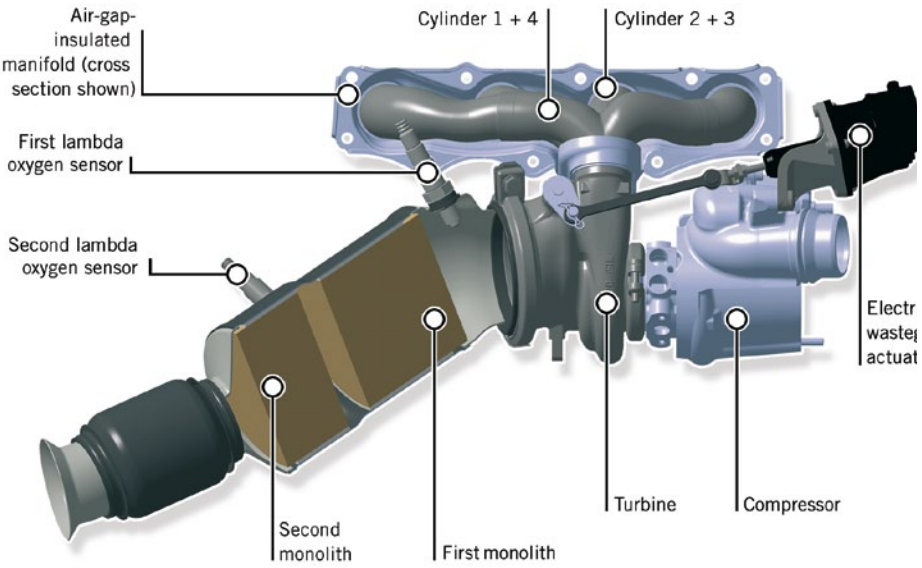


❼ Optimised flow to the catalytic converter with comparison of exhaust gas temperature

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converter without any direct contact with the walls. The flow to and through the wastegate ports was maximised initially by means of consistent CFD optimisation measures. The maximum wastegate opening angle was increased from 25° to 45°, using an electric wastegate actuator that offers as well other functional advantages. By implementing these measures, the proportion of exhaust gas routed through the wastegate considerably was increased from 40 to 70 %. Central routing of the exhaust gas from the wastegate ports to the front surface of the catalytic converter with minimum heat losses can be achieved by means of appropriate orientation of the wastegate inside the turbine housing and positioning the catalytic converter close to the turbine.

According to the CFD simulation, this package of measures offers a means of increasing the temperature of the exhaust gas upstream of the catalytic by more than 200 K. The results of the optimisation process have been confirmed within the framework of vehicle tests. The temperature of the exhaust gas following a cold start in the optimised variant is up to 220 K higher than in the basic version.



❽ Exhaust-gas turbocharger module

STRUCTURAL DESIGN OF THE EXHAUST-GAS TURBOCHARGER MODULE

Designed as an integral construction, the exhaust-gas turbocharger module for the new SULEV engine is shown in ❽. Apart from the electric wastegate actuator, it is a carry-over of the ULEV module. The four-in-two exhaust manifold comprises sheet-metal shells, whereby the hydroformed inner tubes are made of austenitic stainless steel. The exhaust gas from cylinders 1 and 4 are brought together, as are the gas from cylinders 2 and 3, in order to separate the flow for an optimal charge cycle. The gas enters the twin-scroll turbine via two circular ports for optimum flow behaviour. The compact turbine housing is made of austenitic cast steel and the scrolls are positioned at an angle of 30° relative to one another. Fitted in close proximity to the engine, the close-coupled catalytic converter has been designed as a two-stage construction (first monolith: 900 cpsi, second monolith: 600 cpsi). The primary oxygen sensor has been positioned in such a way that the engine can be operated with closed-loop lambda control within just a few seconds. The secondary oxygen sensor has been positioned between the two monoliths. The electric diverter valve has been integrated into the compressor. The exhaust-gas turbocharger module is assembled with the highest level of automation currently used in high-volume production processes. The exhaust-gas turbocharger module described above is fitted in all of the BMW Group's vehicle series that are equipped with the new engine.

ACHIEVEMENT OF SULEV TARGETS IN THE VEHICLE

The efficacy of the individual measures and their respective contributions towards target achievement are evident in the HC emissions shown in ❾ by way of an example. The ULEV variant of this engine [3], which has already been launched into the market, constitutes the starting point for this study. The aim is to undercut the SULEV limits with an adequate safety margin to ensure compliance with long-term durability requirements. The values are based on measurements carried out on an as-new catalytic converter.

It has been possible to reduce emissions by 40 % by optimising the exhaust-gas turbocharger and shortening the light-off time of the catalytic converter accordingly. The cell density and precious metal loading have been increased in the close-coupled catalytic converter fitted in close proximity to the engine and an additional catalyst has been fitted under the floor, reducing emissions by 60 %. Emissions have been reduced by another 50 % thanks to the controlled valve operation and adaptations to the application during start-up and in catalyst heating mode, for example. The development target amounting to 50 % of the SULEV limit was achieved by combining the measures described above with the catalytic converter in as-new condition and without a secondary air system.

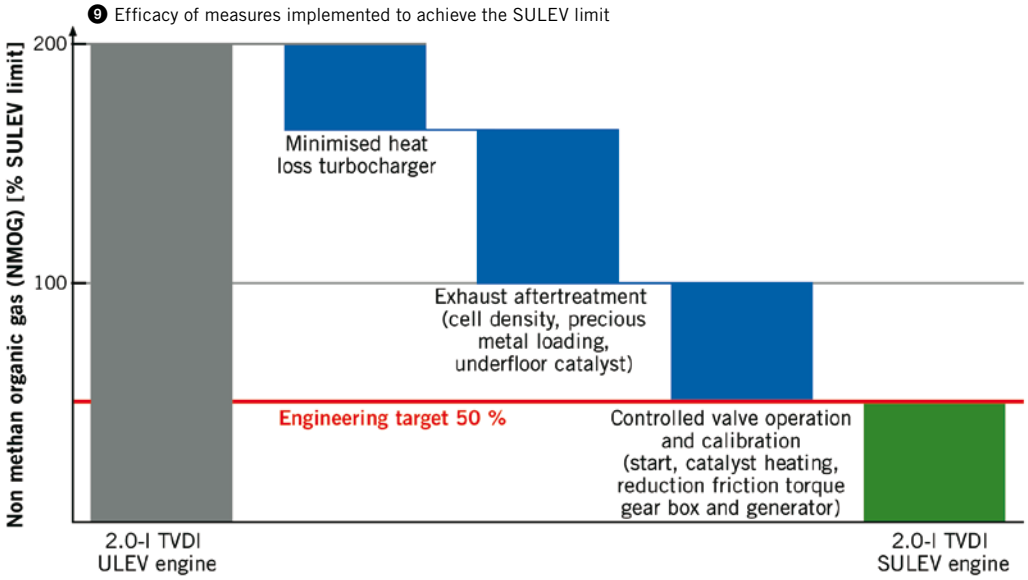
A so-called Prem-Air coolant radiator with a special catalytic coating on the surfaces that come into contact with the cooling air has also been used. This coating converts harmful ozone molecules in the air passing through the radiator into oxygen molecules. US legislation acknowledges the use of ozone-reducing Prem-Air radiators by raising the limit for exhaust gas components that produce ozone.

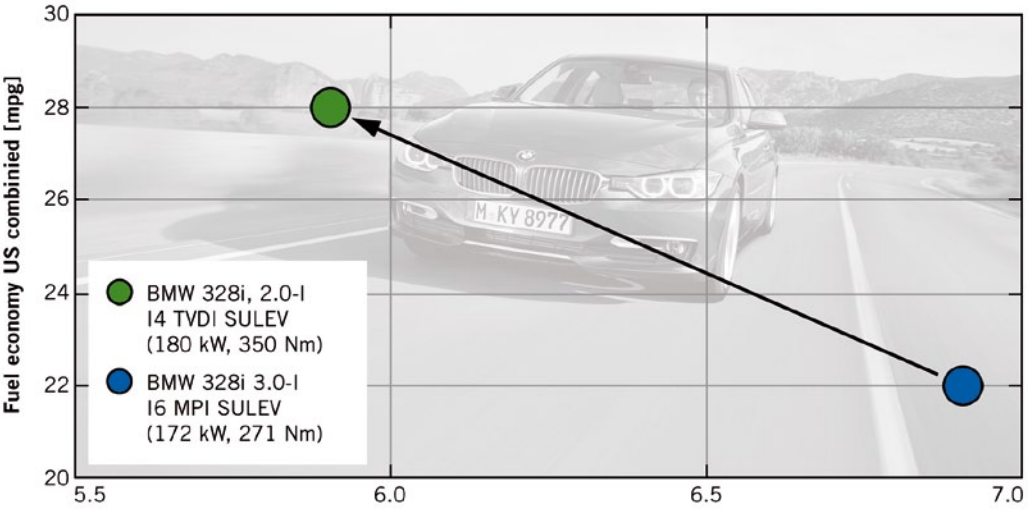
DRIVING PERFORMANCE AND FUEL CONSUMPTION

The SULEV version of BMW's new 2.0-l four-cylinder gasoline engine also offers the dynamic driving performance that is typical of BMW in conjunction with low fuel consumption. The engine delivers a maximum torque of 350 Nm and a maximum power of 180 kW. These figures

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❾ Efficacy of measures implemented to achieve the SULEV limit





❿ Driving performance and fuel consumption

represent a 30 % and 4 % improvement over the respective values of the predecessor engine. At the same time, the useable engine torque range is greatly enhanced. Compared with its predecessor, the new 328i PZEV boasts significant improvements in driving performance, as well as a substantial reduction in fuel consumption, ❿, proving that it is possible to comply with the world's most stringent exhaust emissions legislation without making any sacrifices in terms of dynamics or efficiency.

SUMMARY

BMW's new SULEV 2.0-l four-cylinder gasoline engine demonstrates the potential offered by Twin Power Turbo technology for compliance with the world's most stringent exhaust emissions legislation while simultaneously achieving the EfficientDynamics targets of lower consumption and greater driving pleasure. The degree of freedom offered by the TVDI combustion concept can be used to enable catalyst heating mode with a very high exhaust-gas enthalpy flow rate and ultra low emissions, as well as excellent running smoothness through the entire life cycle of the vehicle. A consistent reduction of exhaust gas heat losses ensures that the catalytic converter lights off as quickly and cuts the heat sink effect of the exhaust gas turbocharger down to a minimum. The new BMW 2.0-l four-cylinder gasoline engine is the first turbocharged SULEV engine worldwide that requires no secondary air system and it is currently being launched into the US American market in the new BMW 328i PZEV.

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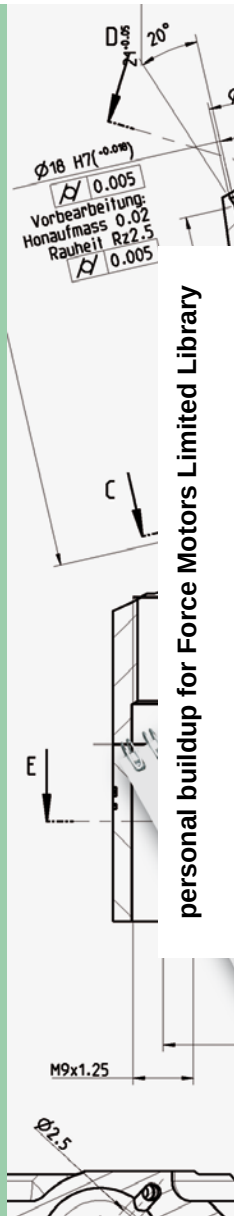
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VARIABLE COMPRESSION RATIO FOR GASOLINE ENGINES

Downsizing in combination with turbocharging enables a sustainable CO_2 emission reduction. In order to mitigate knock at higher engine loads the compression ratio has to be diminished with increasing boost pressure levels. To alleviate disadvantages of reduced efficiency at part load FEV developed a two-stage variable compression ratio mechanism.



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MOTIVATION

In order to meet the CO_2 fleet threshold value of 130 g/km in the vehicle inertia class of 1372 kg introduced by the European Commission, manufacturers already started offering boosted engines with reduced displacement. Lower throttle and friction losses of these smaller turbocharged engines enable a CO_2 reduction between 10 and 20 % – depending on vehicle mass and downsizing level.

The application of downsizing in context with boosting presents a disadvantage because of the higher knock sensitivity at higher engine loads, thus the compression ratio (CR) has to be reduced. Despite the fact that the majority of the turbocharged engines are equipped with direct fuel injection, today's boosted engines are designed for RON 95 gasoline with a compression ratio lowered by 1 to 1.5 units in comparison to naturally aspirated engines.

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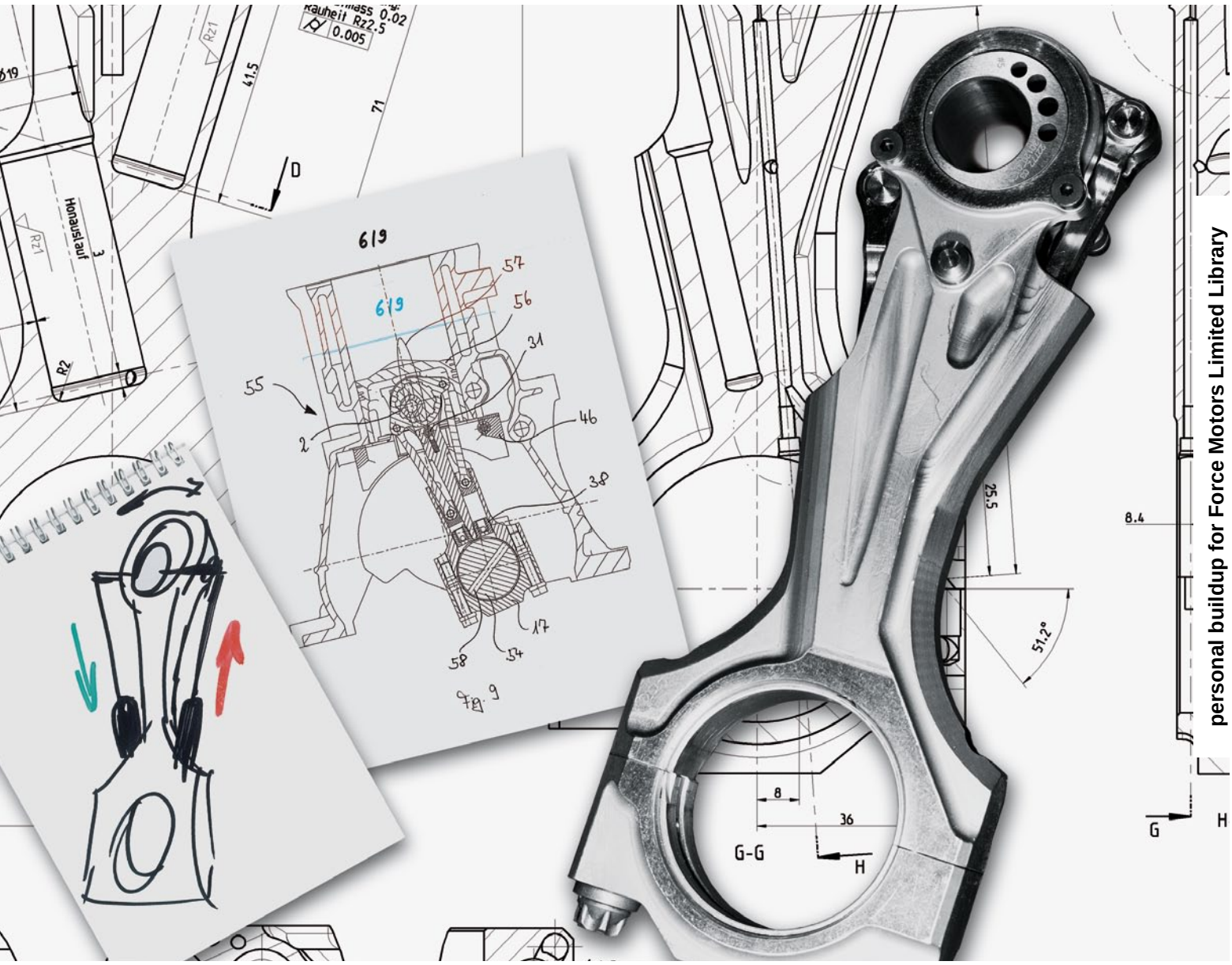
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Decreased pressures and temperatures at the end of compression shift the knock border line to higher loads. Hence, the more beneficial location of the center of combustion offers improved efficiency at full load. However, this full load benefit is at the cost of reduced thermal efficiency at low engine loads.

Besides downsizing in combination with boosting, several other means to reduce fuel consumption of gasoline engines can be considered. The modular application of a variable compression ratio mechanism lends itself to resolve this conflict.

SYSTEM PROPERTIES OF THE TWO-STAGE VCR SYSTEM

The VCR systems with variable kinetic connecting rod lengths entail variable powertrain components which can be used instead of the conventional components and thus only require minor modifications to existing engine architectures.

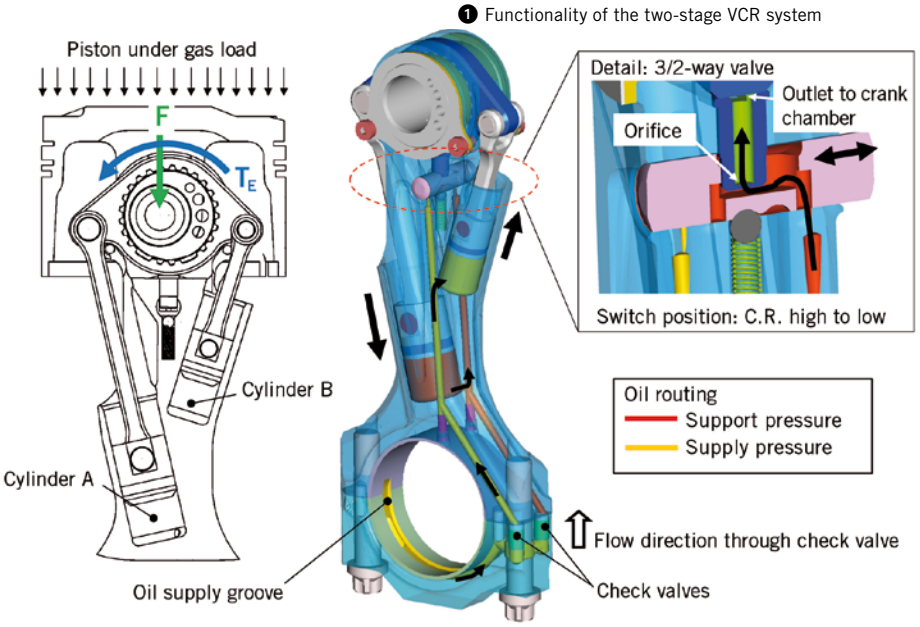
The presented connecting rod belongs to the group of VCR systems with variable kinematic lengths and has been developed over the past few years. ❶ (left) shows a cross section of the actual connecting rod. The small end is equipped with an eccentric sleeve which houses the wrist pin. By rotating the eccentric sleeve the effective connecting rod length and thus compression ratio can be varied. Further details about the working principle can be found in [1, 2].

To adapt the system to existing engine architecture requires only relatively small changes to existing parts as compared to the requirements of other VCR systems at acceptable additional manufacturing costs. The influence on the lubrication circuit of the engine is so small that the oil pump capacity does not need to be increased.

The reduction of moving masses is part of current development activities. For engines with a cylinder displacement of approximately 0.4 l the oscillating mass increases by 30 to 50 % depending on stroke-to-bore ratio. Simulation results have shown that optimising the design and utilising high-strength steel significantly reduces the mass of the conrod.

RESPONSE TIME

In order to clarify the feasible two-stage VCR response time will satisfy the thermodynamic requirements a CR switching as part of a load step was simulated. In this particular case the full load acceleration in 6th gear at 2000 rpm was investigated. The boost pressure trace is based on measurements taken from a two-stage turbo-



❶ Functionality of the two-stage VCR system

charged DI gasoline engine. ❷ shows the trace of the boost pressure and the derived trace of an efficiency optimised compression (ideal C.R.) as a function of time.

In the case shown, the switching point from high to low compression ratio is located exactly at the beginning of the load step. The switch-over is completed after 0.6 s as illustrated in the simulated actual compression trace (actual C.R). The CR switch-over takes place much faster than boost can be generated by the highly dynamic and optimised boosting system. Therefore, further reduction in response time would not result in further improvement. In fact, faster switch-over efficiency may lead to unintended torque reductions. The influence of the oil temperature on the response time is relative small. At an oil temperature of 0 °C the response time going from high to low CR increases only by a small amount compared to the response time at warm engine.

CO_2 POTENTIAL OF A TWO-STAGE VCR SYSTEM

Investigations so far have shown that the fuel consumption improvement potential for a continuous compression ratio adjustment over the European NEDC Cycle is between 6 and 8 % [3]. With a two-step VCR, due to knock limitation occurring now at part load, the CR needs to be switched back to the lower value as designed for full load operation earlier in the upper area of the load map. Thus, the fuel efficiency is slightly reduced.

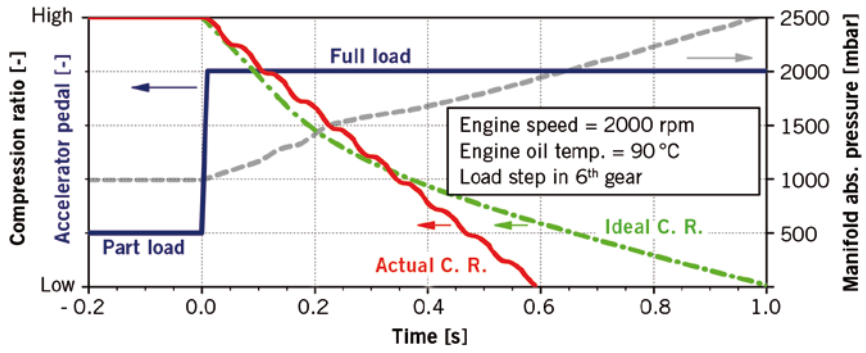
In order to assess the fuel consumption reduction potential of the two-step system, a cycle simulation using the currently valid NEDC Cycle and the WLTP Cycle which is under discussion was performed. The WLTP is marked by a longer test cycle time and higher dynamics with more pronounced accelerations. A future mid-size vehicle equipped with a 2.0-l four-cylinder engine was targeted and defined as follows:

: inertia class of 1372 kg
: turbocharged DI-Engine
: 180 kW/350 Nm (RON 95)
: manual transmission, start/stop system.

For this targeted vehicle powertrain configuration, ❸ shows a possible CR map for a continuously working VCR system (left) and for the two-step VCR system using a minimum CR of 8 and a maximum CR of 12 (right). The transition from 12 to 8 was defined for minimum fuel consumption during part load and offers operation with beneficial high compression ratio over the entire NEDC. It can be seen that using current downsizing concepts with longer gear ratios the engine speed can be kept below 2000 rpm across almost the entire test cycle. The de-throttling effect due to displacement reduction is complemented with lowered parasitic friction losses by the downspeeding. The NEDC fuel consumption simulation shows an advantage for the two-stage VCR system in comparison to the base engine with a 9.6 CR of about 6 %, ❹).

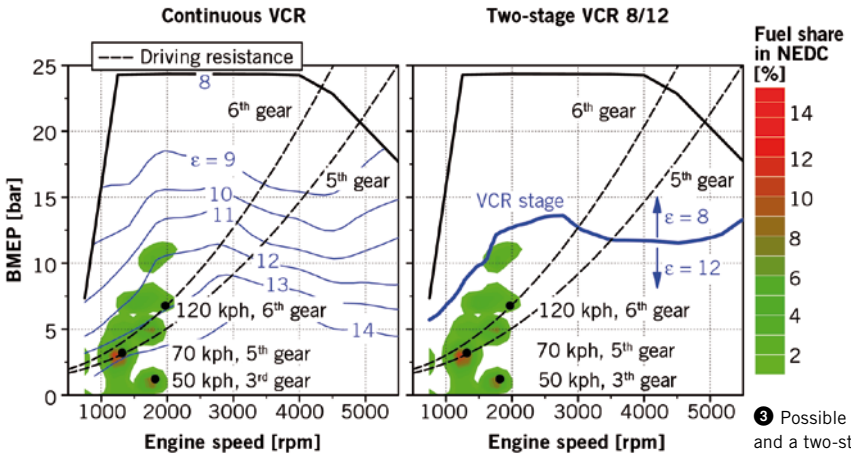
The adjustment of the CR range results in changes regarding the transition function in the engine map. With an increased maximum CR the knock limit is reached at lower part load. With an increase in minimum CR an efficiency benefit can be achieved in the areas not relevant to knocking. However, at full load the increased minimum CR leads to a reduced achievable BMEP level due to knock limitation.

The potential fuel consumption improvement in the WLTP requiring higher engine loads is still at 5 %. For



❷ CR actuation and boost pressure build-up after positive load step

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❸ Possible CR map with a continuous and a two-stage VCR system

both cycles the combination of 8/13 has the greatest fuel consumption benefit. This is caused by the higher loads during the WLTP; a switch-over to the lower CR is required. Therefore the selection of the maximum CR regarding the fuel consumption result has a greater impact on the WLTP than on the NEDC.

Overall, it can be stated that for the two-step VCR system there is a low sensitivity regarding the selection of the compression ratio range as well as the influence of the driving profile.

POTENTIAL WITH OTHER FUTURE CO_2 TECHNOLOGIES

Further investigations were conducted to assess the CO_2 emissions benefit of the two-stage VCR system in combination with other future engine technologies:

: extreme downsizing concept with further reduced displacement of 1.5 l while full load torque is kept constant

: Gasoline Controlled Auto Ignition (GCAI)

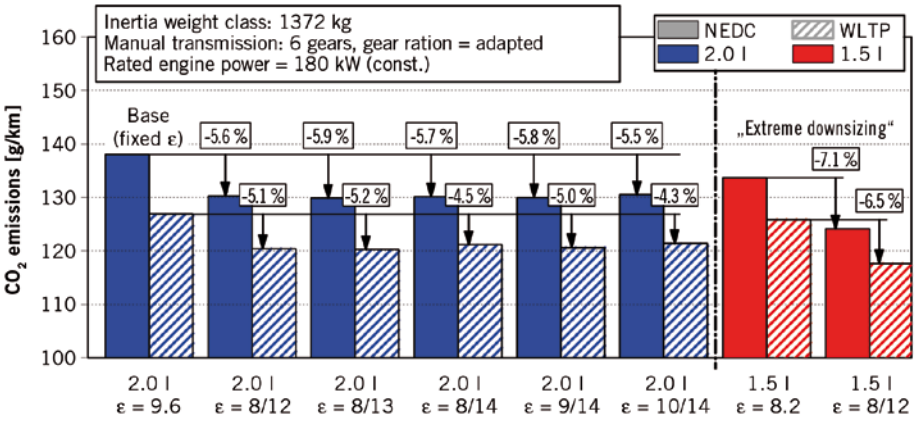
: alternative fuels with high knock resistance.

EXTREME DOWNSIZING

The example of further downsizing (1.5 l) results in 120 kW/l specific power to offer identical drive power. While the fuel consumption benefit of the extreme downsizing without VCR over the NEDC still reaches about 3 to 4 %, it has to be stated that with the two-step VCR fuel efficiency can be increased between 6 and 7 % for both investigated driving profiles. In comparison to the base variant with 2.0 l displacement the further potential of a two-step VCR will be higher for the extremely downsized engine with operation at higher engine load. Minimum as well as maximum CR has to be analysed and a reduction of the upper compression ratio value by one unit from 13 to 12 represents the best trade-off while a CR of 8 is still the optimum for the lower stage.

COMBINATION OF GCAI AND TWO-STAGE VCR SYSTEM

The GCAI combustion system is based on the recirculation of large, hot residual gas fraction in the combustion chamber. During lean engine operation a de-throttling effect can be achieved while significantly minimising the NO_x formation due to the relatively cold combustion. The GCAI operation is limited at low engine loads caused by misfiring due to the unstable self-ignition. At higher loads, steep cylinder pressure gradients and high NO_x emissions narrow the GCAI operation area. This leads to a reduced fuel consumption advantage. The main reason for that finding is the short time during which the GCAI mode can be successfully applied, ❺. If the GCAI combustion system is combined with a two-step VCR system, then the fuel consumption benefit over the NEDC increases from 4 to 12 %. This increase is caused by the much longer time period of the GCAI operation in combination with the higher



❹ CO_2 potential of the two-stage VCR system in NEDC and WLTP

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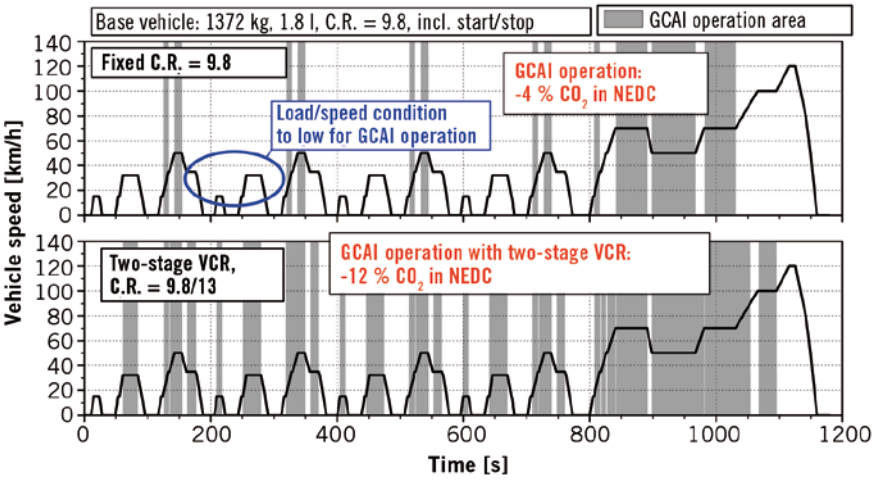
compression ratio as well as the thermodynamic advantage of the higher CR at lower loads.

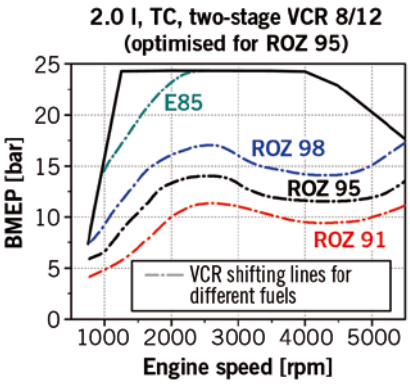
ALTERNATIVE FUELS WITH HIGH KNOCK RESISTANCE

Achieving a drastic reduction in CO_2 emissions make alternative fuels a suitable option. Besides CNG, Ethanol represents a meaningful alternative to conventional gasoline fuel combining the CO_2 reduction potential gained from a partially closed carbon cycle with high knock resistance. This provides the opportunity to select a higher compression ratio and significantly enhance the efficiency over the entire engine map particularly for boosted applications [4].

However, this is only possible for mono-fuel applications. Since Ethanol is not consistently available over the entire existing infrastructure, the compression ratio for dual-fuel applications is designed for the lower gasoline octane number. As a result, the fuel consumption improvement potential is not being fully taken advantage of while using these knock resistant fuels. The utilisation of the two-stage VCR system would help to further optimise the CO_2 emissions for these bi-fuel applications by simply recognising the Ethanol content and adjusting the optimum compression ratio, ❻, accordingly. In addition, a similar ECU functionality is possible for markets offering a wider variety of octane ratings (for instance Japan and China) or regarding variation of boundary conditions (e.g. high intake air temperature).

❺ Combination of GCAI and two-stage VCR system





❻ Possible variation of the two-stage VCR system transition depending on fuel quality

SUMMARY

In order to comply with the strict CO_2 emission worldwide, manufacturers will have to introduce technologies other than downsizing concepts involving boosting and direct injection. Further increase of compression ratio to enhance the thermodynamic efficiency of the engine at part load is limited due to knock sensitivity. One possibility to resolve this conflict is the introduction of a variable compression ratio (VCR) mechanism.

The presented two-stage VCR system can be integrated into existing engine families with competitive additional costs. The transition duration is sufficiently short especially for boosted engines and almost independent from boundary conditions.

The VCR system can be used to allow for fuels with different octane ratings by shifting the compression ratio transition strategy accordingly. The presented system expands the CO_2 reduction potential of other future technologies such as multi-stage boosting and controlled auto-ignition and thus offers an important contribution to reduce fuel consumption on gasoline powertrains.

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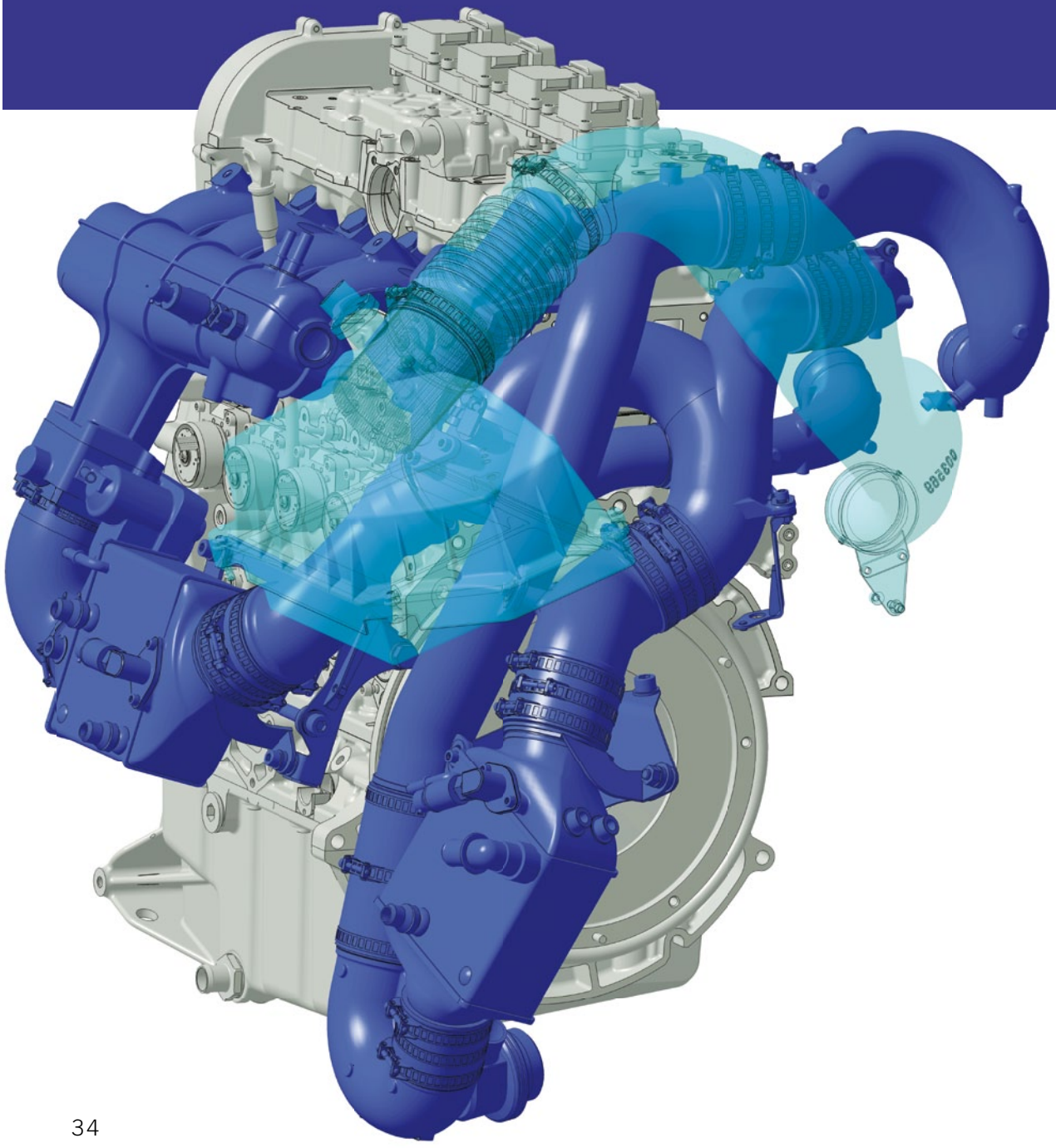
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POTENTIALS OF ENGINES WITH VARIABLE COMPRESSION RATIO

Especially in combination with high-pressure boosting, variable compression ratio and gasoline direct injection are intended for reducing fuel consumption. The two strategies both aim at significantly increased SI engines' torque and power density enabling optimum downsizing and downspeeding. Additionally cooled EGR can provide synergies to VCR in the more and more important unthrottled operating area used by downsizing/downspeeding engines. AVL has analysed the potentials.



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MARKET SITUATION

Future CO_2 legislation is one of the key challenges for future passenger car powertrains [1]. Europe and Japan lead the reduction of CO_2 emissions; China has announced further stringent CO_2 reductions, US follows with a five to ten year offset. Many OEM's meet this challenge with downsizing/downspeeding their engines [2]. Audi, BMW, VW and Daimler already have a considerable amount of charged engines in their portfolios. And this share is significantly increasing in the future – up to a share of 100 % of charged engines. Other OEMs' like Ford, Fiat, PSA and GM follow that trend.

MEGATREND AND RISKS

The actual 1st generation of downsized engines reflects a displacement reduction of approximately 25 % of the replaced NA engines and, together with more widespread gearboxes, fuel consumption (FC) in NEDC can be improved by approximately 15 % (without consideration of "blue technology" like start-stop, intelligent generator, specific vehicle measures). By further enhanced levels of downsizing, strong downsizing, additional significant benefits in NEDC FC can be achieved, since main operation areas of the driving cycle become situated within the sweetspot of the BSFC map, ❶ (left).

However, strong downsizing requires high levels of specific power and BMEP and thus boosting, leading to high BSFC with state-of-the-art technology. Thus, fuel consumption becomes more depending on driving style, vehicle category, transmission type and fuel RON and may even turn to negative results, ① (right), [3].

Besides this matter, with current technology high levels of boosting and BMEP increase the risk of irregular combustion, i.e. unwanted self-ignition prior to spark timing, especially at low engine speeds [4].

VARIABLE COMPRESSION RATIO

Therefore new approaches are required to apply strong downsizing without these potential risks in terms of FC. Addressing one of the main thermodynamic boundaries of gasoline engines, variable compression ratio (VCR) is a possibility to expand the BSFC sweetspot towards higher torque and power and to minimise potential risks of irregular combustion. For the work discussed here an engine equipped with a variable compression ratio system MCE-5 was chosen [5].

With its 6:1 minimum compression ratio and its two-stage turbo system, the MCE-5 VCRi engine allows to reach 40 bar BMEP at 1800 rpm with a peak cylinder pressure around 110 bar and a BSFC of 308 g/kWh. The mechanism has been designed with a nominal peak pressure of 130 bar and maximum engine speed of 6500 rpm.

BOOSTING AND EGR SYSTEMS

The engine is equipped with a two-stage TC boosting system with intercoolers for both stages and a new high-pressure external EGR system with feed to the individual cylinders [6]. ❷ (top) shows the

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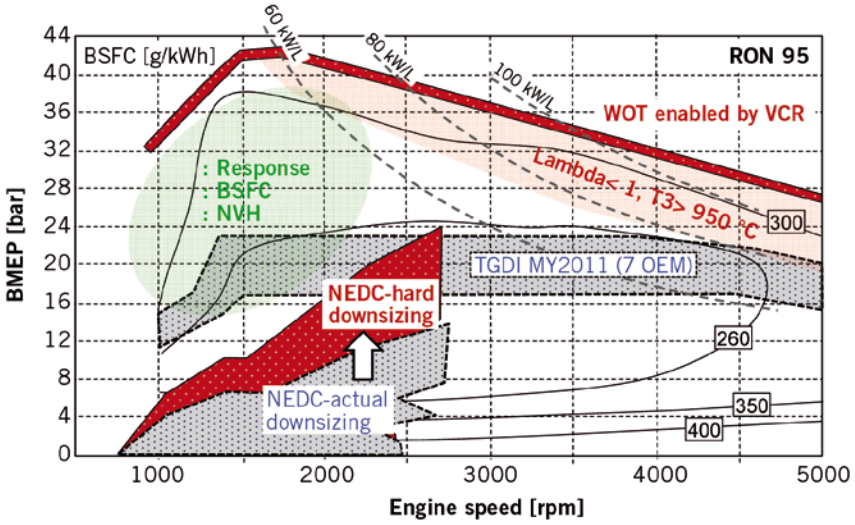


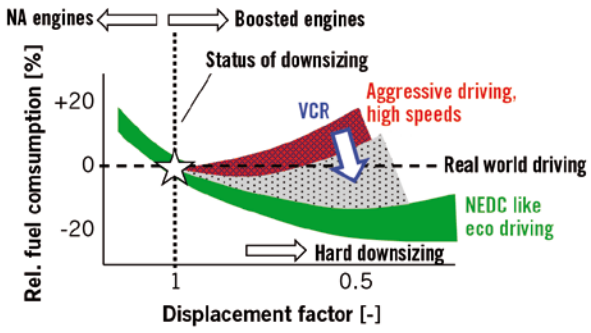
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❶ Status and projection of downsizing with VCR; NEDC area versus BSFC sweetspot (left); WOT-curves: increasing risk of real-life FC being affected by operation conditions with state-of-the-art technology (right)

pressurised inlet system comprising of a big LP- and a small HP-compressor with an electrically actuated HP compressor bypass flap. Since performance borders with BMEP >40 bar should be exploited, any risks of overheating the HP compressor should be covered and therefore an intermediate intercooler was installed. However, the increased pressurised volume compromises the transient response. Both intercoolers are operated in series once HP compressor is bypassed. The IC's are liquid cooled and of equal size.

The exhaust system comprises a compact low-volume cast exhaust manifold with integrated HP turbine housing and the electrically actuated pressure control valve. The separate LP turbine housing with integrated wastegate is flanged to the exhaust manifold. The system was designed for 1050 °C with 1030 °C used as maximum value for development. However, despite Lambda=1, exhaust temperatures >1000 °C were only needed in small map areas at speeds >3500 rpm and specific power >90 to 100 kW/l (RON95 fuel) due to VCR.

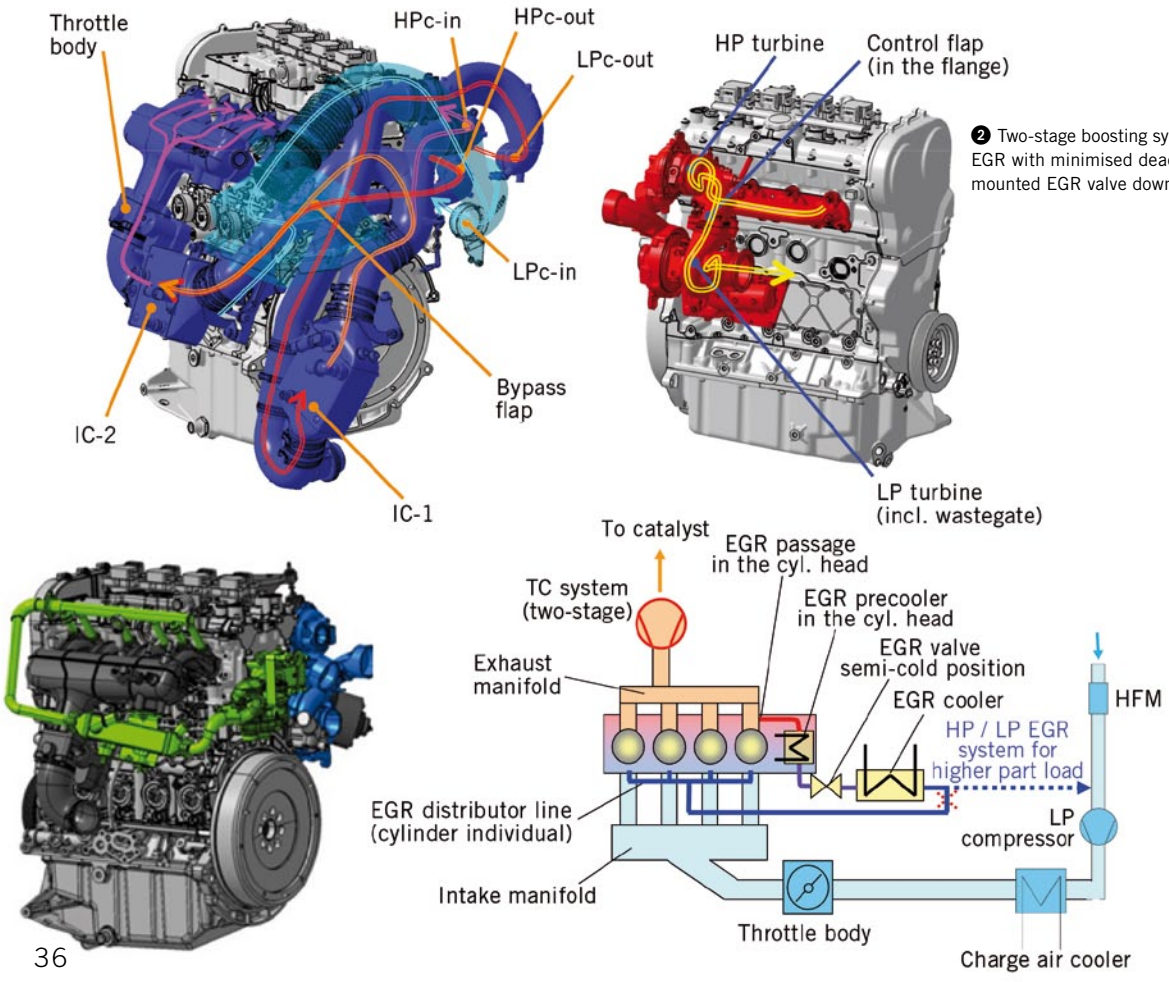
Although VCR provides advantages for knocking and thus BSFC at high BMEP, also the additional effect of cooled external EGR should be assessed. Furthermore, the effect of EGR on BSFC in the unthrottled sweetspot area of the map was of interest. Thus a high-pressure EGR system with cylinder individual feed and with the option to apply a feed upstream the LP-compressor, i.e. HP-LP system, was selected. Specific focus was put on minimum dead volume attached to the exhaust manifold in view of transient response so that the EGR valve was mounted in a semi-hot position at the exit of a cylinder-head integrated EGR pre-cooler, ② (bottom).

COMBUSTION SYSTEM

In order to show the potential of future GDI TCI engines, the variable compression ratio engine (1.5 l displacement, bore 75 mm, stroke 84 mm) was equipped with the latest generation of AVL's combustion system, ❸ (top):

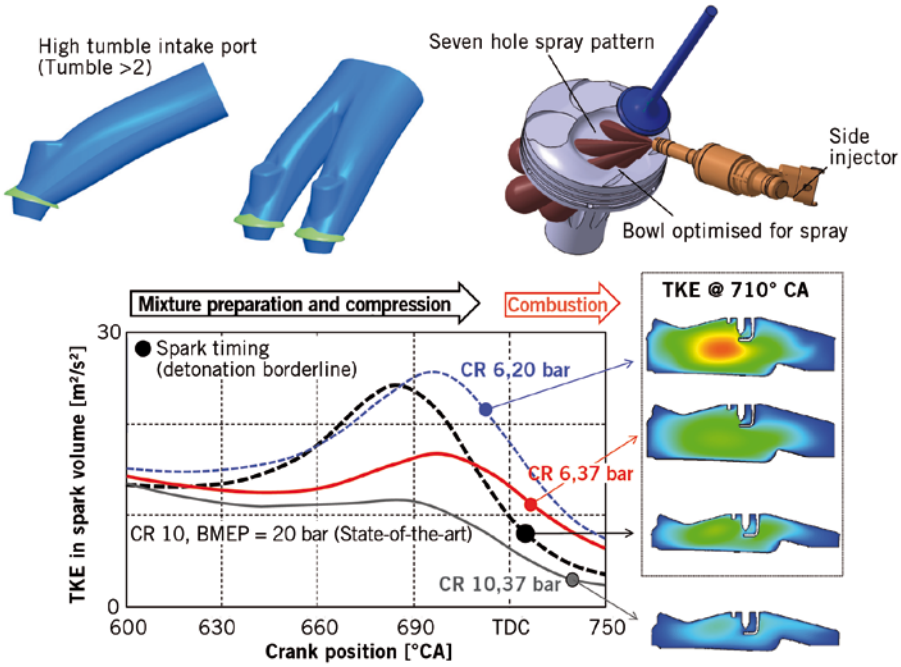
: high tumble intake port for fast combustion and excellent EGR tolerance
: dual cam phasing with wide range cam phasers
: multihole fuel injectors with optimised spray layout and reduced penetration
: high compression ratio piston and combustion chamber with optimised bowl
: smooth combustion chamber shape for reduced knock tendency.

The tumble level of the variable compression ratio engine was chosen rather high. Simulations showed that with increased compression ratio the turbulence level in the combustion chamber at spark timing is lower than with lower compression ratio, ③ (bottom). Additionally BMEP levels higher than 30 bar raised the demand for a highly efficient combustion. On the other hand lower compression ratio allows for earlier spark timings at high loads due to reduced knock tendency. This earlier spark timing enables a more efficient use of the turbulence. The tumble dissipates into turbulence typically around 30° CA before TDC. Thus with late spark timing the efficiency of turbulence to enhance



❷ Two-stage boosting system and external cooled EGR with minimised dead volume and hot mounted EGR valve downstream a pre-cooler

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❸ Combustion system of the 1.5-l GDI TCI test engine with variable compression ratio; effect of compression ratio on turbulent kinetic energy (TKE) during compression and ignition

combustion is lower than with advanced spark timing.

RESULTS

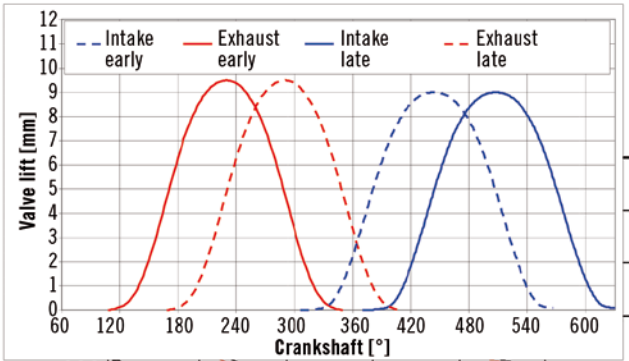
Any engine that can be operated at rather high compression ratio (up to 15:1 in this case) runs into compromises regarding combustion chamber shape and dead volumes. In the optimisation process a design with limited valve pocket depth was chosen limiting the maximum valve overlap.

In order not to limit fuel consumption reduction potential, a new approach for the operating strategy was required instead of the well-established "late/late" strategy with retarded positions of exhaust and intake valve events.

The new approach was a strategy known from HCCI engines – operation with negative overlap. ❹ shows pV diagrams and results achieved with this strategy compared to the standard late – late strategy for the operating point 2000 rpm and 2 bar BMEP. Improved fuel consumption could be combined with an optimised design of the combustion chamber without crevice volumes by utilising this operation strategy. Additionally this strategy allows a very smooth application of cam positions throughout the whole engine map.

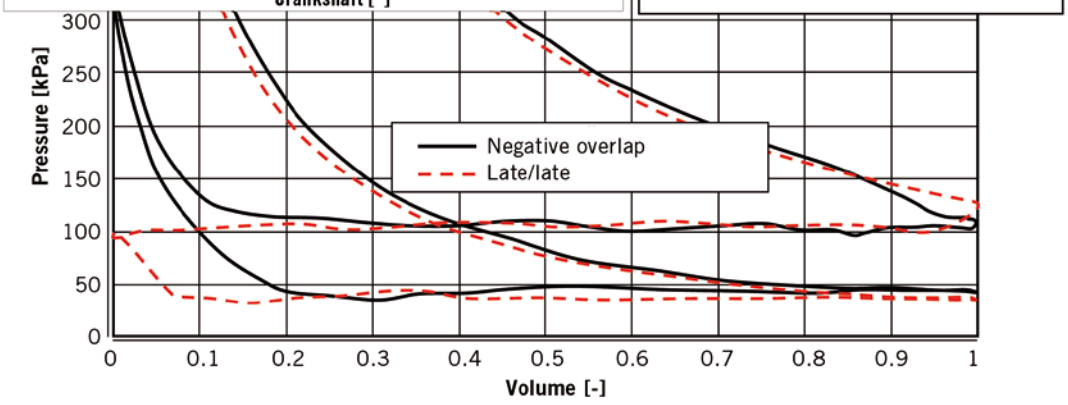
The engine was optimised with regard to the parameters compression ratio, wastegate operation, cam timing and EGR. ❺ (left) shows exemplarily such a stepwise optimisation for the operation point 2000 rpm and 8 bar BMEP.

As the test engine was not yet optimised for lowest friction – owing principally to non-optimised ancillary equipment – the BSFC at standard compression ratio, closed wastegate and closed EGR valve is not extremely low. By optimisation of compression ratio, wastegate opening and EGR, very attractive fuel consumption – despite the slightly

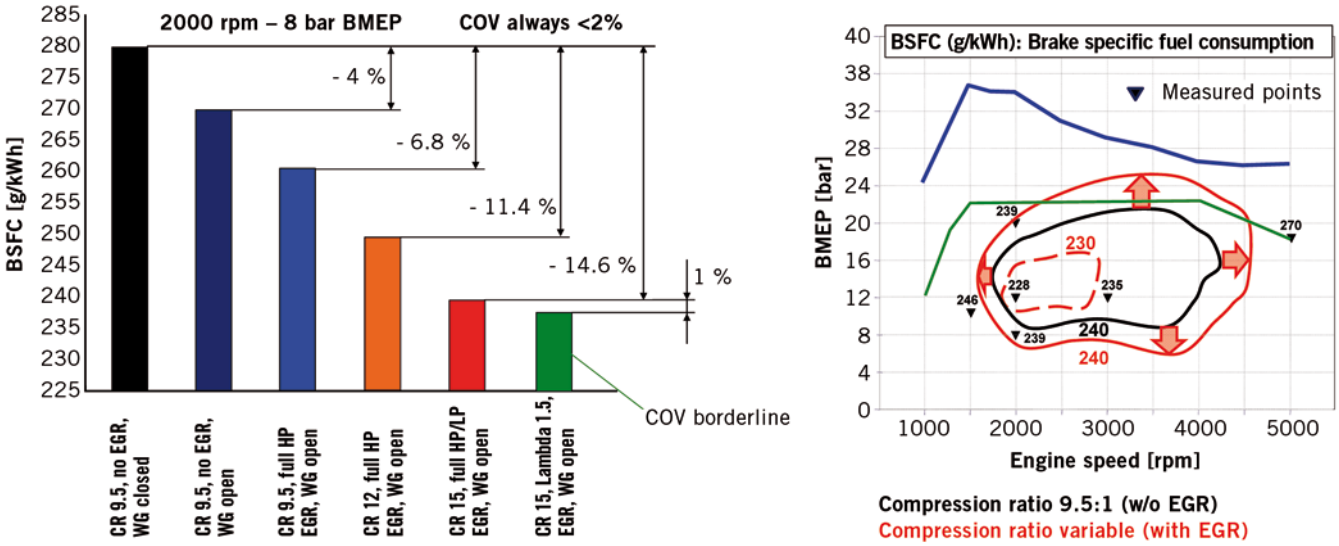


❹ Part load operating strategy: late/late versus negative overlap; 2000 rpm and 2 bar BMEP

	Neg. overl.	Late/late
IMEP (bar)	2.61	2.69
COV (%)	1.2	1.0
IMEP low (bar)	-0.56	-0.65
IMEP high (bar)	3.18	3.33
BSFC (%)	97	100



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❺ Part load fuel consumption optimisation: effect of compression ratio and EGR

increased friction – can be achieved (239 g/kWh). The optimised point shows a very high compression ratio enabled by the high amount of cooled EGR suppressing knock in combination with a "negative overlap" cam strategy. Due to the high turbulence level, the combustion stability always is on a very good level. Even lean operation at the stability limit does not show a significant improvement in BSFC compared to the chosen stoichiometric and EGR strategy.

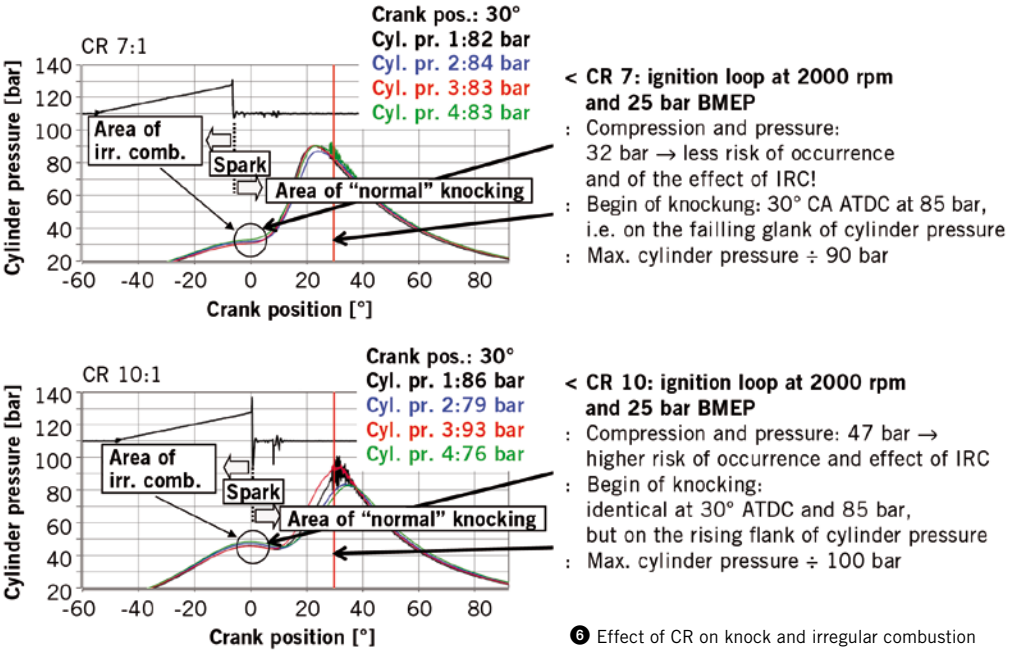
With this strategy of increasing compression ratio and adding cooled EGR, an estimation of the BSFC improvement in the whole map was done by testing individual load/speed points, ⑤ (right). The comparison was done with the baseline engine with compression ratio 9,5:1 and w/o EGR. The main emphasis was put on the unthrottled area being important for downsized/downspeeded engines.

The new strategy allowed expanding the low BFSC area (line of 240 g/kWh in ⑤ (right)) significantly. BSFC values below 230 g/kWh were recorded.

For a typical TGDI engine with 22 bar BMEP at full load (single stage charging) this would mean operating the engine at full load at BSFC values of less than 240 g/kWh.

For the VCR engine with a calibrated full load curve at maximum 35 bar BMEP [5] still the majority of the operating area is run at very low BSFC.

A further benefit of a variation in compression ratio is the possibility to suppress knock and irregular combustion. ❻ shows cylinder pressure diagrams of the four cylinder engine operated at 2000 rpm and 25 bar BMEP – once with a compression ratio of 7:1 and once with 10:1.



❻ Effect of CR on knock and irregular combustion

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With standard compression ratio (lower figure) the cylinder pressure at the end of compression is 47 bar. Due to this high pressure the probability of autoignition increases. Additionally combustion has to be retarded due to knock. As the occurrence of knock is most probable at approximately 30° CA after TDC, the knock occurs at a time when the cylinder pressure still shows a rising flank. This indicates that a high amount of mixture is still unburnt. Any knock event thus shows a rather high peak pressure.

With a lowered compression ratio the cylinder pressure at end of compression is 32 bar. Due to this lowered pressure the probability of autoignition decreases. Combustion can be advanced due to lower knock tendency. The occurrence of knock is equally located at approximately 30° CA after TDC, but the knock occurs at a time when the cylinder pressure shows a falling flank. The pressure increase due to knock is low. As future engines have to meet very stringent particle number legislation, a special focus was set on this emission component.

Vehicle emission results achieved during catalyst heating condition (1200 rpm, 1 bar BMEP, conditioned coolant, double and triple injection) show, compared to latest development Euro 6 engines with fixed compression ratio, very low PN values. At low compression ratios (below 10:1) the free spray length increases and the more open combustion chamber supports the avoidance of liquid fuel impinging on cold walls. The absolute particle number achieved supports the achievement of lowest emission levels.

SUMMARY

A variable compression ratio device (VCR) with gasoline direct injection can be used as enabler to increase compression ratio in important operating areas without limiting high load operation. A low compression ratio at full load allows operation without enrichment. Additional cooled EGR allows for a further improved tradeoff between compression ratio and knock due to the knock suppressing effects of cold EGR. Due to that a further improvement in BSFC in the sweet spot of the engine map can be achieved. BMEP levels above 40 bar (318 Nm/l) can be achieved at low speeds without severe knock or irregular combustion. Specific cam timing strategies were developed to overcome the issue of increased crevice volumes for a high compression ratio combustion chamber. A variation in compression ratio allows for an adaption of combustion chamber shape (increased free spray length) for improved particle emissions in start and catalyst heating.

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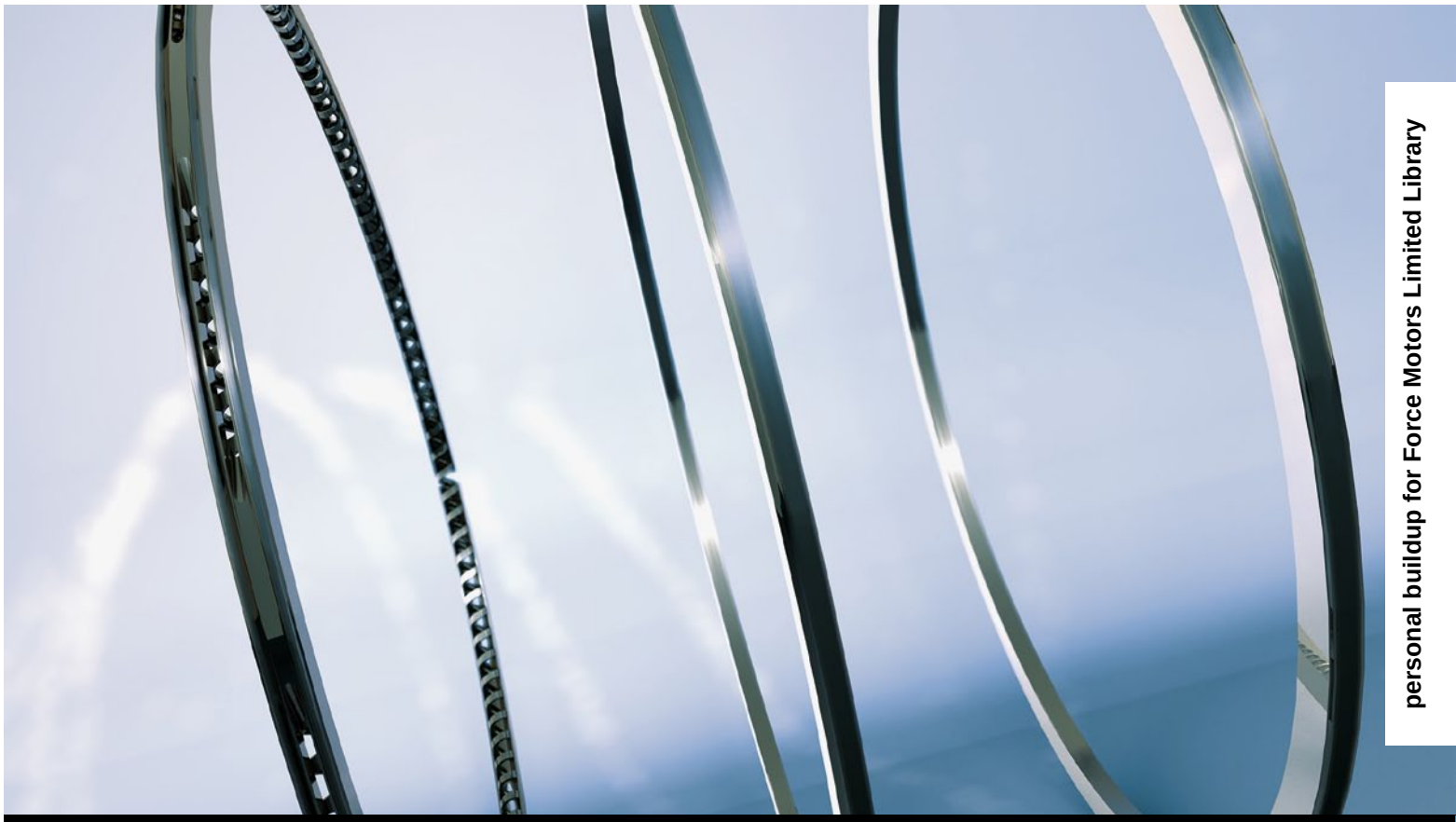
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PISTON RING COATING REDUCES GASOLINE ENGINE FRICTION

Federal-Mogul has developed a coating system for piston rings under the name of Carboglide which decreases wear in highly charged gasoline engines and further reduces friction losses which occur between the ring pack and the cylinder running surface. The tailored composition of the specific layer structure of the carbon-based Carboglide coating in combination with corresponding modified piston ring designs yields a potential to improve fuel efficiency by up to 1.5%.

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REQUIREMENTS

Manufacturers of internal combustion engines continually face the challenge to reduce fuel consumption and emissions by improving engine efficiency. In particular this necessitates a substantial reduction of mechanical frictional losses. The components of the power cylinder and crankshaft drive are at the center of these development efforts. The piston ring pack has a significant potential for bringing down friction losses due to its fairly high (24 %) share of mechanical frictional losses, ❶. At the same time measures such as direct injection and turbocharging among others, which increase engine performance, intensify the requirements for the functional behaviour of piston rings.

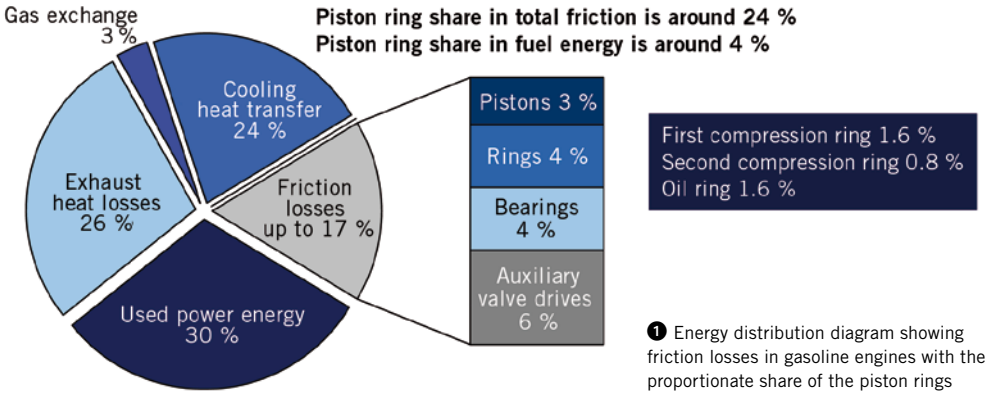
When considering measures to optimise the tribological system of piston ring and cylinder surface, piston ring coatings play an increasingly important role as they can directly influence the wear and friction behaviour and the resulting scuff resistance. By introducing Carboglide Federal-Mogul is providing a coating for piston ring applications which meets the most stringent requirements for functional behaviour and offers a considerable potential to reduce frictional losses.

CARBON-BASED PISTON RING COATINGS

Carbon-based coatings (DLC, Diamond-like Carbon) have a long track record in applications for cutting tools and components that are tailored for the most severe tribological requirements [1]. Diamond-like carbon coatings have only a low tendency for adhesive wear or tribochemical reactions. In most tribological systems these properties result in a lower coefficient of friction under boundary lubricated conditions when the coating is compared to other coating systems (such as metallic layers, hard coatings and others).

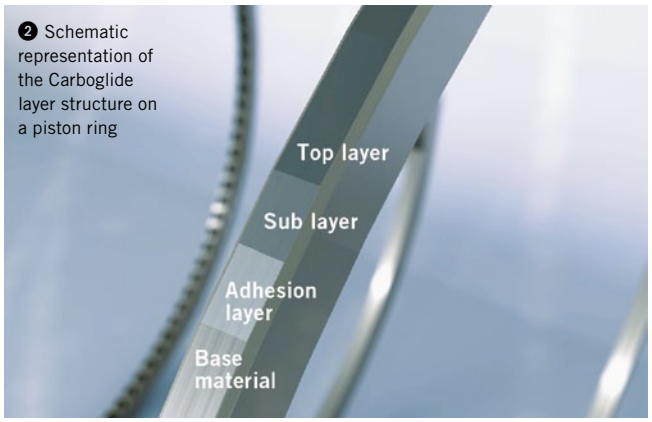
Using carbon-based coatings for piston rings necessitated a significant further development of this coating technology to meet the tougher requirements of surface fatigue resistance and coating fatigue. Based on the development of the GOE245 metal-doped, hydrogenated carbon coating the first series application for highly loaded passenger car gasoline engines succeeded in 2006. GOE245 significantly reduced the previously observed cylinder surface scoring which led to scuff-mark formation under the harsh tribological conditions in these AlSi cylinder surface engines. As the damage mechanism is most frequently seen under cold start conditions, this carbon-based coating is typically chosen and used as a running-in coating applied onto a wear resistant coating (nitrided layer, PVD layer) [2].

During the subsequent development work the level of coating wear was substantially reduced by introducing the GOE247 coating. This piston ring coating consists of several layers which comprise an amorphous hydrogenated carbon top layer which is applied onto a metal-doped and hydrogenated intermediate carbon layer. The piston ring coating is applied with a layer thickness of 3 to 5 microns and shows a wear resistance in the engine which is higher by a factor of 4 to 5 when compared to the performance of a metal-containing DLC piston ring coating.

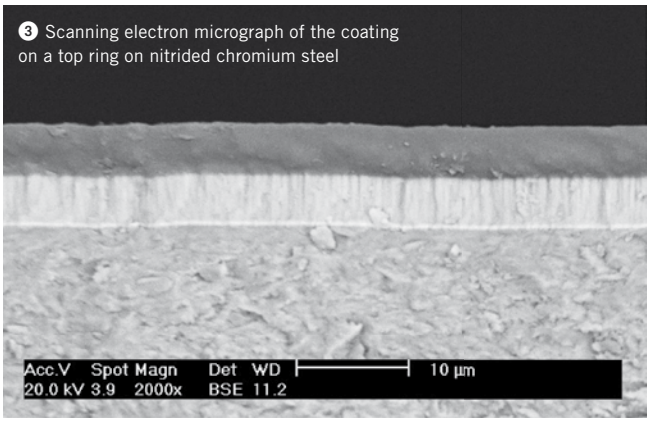


❶ Energy distribution diagram showing friction losses in gasoline engines with the proportionate share of the piston rings

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❷ Schematic representation of the Carboglide layer structure on a piston ring



❸ Scanning electron micrograph of the coating on a top ring on nitrided chromium steel

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COATING STRUCTURE AND PROPERTIES

The combination of direct injection and turbocharging with very smooth cylinder running surfaces, in gasoline engines, significantly increases the demand for piston rings with excellent scuff resistance and low cylinder running surface wear.

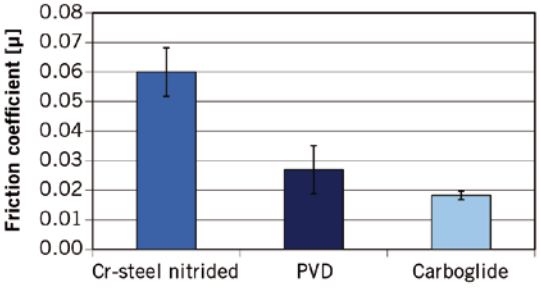
The necessity to reduce fuel consumption and thus CO_2 emissions requires an increasing contribution of the reduction of frictional losses.

In order to meet these tough requirements which translate into very low friction in combination with high wear resistance and superior scuff resistance, a new piston ring coating had to be developed. This was based on Federal-Mogul's application experience with carbon-based piston ring coatings. The new coating, which is known under the Carboglide name, employs a three-layered structure. The coating architecture is characterised by a very thin chromium adhesion layer in the transition zone to the original piston ring surface. This is followed by an intermediate layer and a top layer, ❷. The intermediate layer's structure consists of tungsten- and hydrogenated carbon in which the tungsten carbide is embedded in the form of nanocrystals found in a column-shaped growth pattern. The top layer is an amorphous diamond-like carbon layer which is characterised by a specific ratio of sp2 and sp3 bonded carbon. By adjusting the layer thickness ratio of the top layer to the intermediate layer to a value between 0.7 to 1.5 while adjusting the relation between top layer and total coating thickness to between 0.4 and 0.6 the level of residual coating stress is modified so that the total coating thickness is roughly 10 microns, ❸. Furthermore the high level of coating hardness, measuring between 1800 and 3100 HV0.02, in combination with a coating thickness, which is greater by a factor of 2 to 3 in comparison to conventional DLC coatings, leads to a significantly improved wear resistance of the coating.

Owed to its layered structure, Carboglide shows excellent friction properties. A comparison with the coefficient of friction of existing piston ring coatings for gasoline engines (nitrided, PVD coatings) clearly shows the potential frictional benefit [3], ❹. In addition to that the tribological behaviour was evaluated under boundary lubrication conditions, using a so-called overload test. The results shown in ❺ demonstrate that the scuff resistance is nearly twice that of a PVD coated piston ring running surface.

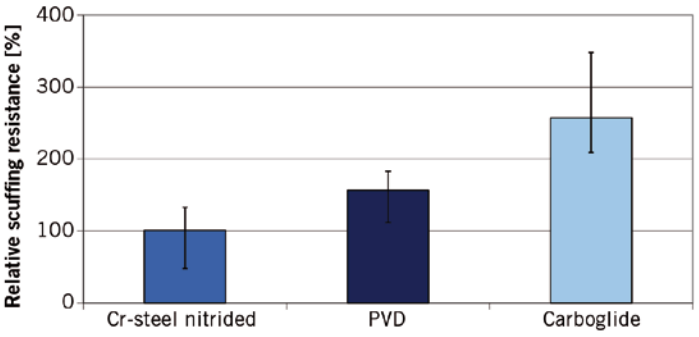
FUNCTIONAL BEHAVIOUR DURING ENGINE OPERATION

The beneficial functional behaviour of the new coating was extensively tested under real engine conditions. The first series applications in highly charged gasoline engines demonstrated the durability under "field conditions". The development of a new generation of piston ring coating that serves to significantly reduced frictional losses has been successfully confirmed by coating applications in highly loaded gasoline engines. Frictional loss tests carried out in a fired single cylinder engine with floating liner showed up to 20 % reduction in ring frictional losses using Carboglide in the first and third grooves, ❻. The excellent thermal and chemical durability was confirmed via endurance testing in a highly charged gasoline engine with cast iron cylinder running surfaces and inadequate lubrication levels, ❼. As can be seen, the nitrided steel top ring (first groove) shows strong traces of seizure as well as material

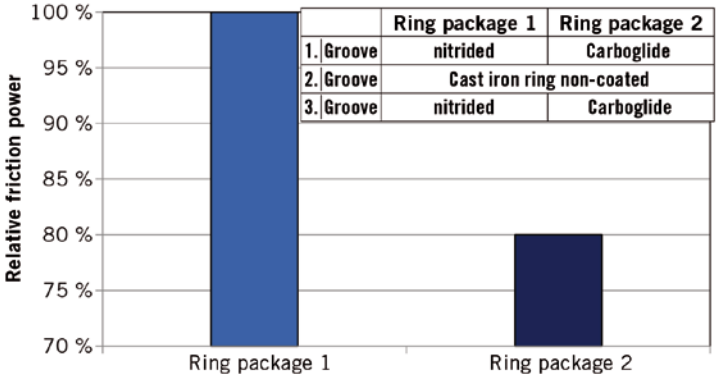


❹ Coefficients of friction of typical gasoline piston ring coatings as measured with an oscillating, vibrating frictional wear tribometer (cast iron cylinder, stroke 30 mm, frequency 10 Hz, normal force 450 N, ester oil lubrication, temperature 190 °C)

build-up in combination with cylinder scoring, while the Carboglide piston ring running surface shows a clear uniform wear pattern without any cylinder wall damage occurring. The positive effect becomes particularly evident in evaluating cylinder wear. In cast iron cylinders Carboglide significantly reduces cylinder wear in the top dead center area. Non-engine rig tests confirm a cylinder wear that is lower by a factor of up to 3 compared to PVD coated piston rings.



❺ Relative scuff resistance of nitrided, PVD and Carboglide coated piston rings, (cast iron cylinder, stroke 4 mm, frequency 40 Hz, normal force continuously increasing to 700 N, lubrication Castrol Edge 5W30, temperature 120 °C)

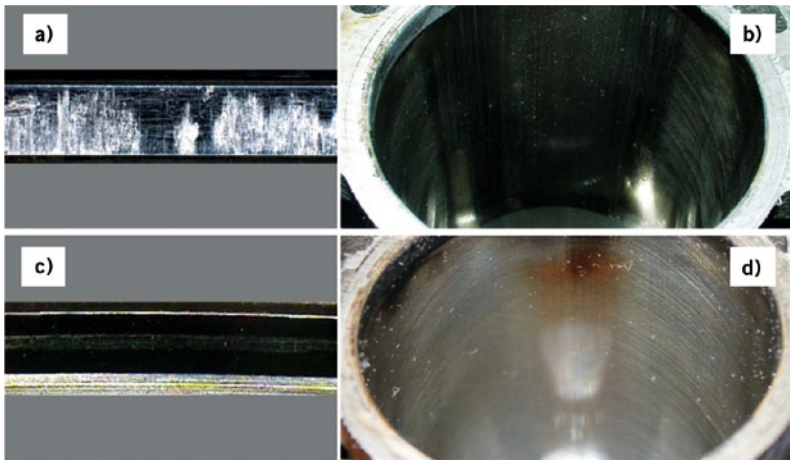


❻ Relative frictional losses of a piston ring package with Carboglide coated top and oil control ring in fired single-cylinder engine in comparison to a conventional nitrided ring pack

SUMMARY AND OUTLOOK

The new coating coating combines extremely low friction values with high strength and durability of piston rings and cylinder running surfaces. By using this coating, the ring pack's frictional losses can be reduced by up to 20 %. It significantly protects the cylinder running surface against scoring, increased wear and scuffing during inadequate lubrication. Carboglide makes a substantial contribution to the development of high performance gasoline engines with even better fuel economy by up to 1.5 % with consequently lowering CO_2 emissions.

Future development work will focus on the combination with the design advantages of LKZ oil control rings addressing both frictional losses and low oil consumption [5]. Due to the low friction and the high wear resistance of Carboglide the combination with LKZ further reduces the frictional losses of the ring pack. Initial customer tests in diesel engines for delivery vans already resulted in a significant reduction of fuel consumption of around 1 % during the New European Drive Cycle. This potential with oil control rings in diesel engines will be examined during further research.



❼ Comparison of the running surfaces of nitrided (a) and Carboglide (c) coated top rings and the corresponding cast iron cylinder running surfaces (b + d) after a 525 h endurance test in a turbocharged gasoline engine

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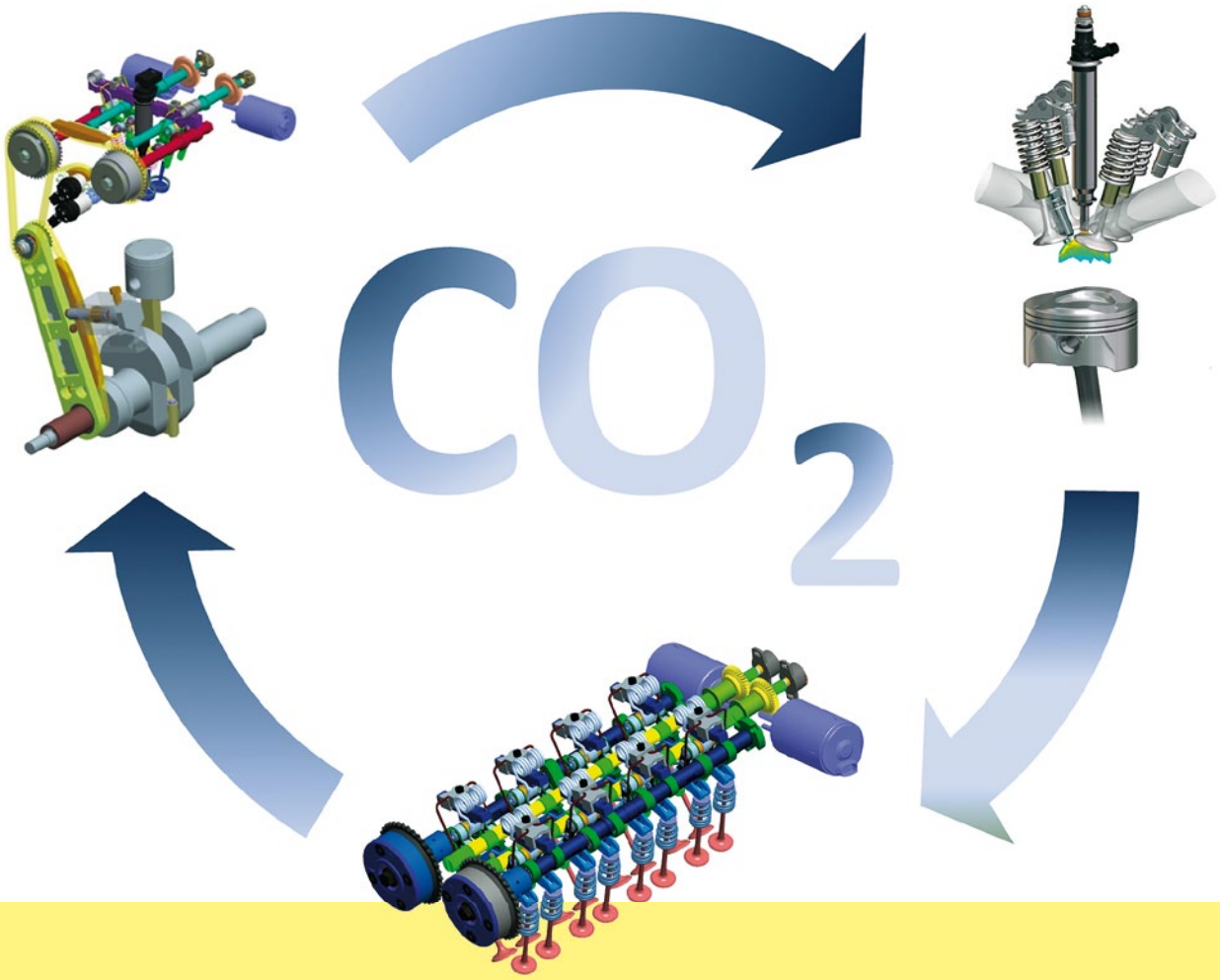
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POTENTIALS AND LIMITS OF CO_2 EMISSIONS OF GASOLINE ENGINES



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PART 2: NEW COMBUSTION PROCESSES

In MTZ 4 the Technical University of Kaiserslautern and Karlsruhe Institute of Technology (KIT) have presented the mechanical possibilities to further develop homogeneously and stoichiometrically operated gasoline engines. This second part of the article describes new combustion processes to increase engine efficiency. For this purpose stoichiometric and stratified lean operation mode as well as future combustion processes like HCCI and their potentials are presented. The described potentials and limits of both articles are combined and contrasted with each other.

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MOTIVATION

Already in [1] the possibilities and limits of thermodynamic approaches to optimise process on homogeneously, stoichiometrically operated gasoline engines via valve train variability were evaluated. Therefore the application of cam phasers, discretely staged valve lift shifters and fully variable systems were separately examined. Furthermore the potential to reduce fuel consumption with cylinder deactivation in throttle free engine operation was demonstrated; as well as the technology combination of fully variable compression ratio and mechanically fully variable valve train on inlet and outlet sides. This present article extends these statements by possibilities and limits to further develop combustion process.

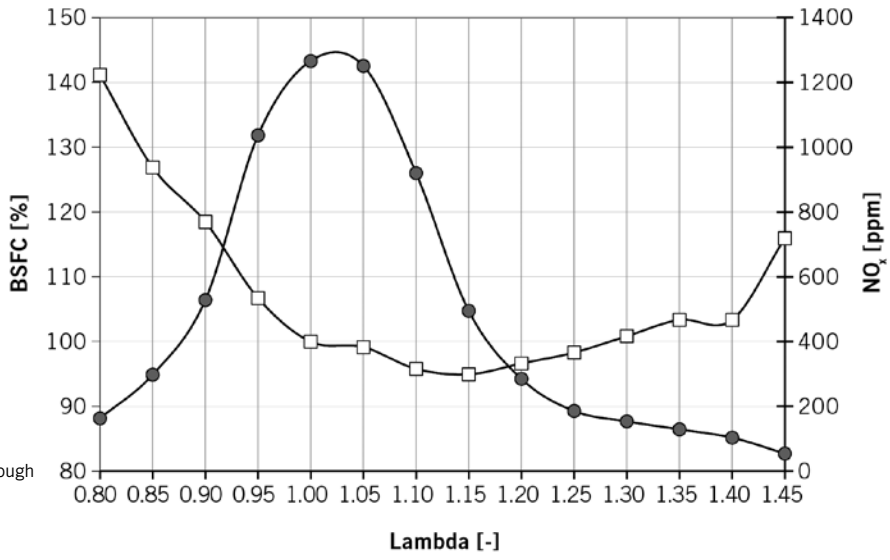
FULLY VARIABLE VALVE TRAIN WITH HOMOGENOUS LEANING

Fuel consumption of gasoline engines is reduced when operated homogenous lean, respectively with excess air. This homogeneous-lean mode is adjusted through higher inlet valve lifts which effectively decrease the charge-cycle-work losses compared to stoichiometric throttle-free operation. Homogeneous leaning results in charge dilution which reduces the residual gas compatibility of the engine. Thus, full charge cycle potential in lean operation mode cannot be exploited. For this purpose smaller inlet and outlet spreads would be necessary which are not applicable through the reduced residual gas capability. On a 3.0-l engine the effective fuel consumption is reduced by approximately 7.5 g/kWh or 2.5 % at n = 2000 rpm, BMEP = 2 bar through homogenous-lean engine operation mode with throttle-free load control, ❶.

During homogenous-lean operation mode, efficient exhaust gas after treatment by three-way catalyst is no longer useable to reduce NO_x emissions. Thus, additional extensive exhaust gas after treatment is necessary, e.g. NO_x storage catalyst or SCR. From today's perspective the transfer into mass production is irrelevant due to the small advantages in fuel consumption and the costly emission control.

FULLY VARIABLE VALVE TRAIN AND LAMBDA-SPLIT PROCESS

The idea of lambda-split process is to run a part of the engine's cylinders stoichiometrically with $\lambda = 1$ while the remaining cylinders are operated in homogenous lean mode with $\lambda > 1$. Applying lambda-split process to the subsequently analysed six-cylinder engine means, three cylinders are run stoichiometrically whilst three cylinders are operated in homogenous-lean mode. Lean operation mode is applied through reducing the fuel injection into these cylinders while keeping the amount of air constant.
In so doing, the load on these cylinders decreases. In order to maintain torque output of the engine, load and hence efficiency of the three cylinders which run stoichiometrically increases. Within the leaned cylinders, on the one hand effi-



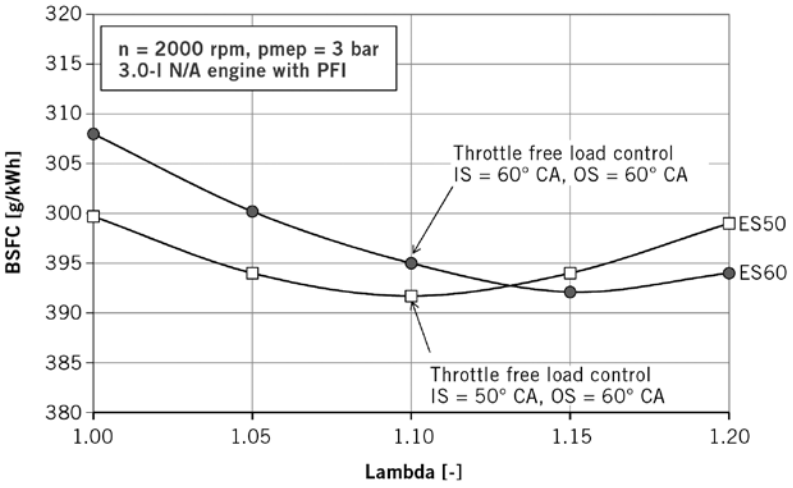
❶ Effective specific fuel consumption through homogenous leaning

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ciency increases through homogeneous lean burning, but efficiency decreases through shifting the load point to a lower indicated load. In total, the advantages of the load point and efficiency shifting overweigh, so that savings in fuel consumption of 2.5 % could be gained, ❷.

At higher load points with increased residual gas compatibility, fuel consumption advantages of up to 8 % could be reached [2]. Thus, this technique is particularly suited in combination with throttle-free load control to improve fuel consumption at higher loads (customer fuel consumption). However, after treatment of exhaust emissions also has to be adapted to this engine operation mode.

❷ Lambda-split process and its advantages in fuel consumption



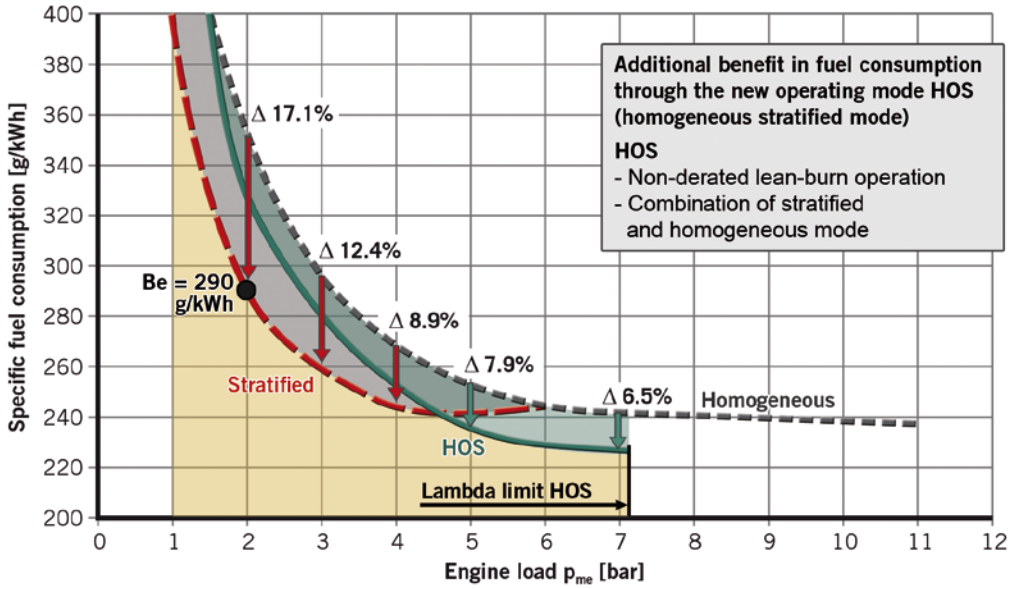
SPRAY-GUIDED GASOLINE DIRECT INJECTION

The fuel consumption which can be achieved today by spray-guided gasoline direct injection is described by Waltner et al. in [3] using a 3.5-l naturally aspirated engine with multi-injection through piezo injectors. The determined consumptions as well as the potentials compared to homogeneous stoichiometric engine operation are displayed in ❸. When relating these results to a 2.0-l naturally aspirated engine, the calculated outcome is approximately 300 g/kWh at an engine operating point of n = 2000 rpm, BMEP = 2 bar compared to the 3.5-l naturally aspirated engine with 290 g/kWh. The reason is the increased power consumption of the auxiliaries. In a 2.0-l naturally aspirated engine, this operating point corresponds approximately to the engine performance of NEDC's current average speed.

Since the average driving speed actually driven on the street is approximately double this amount, a higher average engine performance is required in every day vehicle operation. The engine operating point most applicable to this situation is at n = 2000 rpm, BMEP = 5 bar. In a naturally aspirated engine, the specific fuel consumption at this particular operating point lies at 255 g/kWh which corresponds to a specific efficiency of 33 %. When relating this efficiency to the actual efficiencies between 25 and 30 % achieved today in everyday vehicle operation with homogenous gasoline direct injection and charging, an efficiency increase of 10 up to over 30 % can be attained through the use of spray-guided direct injection with stratified charge and piezo injectors. Through more intensive research and further development of this combustion process, focusing especially on the injection technique used here, but also through improved fuels, efficiencies of more than 40 % may be further attained. To date, spray-guided direct injection opens up the highest potential so far for fuel economy that can be achieved with today's approved technologies.

Similarly to direct injection with homogenous mixture preparation, fur-

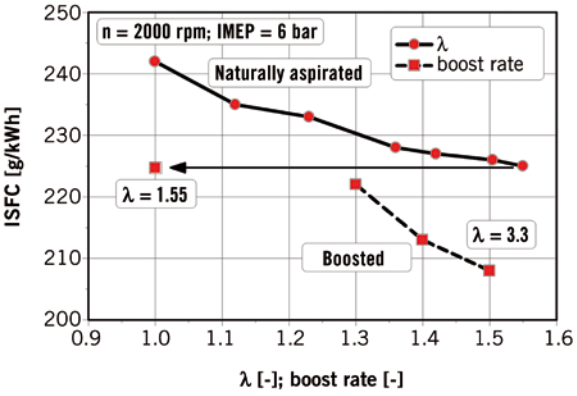
❸ Spray-guided direct injection and its advantages in fuel consumption [3]



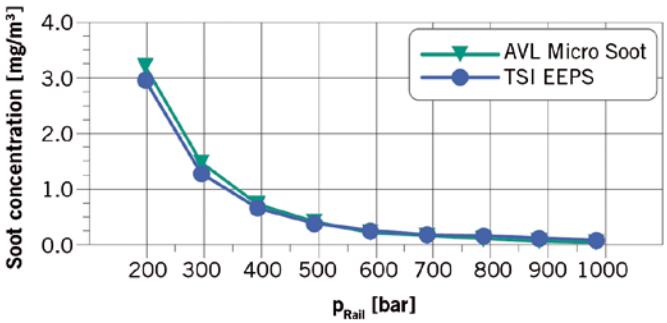
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ther improvements can also be accomplished in this mode through charging. Indeed, initial tests in [4] with especially high injection pressures of up to 1000 bar, using multi-hole diesel-nozzles and charging, have shown that in stratified charge mode, efficiency increases of over 40 % (indicated) and over 35 % (effective) with further fuel consumption savings (bi = 208 g/kWh respectively be ≈ 230 g/kWh) are possible, ❹. In so doing the engine was run at a degree of charge of 1.5. Thus, at an indicated mean pressure of IMEP = 6 bar (BMEP = 5 bar), a very lean air ratio of λ = 3.3 arose.

It was also evident that the particle emissions (weight and number) in particular stayed clearly below future limits [5]. ❺ demonstrates the soot concentration for the same engine operating point with IMEP = 6 bar and an engine speed of n = 2000 rpm. The engine was run here using stratified injection in naturally aspirated mode and was thus operated without charging. The injection pressure was varied in steps of 100 bar, starting from 200 bar proceeding to a maximal injection pressure of 1000 bar. A clear decrease in soot concentration was observed through increasing the injection pressure. The drop in soot concentration from approximately 3 to 0.1 mg/m³ was a result of the reduced number of particles with large diameters. All in all, the number of particles decreases by 88 %, while the particle weight was even reduced by 97 % through an increase of the injection pressure. ❻ displays the overall mass of particles for an engine load variation in stratified mode at an engine speed of n = 2000 rpm for injection pressures of 200 bar and 1000 bar utilising the two injection nozzles applied in [5].



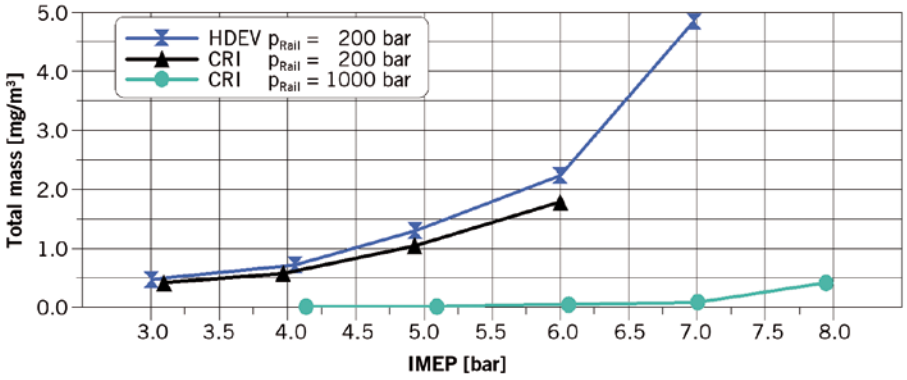
❹ Specific fuel consumption using stratified spray-guided gasoline direct injection with charging (n = 2000 rpm, BMEP = 6 bar, injection pressure = 1000 bar)



❺ Soot concentration when varying the injection pressure, using stratified spray-guided gasoline direct injection with charging (n = 2000 rpm, pmi = 6 bar) [5]

It is evident that the overall mass of emitted particles at an injection pressure of 1000 bar is considerably lower than that at an injection pressure of 200 bar. In fact, up to a mean pressure of IMEP = 6 bar, the particle mass remains below 0.05 mg/m³ at an injection pressure of 1000 bar, and only starts to increase at IMEP = 7 bar to 0.5 mg/m³ at IMEP = 8 bar. Given today's current practice of maximum injection pressures of 200 bar, such a result can only be achieved at an indicated mean pressure of 3 bar.

In so doing a reduction of NO_x emissions could likewise be achieved compared to an operation with injection pressures of 200 bar, although further testing is here required. In particular, exhaust gas retention and exhaust gas recirculation strategies are relevant here. In light of this aspect, a combination of direct injection and fully variable valve control on inlet and exhaust side is promising. In fact, tests on stratified exhaust gas recirculation [16] have shown that NO_x emissions can be reduced by far more than 90 %, in some cases even up to 99 %.

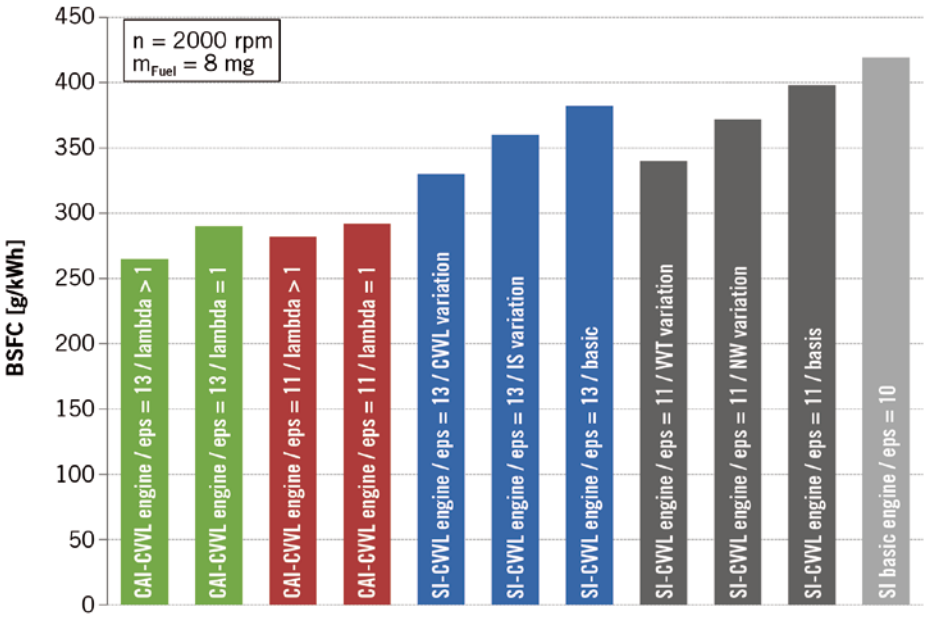


❻ Overall mass of particles using stratified guided gasoline direct injection and load variation (n = 2000 rpm) [5]

HOMOGENEOUS CHARGE COMPRESSION IGNITION

A mechanically fully variable valve train on the inlet and exhaust side and a variable compression ratio are the best preconditions for homogeneous charge compression ignition (HCCI) operation in a wide engine map range. The conflict between high effective compression ratio at part load and reachable high mean pressures can thereby also be resolved. According to Gottschalk [7], a consumption of 270 g/kWh at part load can be achieved when injecting 8 mg fuel in a

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❼ Comparison of efficiency effects in combustion process and degrees of freedom in valve train at the part load point m_{Fuel} = 8 mg, n = 2000 rpm [7]

turbo charged engine with a geometrical compression ratio of ε = 13:1 in lean engine mode, ❼.

The research described in [8], which examined basic relations and phenomena between injection, mixture preparation, self ignition and combustion in premixed fuel/air preparations, showed that through homogenous charge compression ignition not only fuel consumption can be reduced, but furthermore NO_x emissions can also be drastically decreased. ❽ shows the results of engine performance at loads between IMEP = 4 bar and IMEP = 6 bar at an engine speed of n = 2000 rpm as well as for an operating point at an engine speed of n = 1500 rpm and an indicated mean pressure of IMEP = 6 bar. The engine was operated here in such a way that the NO_x emissions always remained below 10 ppm (0.07 g/kWh), which results in an air ratio of between λ = 1.65 and 1.8. The engine was additionally charged with an externally serviced compressor. Intake pressure and exhaust gas back pressure were similarly modulated in order to adjust the engine operating points in a precise manner. The time at which 50 % of the fuel mass is burned (MFB = 0.5) was set to be at 8 °CA after top dead centre. The variance of the cyclic deviation in indicated mean pressure demonstrated values of approximately 1 %. The midrange maximum pressure gradient fluctuated between 5.5 and 7 bar/° CA. The specific fuel consumption was compared to a similar engine with charging, variable valve control and gasoline direct injection (TVDI).

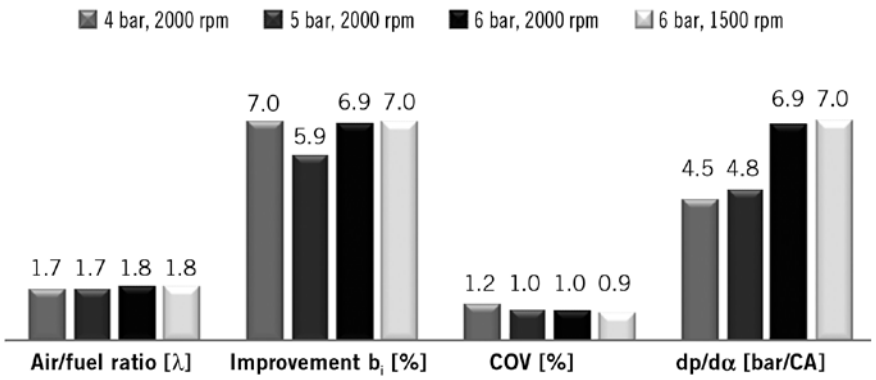
At each of the investigated engine operating points, the advantage in fuel consumption in HCCI mode was between approximately 6 and 7 %. As well as showing a clear reduction of exhaust emissions, in particular of NO_x emissions, this also shows that in comparison to today's current combustion processes, the HCCI combustion technique offers an advantage in terms of fuel consumption. This advantage is not so clearly noticeable compared to stratified charge operation with direct injection and charging.

Nevertheless, this combustion technique remains of interest for the further development of gasoline engines. Particularly the extremely low levels of exhaust emissions might make extensive after treatments of exhaust gas (particle filters; NO_x catalysts) unnecessary. In this context it is by all means also worth considering combining direct fuel injection with stratified charge and HCCI mode, depending on the operating point in the engine map.

SUMMARY AND FUTURE PROSPECTS

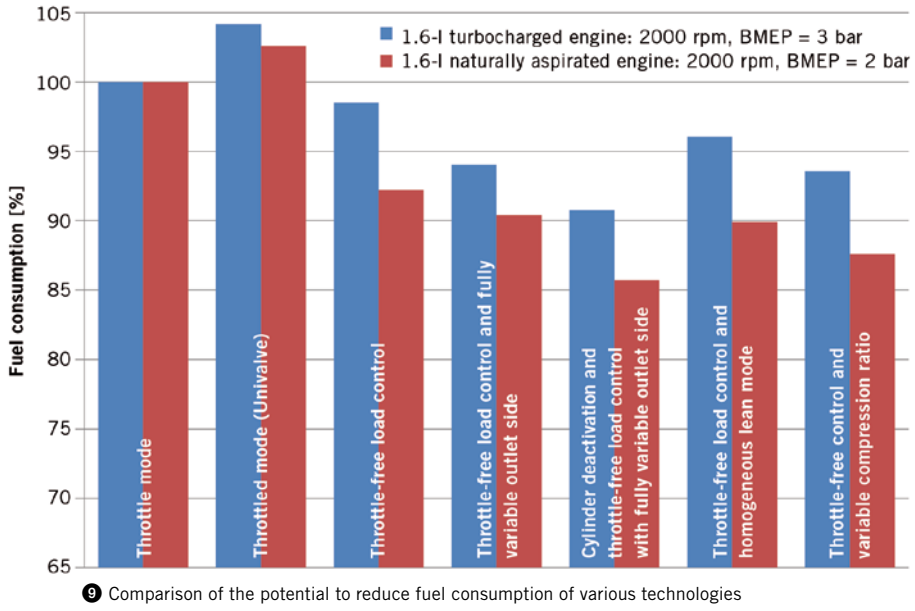
This paper shows that there is still a high potential to reduce the fuel consumption and CO_2 emissions in gasoline engines. When examining future combustion technologies it is noticeable that development departs from conventional gasoline combustion. A combination of different combustion methods may allow the lowest possible fuel consumption and exhaust emissions in the overall engine load map.

Some new combustion technologies also make demands on engine mechanics, e.g. fully variable valve trains for advanced residual gas control respectively variable compression ratio. To investigate the limits of such potentials in the com-

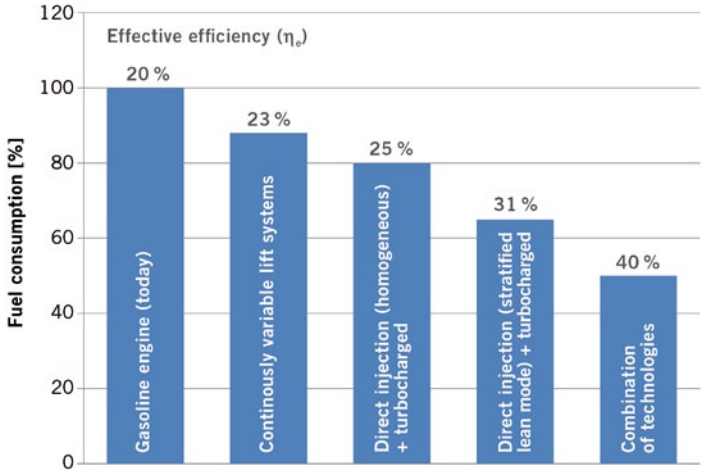


❽ Air ratio, improvement in fuel consumption, variation coefficient of mean pressure deviation and pressure gradient in HCCI mode with pilot injection [8]

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❾ Comparison of the potential to reduce fuel consumption of various technologies



❿ Comparison of the potential to reduce fuel consumption of various technologies

bustion of gasoline engines, thermodynamic and mechanical approaches have to be further advanced together in joint research projects, ❾ and ❿.

When comparing the different measures aimed at improving engine efficiency through deliberate development of injection systems, charging, valve trains, friction and auxiliaries as well as factors in basic engine design (geometry of combustion chamber, charge motion, compression), drive train efficiencies of 40 % and beyond are expected to be attained. In relation to the NEDC, according to which today's new vehicles of the B-segment are consuming on average approximately 6.5 l fuel per 100 km (146 g CO_2/km) [9], 40 % of efficiency results in 4.15 l per 100 km (93 g CO_2/km). It is therefore the case that the required 95 g CO_2/km by the year 2020 for a vehicle of the B segment (e.g. VW Golf) can be achieved simply through a consequent enhancement of gasoline engines, using additional funds that are still lower than the costs of diesel engines and far below the costs of new driving systems (full hybrid, plug-in hybrid, electric vehicles, fuel cell).

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LASER IGNITION OF ALTERNATIVE LIQUID FUELS

Within a research project at the TU Vienna, the potential and mechanism of laser-induced ignition with respect to mixture inflammation and combustion were investigated compared to conventional spark ignition. A single-cylinder research engine with homogeneous charge was operated with bio-ethanol, bio ethyl-tertiary-butyl-ether (ETBE), butanol and blends of these fuels. Their potentials were compared with conventional gasoline. The intention of the investigations was to systematically describe the differences between spark and laser ignition with respect to the thermodynamics of the combustion process and the pollutant emissions.

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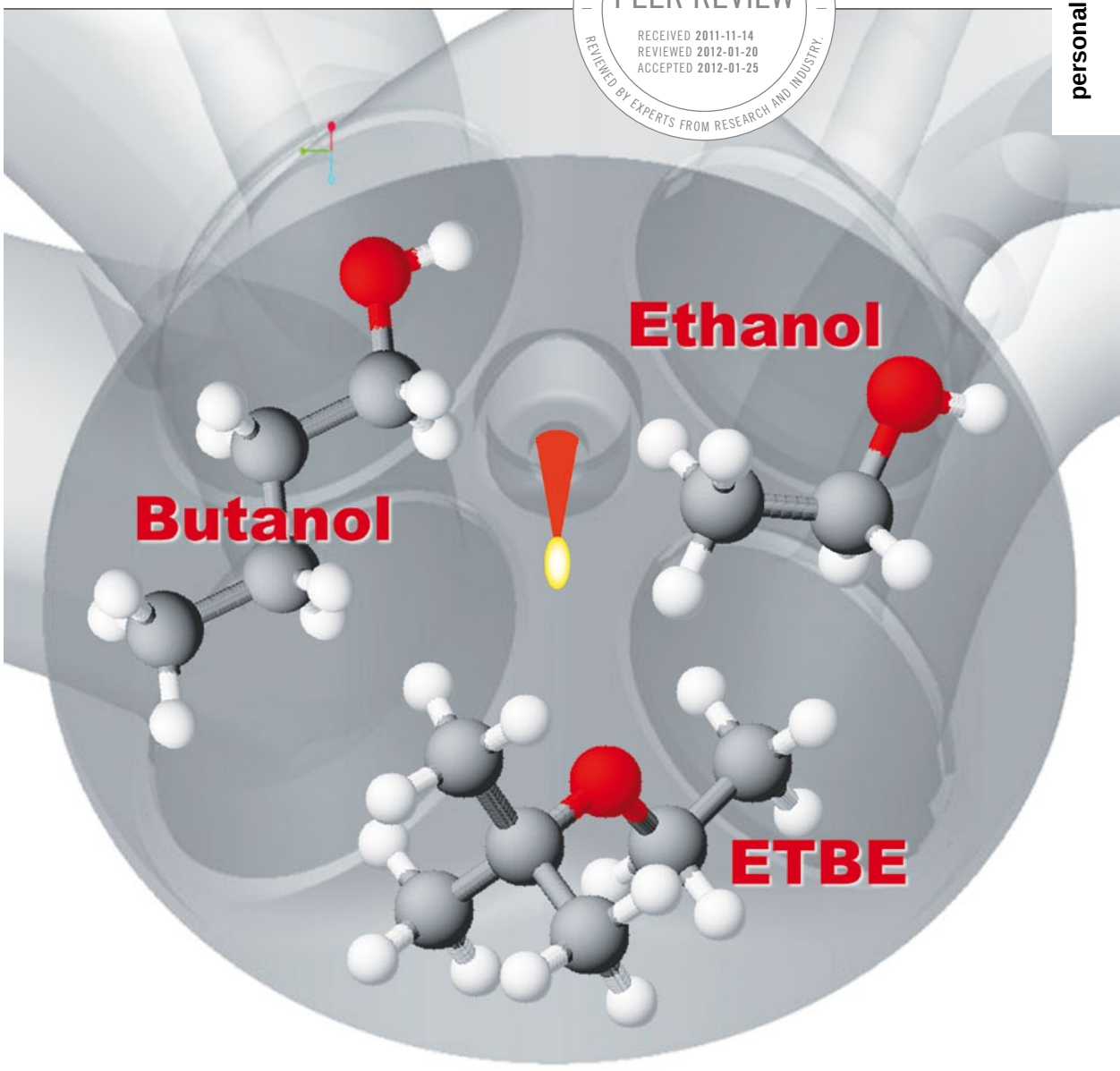
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1 INTRODUCTION
2 INVESTIGATED ALTERNATIVE FUELS
3 EXPERIMENTAL SETUP
4 ANALYSIS OF THE MEASURED RESULTS
5 SUMMARY

1 INTRODUCTION

Due to limited resources of fossil energy carriers and a steadily increasing global demand of energy, alternative fuels will become more and more important. This development is emphasised by the requirements of further reductions of emissions and greenhouse gases [1]. However, a thorough adaption of the engine design to the fuel properties is necessary to exploit the specific potentials of the alternative fuels. In the following, the differences between laser ignition and spark ignition in terms of combustion initiation and energy conversion are discussed for a spark ignition engine with homogenous charge. Additionally, the emission formation with alternative fuels was evaluated.

2 INVESTIGATED ALTERNATIVE FUELS

Alcohols are the most promising alternatives to conventional fuels originating from crude oil. Ethanol is appealing due to its manufacturing from cellulose and the resulting exploitation of bio resources beyond the food chain. Nowadays, ethanol is the mostly used alternative energy carrier for spark ignition engines. Due to a low C/H ratio, a high heat of vapourisation in order to cool the aspirated mixture and a comparatively high research octane number, potentials for the reduction of greenhouse gases and exhaust emissions can be derived [1–3]. Principally, spark ignition engines are capable of using longer chained alcohols, like butanol, ethers (like ETBE) and other highly volatile bio energy carriers.

To determine the potential of longer chained alcohols, experimental and analytical investigations were carried out at the Institute for Powertrains & Automotive Technology (IFA) at the Vienna University of Technology (TU Vienna) in the framework of a multiannual research programme. In the following, the results of the research work are explained. The specific advantages and disadvantages of the ignition systems will be specified. For more details on fuel characteristics (like research octane number, vapour pressure etc.) [7].

3 EXPERIMENTAL SETUP

A single-cylinder spark ignition research engine with port fuel injection was used for the experiments. The four valve cylinder head had a central spark plug. The technical specifications of the engine are summarised in ❶.

3.1 SPARK IGNITION

A high-frequency AC ignition system was chosen to realise the spark ignition. A spark plug with a narrow shaped, platinum electrode that enables a good accessibility of the mixture and a stable combustion (reduction of the quenching effect) has been chosen for this comparison. An increased ignition energy of 250 mJ (approximately twice the ignition energy of a series transistor ignition system) and a typical spark duration of 1 ms were defined for the following investigations. Thus, a reliable ignition was ensured even for lean mixtures.

3.2 LASER IGNITION

As ignition unit, a Nd:YAG-solid state laser with a wavelength of 1064 nm and a beam diameter of 6 mm was used. The laser beam was launched into the combustion chamber via an optical adapter that was screwed into the available spark plug thread. The lens window was made of silica glass with a focal length of 16 mm.

During the focusing event of a short laser impulse in air, an intense spark (ignition plasma) is created in the focal point of the lens window that is somehow similar to the plasma that is created during the discharge between two electrodes [4–5]. Besides the considerable difference of the plasma temperature, the main characteristic is a different efficiency of the energy supply to the combustion gas compared to the spark ignition [4]. For a comparative measurement, the laser plasma (focal position) has been positioned at the same location as the spark of the spark plug. In order to investigate the influence of different ignition systems, the advantage of a more central position of the spark that could have been achieved with the laser ignition was neglected. The laser energy was defined to 60 mJ, which is an empirical value that is based on years of research experience with this ignition system. Thus, a save ignition is enabled, even for very lean mixtures.

3.3 CALCULATED COMBUSTION ANALYSIS

In parallel to the experiments, an analysis of the combustion process based on the measured cylinder pressure was carried out with the simulation software GT-Power. For the validation of the analytic approach, the volumetric efficiency was determined on the test bench with a hot film (heated area) sensor and the cylinder pressure with a pressure sensor. The fuels' enthalpy was defined using a polynomial approach for the liquid and vapour phase [8, 9]. The specific coefficients for gasoline EN 228, ethanol, butanol and ETBE were taken from the literature [10–13].

4 ANALYSIS OF THE MEASURED RESULTS

4.1 COMBUSTION ANALYSIS

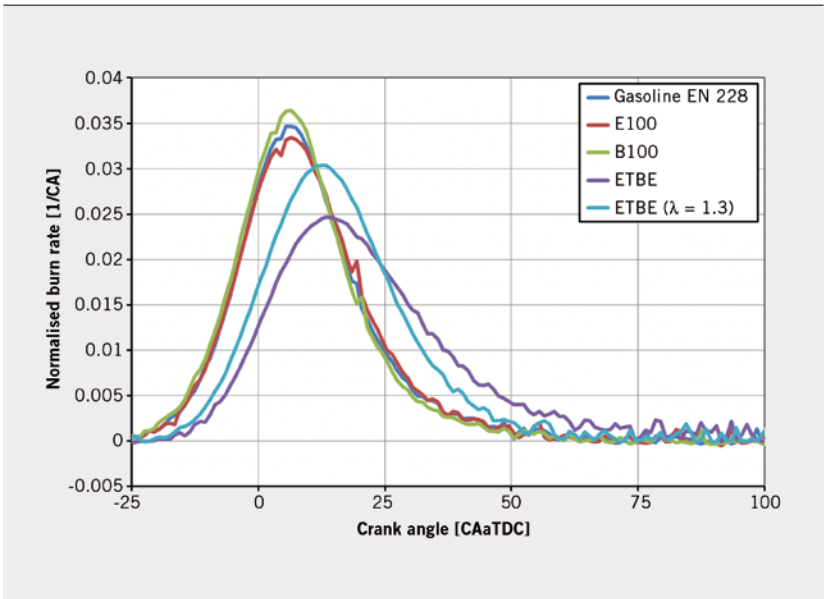
The described combustion analysis has been carried out at a part load operating point with a relative air/fuel ratio of $\lambda = 1.4$ for the fuels gasoline EN 228, ethanol (E100), butanol (B100) and ETBE in order to gain a deeper understanding of the combustion process. The ignition of the mixture was realised with the described laser. ❷ shows the burn rates that were derived from the measured cylinder pressure. For ETBE a later ignition at higher pressure and compression temperature was necessary to ensure the combustion stability. Additionally, a less lean mixture with a relative air/fuel ratio of $\lambda = 1.3$ was investigated. A burn rate with an optimal position of 50 % mass fraction burned with respect to combustion efficiency could be achieved with the laser ignition for the fuels gasoline EN 228, E100 and B100.

The temperatures during compression and combustion were calculated in dependency of the different physical properties of the

SINGLE-CYLINDER RESEARCH ENGINE WITH PORT FUEL INJECTION	
Stroke	85 mm
Bore	88 mm
Displacement volume	517 cm^3
Compression ratio	11

❶ Technical specifications of the test engine

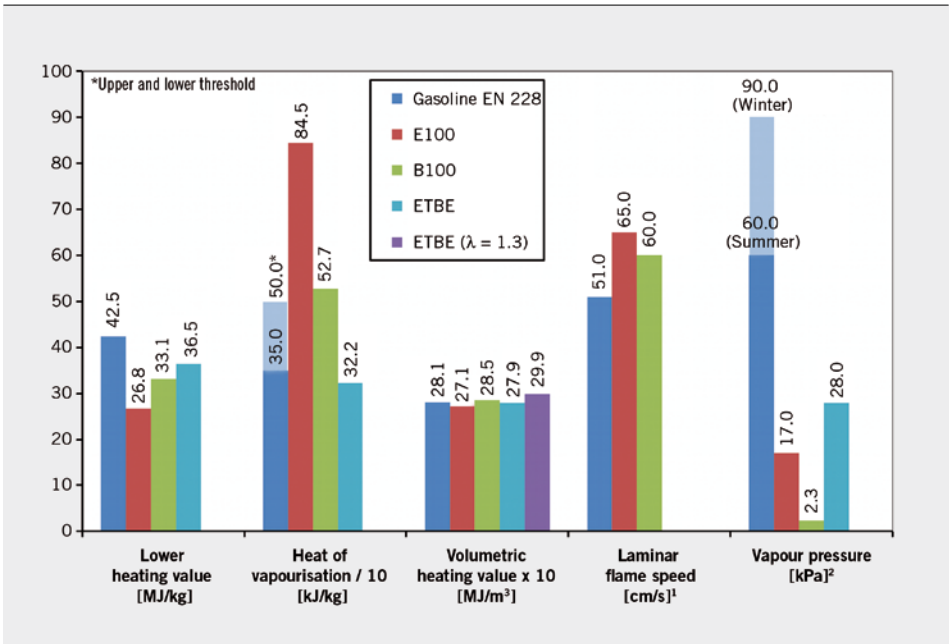
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❷ Burn rates derived from the measured cylinder pressures for the investigated fuels (n = 2000 rpm, IMEP = 4 bar, λ = 1.4)

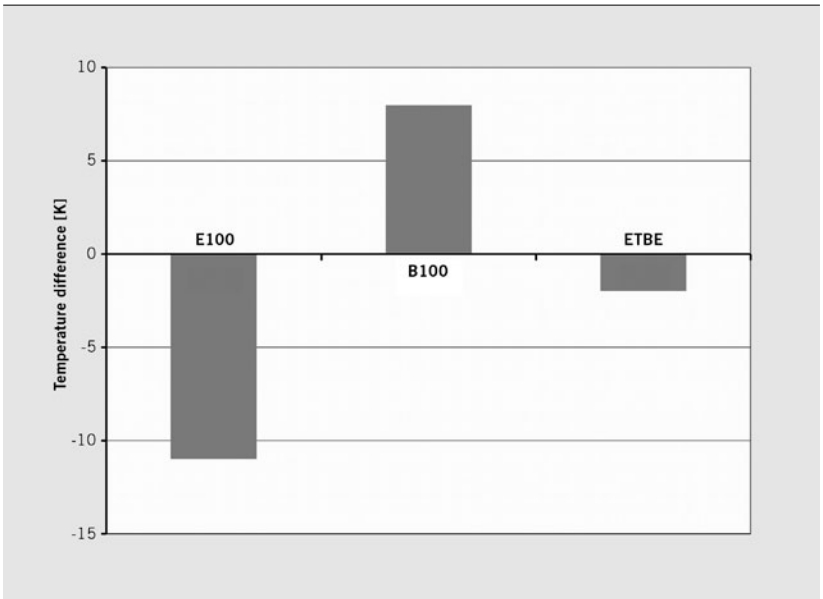
particular air/fuel mixtures in order to analyse the combustion and the emissions. ❸ illustrates some characteristic fuel properties, namely the lower heating value, the heat of vapourisation, the volumetric heating value of the mixture, the laminar flame speed and the vapour pressure according to DIN EN 13016. The laminar flame speed was taken from a reference where ISO-octane, E100 and B100 were investigated in one measurement series at an initial temperature of 393 K [14]. The illustrated values are the maximum laminar flame speed at an equivalence ratio $\Phi = 1/\lambda = 1.1$. ETBE, on the other hand, showed similar values of the maximum laminar flame speed as gasoline [15]. Due to the different temperature ranges in the two references, these values are not included in ③.

In a first step, the temperature of the aspirated mixture was calculated from the measured volumetric efficiency and the measured cylinder pressure. Due to the different values of the heat of vapourisation and the stoichiometric air/fuel ratio of the investigated fuels, a more or less intense mixture cooling follows. Particularly the two alcohols have a significantly higher heat of vapourisation compared to conventional gasoline. ❹ shows the calculated temperature difference of the aspirated mixture at the start of compression in comparison to gasoline EN 228. It can be noticed that there is a moderate mixture cooling of 11 K for ethanol, of 2 K for ETBE and that there is a temperature increase of 7 K for butanol. These values, which are significantly below the theoretical values (such as for ethanol approximate 60 K at λ = 1.4 can be estimated), can be explained



❸ Fluid and mixture properties for the different fuels
¹ T_{init} = 393 K; p_{init} = 1 bar; Φ = 1.1; ISO-octane instead of gasoline EN 228 [14]
² Maximum allowable values for gasoline according to EN 228

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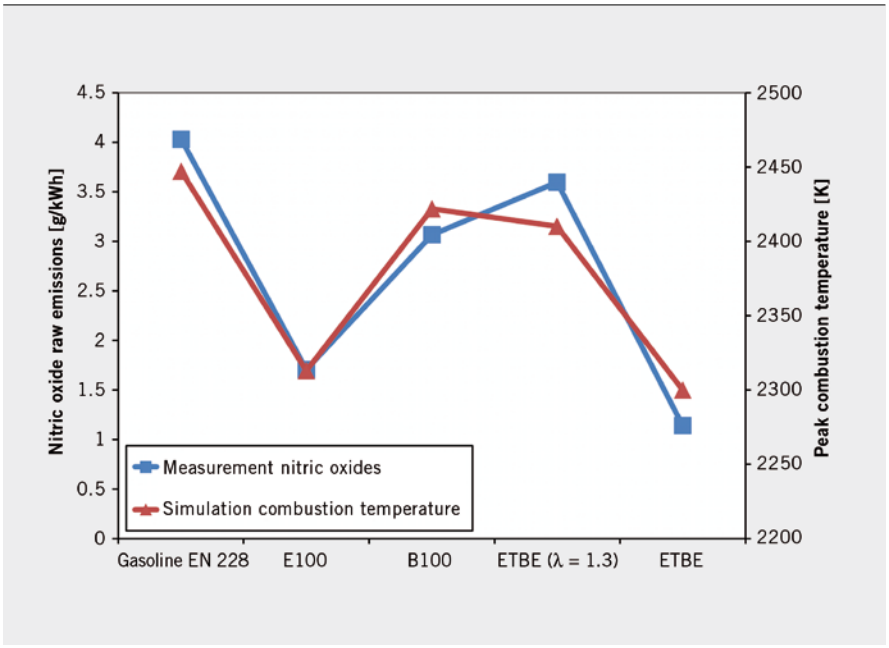


❹ Calculated mixture cooling for the investigated fuels relative to gasoline EN 228 (n = 2000 rpm, IMEP = 4 bar, λ = 1.4)

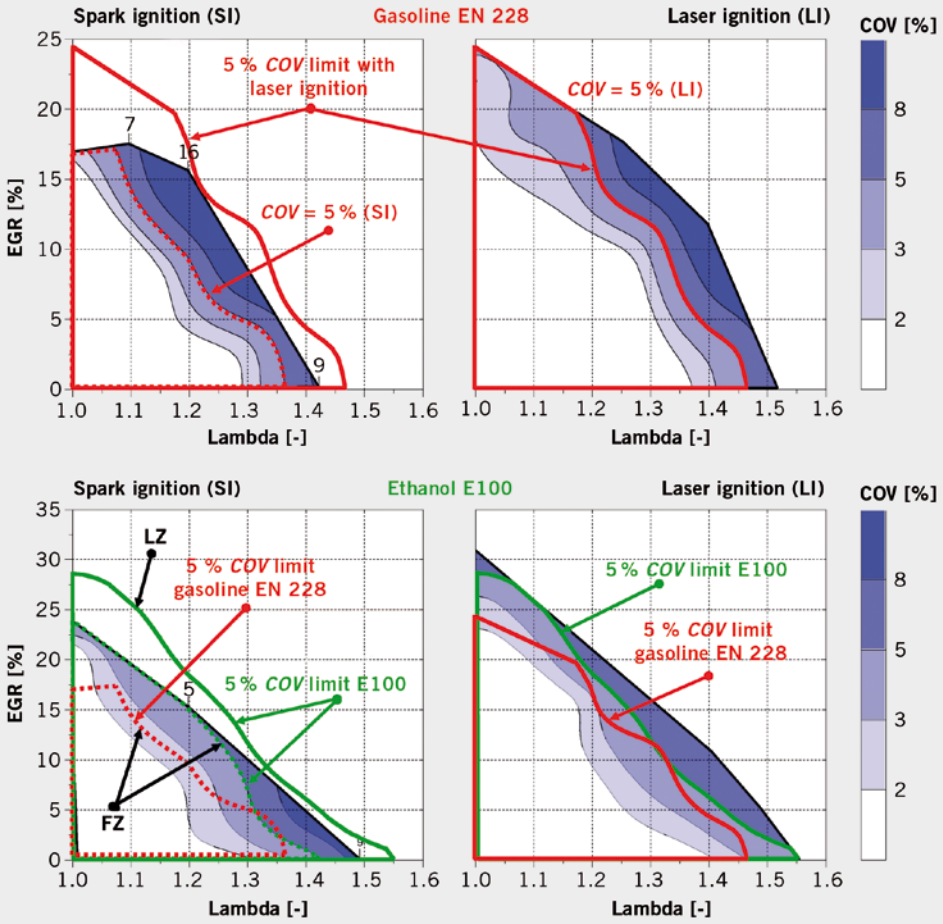
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with the wall film formation on the surfaces of the inlet port and the inlet valve disks and the resulting heat fluxes between the structure and the liquid film. This is typical for spark ignition engines with port fuel injection [16, 17]. Butanol, on the other hand, has an extremely low vapour pressure. Therefore, incomplete fuel evapouration at inlet valve closing can be assumed.

In ❺ the calculated peak combustion temperatures are compared with the measured nitric oxide emissions. Generally, a good correlation can be observed. It becomes obvious that in spite of almost identical burn rates different peak temperatures are calculated and different nitric oxide emissions are measured for gasoline EN 228, E100 and B100. Besides the mixture temperatures at the start of compression, ④, the maximum combustion temperature is determined by the volumetric heating value of the mixture, the specific heat capacity of the burned gas and the dissociation for a given burn rate. As shown in ③, the volumetric heating value of the mixture is comparable for the investigated air-fuel mixtures. Only for E100, the value of this parameter is moderately smaller. For ETBE and a different relative air/fuel ratio of λ = 1.3 it is accordingly higher. The mixture properties are influenced by the fraction of complex molecules with high specific heat capacities, such as the fraction of fuel vapour in the unburned mixture and the fraction of the products CO_2 and H_2O in the burned gas, see for example [11]. The calculation of the burned gas composition shows that the overall fractions of CO_2 and H_2O increase with a lower stoichiometric air/fuel ratio. Particularly for E100 favourable mixture properties in terms of peak com-



❺ Comparison of the calculated peak temperatures with the measured nitric oxide emissions (n = 2000 rpm, IMEP = 4 bar, λ = 1.4)



❻ Comparison of engine running smoothness (COV) with lean gasoline EN 228 and ethanol (E100) mixtures (n = 2000 rpm, IMEP = 4 bar, optimal consumption ignition timing)

bustion temperatures and nitric oxide formation can be derived from that. With this argumentation, the lower combustion temperatures of ethanol and butanol compared to gasoline EN 228 can be understood. The different temperatures of the two ETBE mixtures mainly result from the different volumetric heating value of the mixtures and the late combustion.

4.2 INFLUENCE ON THE CHARGE DILUTION

A significant reduction of the CO_2 emissions is possible with charge dilution at part load operation for spark ignition engines. The exhaust gas recirculation (EGR) combined with a stoichiometric combustion process ($\lambda = 1$) is state-of-the-art today. The potential of these concepts is mainly limited by the increasing cyclic variations of the combustion. For the following investigations, the limit of the engine's running smoothness was defined at a COV (variation coefficient of the indicated medium pressure) value of 5 %. In the right top diagram of ❻, there is a characteristic map for laser ignition shown with an operation limit of COV = 5 % (red line). In the left top diagram of ⑥, this limit is illustrated for spark ignition as a dotted line as well. The high potential in terms of charge dilution and fuel saving at part load operation is clearly visible when igniting the mixture with a focussed laser beam. The bottom diagrams of ⑥ describe the engine's running smoothness using pure ethanol. In the right bottom diagram of ⑥, the additional regions with an increased air/fuel ratio or EGR rate are illustrated for ethanol compared to gasoline EN 228 when operating the engine with the laser ignition system. However, the differences between spark ignition, dotted lines in ⑥, and laser ignition for E100 operation are smaller than for gasoline EN 228, left top diagram of ⑥. This observation can be explained with a shorter combustion delay when using ethanol. Especially in case of the spark ignition, an extended range of smooth engine run can be achieved.

In the upper diagrams of ❼, the engine maps for butanol are shown. With both, spark and laser ignition, only an engine operation with a less lean mixture is possible compared to gasoline EN 228, red lines in ⑦. However, the same maximum EGR rate is reached. The potentials of a higher charge dilution with laser ignition are plotted in the left top diagram in ⑦ in comparison to spark ignition. The use of pure ETBE reduces the usable part of the engine map most clearly compared to gasoline EN 228, bottom diagrams in ⑦. However, the potential of laser ignition compared to spark ignition in

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terms of a lean combustion is significantly higher for ETBE than it is for ethanol or butanol, left bottom diagram in ⑦. The advantages of the laser ignition compared to the spark ignition are most evident in terms of lean combustion and EGR for an engine operation with gasoline EN 228 and ETBE. This can be explained with the relatively low laminar flame speed of the air-fuel mixtures compared to ethanol and butanol, ③.

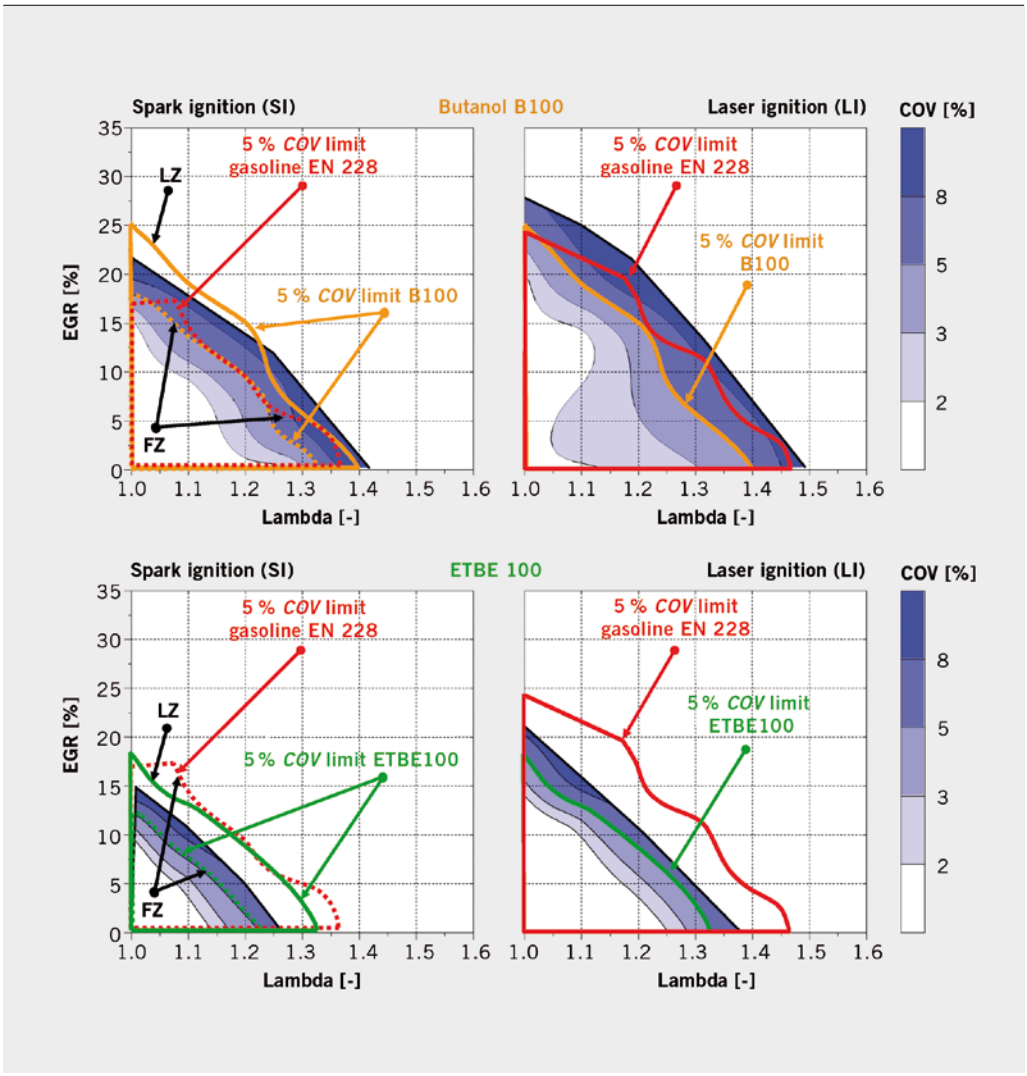
4.3 INFLUENCE ON COMBUSTION AND EMISSIONS

In the upper diagram in ❽, the position of burned is summarised for all investigated fuels and blends. With the exception of ETBE, the laser ignition always enables optimised spark timing with respect to the combustion efficiency. For the spark ignition, on the other hand, spark timing with optimal combustion efficiency can only be realised for pure ethanol. In the lower diagram in ⑧ the COV values are compared. Once again, the favourable effect of the laser ignition on the engine process with lean mixture becomes obvious.

Finally, ❾ shows the results for exhaust emissions (raw emissions) at part load and stoichiometric combustion for the three discussed alternative fuels compared to gasoline EN 228. In general, emissions are remarkably lower when operating the engine with the alcohol fuels. In addition to the fluid properties the burn rate and the resulting peak temperature are decisive. The extinction of the flame determines the amount of hydrocarbon emissions. Therefore, a high-laminar flame speed, such as for alcohol fuels, and a fast burning of the mixture at high temperatures are favourable. For stoichiometric engine operation no significant differences between spark ignition and laser ignition could be identified in terms of emissions. Additional measurements with blends of gasoline EN 228 with acetone, toluene and ISO-octane show similar behaviour with respect to combustion as ETBE. In a subsequent research project, the use of bio-gasoline, furans and other alcohols as blending components will be investigated [7].

5 SUMMARY

The combustion is mainly determined by the physical and chemical fuel properties. Analytical investigations were carried out to explain the mixture cooling and the peak combustion temperatures for the investigated fuels gasoline EN 228, ethanol, butanol and ETBE on the basis of their specific fluid properties. One of the main advantages of laser ignition is a rapid ignition of a homogeneous air-fuel mixture, especially at lean operation and higher residual gas content. An increasing fraction of ethanol decreases the advantage of the laser ignition. Therefore, hardly any differences are noticeable



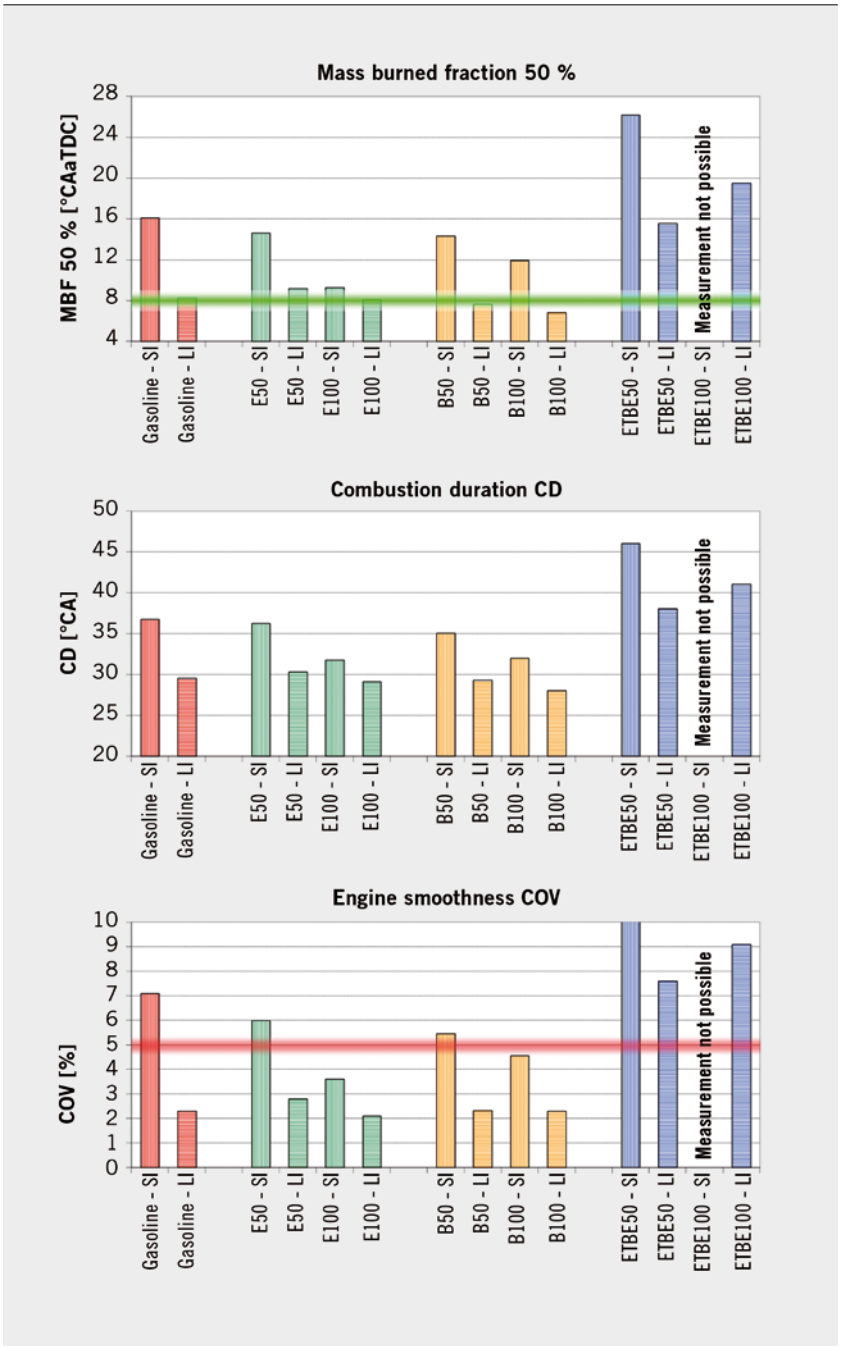
❼ Comparison of the engine running smoothness (COV) with lean butanol and ETBE mixtures (n = 2000 rpm, IMEP = 4 bar, optimal consumption ignition timing)

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between laser and spark ignition for pure ethanol operation. In contrast to ethanol, an engine operation with leaner air/fuel ratios or higher residual gas content rates could not be observer when blending gasoline EN 228 with ETBE or butanol.

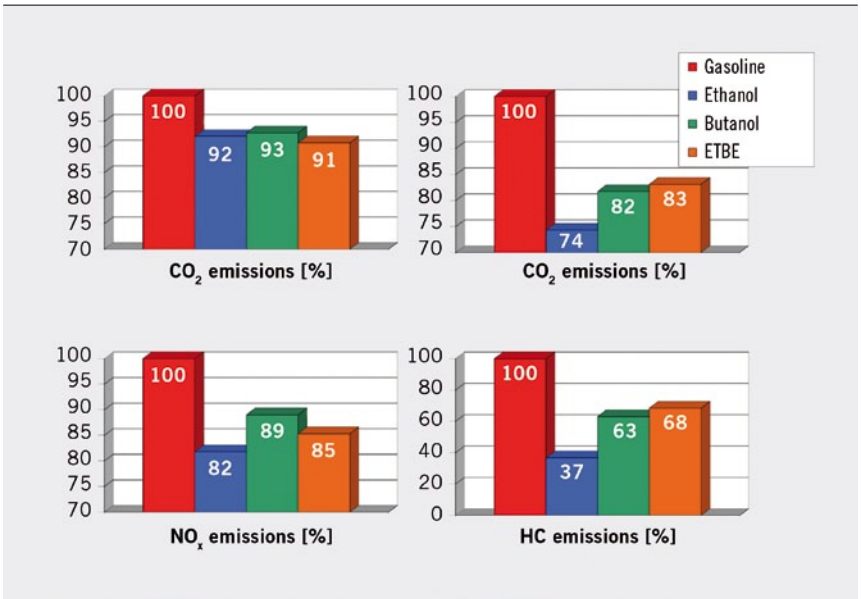
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❽ Comparison of the combustion parameters for different alternative fuels (n = 2000 rpm, IMEP = 4 bar, without EGR, λ = 1.4, optimal consumption ignition timing)

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❾ Emission potential compared for ethanol, butanol, ETBE to gasoline EN 228 [1] (n = 2000 rpm, IMEP = 4 bar, without EGR, λ = 1.0, optimal consumption ignition timing, spark ignition)

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STRUCTURAL DAMPING IN MECHANICAL JOINT CONNECTIONS

AUTHORS



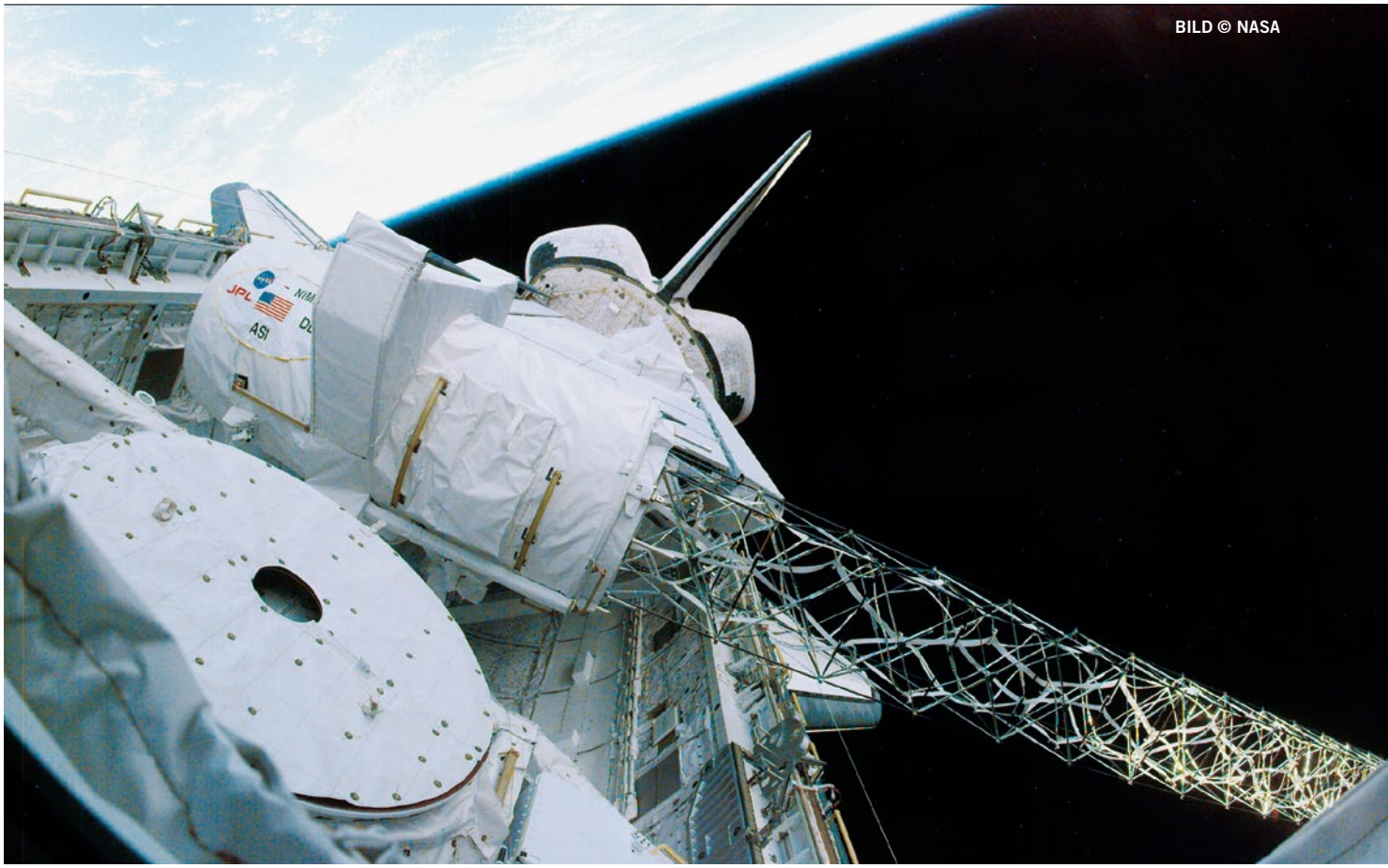
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Vibration properties of many assembled mechanical systems depend on frictional damping in bolted joints. The nonlinear transfer behaviour of the frictional interfaces often provides the dominant damping source in built-up structures and plays an important role in the vibration response of structures. At the University of Stuttgart, three different approaches were compared for modelling damping, based on a truss structure, a motor vehicle control unit and a car engine.

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1 INTRODUCTION
2 POINT-WISE JOINTS
3 AREA CONTACT JOINTS
4 SUMMARY

1 INTRODUCTION

The dynamic properties of most assembled mechanical systems depend on the frictional damping in joints. The non-linear transfer behaviour of the frictional interfaces often provides the dominant damping mechanism in assembled structure and plays an important role in the vibratory response of the structure. This article gives a short overview of three different approaches to damping modelling based on the contact distribution in joints. It discusses the modelling by local joints, by thin layer and zero thickness finite elements method (FE method). The parameters of the joint models are identified from experiments with generic test beds. The applicability of the proposed methods is discussed and documented by a deployable truss structure, an assembled control unit and an automobile engine.

2 POINT-WISE JOINTS

For improving the performance of systems, many studies have been carried out to predict, measure, and/or enhance the energy dissipation of friction. Here, the techniques for describing the nonlinear transfer behaviour of bolted joint connections are used. If the joint dimensions are small, compared to the shortest wavelength of structural deformation, point-wise joint models are sufficient. They include classical Coulomb and practical engineering models, such as Masing, LuGre, or Valanis models [1–3]. These phenomenological frictional models assume the friction force to be a function of relative velocity and/or of other internal state variables.

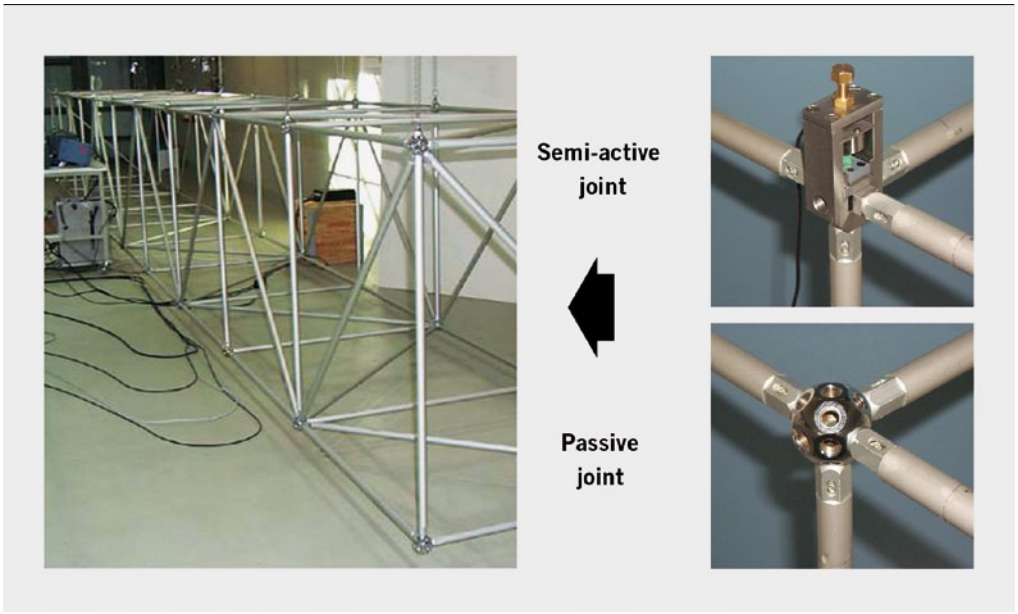
Constitutive friction models describing the nonlinear transfer behaviour of joints are based on a stress-strain behaviour between the joint surfaces. The models deal with the inherent nonlinearity of contact forces, such as Hertzian, Mindlin contact, and the nonlinear relationship between friction and relative velocity in the friction interface [4]. The research activities in this area are a combination of theoretical, numerical, and experimental investigations. Various solution techniques, commonly applied to friction-damped systems, are here discussed. Recent applications are outlined with regard to the use of joints as semi-active damping devices for vibration control.

Several application areas for friction-damped systems due to mechanical joints are presented in [5, 6]. The method is applied to an adaptive mast structure used for the Shuttle Radar Topography Mission, ❶. The mast is assembled with spherical space frame nodes and includes two semi-active friction joints. The displacement of mast tip due to an impulse is shown in ❷. The top plot shows the displacement of mast under free vibration. The bottom plot shows vibrational response, using the semi-active joints which are actuated by a control algorithm to maximise the damping [6].

3 AREA CONTACT JOINTS

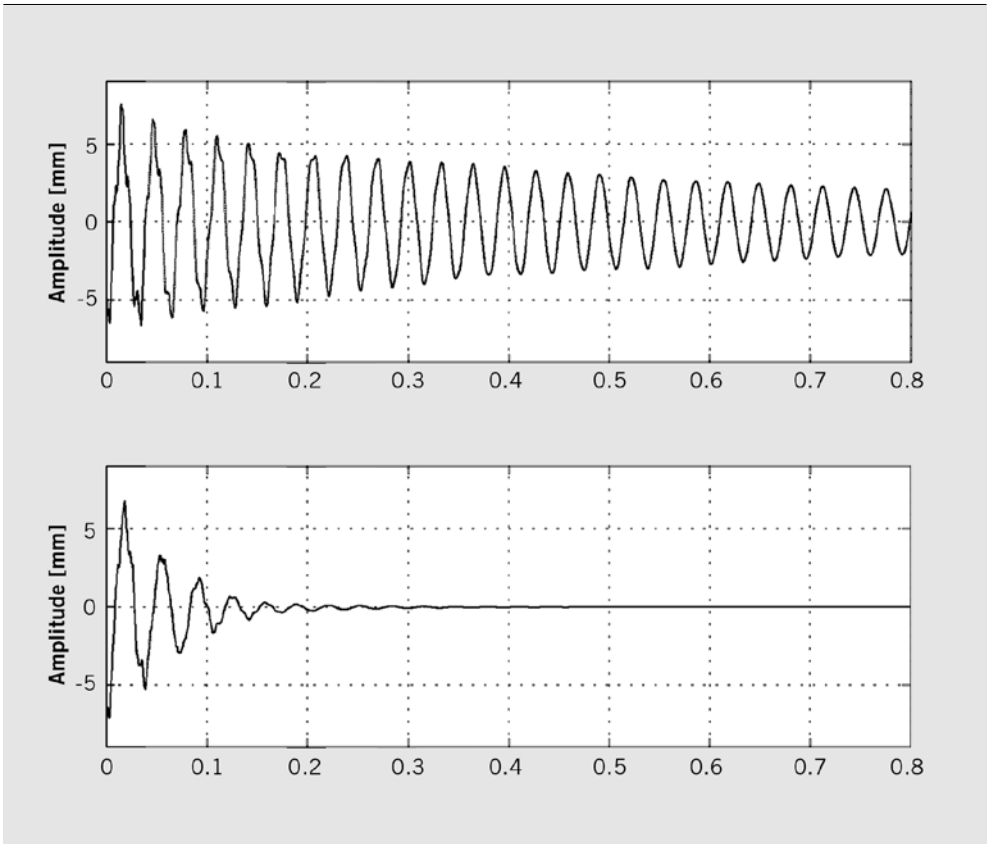
3.1 THIN-LAYER ELEMENTS

Assembled metallic structures with dry bolted joints can be modelled by thin layer finite elements in order to avoid a contact approach. The interface energy dissipation is found experimentally from a generic isolated joint test bench. The thin layer elements contain stiffness and damping parameters obtained from hysteresis measurements. The linearised approach provides a reliable prediction of modal damping ratios even in the design phase of assembled structures [7]. However, constant hysteretic damping estimation can be carried out only in frequency domain, since in time domain it leads to non-casual behaviour [8]. Also, joined surfaces should always remain in contact, since this would cause non-linear behaviour and would compromise the linear approximation of this approach.



❶ The SRTM (Shuttle Radar Topography Mission) mast is assembled with spherical space frame nodes and includes two semi-active friction joints (point contact)

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❷ Displacement of the mast tip: free vibration with passive joint (top) and controlled vibration using semi-active joint (bottom)

During the calculation of vibration characteristics of a structure with FE method the model of constant hysteresis is used to represent the damping in structure's equation of motion:

EQ. 1 $$M\ddot{u} + K^* u = F$$

M is the mass matrix, K* is the complex-valued stiffness matrix and u is the displacement vector. Eigenvalues and eigenmodes can be determined by performing a numerical modal analysis with FE software. The damping is included in the stiffness matrix in frequency domain by augmenting it with the complex-valued product of experimentally determined dissipation multipliers α_i and β_i for the material and the joint damping and the associated elements stiffness matrices:

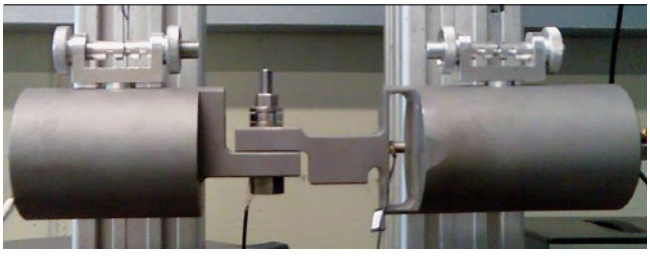
EQ. 2 $$K^* = K + jD = K + j\sum_{i=1}^{n} \alpha_i K_i^{(\text{material})} + j\sum_{i=1}^{m} \beta_i K_i^{(\text{joint})}$$

The material damping is determined from frequency response function (FRF) of homogeneous material samples [7]. The damping as well as tangential stiffness of the joint are acquired from a joint patch damping experiment shown in ❸. The set-up consists of two masses connected with a lap joint, where the normal force is applied by a bolt with force measurement ring. One of the masses has a build in leaf spring, so by exciting the system at its resonance, tangential forces up to 4000 N can be achieved. Acceleration of both masses is measured with accelerometers and by integrating acquired signal twice with respect to time, the relative displacement in the joint is calculated. The dissipation and tangential stiffness of the joint are then calculated from the plotted hysteresis curve of the system.

The method has been applied to dynamic simulation of a motor-block-oilpan combination, ❹. The method shows an improved damping modelling in comparison to a standard Rayleigh damping assumption, and allows differentiating between the modes with higher damping caused by deformations in the joint and modes with lower damping which can be attributed mainly due to the dissipation of material.

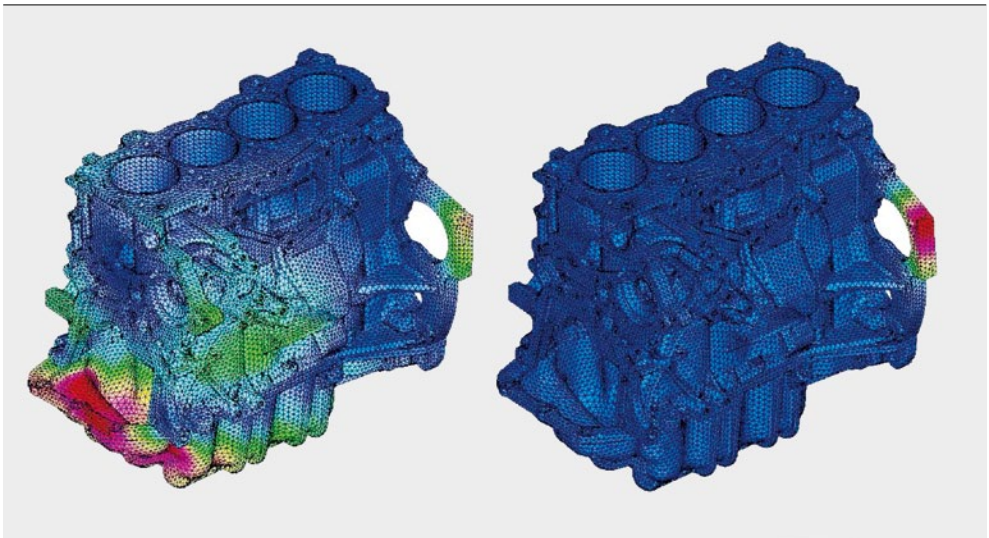
3.2 ZERO-THICKNESS ELEMENTS

Contact interfaces of joints in built-up structures can be modelled by zero-thickness finite elements which incorporate nonlinear tangential and normal contact models. A new model incorporates the



❸ Test set-up (generic experiment) for determination of joint patch parameters

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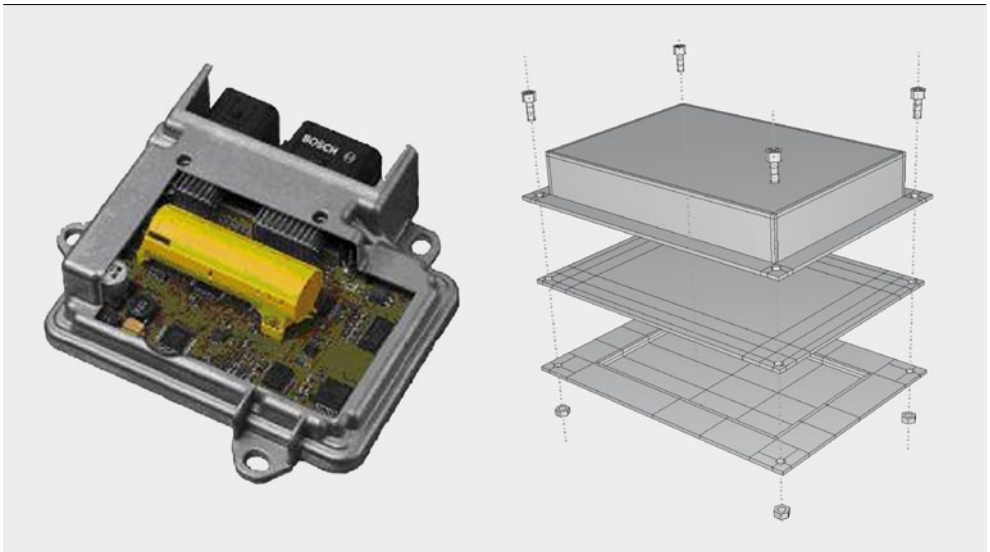
❹ FE model of a motor block with an oil pan attached by thin layer elements; left: mode 1 (modal damping 0.157%); right: mode 3 (modal damping 0.049%)

Hertz and Mindlin contact of a single spherical asperity generalised by a statistical approach for rough surfaces according to Greenwood [9, 10]. The method is applied to an electronic control unit consisting of three aluminium members joined with four bolts, ❺. This approach is able to simulate the behaviour of structures with non-homogeneous contact distribution, even in joints susceptible to uplift. The graphical presentation of contact law used in the simulation is shown in ❻. Nonlinear power law is used for the joint contact modelling in normal direction and Coulomb friction with microslip is implemented to model the tangential contact. The method has a drawback – long calculation times, limiting the application of the approach to smaller finite element models.

4 SUMMARY

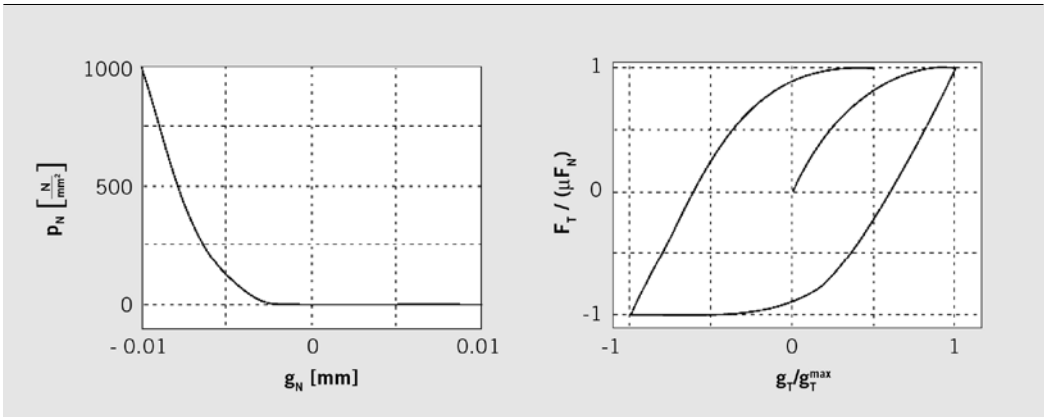
This article gives an overview of various joint types, characteristics and models used for their simulation. Three modelling methods, point-wise contact using Jenkins friction element, thin layer element method with hysteretic damping and linear stiffness and zero thickness element method with nonlinear normal and tangential stiffness are presented. The application of these methods is shown on real life structures. The presented approaches can be used in various cases; however, each of them has advantages in some applications and limitations in others.

Point-wise modelling can be applied for geometrically linear problems and are most suitable for coincident meshes in the substructure interface. This kind of modelling offers the possibility to implement various different types of friction and contact models, which can also be adapted or substituted in the modelling process. In addition, nonlinear effects like slip-stick and separation of each node pair can be modelled. A full 3-D contact analysis for contacting surfaces is possible. However, for a large number of contact pairs the numerical effort to evaluate each element increases drastically, resulting in a decrease of efficiency. Furthermore, numerical stability and data handling for a large number of contact elements often demands additional preprocessing effort.



❺ Electronic control unit with non-homogeneous contact pressure distribution in joints (left) and simplified geometry of an electronic control unit (right), the contact area is represented by zero-thickness elements

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❻ Plot of the normal contact pressure versus normal relative displacement in the joint (left) and hysteresis curve representing normalised tangential contact force versus normalised tangential displacement in the joint (right)

The approach utilising thin layer elements with hysteretic damping can be implemented relatively easy with existing FE software. The advantage of the method is the ability to predict the damping and eigenfrequencies of a structure using experimentally determined joint parameters. The technique can be applied to large structures, such as an automotive powertrain with several million DOFs, and simulation time equals to that of the corresponding complex eigenvalue analysis. The drawback of the approach is that it is a linear method, thus allowing the user to simulate only structures with minimal dissipation in joints and can be used only in frequency domain.

The proposed method with zero thickness elements uses nonlinear normal and tangential stiffness to describe the dynamics of joints. The Hertzian microscopic model is generalised with statistical approach to model the normal contact. A model based on Mindlin's approach is used to account for tangential contact. This technique allows modelling the dynamics of structures in time and in frequency domain under consideration of various contact pressures, excitation amplitudes and frequencies. The limitation in this case is also extremely long calculation times, especially when simulating in the time domain.

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