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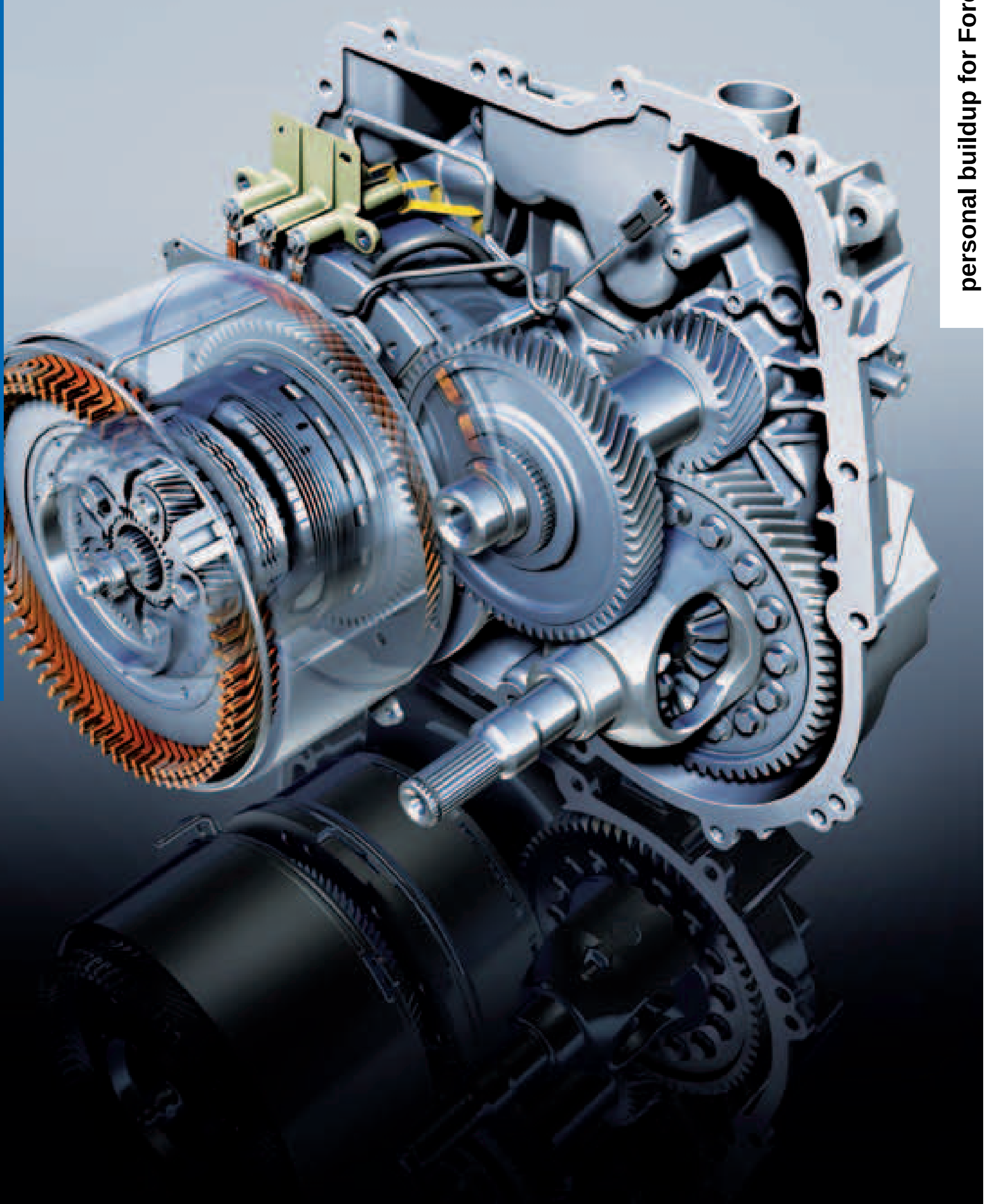
05 May 2011 | Volume 72

DOWNSIZING for Future Gasoline Engines

V6 DIESEL ENGINE from Mercedes-Benz

DIE AND POWDER Forging Materials for Automotive Connecting Rods

/// INTERVIEW
Saita Kanai
Mazda



ENERGY MANAGEMENT IN THE POWERTRAIN

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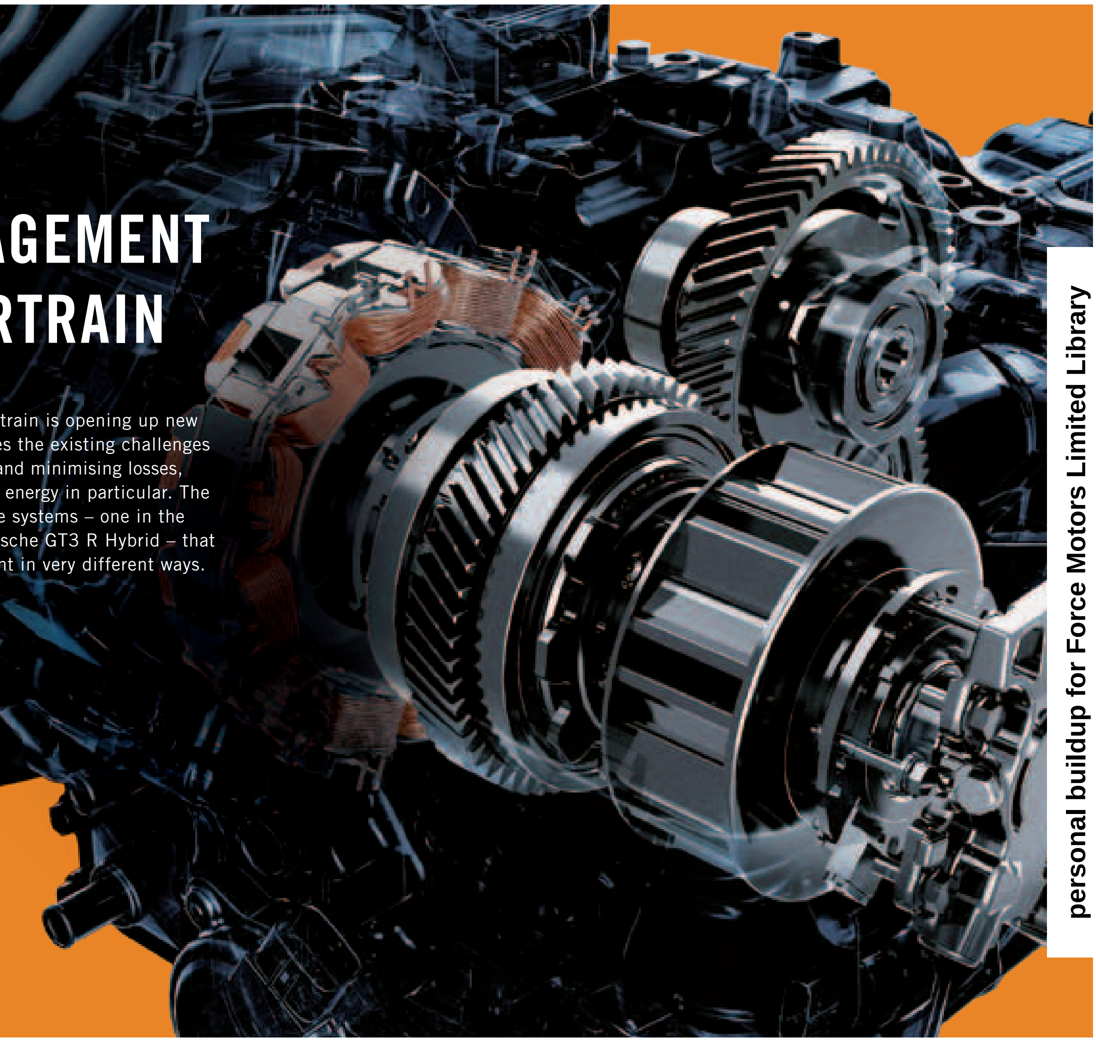

COVER STORY

ENERGY MANAGEMENT IN THE POWERTRAIN

4, 12, 16 I The electrification of the powertrain is opening up new potentials for energy management. Besides the existing challenges of using energy as efficiently as possible and minimising losses, the focus now is also on recovering wasted energy in particular. The cover story of this issue presents two drive systems – one in the Opel Ampera and the other one in the Porsche GT3 R Hybrid – that exploit the potential of energy management in very different ways.

COVER FIGURE © General Motors **FIGURE ABOVE** © Lexus

ENERGY SOURCES

Dear Reader,

Your everyday work involves finding ways to further improve the energy efficiency of the powertrain. As a developer of engines and drive systems, you are faced with the challenge of using the energy available in the vehicle with the minimum of losses. There are still plenty of possibilities at your disposal. Internal combustion engines have not yet achieved their maximum efficiency. Longer gear ratios can reduce engine speed and, in this context, result in lower throttling losses due to higher specific loads. The thermal potential of exhaust energy is largely unexploited. And, not least, electrification opens up new perspectives.

At the same time, against a background of conserving resources and reducing emissions, engineers need to consider in which form energy is to be stored in the vehicle. The search for alternatives to oil leads not only to the question of availability but also to the efficiency chain, which already begins when the fuel is produced or when the electricity is generated. With regard to the well-to-wheel CO_2 balance, both electricity and fuel have great potential. Provided, that is, that only "green" electricity is used in electric vehicles. Or, as an example on the fuel side, cost-efficient generation of BTL fuels from waste wood is implemented. The latter in particular would, however, require significantly higher investment in research.

It is therefore more realistic for oil, gas and coal to remain our main sources of energy for the time being, with oil retaining its role as the principal energy source for vehicles. Which means that, while we are still on the way towards regenerative energy sources, the challenge remains to use it as efficiently as possible. Or, as Karl-Heinz Schult-Bornemann recently put it: energy efficiency will become our most important energy source.

Examples of the further development of energy management in the powertrain can be found in the cover story of this issue.

RUBEN DANISCH, Vice Editor-in-Chief
Wiesbaden, 24 March 2011

VOLTEC – THE PROPULSION SYSTEM FOR CHEVROLET VOLT AND OPEL AMPERA

The Chevrolet Volt and the Opel Ampera are electric vehicles with extended range. They operate as pure electric vehicles as long as there is useful energy in the battery. However, unlike a battery electric vehicle they do not suffer from lost vehicle utility when the battery is depleted. Volt and Ampera can continue operation by using an internal combustion engine.

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MOTIVATION FOR VEHICLE ELECTRIFICATION

The electrification of the automobile is driven by three major goals: (1) further efficiency improvements for vehicles, (2) the desire for local emission-free transportation and (3) the independence from fossil energy sources.

With production of the Chevrolet Volt under way and the Opel Ampera following soon into the European market, General Motors enlarged its portfolio of electrified vehicles with a very important product.

General Motors has significant background in the electrification of the automobile. In 1996, GM introduced the EV1, the first modern production battery-electric vehicle (BEV). The second generation of EV1 was equipped with nickel-metal hydride batteries. The energy density of the battery limited the range of the vehicle to slightly above 100 km in city driving. Some EV1 drivers described the limited range and the fear of being stranded with a depleted battery as “range anxiety”. There are solutions needed to overcome the range anxiety phenomenon, as meeting customer expectation is key to the market success of electrified vehicles.

INCREASED VEHICLE EFFICIENCY

In all the major markets, regulations are put in place to reduce vehicle fuel consumption and thereby the emission of CO_2 from the fleet of vehicles. In addition to these regulations, economic and social consciousness of developed-market consumers towards vehicles with lower energy consumption is also growing.

To achieve these challenging targets, the entire automotive industry is reducing the road load demand of their vehicles and is improving the internal combustion engines and drivetrains by adding new technologies.

The conventional powertrain still has a significant potential for efficiency improvements. However, applying all the possible improvements gets the vehicle to a threshold that is defined by the laws of physics, ❶.

Electrification in the form of hybridization allows overcoming this threshold as we tap into additional areas for system efficiency improvements, like stopping the engine at idle, recuperating energy during braking and optimized shifting of the engine load and speed to areas of the highest powertrain efficiency. Limited electrical driving is also enabled by strong hybrids.

The next step takes us to the displacement of petroleum by adding energy to the vehicle from the electrical grid. This of course requires significantly larger batteries and an electric infrastructure, which can deliver the grid power efficiently and effectively.

EMISSION FREE LOCAL MOBILITY

The reduction of tailpipe emissions has come a long way since the introduction of catalytic converters in the 1970s. The exhaust aftertreatment decreased tailpipe emissions tremendously, and the refinement of these systems has continued. Despite this fact, there is still a strong desire to enable zero-emission driving in certain areas.

The addition of electrification can enable short to midrange pure electric driving for both efficiency and local emission free mobility. Electric vehicles with extended range can be designed in a way that conserves electric operation for specific areas. For example, the ability to hold charge and utilize it in city centers is an important feature of the Opel Ampera, which helps preserve the environment and meets local requirements for zero emissions.

DISPLACE PETROLEUM

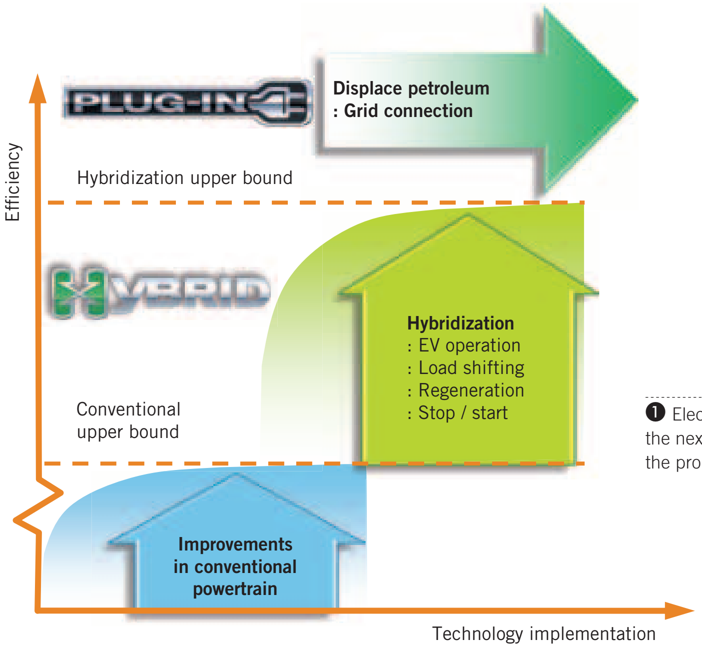
The automotive industry today relies nearly entirely on fossil fuel. 96 % of the energy used to move all vehicles globally stems from fossil fuels. Due to the distribution of crude oil around the globe and because of the fact that oil reserves are limited, there is a strong political desire in several nations to reduce reliance on imported oil.

Electricity can be generated from several energy sources including renewable resources. This diversity of sources and the fact that the distribution grid already exists is a big advantage for electricity as automotive fueling option. Of course, an infrastructure for the vehicle charging beyond home charging needs to be established.

CLASSIFICATION OF ELECTRIFICATION

The electrification of the automobile spans a wide range of technologies. It starts with micro-hybrid systems, which stop the combustion engine automatically when the vehicle comes to a halt. Stronger electric motors, that are connected to the drivetrain during vehicle operation allow electrical assistance to the combustion engine and enable recuperation of energy during vehicle decelerations. These systems, usually referred to as mild hybrid systems, do not enable electric driving capability. ❷ gives an overview on the major attributes that help distinguishing between the different types of electrified vehicles with higher capability.

Hybrid-electric vehicles (HEV) are defined by SAE [1] as vehicles with two or more energy storage systems, both of which must provide propulsion power – either together or independently. An HEV typically uses a larger combustion engine that covers most of the propulsion effort during high power requirements. The electric motor is usually the smaller of the propulsion systems and is dimensioned for the transient component of the vehicle operation and the opportunity to recuperate energy during vehicle braking.



❶ Electrification enables the next level of efficiency of the propulsion system

A plug-in hybrid-electric vehicle (PHEV) is defined as a hybrid vehicle with the ability to store and use off-board electrical energy in the battery. It requires a bigger battery and a charger for connection to the electric grid.

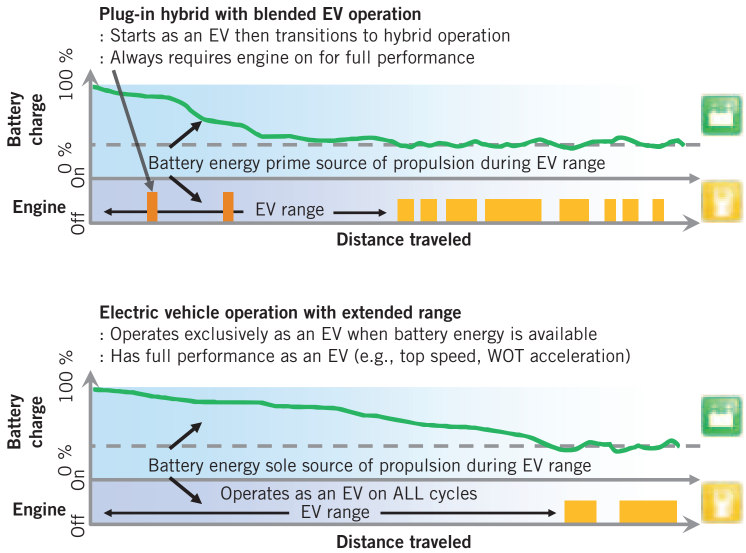
PHEVs are limited in their electric driving capability by the installed power of the electric motor and its thermal capacity as well as by the power capacity of the battery, the power electronics and the kinematics of the hybrid transmission. Plug-in hybrids start with initial electric operation. However, as the sizing of the electric motor is typically not done for the entire range of vehicle power requirements, a plug-in hybrid requires the combustion engine to come on, upper part of ❸, when the power demand is bigger than what the electric motor can deliver. The power of the electric motor and the combustion engine are blended to meet the vehicle's performance demand. The start of the combustion engine is dependent on the power request of the driver and therefore cannot be predetermined or prescheduled. The emission control system of such an engine needs to be prepared for this type of operation to manage the light-off behavior of the aftertreatment system.

An electric vehicle with extended range (E-REV) provides full electrical drive capability until the charge of the battery is depleted, comparable to a battery electric vehicle. An E-REV has full performance in this operation, including wide-open throttle acceleration and top speed. The cold-start emissions can be well-managed as the combustion engine has scheduled starts.

A battery-electric vehicle (BEV) is solely powered by electrochemical energy stored in the rechargeable battery pack. The electricity stored in the battery is used to power an electric motor that drives the wheels and as such, like the E-REV, the BEV must have sufficient power capacity

VEHICLE TYPE	ELECTRIC POWER	ONBOARD ELECTRIC STORAGE	GRID RECHARGE	ELECTRIC DRIVING CAPABILITY	RANGE LIMIT
HEV	Low to med	Low	No	Very limited	Gasoline
PHEV	Med	Med	Yes	Urban only	Gasoline + battery
E-REV	High	High	Yes	Full	Gasoline + battery
BEV	High	Highest	Yes	Full	Battery

❷ Attributes of electric vehicles



❸ Operation of a PHEV (top) and an E-REV (bottom) during charge depletion mode

in the driving motor, inverter, interconnects, battery, and thermal systems. The driving distance of the BEV is limited by the energy that is available in the battery, which with today's technology provides much less range than a conventional engine. Fuel cell-electric vehicles using hydrogen, are yet another solution for electric vehicles with a significant larger range.

VEHICLE TYPE	5-door, front-wheel-drive hatchback
CATEGORY	Electric vehicle with extended range capability
CHASSIS	Independent McPherson struts front Compound crank torsion-beam axle rear Four-wheel disc brakes Full regenerative brakes Electric power-assist steering
SEATING CAPACITY	4
CURB WEIGHT [KG]	1715 kg
OVERALL LENGTH [MM]	4498
WHEELBASE [MM]	2685
WIDTH [MM]	1798
HEIGHT [MM]	1430
CARGO VOLUME [L]	301
BATTERY	T-shaped module with integrated thermal management Lithium-ion manganese-spinel (LiMn204)
TOTAL RATED ENERGY [KWH]	16
TOTAL USABLE ENERGY [%]	65
RECHARGE TIME	< 4 h @ 230 V / 16 A
FUEL TANK [L]	35.2
POWER [KW / HP]	111 / 150
TORQUE [NM]	370
TOP SPEED [KM/H]	161
ACCELERATION (0 TO 100 KM/H) [S]	9
EV RANGE [KM]	40 – 80 (MVEG cycle)
TOTAL RANGE [KM]	> 500

❹ Vehicle specifications

PHYSICS OF ELECTRIFICATION

A very important aspect of automotive fuels is energy density. When the mass and volume of the tank or the battery is taken into account, the energy density of liquid fuels is about one hundred times higher than that of batteries storing electricity. Even when considering the vehicle efficiency of a battery-electric vehicle being up to four-times higher and the storage density of batteries improving significantly, we still have to assume a factor of 15 to 20 between the batteries and the conventional liquid fuels for the "tank" range. This results in limited range for a battery-electric vehicle.

The time needed to recharge the battery is also a limitation for plug-in electric vehicles. The charging rate is determined by two facts: (1) the available electrical power, and (2) the impact on the durability of the battery cells. Quick-charging is possible, but limits the life of the battery.

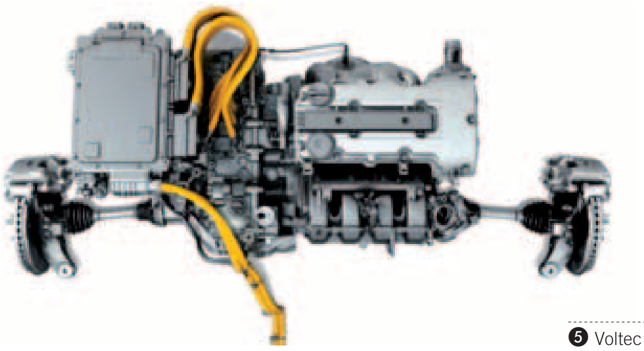
Lithium-ion batteries have become the standard for electric vehicles that store energy from the grid because of their superior energy density. However, lithium-ion battery cells require operation in a defined temperature range. Low temperatures cause the chemical reactions in the electrolyte of the battery to slow down, with the power limit of the battery decreasing significantly below 0 °C. Meanwhile, high temperatures increase the chemical degradation of the battery and therefore reduce battery life. In order to keep the battery temperature in the optimal range, thermal management systems are used.

In addition to the temperature operating range of the cells, ambient temperature also influences the energy requirement of the vehicle for heating and air-conditioning of the passenger compartment. The energy requirement for the comfort of the passenger and for other vehicle systems such as lighting and windshield wipers can be significant.

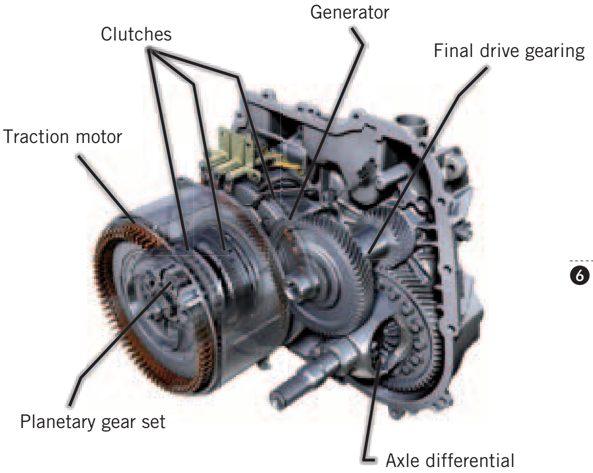
The cost of lithium-ion battery cells is expected to come down significantly; however, this will be an evolutionary process and will also be limited by the price of the raw materials used in the components.

THE VEHICLE

The Chevrolet Volt and the Opel Ampera, ❹, are five-door hatchback vehicles that provide space for four passengers. They



❺ Voltec propulsion system



❻ Voltec electric drive – cut-away

are powered by the Voltec system that drives the front wheels.

The car is a full-performance electric vehicle with extended range (E-REV). It operates as an electric vehicle as long as the battery is above the minimum charge level and the engine is not needed for heating the passenger compartment. The electric range of the vehicle depends on the terrain, driving techniques and the ambient temperature. When the battery reaches a minimum state-of-charge, the internal combustion engine starts and provides the system with electricity. This increases driving range while still providing the same vehicle performance. Volt and Ampera have a “Drive Mode” button that allows the driver to select between a Normal, Sport and Mountain Mode. In Sport Mode, the vehicle changes the settings to a quicker accelerator throttle response. Mountain Mode automatically adjusts the system to provide needed power in mountainous environments, when vehicle performance could otherwise be compromised. Mountain Mode changes default settings to sustain a sufficient state of charge so in the event supplemental power is needed, it can be available from the battery. The battery can be recharged using a household power outlet.

VOLTEC MULTI-MODE ELECTRIC TRANSAXLE

The Voltec propulsion pystem, ❺, [2] utilizes multiple operating modes that are more efficient than fixed-ratio transaxles. This allows for increased driving range in electric-drive mode and lower fuel consumption in the extended range operation.

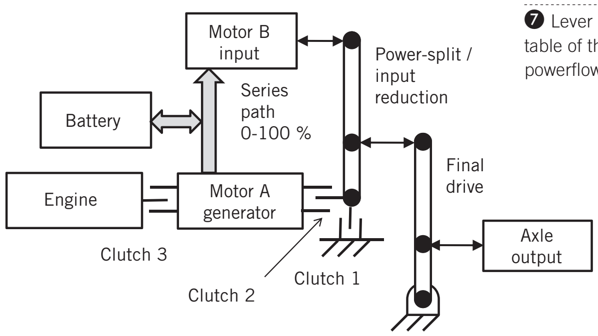
The powerflow of the Voltec electric drive, ❻, comprises two electric motors, one planetary gear, two rotating clutches and one stationary clutch. The electric traction motor with 111 kW maximum power and 370 Nm of maximum torque is arranged on one common axis with the 54 kW generator. Both electric motors are permanent-magnet type. A geared final drive connects the planetary gear set to the front wheels of the vehicle.

The clutches allow for two modes of electric vehicle operation and two additional engine-on modes of operation when the vehicle is in range extended operation.

❼ shows a lever diagram of the Voltec powerflow. The lever diagram represents the relationship between torque levels and rotation-speed levels with force and displacement of the lever elements in the free-body diagram. ❽ shows the architecture with the planetary gear and the coupling of the elements of the powertrain system.

The innovative Voltec drive unit has four distinct operating modes. Depending on the clutch setting, either one electric motor or both are used to drive the vehicle. The engine can be coupled with the generator as well.

In Electric Mode 1 only the 111 kW primary propulsion motor is used to propel the vehicle. The motor is supplied with electrical



		Clutch 1	Clutch 2	Clutch 3
Electric Mode 1	Single-motor EV drive	Closed	Open	Open
Electric Mode 2	Two-motor EV drive	Open	Closed	Open
Extended Range Mode 1	Series drive	Closed	Open	Closed
Combined Drive	Output power-split drive	Open	Closed	Closed

❼ Lever diagram and clutch state table of the Voltec electric transaxle powerflow energy from the battery in the vehicle and operates on the sun gear of the planetary gear set. The latter functions as a reduction gear as the ring gear is held by a brake (C1).

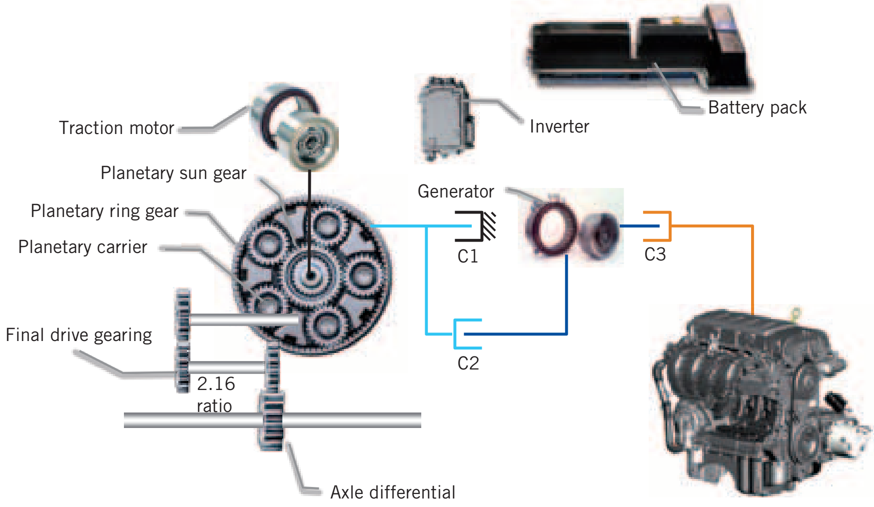
As a range extended vehicle is by definition equipped with a second electric machine, this second motor can be used to improve the efficiency of electric operation. The second motor is decoupled from the internal combustion engine, which is not running. In Electric Mode 2, only clutch C2 is applied. The two electric motors operate on the planetary gear where the second motor is now counteracting the torque on the ring gear. The torque and the speed of the two electric motors is determined by the linear relationship defined by the planetary gearset.

With the two-motor EV arrangement, the motor speeds of the two electric machines can be adjusted continuously to optimize for the greatest overall efficiency, greatest tractive effort, and lowest noise and vibration (NVH) level without any torque interrupt. The increase efficiency improves mileage in the U.S. highway cycle by one to two miles of additional EV range.

When the battery reaches a minimum state-of-charge, the internal combustion engine turns on and operates with the generator in series mode. In Extended Range Mode 1, clutches 1 and 3 are closed. The internal combustion engine is driving the generator that provides electricity to the drive motor. In this series-drive arrangement there is no mechanical connection from the engine to the wheel and only the electric traction motor drives the vehicle.

The engine speed is varied with the vehicle speed to allow for a connected natural feel and sound response from the engine. NVH and engine operating speed control were greatly refined as part of vehicle development and specific speed profiles to maintain natural haptics are utilized. Since the engine is fully decoupled, it runs largely unthrottled to achieve good efficiency.

The combustion engine delivers the average cycle power, but the dynamics of the vehicle remain largely electric. This is enabled by an energy reserve in the traction battery, which is employed to maintain the EV feel, even in extended range operation. High throttle launches start out fully electric, with combustion engine operation starting at 30 km/h and following an NVH profile to deliver EV-like performance within the battery energy reserve window.



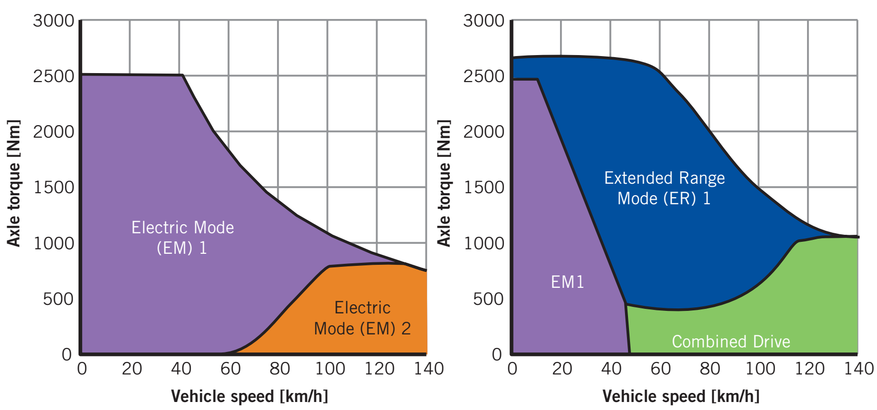
❽ Voltec – kinematic architecture

While the series arrangement is efficient for lower-speed operation, it clearly has efficiency disadvantages at higher vehicle speeds due to (1) the efficiency loss of the electric motor operated at high speed and (2) to the dual energy conversion of the series path from mechanical energy to electrical energy and back to mechanical energy at the axle.

A significant improvement in efficiency can be achieved using an output-split powerflow. In Combined Drive Mode, clutches 2 and 3 are closed. This powerflow also has no direct connection (fixed gear ratio) of the engine to the wheel. The generator in combination with the combustion engine and the electric traction drive motor act on the planetary gear and counteracts the torque that is transmitted to the final drive.

During highway driving, efficiency can be improved by 10 to 15 % over the series power-flow arrangement. Under this operation, mechanical power from the engine is partially converted by the generator to supply electric power to the traction motor. This configuration still enables full speed and load selection for efficiency, and the motors operate to control the electric power conversion and application of power to the planetary. Both electric motors must operate and only when the traction motor reacts torque on the planetary is torque transmitted to the wheels.

❾ shows the operation areas for the different modes in the tractive effort map. The system reconfigures the powerflows during driving to optimize efficiency, as well as power and NVH requirements, and this is fully transparent to the driver.



❾ Operating space of the four modes in the tractive effort map

THE ENGINE

The engine in a fully capable electric vehicle with extended range needs to provide the average sustained electric power to drive the car in the extended range operation. The targeted maximum vehicle speed and the sustained gradeability determine the size of the engine. During the architectural phase of vehicle development, several engine options were investigated. The engine needs to be light, small, efficient, and provide excellent noise and vibration behavior (NVH). While Diesel engines have high efficiency, balancing NO_x emissions, NVH, and efficiency drive significant effort and mass into the system. Boosted engines increase the package requirements due to the need for a charge intercooler system. Therefore, a naturally aspirated spark-ignition engine was selected. The chosen 1.4 l four-cylinder engine is a member of GM family 0, generation 3 [3], ❿ and ⓫.

Limiting maximum engine speed improved the noise and vibration behavior of the engine; however, it also drove a slightly larger displacement.

The engine operates in stoichiometric operation with a three-way catalytic converter. The engine-start emissions are optimized by using a start procedure that balances engine-out emissions and throughput to achieve a very fast light-off of the catalyst. This is possible as the vehicle is still driving in electric mode when the engine start occurs. Plug-in hybrid vehicles that need to use the torque from the engine to achieve required vehicle acceleration cannot utilize such a strategy.

The engine accessory drive was also simplified. The starter and alternator were deleted and a simple Poly V belt drives the water pump. Oil filter housing, throttle body and oil pan were modified to fit the package requirements. The operating strategy for the engine is shown in ⓬. For maximum efficiency, the efficiency maps of both the engine and generator need to be considered to achieve the minimum fuel consumption.

❿ 1.4 l four-cylinder gasoline engine

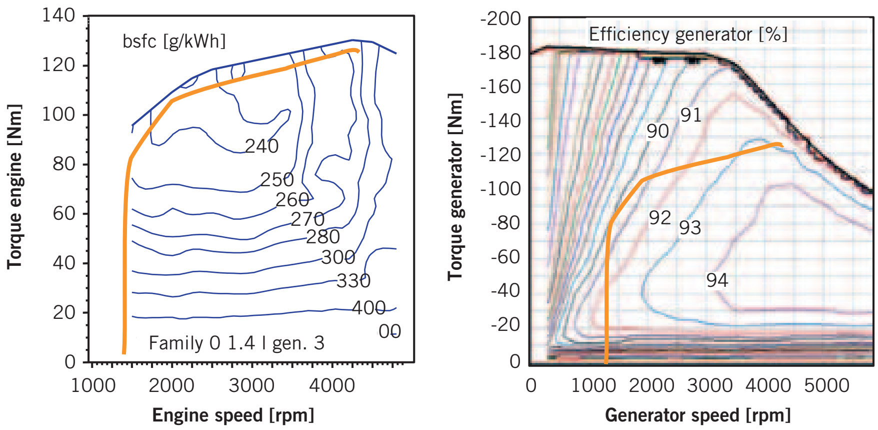
LITHIUM-ION BATTERY PACK

The battery for an electric vehicle needs to have high energy capacity, safety, performance, durability, reliability and high package density. The electrical energy storage used in the Chevrolet Volt and Opel Ampera is a lithium-ion battery containing 288 prismatic cells. LG Chem manufactures the cells. They use a carbon-graphite anode and manganese spinel cathode, separated by a ceramic-coated safety-reinforced separator (SRS). The cells are encased in polymer-coated aluminum housings (pouch cells) and stacked vertically in modules. Linked modules are combined to the entire battery pack. The battery is assembled at GM's own battery assembly facility in Brownstown Township, near Detroit, the first such facility in the U.S. built by a major automaker.

The total rated energy is 16 kWh; of which 65 % are useable when the battery is cycled between its state-of-charge limits. The battery's top and bottom "buffer zones" help to ensure long life. It operates at a nominal voltage of 360 V. The battery pack uses a liquid based thermal management to ensure long-term durability and reliability. The circulating liquid passes through a series of internal heat exchangers in the battery module. The battery can thereby be cooled or heated. It is designed to provide reliable operation, when plugged in, at ambient temperatures as

ENGINE TYPE	In-line four-cylinder gasoline
VALVES / CYLINDER	4
DISPLACEMENT [CM³]	1398
BORE / STROKE [MM]	73.4 / 82.6
STROKE / BORE RATIO	1.12
BORE DISTANCE [MM]	78
CONNROD LENGTH [MM]	130.3
COMPRESSION RATIO	10.5
POWER [KW @ RPM]	63 / 4800
MAX. TORQUE [NM @ RPM]	130 / 4250
FUEL SYSTEM	MPFI
ENGINE MANAGEMENT SYSTEM	GM

⓫ Engine specifications



⓬ Engine operation during charge sustaining

low as -25 °C and as high as 50 °C. The battery power management monitors the battery in real-time for optimum operation. The thermal and power control were developed within General Motors.

The complete module, ⓭, weighs 180 kg and is designed in a T-shape to fit under the middle tunnel and back seat of the vehicle. It is 1.7 m long and serves as a semi-structural element of the vehicle structure, increasing the bending stiffness of the vehicle body.

SUMMARY

Increasingly stringent requirements for lower CO_2 emissions and the demand for a diver-sification of energy carriers are driving the development of future vehicles. The electrification of the powertrain systems will play a major role in meeting these requirements.

Battery-electric vehicles are a part of the solution. However, limited energy density, the time required to recharge the battery, and the cost of the batteries make BEVs a solution primarily for urban applications.

The internal combustion engine is an enabler to extend the operating range of an electric vehicle. The Chevrolet Volt and the Opel Ampera operate as a pure electric vehicle as long as there is useful energy in the battery. A 1.4 l gasoline engine extends the range. The benefit of an electric vehicle without range limitations is a clear customer benefit.

Chevrolet Volt and Opel Ampera are full capable electric vehicles delivering zero emissions for normal daily driving. They are also able to make longer trips on weekends, holidays and in emergencies. Volt and Ampera have the ability to be only vehicle in a household to fulfill all personal transportation needs.

The Voltec powertrain represents what is possible in delivering a practical electric vehicle. General Motors is committed to an expanded portfolio of electrified solutions for our vehicles including Stop Start, hybrids, PHEV, EREV, BEV, and fuel cell-electric vehicles. Each technology offers a viable technical solution, which will coexist over the coming decades to meet customer needs for exciting and affordable personal transportation.

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⓭ Battery pack of the Chevrolet Volt and Opel Ampera

THANKS

The authors thank the entire Chevrolet Volt / Opel Ampera team for their outstanding performance, which allowed the delivery of these products on time to the market. We also thank all our suppliers and business partners for their exceptional support.

"LOWER CO_2 EMISSION EFFICIENTLY"

In the context of the Geneva Motor Show, MTZ spoke with Saita Kanai, Mazda's Senior Managing Executive Officer, in charge of R&D and Program Management, about the company's new Skyactiv technologies and automotive future perspectives. To lower CO_2 emission efficiently is a crucial point for him.

Saita Kanai is Director; Senior Managing Executive Officer, in charge of R&D and Program Management; President, Mazda Engineering & Technology Co., Ltd. After graduating in Mechanical Engineering at Tokyo Institute of Technology he joined Mazda in 1974. Within the Mazda Motor Corporation, Kanai always worked in the sector of vehicle engineering. Since 2005, Saita Kanai is in charge of R&D. Thus, he was closely involved in the development of Mazda's latest Skyactiv technologies which encompass a diesel and gasoline powertrain, transmissions, body and chassis. All those components are designed for lower emissions as well as improved performance and fuel consumption.

MTZ _ The new Skyactiv-G 2.0 l petrol engine is said to consume 15 % less fuel than its predecessor and to have 15 % more torque. That sounds good, but it is not revolutionary in times of downsizing and engines with 1.2 to 1.6 l. What vehicle classes are you targeting with this engine?

KANAI _ The concept of downsized engines is popular today, we know that. However, we have 2.0 l engines or engines over 2.0 l. We don't cover 1.2 or 1.6 l turbocharged engines. But nevertheless, we will be able to achieve the same results compared to downsized 1.2 and 1.6 l engines. With a high compression ratio and reductions in friction, we believe that our engine's efficiency is equal to, or even better than, that of downsized engines.

Will we see smaller engines like 0.8 l engines or three-cylinder engines from Mazda in the near future?

We are not considering that at the moment.

Friction reduction is one of the issues. Possibilities for reducing friction include high-tech coatings and bearings. What solution did you choose?

Regarding friction reduction, we are pursuing various approaches. For example, we use high-tech coatings and bearings. The smaller the diameter is, the less friction is caused. Therefore, we reduced the size to avoid unnecessary space.

Mazda expects the internal combustion engine to remain at the heart of the car for the near future. Is the development of the Mazda 2 electric vehicle for the Japanese market the starting signal for a global electrification strategy?

With new restrictions in the US market starting from 2018, we have to provide electric vehicles and we need to prepare for that kind of regulation. Until then, we need to learn and we will benefit from the experience we are gaining with our current technology and by further developing this technology over the next few years.

„We will benefit from the experience we are gaining with our current technology."

Mazda's Skyactiv-D diesel engine is said to consume 20 % less fuel than the current 2.2 l diesel engine. That is around 4 l in a Mazda 6. Electric vehicles in Japan, diesels for Europe – when will we see a combination of both?

To combine the diesel engine with an electric motor to create a diesel hybrid system is a little difficult. For the diesel engine, we use turbocharging and the engines operate at a high pressure. For diesel hybrids, the development costs are quite high. Therefore, it is highly unlikely that the diesel hybrid will establish itself as a stand-alone concept in the market.

Why do you think PSA Citroën is developing a diesel hybrid engine?

They want to test the price in top-end technology. But at the end of the day, it depends on how widespread it is in the market, how popular it is in the market.

In terms of CO_2, where do you see the ideal figure for an internal combustion engine for a midsize car with a weight of up to 1500 kg in 2020?

I wouldn't like to identify a concrete figure for CO_2 emissions in 2020. For one of our new midsize vehicles combined with the new Skyaktiv-D engine, we have a plan to achieve a CO_2 emission of only 105 g/km in the near future. Thus, with the Skyactiv-D, we will make a big leap forward with regard to reducing CO_2 emissions. At the moment, we are looking for the best idea on how to repeat this success. We are aiming at a further improvement of more than 15 to 20 % CO_2 reduction by 2020.

Your current diesel engine is a turbo with variable turbine geometry (VTG). Do you expect the current trend towards two-stage or multiple-stage turbocharging to continue?

Generally speaking, for our new Skyaktiv-D engine we will use two-stage turbocharging, and we think that this will cover most applications for normal vehicles and for the ordinary customer. A further step in turbocharging could be interesting, but is probably useful only for motor racing.

Electric mobility needs renewable energies as a power source. An alternative source of energy is the fuel cell. What is the status of this technology in your company?

Our development work on the fuel cell has been suspended for the time being, because it seems highly unlikely that the fuel cell will become very popular on the market in the near future.

What do you mean by "in the near future"?

In the next ten or twenty years. Even if fuel cell vehicles might be introduced by others, the market is going to reject them, as the cost and infrastructure are always critical factors.

Mazda and rotary engines – a story that lasted a long time and seems to be over today. Are you currently working on further improvements to the legendary rotary engine or is this topic completely out of consideration?

We are still working on that and developing it, but it is too early to talk about market introduction timing. However, we are working on that kind of technology right now and we shouldn't let others forget about rotary engines. We will introduce it as soon as possible, or at least we will soon be able to talk about the technologies we are working on.

"The market is going to reject the fuel cell vehicle."

Can you imagine rotary engines being used as range extenders?

It could be one of the possibilities, but that is not the entire usage of a rotary engine. We are developing and doing research to enhance the efficiency of the rotary engine up to the level of Skyactiv-G. Probably, we will be able to raise the capability of the rotary engine to speeds of up to more than 8000 rpm, since there is a possibility that the displacement of the next-generation rotary engine increases. I think it surely has the potential to be used as an engine for sports cars. Fuel economy will be equivalent to that of our Skyactiv-G, so it will be a very nice car to drive.

Ford has sold a majority of its 11 % share to Mazda. Does this have any consequences regarding the engines or platforms that Mazda received from Ford in the past?

This has no effect at all. The cooperation was beneficial for both parties and we will

Kanai does not believe in the diesel hybrid as a stand-alone concept

continue to cooperate in several areas in the future. For example, we are jointly building a pick-up truck with Ford in Thailand right now. We announced our agreement on the model change programme of this product last year. Additionally, we are working together with other OEMs such as Suzuki and Toyota in different areas. For example, we are tying up with Toyota regarding hybrid technology.

In Europe, we are now starting to talk about what will come after 2020 and about the new emission regulations. What is the current state of the discussion in Japan?
The Japanese government is pushing electric vehicles by a kind of incentive.
By contributing just under 10,000 euros for each electric vehicle, the government is currently trying to increase the number of electric vehicle buyers. Of course, this strategy cannot be a long-term concept because the Japanese government cannot

"The most important question should be how we can lower CO_2 emission efficiently."

afford this even if the amount of the incentive is reduced. What we really need are long-term solutions, so an alternative must be found. In general, the most important question should be how we can lower CO_2 emission efficiently.

Mr. Kanai, thank you very much for this interview.

INTERVIEW: Roland Schedel and Johannes Winterhagen
PHOTOS: Mazda

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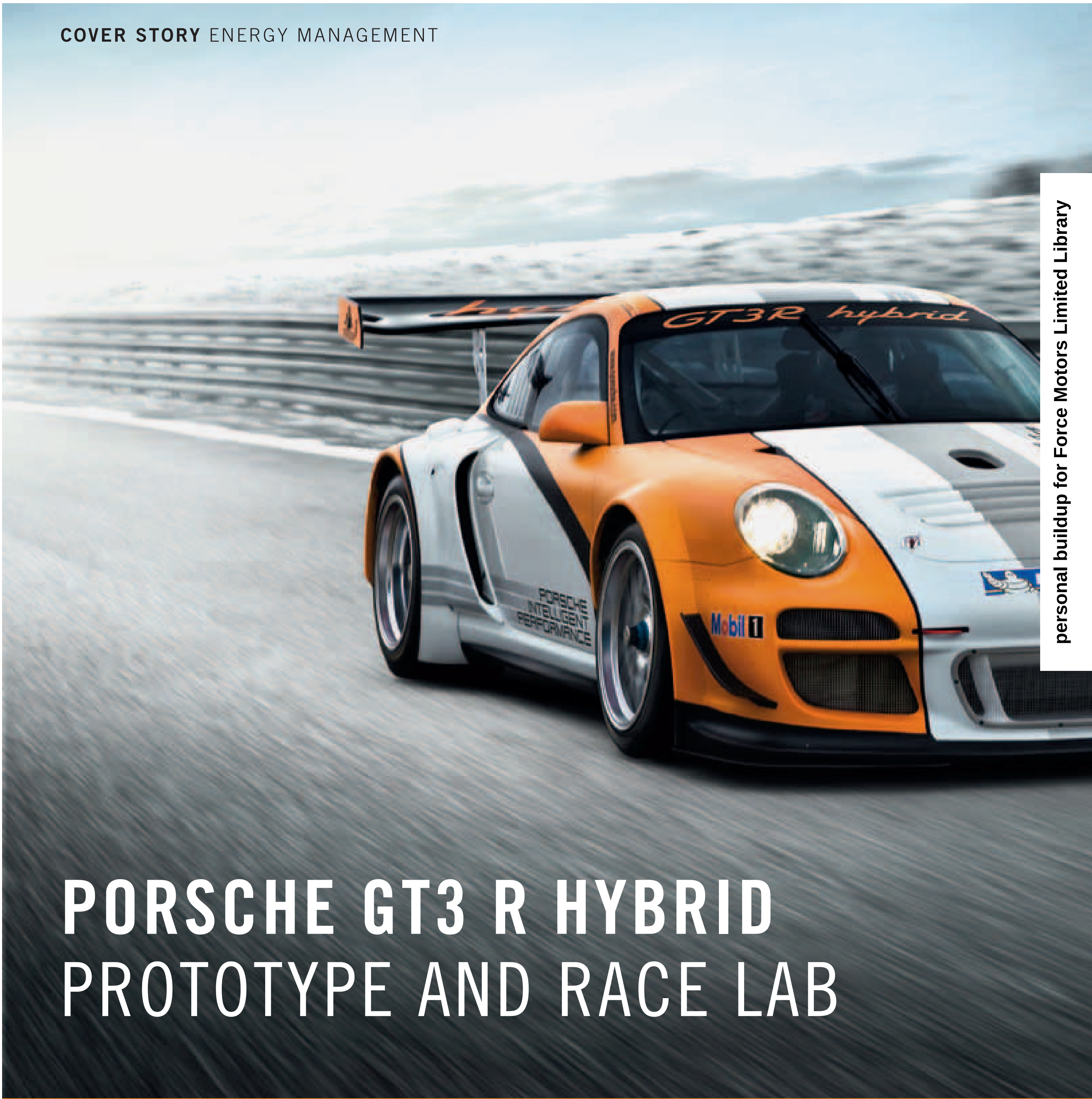

PORSCHE GT3 R HYBRID
PROTOTYPE AND RACE LAB

Hybrid technology is one of the key technologies for coping with changing future requirements in the area of individual mobility. This technology is already being used in small, mid-size and upper class vehicles and is perceived by the public as environmentally friendly. To date, however, no hybrid drive concepts suitable for mass production have been available for sports cars, such as the Porsche 911. In order to answer questions about the optimum design of hybrid drive systems for high-performance vehicles and about the use of future technologies in motor racing, Porsche Motorsport therefore developed the GT3 R Hybrid.

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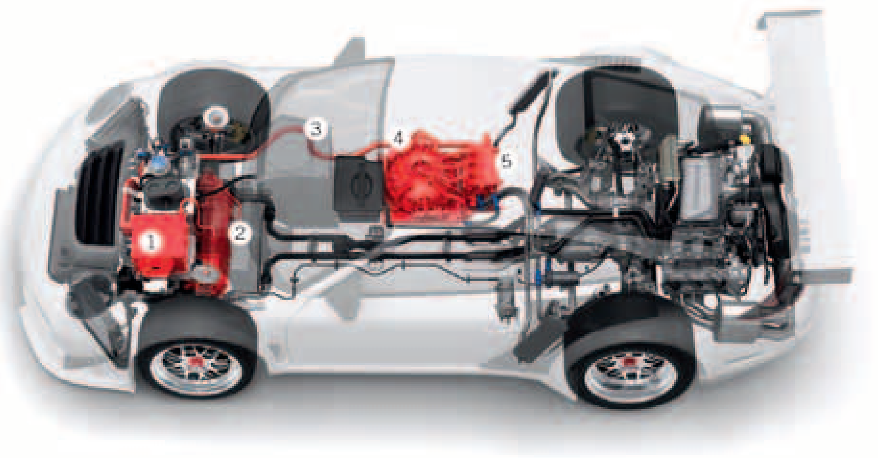
OBJECTIVE

The Porsche GT3 R Hybrid is a thoroughbred race car that acts for the engineers as a laboratory on wheels and is therefore also known as the "Race Lab".

Porsche has a long tradition of developing new technologies in motorsport and then transferring these to mainstream car production. Development work in motorsport is also highly efficient and provides the following advantages:

: rapid rates of development, as the team is small and can work within an efficient project structure
: results-focussed specification management within a tough competitive environment
: tight adherence to deadlines due to fixed race schedules
: fast and permanent troubleshooting.



1. Power electronics
2. Portal shaft with two electric motors
3. High-voltage cable
4. Electrical flywheel battery
5. Power electronics

❶ 911 GT3 R Hybrid, system overview

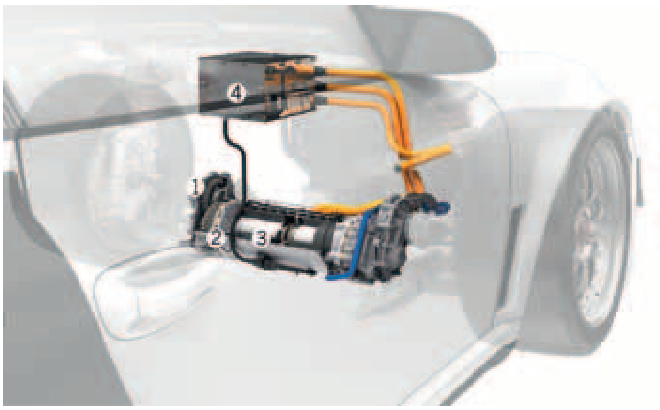
In developing the 911 GT3 R Hybrid the team pursued the following objectives:

: to develop and intensify know-how in important future technologies, such as energy storage and electric drive systems
: to design systems and components for racing use, while at the same time identifying potential areas for a transfer of the technology to road car production
: to develop an energy-efficient and simultaneously performance-oriented operating strategy
: to validate the hybrid system through using it in a 24-hour race
: to underline Porsche's technological expertise in the sports car market.

VEHICLE CONCEPT

The 911 GT3 R Hybrid is based largely on the 2010 model of the 911 GT3 R and has been appropriately adapted for use of the hybrid drive system, ❶. The conventional drivetrain in the back of the vehicle consists of a four-litre, six-cylinder flat engine and delivers 353 kW. Power is transferred via a sequential six-speed transmission from the 911 GT3 RSR. The electric drive unit is located on the front axle and is made up of two motors each delivering 60 kW. Drive torque is transferred to the front axle's drive shafts via a fixed gear reduction unit. Serving as the storage system is an electric flywheel, which is housed on the passenger side near to the centre of gravity. Storage system and electric traction motors need one power electronics system each. These are linked to each other via high-voltage cabling, the DC intermediate circuit. The hybrid system implemented here is a parallel hybrid with added traction. In the 911 GT3 R Hybrid this system provides the following benefits:

: improved weight distribution and thus optimised vehicle balance
: temporary four-wheel-drive functionality
: less mechanical complexity thanks to decoupling of the drivetrains
: less complex application.



1. Transmission support
2. Multiple-plate clutch
3. Electric motor
4. Power electronics

❷ Electric drivetrain

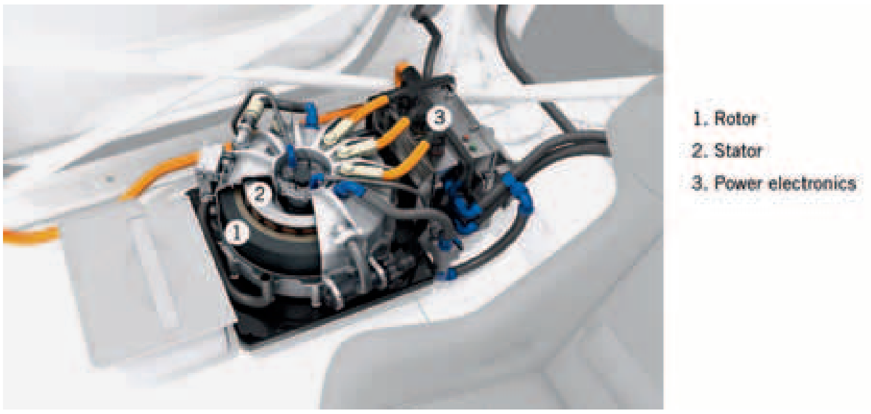
ELECTRIC DRIVETRAIN

The electrically driven front axle essentially integrates within one housing unit two electric motors, each connected to the front axle wheel via a multi-plate clutch, a fixed gear reduction unit and a short drive shaft, ❷. Each electric motor has a position sensor that determines the rotor's precise position for optimum operation. Based on rotor position, rpm speed and the demand for torque from the driver, a power electronics system regulates the electric motors' current flow. The motors and the power electronics have a separate low-temperature water-cooling circuit with a cooler positioned in the middle of the car at the front. The cooling ducts have been integrated directly into the housing.

The electric drivetrain is a reversed portal axle arrangement, which is attributable to the axis of the motor output shaft being offset from that of the wheel drive shaft and to the requirement for a low centre of gravity.

The two permanent magnet synchronous motors arranged side-by-side are not mechanically linked and can thus individually each drive one front-axle wheel. No differential is therefore needed. The mechanical power output per motor is 60 kW at a maximum of 15,000 rpm and 80 Nm of torque on the electric motor's output shaft. Thanks to the gear reduction there is thus additional drive torque available at the wheel of up to 650 Nm. During braking the motors work in generator mode. The energy recovered in this way is transferred via the DC intermediate circuit to the flywheel storage unit.

In the event of a malfunction, such as interwinding in the permanent magnet motor, high fluctuations in torque occur,



❸ Electrical flywheel energy storage

which, when the vehicle is being pushed to the limit, can lead to unstable driving conditions. For safety reasons, a hydraulically activated multi-plate clutch has therefore been integrated on the planetary gearbox's external annulus gear. This cuts the transmission of power between motor and wheel. This function can be triggered both by the driver via a control in the cockpit and by automatic monitoring circuits (insulation monitor, main circuit breaker, diagnosis algorithm in the hybrid control system).

ELECTRICAL FLYWHEEL ENERGY STORAGE

The electrical flywheel storage unit stores the electrical energy recovered during braking in kinetic form, as kinetic energy. The storage unit consists of safety housing made of aluminium, a stator and a rotor made of a synthetic material with infiltrated magnetic particles. A power electronics system controls the energy flow.

The storage unit is in principle an electric motor, ❸, that works with an external rotor as a flywheel. The magnetic particles in the rotor are aligned in such a way that, as with conventional permanent magnet electric motors, they form individual magnetic poles. The rotor is reinforced with wound carbon fibre so that at maximum rotary speeds of up to 40,000 rpm and the associated centrifugal forces it is able to hold firm.

Whenever the flywheel is being charged, the stator coils get fed with the electrical energy generated by the energy recovery process and create rotary motion through the interaction of the magnetic field building up inside the stator with the permanent magnetic field of the rotor. The electrical energy thus gets converted into kinetic energy. In electric traction mode this effect is reversed and as energy is discharged the rotor is slowed. Both the flywheel storage unit and the associated power electronics system have a separate oil-cooling circuit.

In contrast to traditional electric motors, there is no metal in the flywheel's rotor. This increases safety, as no ferritic fragments can break through the housing in the event of a malfunction. It also means that no iron loss can occur in the rotor, as a result of which the extent to which it heats up can be reduced and its level of efficiency significantly increased.

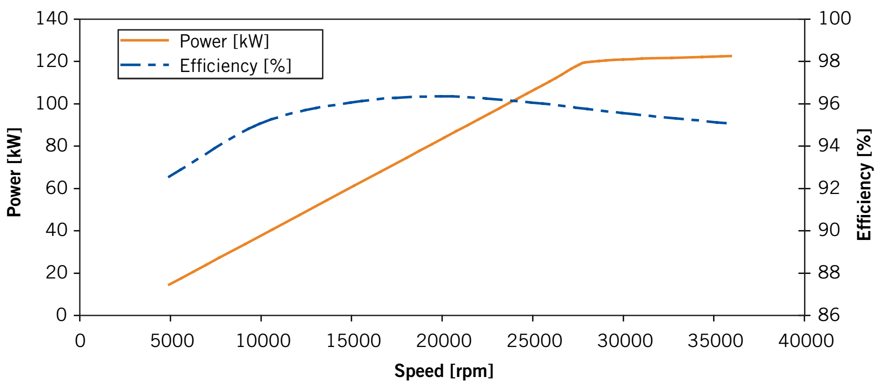
As the rotor's rpm speed goes up, the flywheel becomes more effective, ❹. So that the hybrid system can be permanently run at a high performance level during the race, the flywheel is never discharged to a standstill. Instead, it gets cycled at a constantly high output level of between 28,000 and 36,000 rpm. Because of the constantly high surface speed and resulting air friction, the rotor would heat up at atmospheric pressure to over its permitted maximum temperature. The housing therefore gets evacuated, which additionally also increases the level of efficiency.

The only parts of the flywheel system susceptible to wear are the rotor's ceramic bearings in the housing. These are the only parts that ever have to be overhauled. Depending on the load placed on the system, they should be replaced after a defined period to make the flywheel again as good as new. Other than this, no other signs of aging have yet been seen. Throughout the entire testing and race operations in 2010 there were no mechanical defects at all. Compared to other storage technologies the electrical flywheel has the following significant advantages:

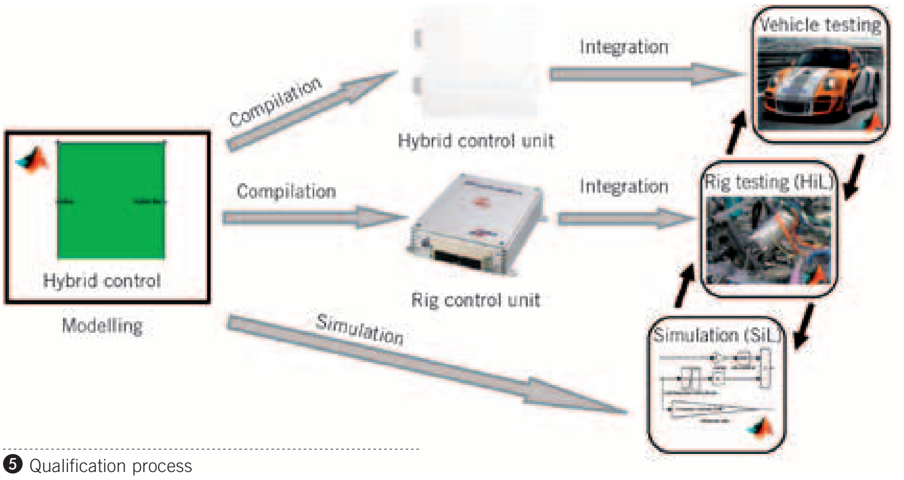
- high efficiency of > 94 % per cycle (charging or discharging)
- high cycle life
- simple and robust mechanical structure and thus less complexity
- reusable materials, making it recyclable
- high level of specific power output
- high duty cycle.

HYBRID CONTROL SYSTEM

The hybrid vehicle functions are centrally regulated from a control system, the hybrid manager. Its task is diagnosis, adjustment of power output and torque and control of the electric motors, the energy store and other electric and hydraulic components. It is also tasked with displaying safety functions for various cases of malfunction and implementing intelligent driving strategies. The control system software is modelled using Matlab Simulink. The soft-



❹ Flywheel energy storage performance figures ware's development and testing is integrated into the Porsche 911 GT3 R Hybrid's development process. As shown in ❺, the test scenarios consist of three levels: total vehicle simulation including hybrid controls (software in the loop), test bench with hybrid components (hardware in the loop) and finally testing in the race vehicle. As it is possible to use the identical basic model of the control system and a continuous tool chain on all levels, the robustness and the quality of the control system improves with each ascending test level. The test results of higher levels can be used for adapting lower levels' test cases and for validating the models. As an example of the hybrid vehicle functions we describe below two of the main ones: "Energy recovery" and "Boost".



❺ Qualification process

ENERGY RECOVERY

Using energy recovery it is possible to recover some of the brake energy, store it and make it usable again for forward propulsion. Alongside that provided by the hydraulic brake system, in generator mode the portal axle's electric motors also contribute part of the desired braking power. This leads to a flow of electrical power towards the flywheel storage unit, which then becomes charged.

The wish to slow down is sensed via the brake pedal and flows as further parameters into the operating strategy in order to generate braking torque in the electric motor. The braking power exerted on the wheel thus results from an interaction of the mechanical brake system and the electric motors' generator output. The main advantages of brake energy recovery are:

: recovery of part of the usually lost brake energy
: less strain on the wheels' brakes and thus far less wear on the brake system.

❻ Controls

BOOST

On the 911 GT3 R Hybrid the internal combustion engine continues to act as the primary drive system. In traction mode the additional electric drive system torque is released to the front wheels based on the driving situation and the energy store's level of charge. On straight sections of track the boost can be used very spontaneously in a short burst with maximum output. Otherwise the additional torque gets regulated based on the driving situation. In order that the boost function can be utilised the hybrid manager, as part of the drive control system, continually analyses the vehicle's key driving data. This includes the rotary speed of the wheels, the angle at which the car is turning and both forward and lateral acceleration. The flywheel energy store's level of charge continues to be constantly monitored. The hybrid manager indicates to the driver when the system is ready to use and how much electrical boost is available via LED lights on the instrument panel. While the boost is being used, the 911 GT3 R Hybrid thus temporarily has four-wheel drive.

As an alternative to being manually operated by the driver, the function can – depending on drive programme – also be automatically activated by the hybrid control system. The extra, recovered electric energy now gets used in a targeted way to relieve the combustion engine in order, with the same lap performance, to increase the efficiency of the entire drive system. The key advantages are:

: tapping into areas of fuel-saving potential

❼ The Porsche GT3 R Hybrid at the Nürburgring

: supplementary drive torque for more acceleration
: spontaneous accessing of power via boost switch for extra momentum in specific driving situations
: improved traction by virtue of temporary four-wheel drive.

OPERATION

The 911 GT3 R Hybrid's cockpit has several hybrid-specific controls that enable the driver to use the system, ❻. Essentially these are the following:
: boost switch on the steering wheel to call up the extra electrical drive torque
: charge status display for the flywheel storage
: recommended boost LED display
: map switch to launch hybrid-specific drive programmes.

The control for manually activating the boost function is ergonomically fitted in typical motorsport style directly on the steering wheel. An indicator on the instrument panel (LCD display) and a strip of LEDs keep the driver informed about the flywheel storage unit's charge level. The driver is also given a recommendation on optimum use of the hybrid system via an LED light likewise on the instrument panel. In manual mode, this shows the driver when the boost function should be used. Also on the steering wheel is a map switch, via which the driver can access specially configured drive programmes stored for a specific race strategy. This implements an "Efficiency mode", which with reduced power output enables extremely efficient driving with low fuel consumption. Related to this, parameters such as the gear change indicators get accordingly adapted for optimum fuel consumption. The LCD display continues to show the driver important performance and diagnosis figures, which get continually analysed by the hybrid manager.

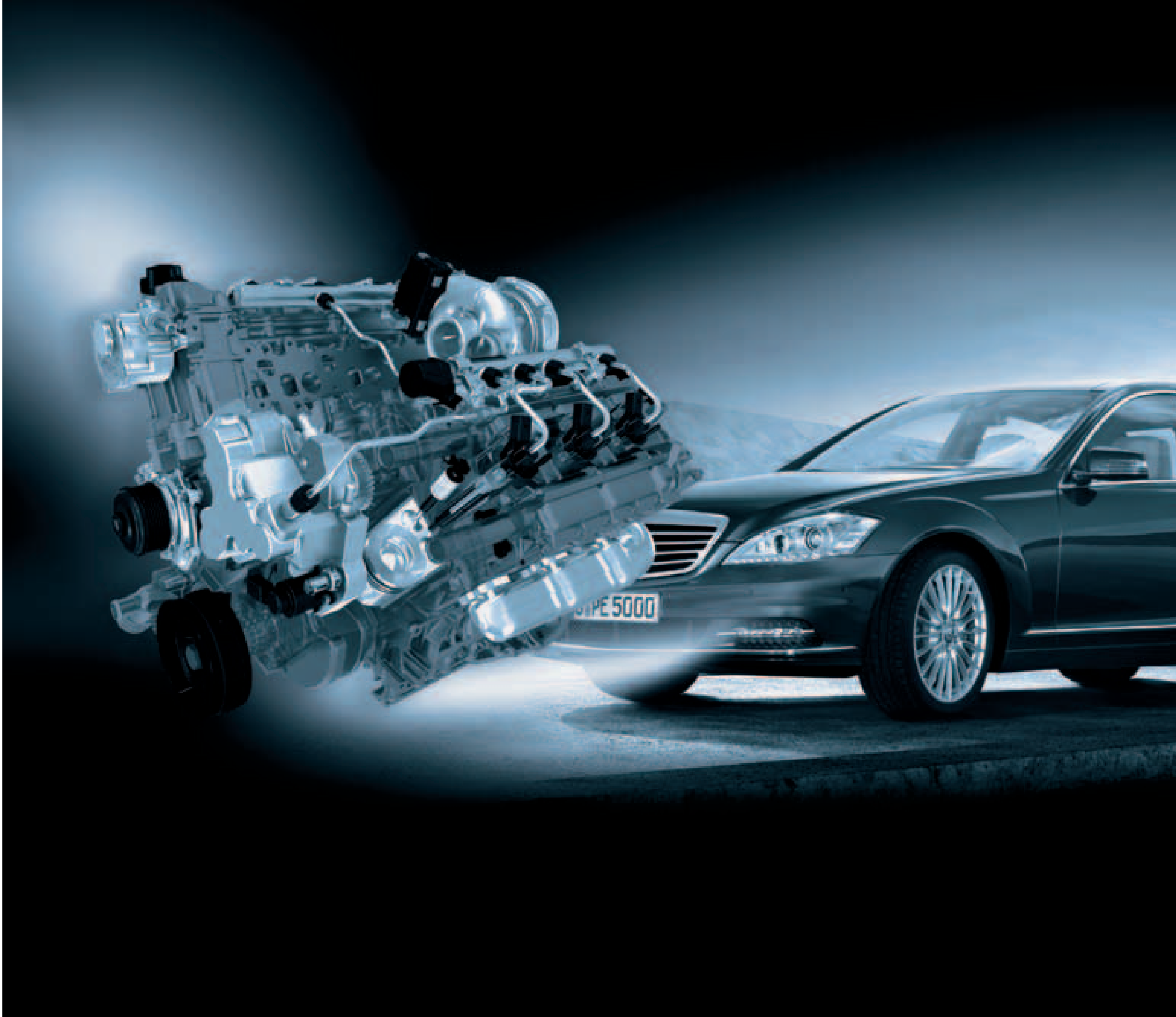
TESTING STRATEGY – RACE USE

As mentioned at the outset, the aim was to test this new technology in racing conditions. On 15th May 2010, the vehicle was thus put through an extremely tough endurance test in the 24-hour race at the Nürburgring, where it impressively demonstrated its performance potential, ❼. For over eight hours the 911 GT3 R Hybrid led the field of 200 cars, before having to retire from the race after 22 hours and 15 minutes with a combustion engine problem. The hybrid system, however, was still working well without any technical issues.

The car completed a second race in September in Atlanta, USA – the 1000-mile race, otherwise known as "Le Petit Le Mans". The focus of development work in the run-up to this race was on optimising the software for automation of the hybrid functionality. The 911 GT3 R Hybrid's final race appearance of 2010 was at the 1000-km race in Zhuhai, China. With its tight corners and relatively long straights, this circuit offered the hybrid racing car significant potential for energy recovery. With the frequent phases of acceleration the car was able to intelligently utilise the hybrid system's traction advantages. The 911 GT3 R Hybrid ended the race as the fastest and most fuel-efficient GT vehicle on the circuit, winning the race in the GT class with a three-lap lead and one less fuelling stop.

OUTLOOK

The 24-hour race enabled the "Race Lab" to prove its potential. The electric drivetrain worked robustly and reliably. For the two following races the team was able to develop and optimise further software functions. The valuable knowledge gained in these tests is now being incorporated into a second-generation race car. Main areas of focus here include increased system integration and reduced system weight with the aim of preparing high-performance hybrid drive systems for use in mainstream production vehicles.



THE NEW V6 DIESEL ENGINE FROM MERCEDES-BENZ

Mercedes-Benz has further enhanced the efficiency of its V6 diesel engine with the internal code OM 642. By using a turbocharger with anti-friction bearings for the first time in a passenger car diesel engine and an extensive package of measures to reduce fuel consumption, the manufacturer has succeeded in combining high power output and torque with high fuel economy.

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HIGHER POWER AND TORQUE, LOWER FUEL CONSUMPTION

The Mercedes-Benz product portfolio has included a 3.0-l six-cylinder diesel engine since 2005. Since then, far beyond one million customers have opted for a V6 diesel engine in the various vehicle applications. Stricter emission regulations coupled with own requirements for ecologically compatible products in the premium class call for a further reduction in the emissions and fuel consumption in the vehicles offered. This is accompanied by the desire for enhanced performance to ensure their attractiveness to the customer.

This article presents the measures which were implemented during the revision of the engine bearing the internal designation OM 642 LS and which enabled an increase in the engine's power and torque of 18 % and 22 % respectively. With a headline figure of 620 Nm, this V6 diesel sets a new benchmark in its class. Fuel consumption was reduced by up to 21 %.

The new V6 diesel engine has been used in the E-Class and R-Class at Mercedes-Benz since fall 2010, ❶. It is also available in the S-Class in conjunction with the successful Mercedes-Benz BlueTec system. In 2011 the launch in further model series will follow.

MAIN TECHNICAL EMPHASIS

The greater mechanical load resulting from the higher performance as well as the associated rise in temperature in the engine required the component modifications described below. They are summarized in ❷.

ENGINE AND CRANKCASE

Following the example of the four-cylinder diesel engine, pistons with bowl lip remelting are used. This involves using an electric arc to melt the bowl lip. During the subsequent rapid solidification of the aluminum-silicone alloy, the sizes of the silicone crystals and of the intermetallic phases are significantly reduced. Thanks to this finer and more uniform microstructure, susceptibility to cracking is significantly reduced. The strength values increased in this way enable an emission-optimized design of the combustion chamber cavity.

		OM 642 EU4 3.0 L V6 DIESEL	„NEW“ OM 642 LS EU5 3.0 L V6 DIESEL
Number of cylinders	-	V6	
Bank angle	degree	72	
Valves / cylinder	-	4	
Displacement	l	2,987	
Bore	mm	83	
Stroke	mm	92	
Cylinder offset	mm	106	
Compression	-	17.7	15.5
Connecting rod length	mm	163	
Main bearing diameter	mm	76	
Bearing width	mm	23	
Crank pin diameter	mm	64	
Bearing width	mm	16.8	
Piston compression height	mm	45.65	
Rated power	kW	165	195
at rpm	rpm	3800	3800
Rated torque	Nm	510	620
at rpm	rpm	1600 – 2800	1600 – 2400

❶ Comparison of the key data of the OM 642 LS in the R-Class with the key data of the predecessor engine

The mechanical stresses in the piston are reduced by the use of a hub case and the temperature lowered by increasing the flow rate of cooling oil. The higher volumetric oil flow rate through the piston-cooling units is provided by a compound oil pump, the drive capacity of which is lower compared with that of the previous standard oil pump.

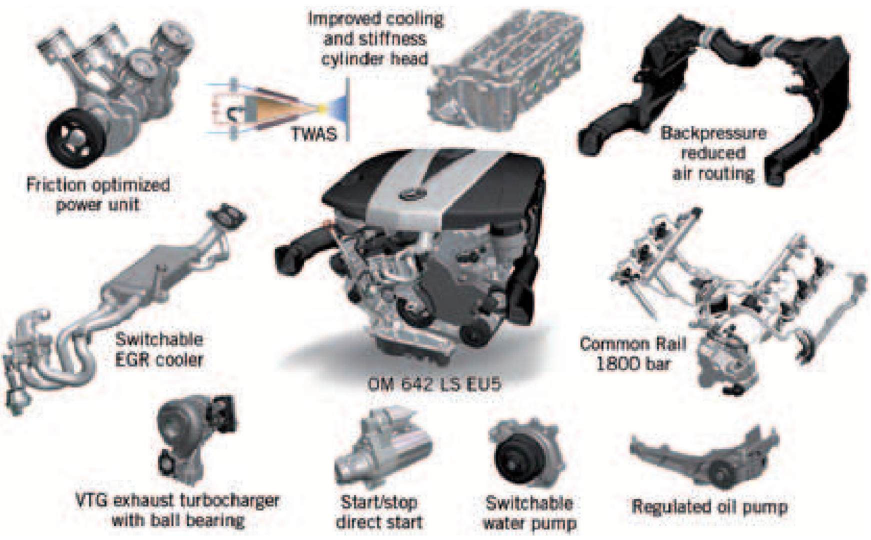
The new Mercedes-Benz precision honing of the cylinder wall was developed for the tried-and-tested aluminum crankcase with molded roughcast cylinder liners manufactured in the core package system. The characteristic structural height of the honed surface was reduced by approximately 50 % in comparison to the predecessor series [1]. The resulting reduction in the oil retention volume of the surface enables the piston ring stress to be reduced. In the process, it was possible to cut the already low engine oil consumption by some 40 %. This was accompanied by a further reduction in the level of wear. The resulting benefit in terms of fuel consumption amounts to approximately 1 % in the NEDC.

With the following development step a thermally sprayed-on layer on the cylinder wall is used for the first time in a production diesel engine in order to further reduce friction losses. Thus the molded, precision-honed roughcast cylinder liners are replaced. Extremely positive experience of sprayed-on cylinder walls in series production has been gained by AMG using the crankcase in the successful 6.3 l V8 engine since 2005. The new cylinder wall has led to a further reduction in friction loss and a lowering of engine weight by 4.2 kg. The additional benefit in terms of fuel consumption is 1.5 %.

The twin-wire-arc spraying (TWAS) system developed by Mercedes-Benz is used, ❸. This involves using an electric arc to melt an iron-based cylinder wall material and applying the material to the preconditioned aluminum cylinder wall with the aid of an inert gas. Surface structuring and activation is achieved using a high-pressure water jet process. In the process used to date, the surface is then provided with a honed structure.

The TWAS overlay enables frictionally optimized pairing of the piston ring and cylinder wall material. The TWAS-specific pores create the required oil retention volume in the surface. This enables an even smoother honing structure, which in turn permits a further reduction of piston ring stress and therefore a lowering of friction loss.

As additional measures aimed at reducing friction loss in the engine DLC-coated piston pins (Diamond-like Carbon) and a revised oil catch tray attached to the bearing block are used. The latter reduces windage loss and lowers the level of oil foaming significantly.



❷ Technologies deployed with the OM 642 LS

CYLINDER HEAD

The duct routing and valve gear are carried over from the predecessor. Additional measures were required to combat higher levels of heat introduced into components exposed to exhaust gases as a result of the modified combustion application. Consequently, the outlet valves and seating rings are manufactured from higher quality materials.

Division of the water jacket brings about a reduction in the temperatures at the critical web areas of the inlet and outlet valves and of the exhaust ports. This results in a high coolant flow speed close to the combustion chamber plate and therefore to excellent cooling of the areas at risk. Fluid flow and uniform distribution in the cylinder head are optimized in terms of temperature by means of modified discharge openings in the cylinder head gasket. The critical web temperatures of the inlet and outlet valves are reduced by up to 20 K, ❹ (right). The risk of cracking at the valve land is significantly reduced as a result.

The two-piece water jacket also increases the rigidity of the cylinder head. This enabled the already excellent closing characteristics of the outlet valves to be improved along with the further reduction in their wear performance.

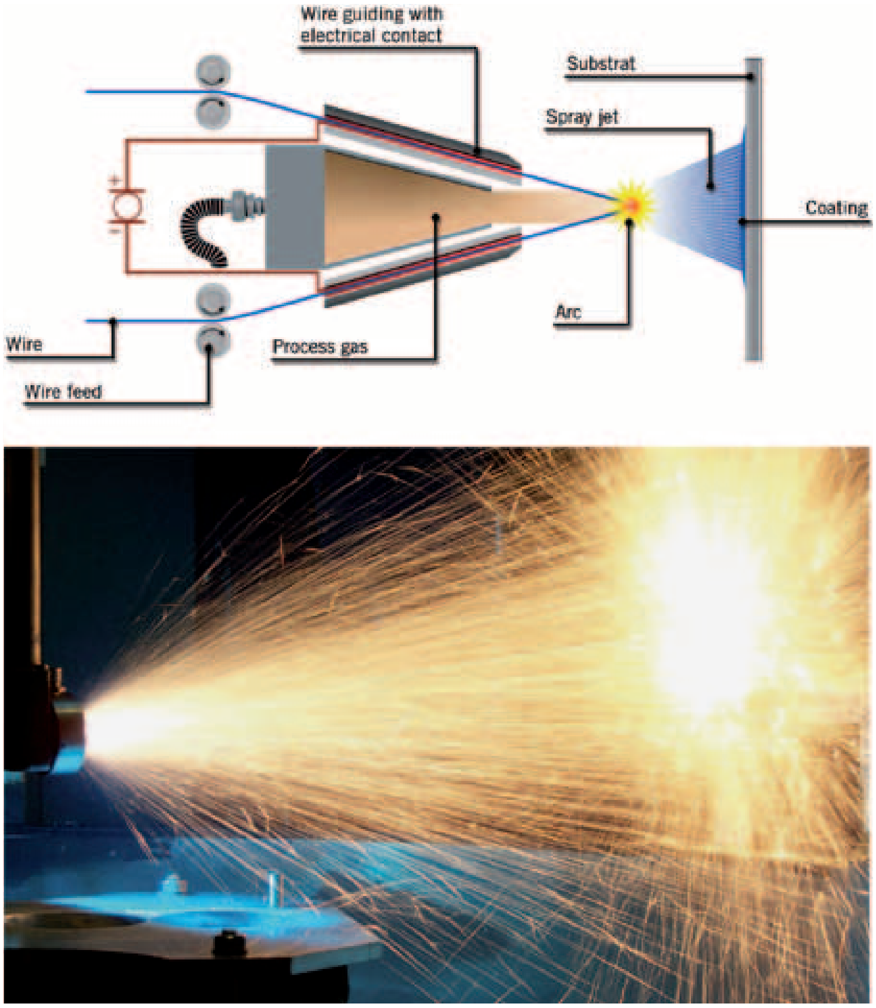
WATER CIRCUIT AND THERMAL MANAGEMENT

To reduce raw emissions during the cold-running phase in particular and to reduce consumption, a pneumatically activated water pump was integrated. The pump action can be prevented by a cylindrical barrel that slides over the impeller. Due to the swifter heating of the cylinder head in particular, the engine reaches the favorable combustion range in terms of emissions at an earlier stage. The emission of hydrocarbons (HC) and carbon monoxide (CO) is reduced in this way. In the zero delivery state, the required operating energy for the water pump is significantly reduced.

AIR INTAKE

The intake of air takes place via the air filters mounted on both sides of the engine. A manifold fitted with two integrated hot-film air-mass sensors (HFM) conducts the purified air to the exhaust gas turbocharger mounted in the inner V.

A 33 % reduction in pressure losses in the air intake system thanks to the use of enlarged flow cross-sectional areas and optimized flow properties contributes to the increase in power and torque. Modifications to the inlet of the HFMs improve the accuracy of the air mass measurement – irrespective of the air filter load condition.



❸ The TWAS overlay process for creating ferrous overlays in aluminum crankcase housings

EXHAUST GAS RECIRCULATION

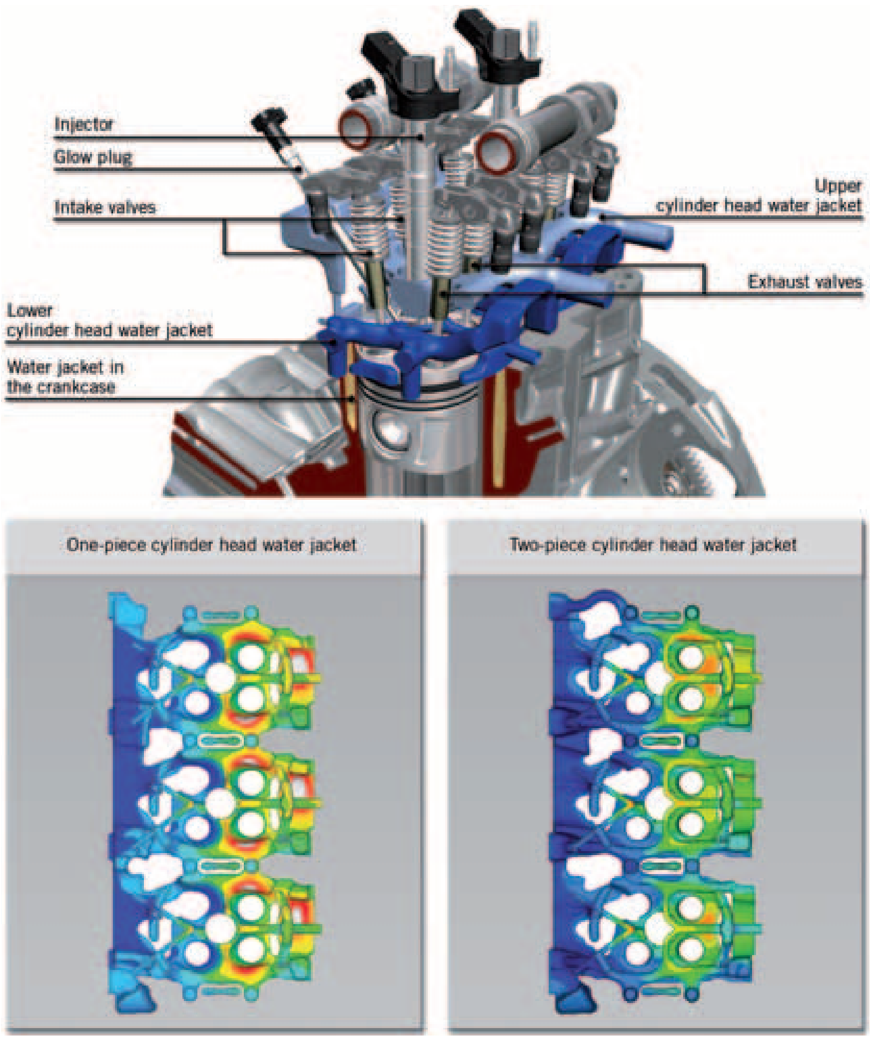
The arrangement of the exhaust gas recirculation with an electric EGR valve, the EGR cooler in the inner V and intake air throttling upstream of the point of release in the charge air distribution line is carried over from the EU4 engine. On the one hand, the formation of nitrogen oxide is effectively reduced by a 60 % improvement in EGR cooling performance. On the other hand, the HC and CO emissions are minimized by operating-point-dependently selectable integrated bypassing of the radiator during the engine warm-up phase.

EXHAUST GAS TURBOCHARGING

In addition to achieving ambitious power and torque figures, the turbocharging concept also places great importance on the issue of vehicle agility. Furthermore, the turbocharging system exerts a strong influence on engine emissions as well as gas cycle efficiency and, therefore, on fuel consumption. Comprehensive evaluation of various turbocharging concepts has identified an advantage in retaining the turbocharging concept introduced, ❺, which uses a single turbocharger, as compared with staged turbocharging concepts.

In order to boost the performance, the turbocharger, ❻, is qualified to withstand exhaust gas temperatures up to T3 = 860 °C while throughput capacity is increased thanks to an enlarged compressor or rather turbine wheel. The resulting higher moment of inertia has been compensated for by reducing the bearing friction to avoid sacrificing agility.

The process of upgrading the turbocharger to cope with higher exhaust gas temperatures required material and design



❹ Cylinder head with a two-piece water jacket; comparison of the temperature distribution in a one-piece and in a two-piece design

modifications on the exhaust gas side. Higher-quality materials are used for the bearing casing and the blade contour ring. The blade contour ring in the turbocharger is no longer bolted. Instead, it is clamped between the turbine and bearing casing.

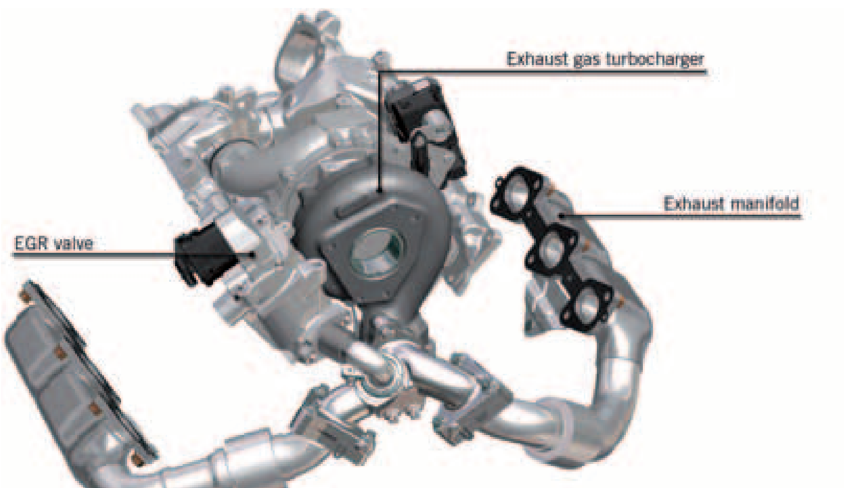
In order to reduce bearing friction, the bearing used was changed from a plain bearing to a ball bearing for the first time in a passenger car diesel engine. As a result, higher turbocharger speeds are reached particularly at low partial loads. The higher moment of inertia is also overcompensated. The maximum permissible turbocharger speed is still determined by the peripheral speed of the compressor turbine wheel. The ball bearing has a lower oil throughput than the plain bearing. The resulting lower heat dissipation is offset by integrating the turbocharger in the cooling water circuit. The reduced bearing friction has a particularly positive impact on transient response. ❼ plots the different accelerations of two E-Class vehicles with plain- and ball-bearing turbochargers respectively from standstill after 4 m, 20 m and 60 m. As can be seen from the graphs, the vehicle equipped with the ball-bearing turbocharger maintains an increasing lead.

The target EGR rate is reached at a higher lambda in the turbocharger fitted with a ball bearing, which results in lower emissions (HC, CO and particulate) as well as fuel consumption benefits (CO_2).

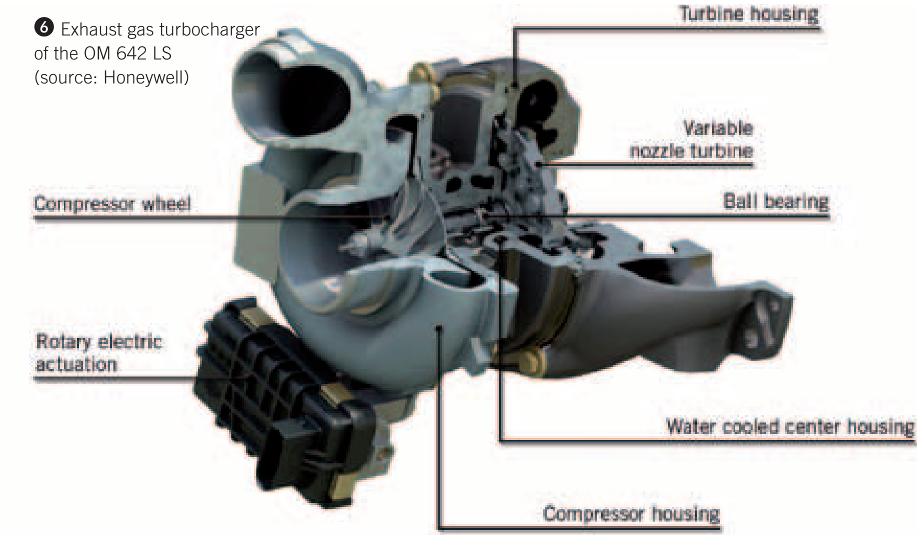
In order to retain the familiar low noise level in spite of the higher mass flow rate, the resonators in the air ducting were optimized and enhanced. The multi-chamber wide-band damper fitted in the front area of the engine, whose effective volume was enlarged by approximately 20 % while keeping the overall dimensions unchanged, is cited as an example of this. The new damper is designed as a two-piece casing made of temperature- and pressure-resistant PA 46 GF40.

INJECTION HYDRAULICS

The OM 642 LS uses Bosch common rail piezo injection technology developed to handle injection pressures of 1800 bar. A key element is the piezo injector, which has proven itself over many years, combined with the latest generation of eight-hole blind-hole nozzle. It excels with ultra-precise timing, high switching speeds and a further improved microquantity capability and delivery stability. High pressure is generated by the established, weight-optimized three-cylinder pump, the drive system of which was carried over unchanged despite the increase in operating pressure.



❺ Exhaust manifold and turbocharger of the OM 642 LS



❻ Exhaust gas turbocharger of the OM 642 LS (source: Honeywell)

A regulated in-tank fuel supply pump is employed as a CO_2 measure. In order to relieve the on-board power supply at extremely low outside temperatures, an on-demand electric fuel filter heater is used. The control function is performed by the existing sensor system in the fuel circuit.

COMBUSTION SYSTEM

The redesign of the compression ratio and combustion chamber geometry played an essential role in the evolution of the OM 642 towards meeting EU5/EU6 emission values. After comprehensive investigations that took account of raw emissions, consumption, combustion noise and cold-start capability, a significant reduction in compression from 17.7 to 15.5 has been realized. The reduction in bowl taper and the use of enlarged radii at the bowl lip enabled the piston bowl for the compression ratio of 15.5 to be configured in a way that allowed particulate emissions to be lowered.

In order to adjust it to the changed combustion chamber geometry and the increase in injection pressure to 1800 bar, the injection nozzle was modified with regard to the spray height angle and the spray side angle. Due to the stricter emissions requirements and higher injection pressure, the blind-hole volume was also reduced. This was accompanied by a reduction in the nozzle hole length and an increase in the flow coefficient (Cv value) in order to improve the spray quality.

The lowered compression ratio necessitates the use of the ceramic glow system already tried and tested in the BlueTec engines. This ensures that cold starts with extremely short glow periods and stable engine operation are achieved under all relevant conditions, and especially in cold weather and at high altitude.

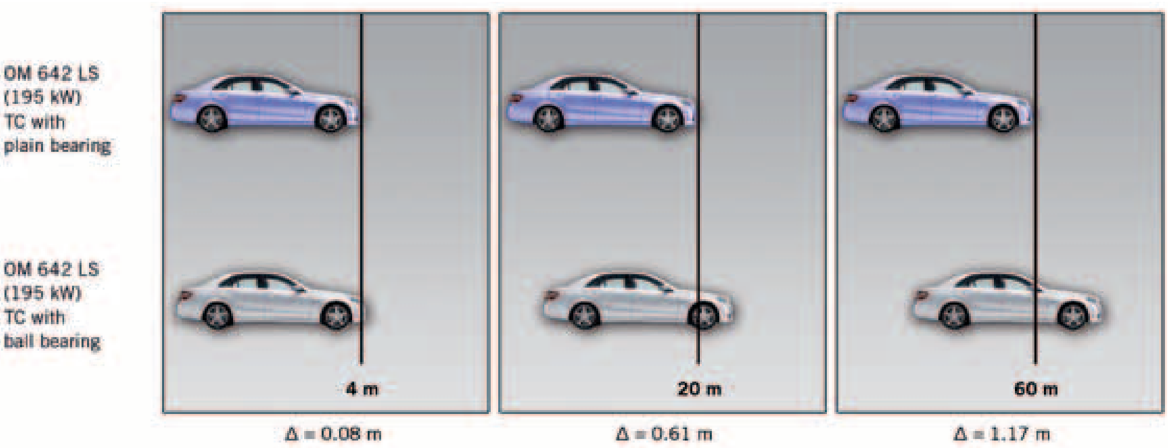
The tried-and-tested inlet port design incorporating a spiral and tangential swirl duct in a non-rotated configuration, which excels by virtue of a high swirl ratio when the spiral swirl duct is shut off (inlet port shutoff) and excellent throughput (filling) in two-channel operation, was carried over unchanged.

EXHAUST GAS AFTERTREATMENT AND EMISSIONS

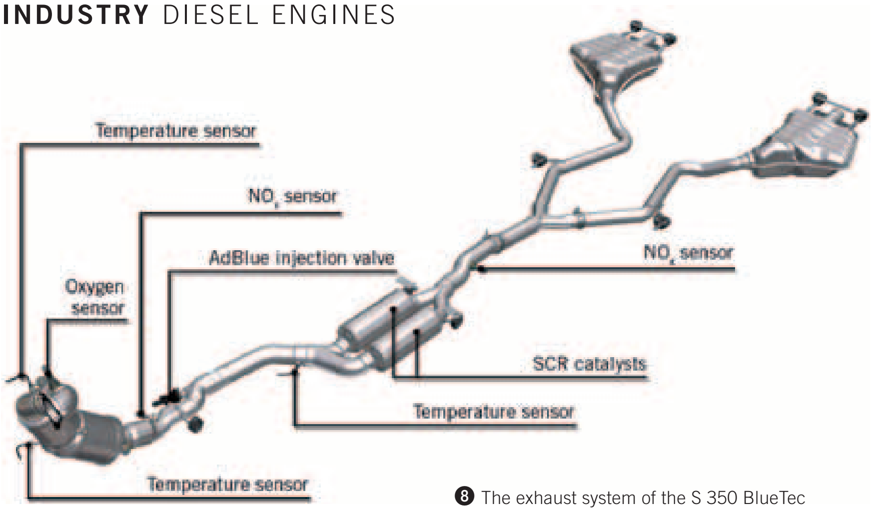
As part of the evolution, a further reduction in emissions was achieved. This means that as well as complying with the current EU5 legislation in the BlueTec version already in use in the S-Class, the OM 642 LS also meets the EU6 limits due to come into force in 2014 and is prepared for worldwide use.

The Mercedes-Benz additive-free particulate filter regeneration strategy was further developed for the OM 642 LS in the S 350 BlueTec. A model-based filter loading analysis in the engine control system that was developed in-house led to an increase in regeneration efficiency. The model's high accuracy enables reduced particulate filter volumes while at the same time retaining regeneration intervals of up to 1000 km. Thanks to a two-stage exhaust gas temperature controller, the desired regeneration temperatures are adhered to with even greater accuracy, ensuring swift and safe soot burn-off and reducing the thermal load on the particulate filter and SCR (Selective Catalytic Reduction) catalytic converter.

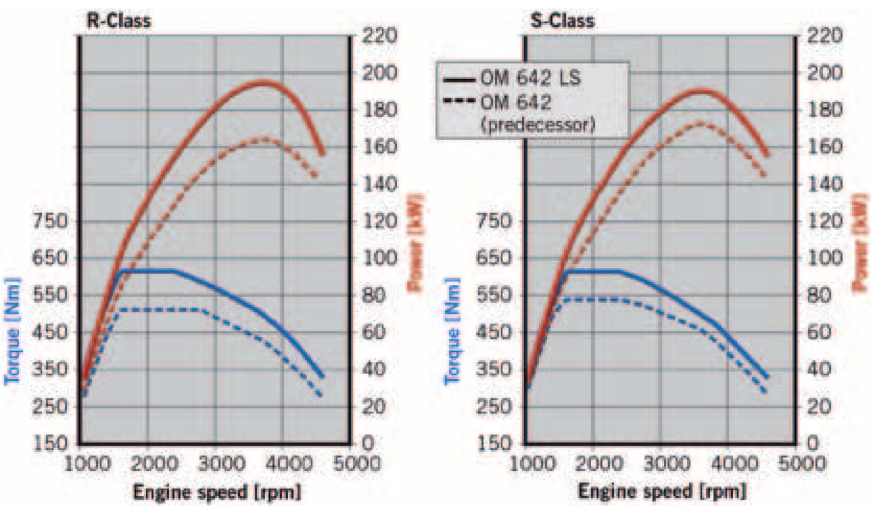
BlueTec is a technology developed by Mercedes-Benz to reduce emissions from diesel engines, particularly of nitrogen



❼ Comparison of exhaust gas turbocharger with plain bearing with a turbocharger with ball bearing (acceleration from stand still – E-Class with automatic transmission)



❽ The exhaust system of the S 350 BlueTec



❾ Power and torque of the OM 642 LS in the R- and S-Class compared with the predecessor

oxides. The technology involves injecting AdBlue, a harmless aqueous urea solution, into the exhaust gas flow. This process releases ammonia, which then reduces up to 80 % of the nitrogen oxides in the downstream SCR catalytic converter to harmless nitrogen and water.

The exhaust system, ❽, which is optimized in terms of emissions and back pressure, features a twin-pipe SCR catalytic converter configuration in the underfloor. This is in addition to the near-engine mounted oxidizing catalytic converter and particulate filter. Along with the temperature and pressure sensors, O_2 and NO_x sensors for controlling and providing OBD monitoring of the emission-relevant functions are also used.

As a result, the new OM 642 LS in the S 350 BlueTec is not only one of the most efficient six-cylinder diesel engines in the luxury class, but also ranks among the cleanest diesel engines in the world thanks to the AdBlue emission control technology.

ENGINE RESULTS AND SUMMARY

With regard to the EU5 applications, the performance compared with the predecessor engine is increased by 18 % to 195 kW and torque by 22 % to 620 Nm, ❾. A power output of 190 kW is achieved in the S-Class designed to meet EU6 limits – despite the higher back pressure in the SCR exhaust system.

Thanks to the evolved engine, optimized transmission and the exhaust gas aftertreatment featuring a DPF and SCR (S-Class), a significant improvement in the emission and fuel consumption figures is achieved, ❿. The higher power and torque also enabled a significant improvement in driving performance. Taking the S-Class as an example, the time taken for the 0 to 100 km/h sprint has dropped by 10 % – from 7.8 to 7.1 s. The top speed is limited to 250 km/h.

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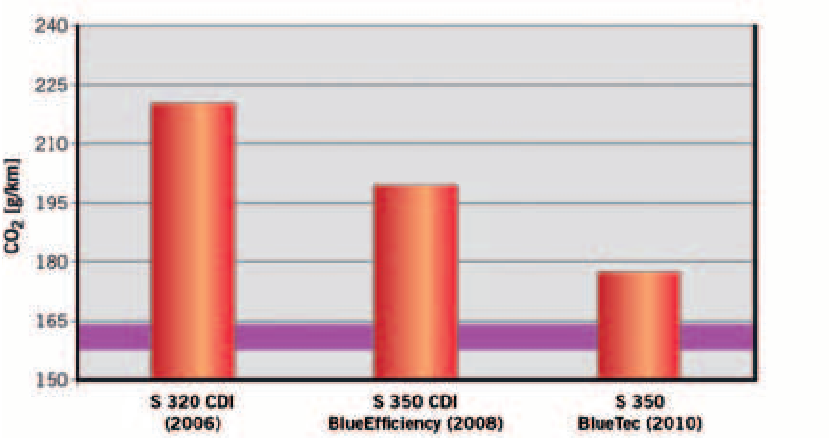

❿ Development of the fuel consumption of the V6 diesel engine in the S-Class

THANKS

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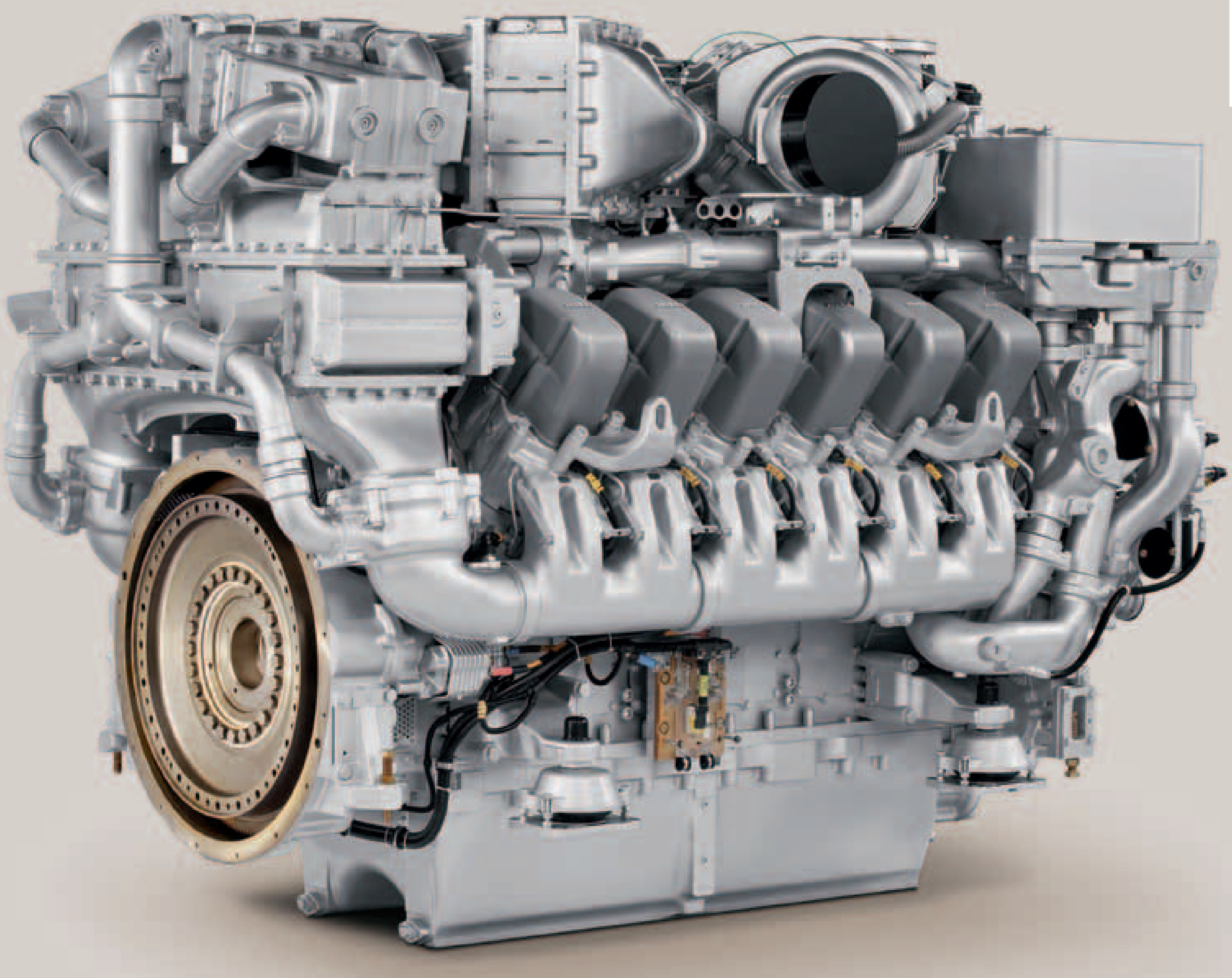
THE NEW MTU SERIES 4000 RAIL ENGINES CERTIFIED FOR EU IIIB

As of 2012, diesel locomotives in Europe must comply with the emission requirements laid down in EU Non-Road Mobile Machinery Directive 97/68/EC Stage IIIB. Compared to the Stage IIIA in effect today, this will require a significant reduction of air pollutants. MTU Friedrichshafen GmbH has developed new engines for rail applications based on the 4000 series which will comply with future emissions standards due to optimisation inside the engine and the use of a diesel particulate filter.



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LOWER EMISSIONS AND FUEL CONSUMPTION, LOWER LIFE-CYCLE COSTS

Since its market launch in 1996, more than 21,000 MTU Series 4000 engines have been sold for marine, power generation, mining, oil & gas and rail applications. Series 4000 engines have been used very successfully in rail applications since 1997. Today, more than 1800 engines are being used world-wide in main line and shunting locomotives. Their combined runtime exceeds eleven million operating hours. In Europe, MTU's main market for diesel locomotives, Stage IIIB of the 97/68/EC emission directive for non-road mobile machinery comes into force in 2012. Compared with the previous Stage IIIA, the limit values for nitrogen oxide levels are down by 43 % and no less than 88 % for particulate matter levels, ❶. Despite the more stringent emission requirements, the primary aim of the product development was to further reduce fuel consumption and life cycle costs of the diesel propulsion system.

REQUIREMENTS ON RAIL PROPULSION SYSTEMS

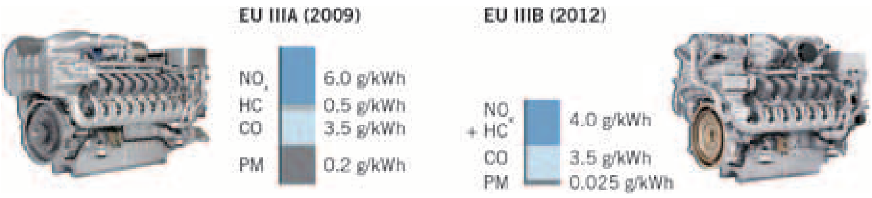
Series 4000 rail engines are being used in shunting and main line locomotives as well as in high-speed trains. In addition to the compliance with the emission requirements, further reduction of the life cycle costs as well as optimum integration into the vehicle (installation space, weight, interfaces) are the key requirements of locomotive manufacturers and rail operators. Diesel engine efficiency is a key factor for the efficiency of the entire railway system. On 2 MW main line locomotives, the fuel costs incurred over the entire engine lifetime can easily amount to 50 times the engine purchase price and add up to several million Euros over the entire lifetime of the engine. Low life-cycle costs can therefore only be achieved by a combination of low fuel costs, low maintenance costs (long maintenance intervals) and high availability and reliability. Integration of the diesel propulsion system into a new or existing locomotive concept poses a particular challenge and requires close cooperation with the locomotive manufacturer from a very early development stage. The following are key requirements to integrate the propulsion system into the limited installation space available: a compact, weight-optimized engine design, low amounts of heat which must be dissipated and compact exhaust gas aftertreatment components.

TECHNICAL ENGINE CONCEPT

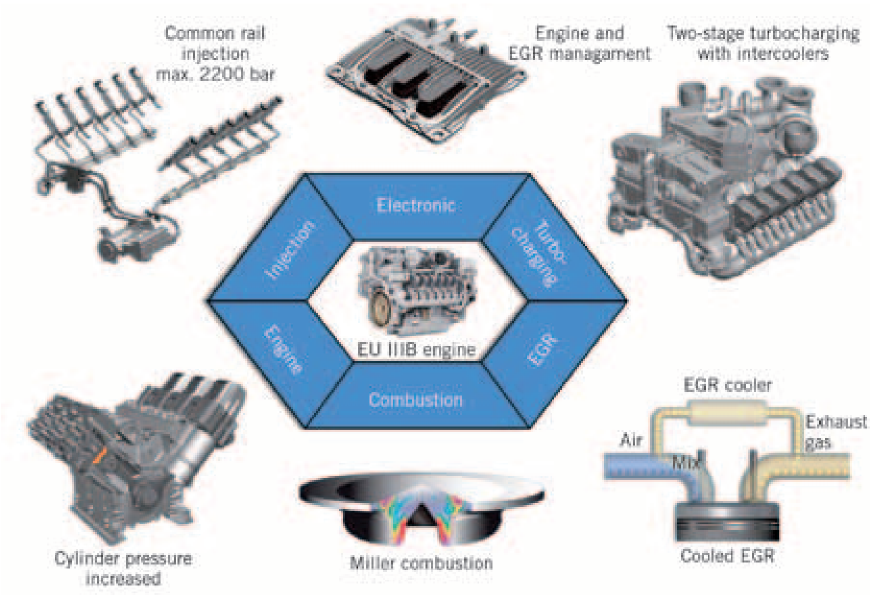
The latest generation of MTU Series 4000 rail engines (Series 4000-04) is a consistent and target-oriented development of the previous versions. Right from the start, the aim was to take over as much tried, tested and qualified technology as possible. The Series 4000-03 introduced into series production in 2009 [1] forms the basis for the EU IIIB engine. The technical concept, a combination of engine-internal technologies and exhaust gas aftertreatment as an integrated system, was established in close cooperation with the locomotive manufacturers.

ENGINE-INTERNAL EMISSION TECHNOLOGIES

MTU's engine-internal concept to comply with EU IIIB uses cooled controlled exhaust gas recirculation as the core technol-



❶ Emission limits for diesel locomotives as per EU directive 97/68/EC for non-road mobile machinery

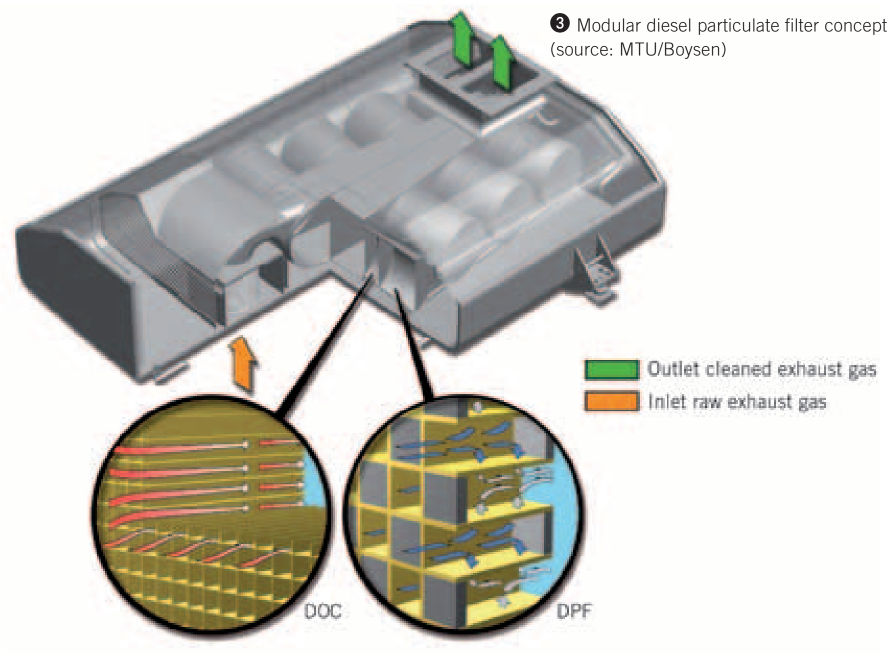


❷ Engine-internal emission technologies

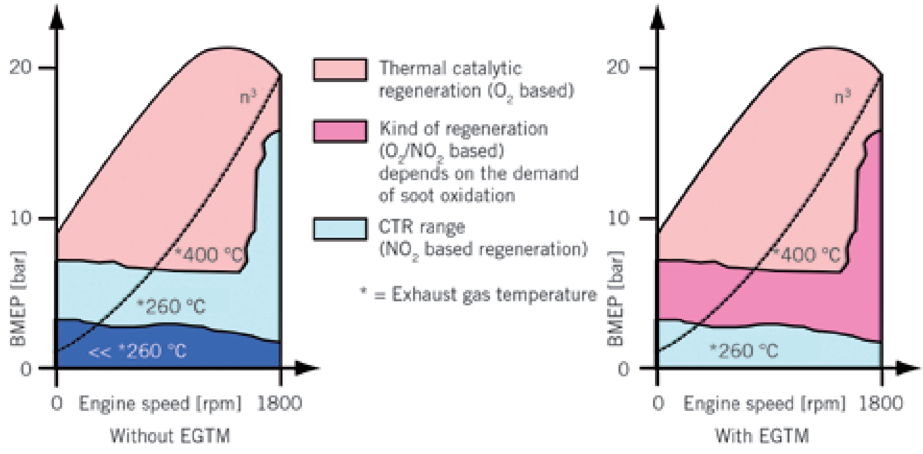
ogy to reduce nitrogen oxides. Together with high turbocharger efficiencies, the donor cylinder concept patented by MTU [2] minimizes the engine's gas exchange losses. It allows exact and stepless setting of exhaust recirculation rates between 0 and approximately 50 % (one cylinder bank as donor cylinders). With the current engine concept, the maximum EGR rate is 25 %. This concept therefore also offers potential for other or future emission stages with higher EGR rates. A two-stage controlled charging system with three MTU turbochargers (two low-pressure and one high-pressure turbocharger) ensures consistent good fresh air supply to the engine at all operating points even under extreme peripheral conditions (intake air temperature, altitude above sea level, back pressure). With the help of the newly developed Miller combustion process, the NO_x emissions limits are complied with while at the same time minimizing fuel consumption. The common rail fuel system with rail pressures up to 2200 bar is capable of multiple injections and ensures low particulate matter raw emissions, thereby providing the basis for a compact and reliable DPF design. ❷ provides an overview over the engine-internal technology portfolio.

EXHAUST GAS AFTERTREATMENT

To comply with the stringent particulate matter limit of 0.025 g/kWh, a closed diesel particulate filter with upstream oxidation catalytic converter is used. When developing this exhaust gas aftertreatment system, installation space optimization as well as low pressure losses were in the center of attention. The DPF system can generally be installed within the installation space for the exhaust gas silencer. In future, silencing and separation will be integrated in one component. Locomotive-side installation of the DPF is done taking into account the individual installation situation. To minimize the production costs for the exhaust gas aftertreatment system when used in very different installation spaces, the DOC and DPF substrates are modular in design. The monoliths are ceramic substrates with a precious metal coating, ❸. The significant reduction in oil consumption, the use of low SAPS oils and of low-sulfur diesel fuels result in long DPF system maintenance intervals which are matched to the main engine maintenance intervals. The modular design allows separate replacement of the individual DOC and DPF substrates, thereby reducing the locomotive downtime during maintenance.



❸ Modular diesel particulate filter concept (source: MTU/Boysen)



❹ Passive regeneration strategy based on the CRT effect (Continuous Regeneration Trap) and exhaust gas temperature management (EGTM)

REGENERATION STRATEGY AND THERMAL MANAGEMENT

The key requirement for DPF operation is safe regeneration under all possible operating profiles and peripheral conditions. This regeneration must be possible with minimum additional fuel consumption and without limitations to use or disadvantages regarding volume or weight. The low level of raw emissions enables the use of a DPF with passive regeneration. Thanks to the CRT effect (Continuous Regeneration Trap), active regeneration such as that using a burner or HC dosing unit is not required. While the exhaust gas temperature is sufficiently high for continuous soot burn-off in the DPF under normal operating conditions, specific climate conditions or continuous low-load operation may require boosting of the regeneration by increasing the exhaust gas temperature. This is achieved by engine-internal thermal management. At engine speeds close to idle, the charge quantity can be reduced by an air-side throttle plate. An additional increase in exhaust gas temperature can be achieved by post-injection and/or delayed main injection. The engine control unit determines which of these measures is employed. The actual DPF load state is monitored redundantly via an emission and soot burn-off model as well as via the differential pressure sensor of the DPF system. This ensures a needs-based and therefore highly efficient regeneration without noticeably increased consumption, ❹.

ENGINE CHARACTERISTICS AND POWER RANGE

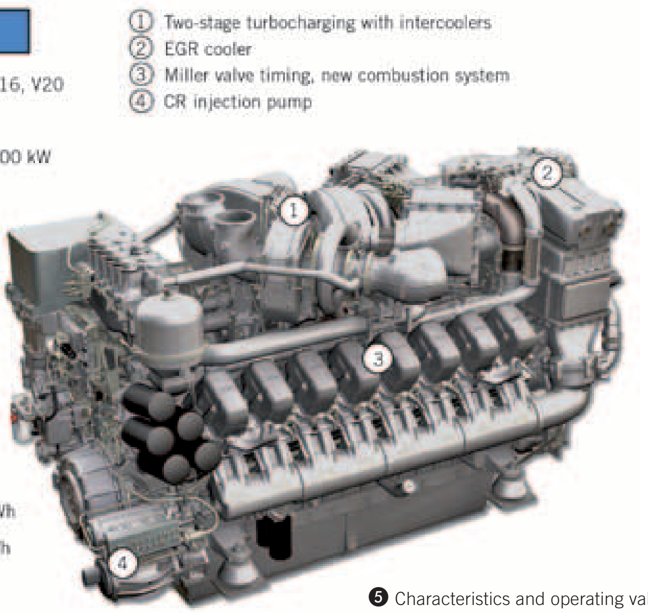
As is the case with the previous engine series, the EU IIIB engines will be available as eight, twelve, 16 and 20 cylinder V-engines. They cover the power range from 1000 to 2700 kW. The maximum cylinder power is 150 kW/cylinder at a rated speed of 1800 rpm. The engine provides sufficient torque in all speed ranges, both for diesel-electric as well as for diesel-hydraulic propulsion systems. ❺ demonstrates the design implementation of the new technologies using the V16 engine as an example and shows the key characteristics of this engine series.

FUEL CONSUMPTION AND LIFE-CYCLE COSTS

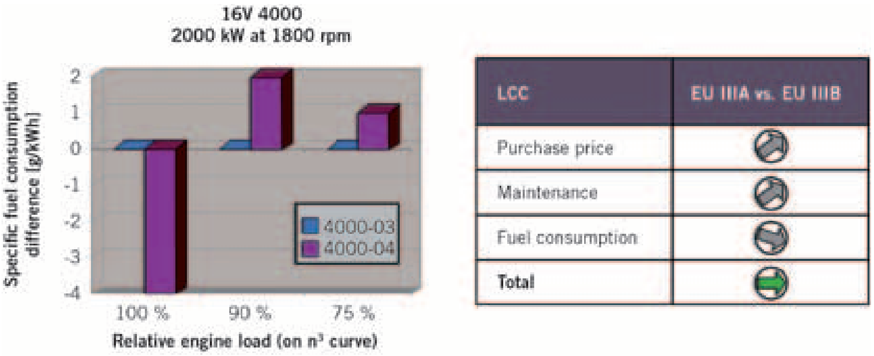
Optimization of the fuel consumption has been the top priority right from the very beginning. Over the next decades, diesel-driven locomotives will only be able to maintain their dominant role in non-electrified rail applications, if the reduction of the typical air pollutants, namely nitrogen oxides and soot particles, can be achieved without an increase in fuel consumption

Engine configuration	
Cylinder numbers	V8, V12, V16, V20
Displacement per cylinder	4.77 l
Power range	1000 – 2700 kW
Max. power per cylinder	150 kW
Rated speed	1800 rpm
BMEP at rated speed	21.0 bar

Engine operating data (at full load and rated speed)	
Boot pressure	4.8 bar
EGR rate	25 %
Air to fuel ratio	25
Fuel consumption	< 205 g/kWh
Raw particulate emission	< 0.1 g/kWh
NO_x emissions	3.5 g/kWh



❺ Characteristics and operating values of the new MTU Series 4000 rail engines



❻ Fuel consumption and life-cycle costs compared to Series 4000 R43

[3]. With the help of the above engine-internal emission technologies and a diesel particulate filter system designed with a focus on efficiency, the specific fuel consumption of a diesel-electric main line locomotive at rated load could again be slightly reduced when directly compared to the current Series 4000 R43 – and that despite a NO_x reduction of 40 % and a particulate matter reduction of approximately 88 %. This effect on the life-cycle cost of the engine compensates for the unavoidable increase in maintenance costs and engine selling price, ❻.

MAIN DIMENSIONS AND WEIGHTS

As a result of the very compact arrangement of the exhaust turbochargers and the intercoolers, the main dimensions remain virtually unchanged when compared to the current engine. This also applies when the diesel particulate filter is taken into account (integrated into silencer); an immense advantage compared to alternative solutions such as the use of an SCR system. An increase in weight compared to the current engine is unavoidable given the multitude of additional systems and components. However, the aim was to limit the increase in weight to a minimum. In this regard, the two-stage turbocharging system is very advantageous. A single-stage concept (examined in the concept phase but not implemented) results in disadvantages regarding the system weight (engine, DPF and heat exchanger). With its high back pressure capability, the two-stage turbocharging system allows a significantly more compact DPF design. In addition, air ratio and EGR rate can be regulated based on the peripheral conditions, which in turn reduces the cooling requirements and therefore the heat exchanger size. The additional engine weight of the third turbocharger and of the two intercoolers is more than compensated for.

SUMMARY AND CONCLUSION

In 2012, the fourth generation of MTU Series 4000 engines for rail applications will be introduced into series production. The engine complies with the stringent EU IIIB emission legislation. The NO_x emissions limits will be met without the need for external engine systems; the particulate matter emissions limits will be complied with by using a diesel particulate filter with passive regeneration only. The chosen concept meets the key customer requirements regarding minimum fuel consumption and life-cycle costs, high reliability and a design optimized with regard to installation space and weight. The technologies used have been tested comprehensively before introduction into series productions, as this is the basis for high reliability and availability.

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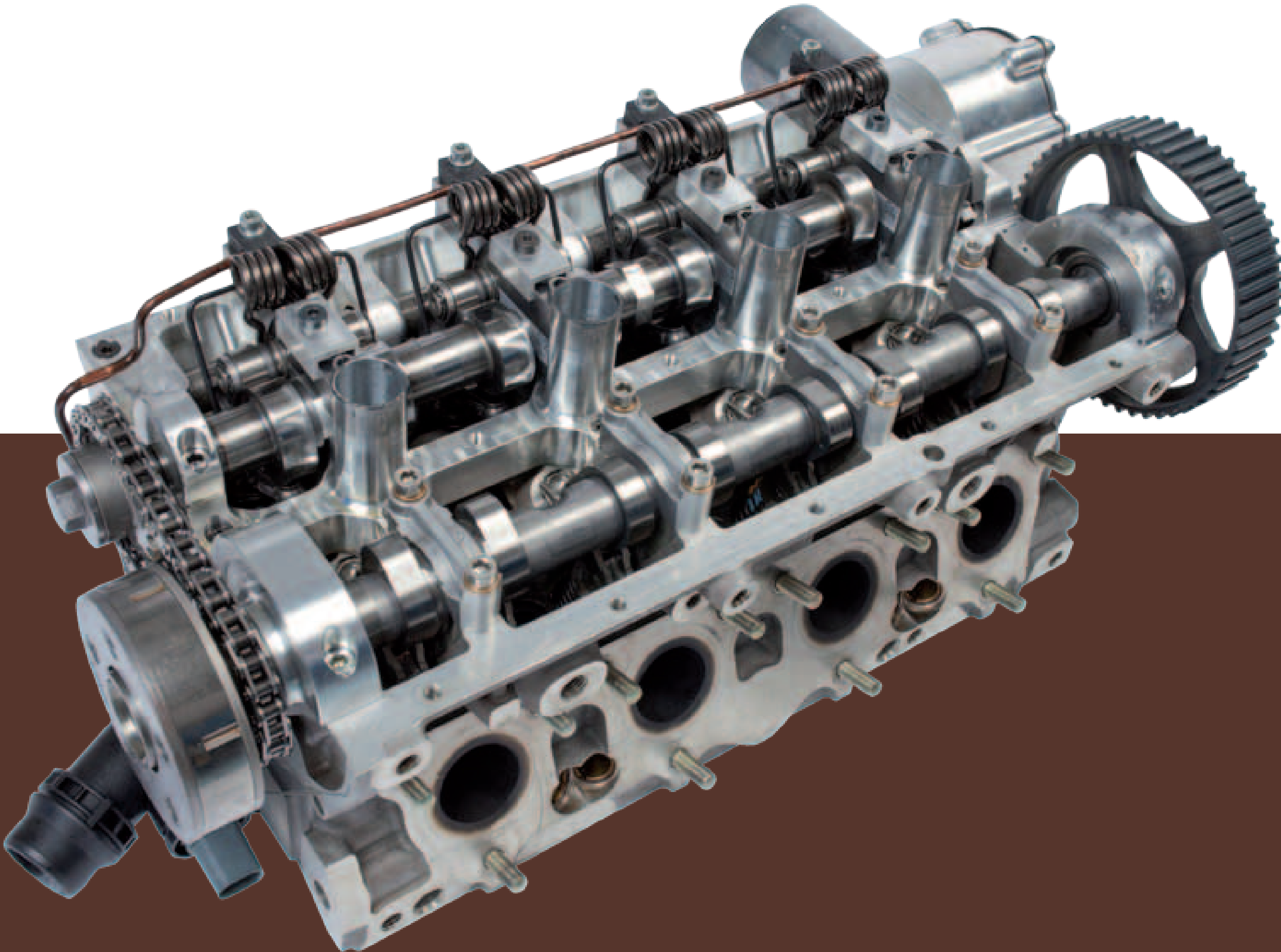
UNIVALVE – A FULLY VARIABLE MECHANICAL VALVE LIFT SYSTEM FOR FUTURE INTERNAL COMBUSTION ENGINES



In order to optimize the gas exchange and combustion sequences, the performance, emissions, and – above all else – the degree of effectiveness of internal combustion engines have to be fully utilized. The only way to achieve this potential is by using a valve train which enables infinitely variable adjustment of the inlet valve lift duration, thus permitting throttle-free load control. Therefore, the Kolbenschmidt Pierburg AG selected the Univalve system, developed by Entec Consulting GmbH, and recently acquired the rights to the new system. In joint effort, the Univalve system is now undergoing final development prior to full-scale serial production.

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MOTIVATION

Without the internal combustion engine, it will be impossible to meet the world population's desire for mobility in the next 50 years. Given the current global inventory of something like a billon vehicles and/or internal combustion engines, alternative drive concepts will have to be developed, and every possible means of improving the internal combustion engine pursued, if a reduction in CO_2 emissions is to be achieved and maximum economic utility from these vehicles obtained.

Variable, and especially fully variable, valve trains result not just in greater fuel efficiency owing to optimized gas exchange, residual gas compatibility and primary drive friction, but also in improved torque and performance, meaning that the consumption potential of downsizing concepts can be further increased. In addition, they are a pre-requisite for the use of new and/or alternative combustion techniques such as HCCI.

Kolbenschmidt Pierburg AG sees the technology field of variable valve trains as a future growth market. Thus, in order to expand its product portfolio for the air pathway of internal combustion engines, it has acquired the fully variable mechanical valve train system Univalve under a cooperation agreement with Entec Consulting GmbH.

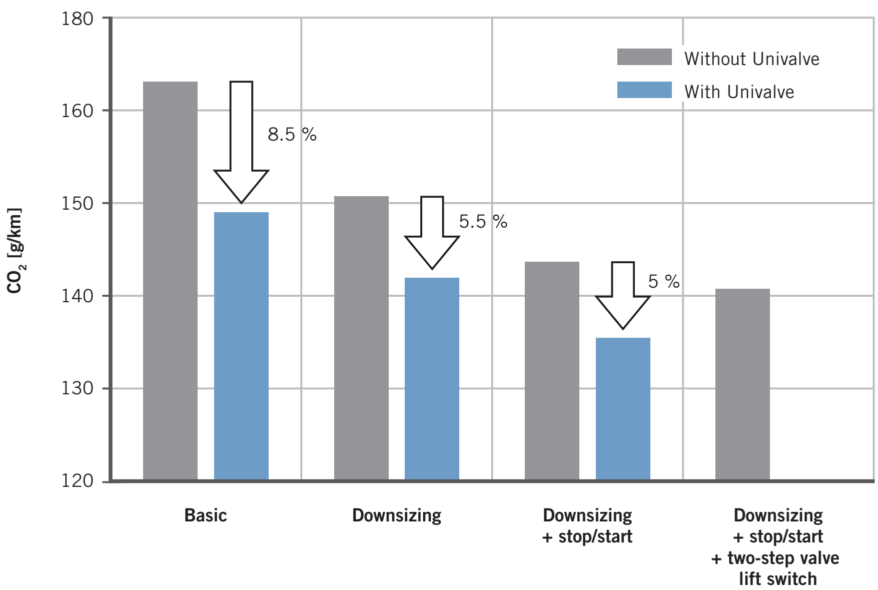
HOW MUCH VARIABILITY DOES AN INTERNAL COMBUSTION ENGINE NEED?

In gasoline engines, the operating point-optimized adjustment of inlet and outlet spread is established with camphasing systems. Since 2005, especially in Japan, a broad array of additional, mechanically fully variable valve train technologies have been employed which are increasingly being used in Europe as well. Competing technologies include servo-hydraulic fully variable systems or two-step valve lift switches.

Thus the question arises as to which concept delivers the greatest utility at an acceptable level of technical and commercial effort and expense, and what fuel consumption advantages remain in technology combinations with alternative measures to reduce CO_2 emissions.

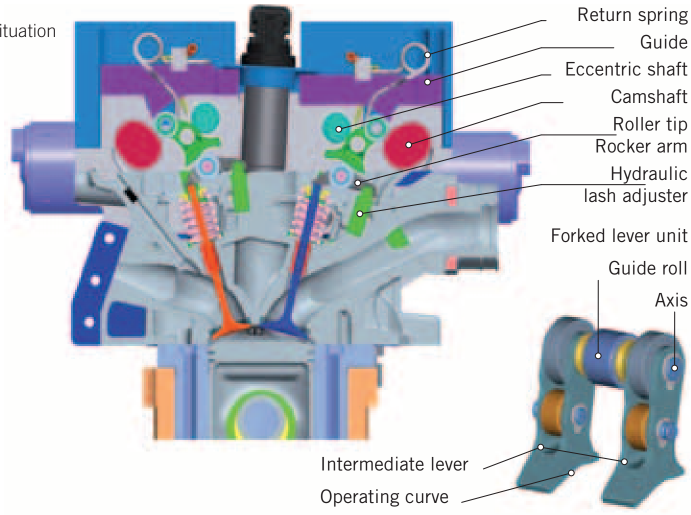
To address this question, a simulation of competing gas exchange systems was carried out on the basis of a series production level 1.6 l naturally aspirated engine with direct fuel injection and a double cam phase adjuster. Besides the basic engine, an equally powerful turbocharged variant with significantly reduced displacement was taken into account. In addition, a stop-start system was considered for each variant.

CO_2 emissions in the new European Driving Cycle (NEDC) were selected as



❶ CO_2 emissions of various technology combinations in the NEDC the assessment criterion on basis of a vehicle in the 1300 kg weight class, ❶.

The simulation shows that a fully variable valve train system on the inlet side offers a CO_2 savings potential of up to 8.5 %. Even when using a downsized engine with a start-stop system, savings of up to 5 % are achieved. Step valve lift switching systems on the inlet side cannot achieve this potential since the design entails certain technical compromises.



❷ Univalve package situation (inlet and outlet)

LAYOUT AND FUNCTION OF THE UNIVALVE

The Univalve is a mechanically fully variable valve train system that enables infinitely variable lift variation from zero to maximum lift as well as simultaneous alteration of the valve lift duration.

Compared with numerous patent applications and actual systems, the Univalve displays a number of advantages with respect to production costs, system friction and functionality in the package, ❷. It should be stressed that integrating it requires only a minor shift in the position of the camshaft. The system is also cheap and easy to install. The valve lift duration at low lift levels can be shortened by up to 70° CA. Moreover, thanks to the rocker geometry of the switching lever, a high rpm ceiling exists.

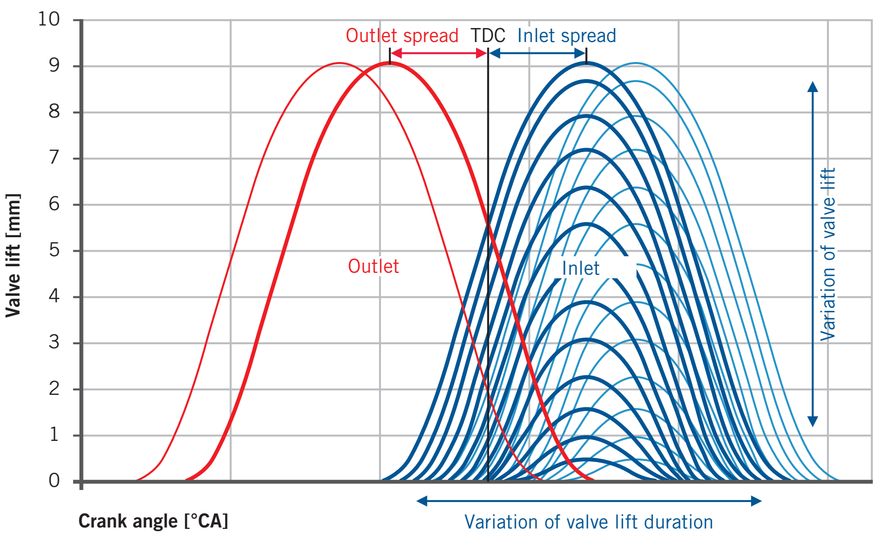
The basis of the system is a standard roller tip rocker arm. An intermediate lever, whose position is defined by the cam contour, a guide and the eccentric shaft (serving as a thrust bearing), transforms the oscillating motion induced by the retreating cam into the desired valve movement via a working contour, ❸.

Since all of the components and contact zones are mounted on roller or needle bearings, the friction moment at maximum lift can be kept at the level of typical systems featuring standard roller tip rocker arms. At low valve lift levels and low loads, friction benefits arise which, in addition to de-throttling, enable the reduction in fuel consumption ob-served in the NEDC. The favorable friction performance also makes it predestined for installation on the outlet side, even with large valve lifts.

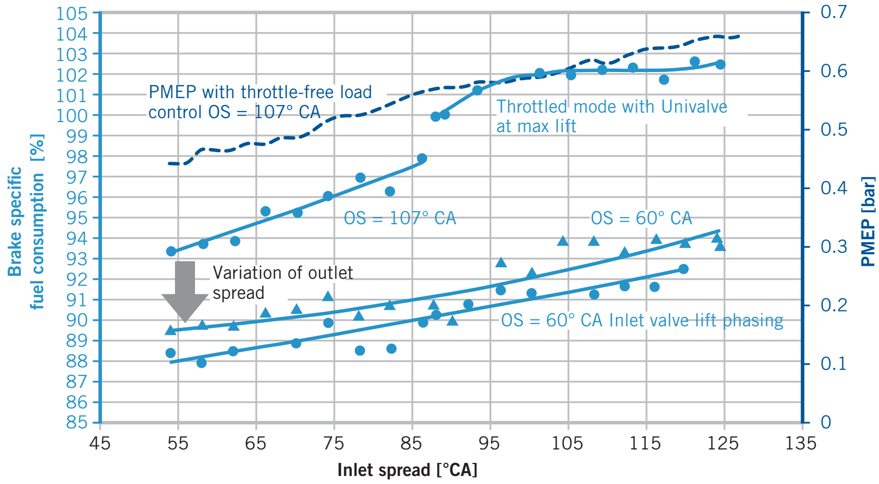
❹ depicts the valve lift curves generated by the rotation of the eccentric shaft. Thermodynamically speaking, the variation of the valve lift duration is decisive. Together with a camphasing system cylinder filling can be limited without dissipative intervention through a strategy of “early inlet valve closing”. The mechanically and acoustically important ramping zones for all lift curves are thus maintained.

Electronically commutated, non-contact electro-motors are used for actuating the eccentric shaft. The design and functional

❸ Univalve assembly



❹ Typical valve lift curves with Univalve



❺ Brake specific fuel consumption at part load of a turbocharged 2.0 l gasoline engine with throttle free load control: n = 2000 rpm, pme = 2 bar; MPI + turbocharger

principle of these drives enables very high performance density coupled with strong resistance to wear during prolonged operation.

Similar actuators are already used today in demanding fields such as commercial vehicle engines, and can thus be produced inexpensively. Closed-loop control of the actuators is based on detection of the position of the eccentric shaft by a contactless sensor, and also uses the rotor's position sensors, which are in any case necessary for operating the actuator. A defined fail-safe position for the eccentric shaft is likewise achievable.

RESULTS IN A GASOLINE TURBO-POWERED ENGINE WITH DIRECT FUEL INJECTION

In an engine with fully variable valve lift system, the torque or filling of the cylinder is controlled by adjusting the inlet valve lift duration (control period), as is the inlet and outlet spread. These parameters influence the charge cycle work which has a direct influence on fuel consumption. This declines steadily when the inlet valve is closed early, ❺. In order to obtain very early closure of the inlet valve, the valve lift duration has to be set for as short a time as possible, which, with a mechanical variable valve lift system, can only be achieved with a small valve lift. At n = 2000 rpm and bmep = 2 bar, the valve lift amounts to approximately 1.2 mm with a valve lift duration of 120° CA. The lowest level of charge cycle work, and thus lowest brake specific fuel consumption, is attained with an inlet spread of IS = 50° CA and an outlet spread of OS = 60° CA. The outlet spread also substantially influences charge cycle work and fuel consumption, and also the internal residual gas fraction, ⑤.

Variable valve train systems that lack the ability to vary the outlet spread are thus at a substantial disadvantage when it comes to reducing fuel consumption.

At an operating point of n = 2000 rpm, it proved possible to reduce fuel consumption of 9 % in a turbocharged 2.0 l four-cylinder gasoline engine with direct injection by throttle free load control compared with a throttled engine, and by an even more impressive 12 % with a port fuel injection engine [1].

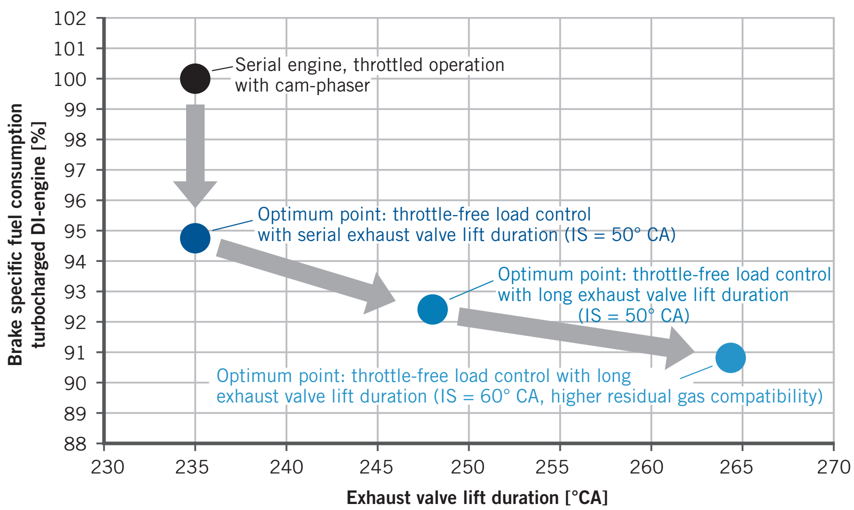
In the process, moreover, residual gas compatibility was successfully increased through an inlet valve lift phasing.

Whereas increased residual gas volume in the partial load has the effect of reducing fuel consumption, with full load operation it is undesirable owing to the increased risk of knocking and reduced cylinder filling.

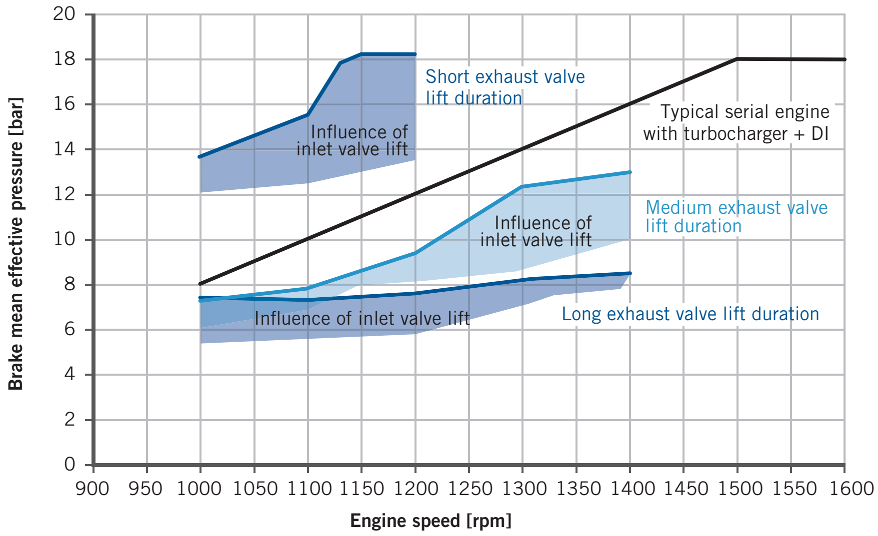
Gas exchange in modern turbocharged four-cylinder downsizing concepts with mono-scroll turbochargers constitutes something altogether different. These engines feature short exhaust valve lift duration control in order to avoid crosstalk between the neighboring cylinders in the firing order. The use of twin-scroll turbochargers, which likewise addresses this problem, is hampered by higher costs and the development of integrated exhaust manifolds [2, 3].

This parameter presents a challenge for throttle-free partial load operation. With a shorter outlet valve lift duration a cam phase adjustment on the outlet side leads to an increase of the internally residual gas fraction but it also results in an undesirable increase in charge cycle work due to higher expulsion losses. Thus, in terms of fuel consumption, the advantage due to throttle free load control is limited, ❻.

Using a mechanical fully variable valve train on both the inlet and outlet sides represents a possible solution for achieving optimum fuel consumption in the partial



❻ Fuel consumption at part load with throttle-free load control and additional fully variable outlet side: n = 2000 rpm, pme = 3 bar; DI + turbocharger



❼ Full load performance

load along with optimum lowend torque. In addition, it makes it possible to set the outlet opening time so as to achieve minimal expansion losses and cylinder expulsion with simultaneous residual gas and consumption-optimized outlet closure. With short outlet camshafts typically found in downsized engines, the additional fuel consumption advantage compared to throttle-free operation can be as high as 4 %, depending on the load point and residual gas compatibility, ⑥.

Regardless of the outlet valve lift duration, full variability on the outlet side has a positive influence on full load performance. The freely selectable inlet valve lift duration enables filling-optimized inlet closure, which means the full load torque in a turbocharged gasoline engine with direct fuel injection in the very low rpm zone (without scavenging) can be increased by approximately 10 % compared with serial engines [1].

In operating points which enable targeted scavenging, a demand-driven prolongation of the inlet valve lift duration and the accompanying increase in the valve overlap, with simultaneous optimum inlet closure, can have a distinctly positive effect, ❼. With fully variable inlet valve system, this enables considerable increases in torque at low rpm levels.

An additional fully variable outlet valve lift system permits adjustment of the outlet valve lift duration to the respective operating state, and thus has a major influence on engine torque.

A long outlet valve lift duration, which in partial load leads to lower fuel consumption owing to the optimized residual gas control, is disadvantageous for full load performance of a turbocharged four-cylinder gasoline engine, ⑦.

Through a massive reduction in the outlet valve lift duration compared to current serial applications, which display a substantial disadvantage with partial load fuel consumption, full load performance can be improved significantly. With a shorter outlet valve lift duration, engine torque at n = 1000 rpm could be nearly doubled. Even at n = 1150 rpm, a brake mean effective pressure of bmep = 18 bar is achieved.

Thus, especially with turbocharged four-cylinder engines, the combination of fully variable inlet and outlet valve lift systems offers an optimum means of addressing the conflicting goals of lower fuel consumption, high low-end torque and improved full-load performance.

SUMMARY

The Univalve fully variable valve train system on the inlet side enables a reduction of up to 12 % in partial-load fuel consumption through throttle-free load control, friction optimization and well-aimed residual gas control. Furthermore, given a demand-oriented design of the eccentric camshaft contour, the Univalve is able to provide a valve lift phasing (asymmetric valve lift) to generate charge motion for an additional reduction in fuel consumption by optimized residual gas compatibility.

The full-load performance of downsizing concepts is improved considerably through careful adjustment of the valve overlap and filling-optimized closure of the inlet valves with the Univalve. Heightened low-end torque can be transformed into additional fuel savings by additional downspeeding.

Additional potential in partial and full load can be exploited by installing the Univalve on the outlet side. Meriting particular mention here is the special case of the four-cylinder turbocharged engine, in which the conflicting goals of lower partial load fuel consumption and higher low-end torque can be successfully resolved.

Univalve thus represents an effective means of meeting the requirements for greater efficiency and improved torque performance in future engine concepts.

OUTLOOK: SOLUTIONS FROM A SINGLE SOURCE

The fully variable valve train Univalve is a new product in the Kolbenschmidt Pierburg AG line-up, complementing the Group's existing portfolio. Thus, in accordance with customer requirements, it is possible to supply either the Univalve valve train system separately or together with complete, ready-to-install cylinder heads.

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EFFICIENT DOWNSIZING FOR FUTURE GASOLINE ENGINES

Gasoline engine downsizing is firmly established as one of the main technologies for achieving fuel consumption and CO_2 reduction targets, with increasing degrees of downsizing being applied in the market place. With advanced downsizing concepts a fuel consumption reduction of 30 % can be achieved, as shown by the Mahle downsizing demonstrator engine with 1.2 l displacement for a 50 % downsizing [1, 2].

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CHALLENGES

The challenges to realise a high downsizing, together with the best possible efficiency for NEDC and real world fuel economy are:

: high low end torque and good transient response at low speeds
: optimisation of knocking combustion at high loads to realise a high compression ratio and a good combustion, as a prerequisite for high efficiency and high low end torque
: further reduction of pumping losses at part load
: a small as possible over fuelling area at high loads and speeds (for thermal protection of the exhaust side).

The related layouts and technologies to achieve further advanced downsizing will be discussed in the following topics.

GASOLINE DIRECT INJECTION AND COMBUSTION CHAMBER LAYOUT

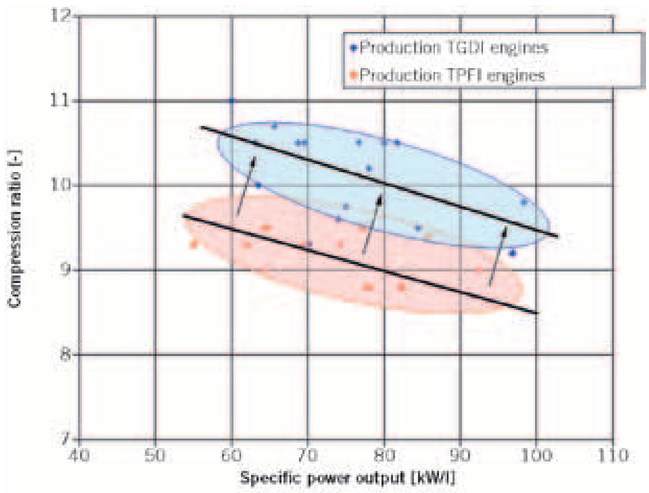
In order to achieve the best improvement in fuel economy the compression ratio needs to be maintained as high as possible.

This requirement has led to the widespread introduction of Gasoline Direct Injection (GDI) specifically for pressure charged engines. Injecting the fuel into the combustion chamber, as opposed to the intake port, reduces the charge temperature in cylinder through the latent heat of vaporisation of the fuel. This has the combined effect of both increasing the charge density (and hence volumetric efficiency) and also improving the knock limit. As a consequence the compression ratio can be increased by approximately one unit for an equivalent specific power output, ❶.

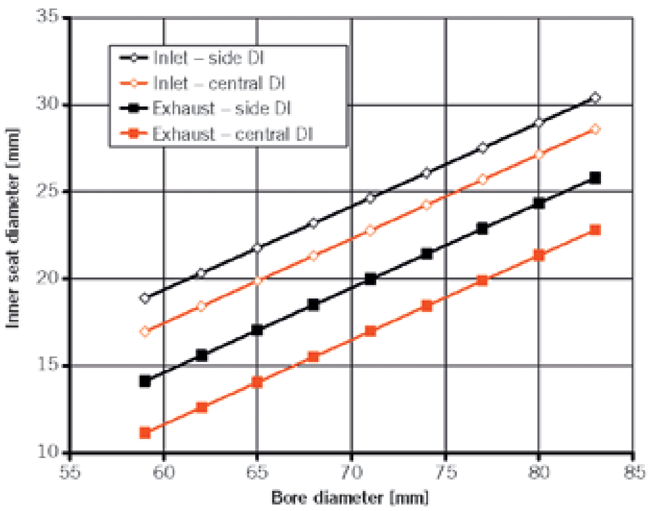
Until recently the majority of production GDI combustion systems have been side injection. There is now however a gradual move to central spray-guided direct injection systems. ❷ shows a typical view of the two different combustion system layouts.

The main benefits of the spray-guided GDI combustion system are largely possible due to the injector location next to the spark plug. This allows more accurate air fuel mixture control within the combustion chamber and at the spark plug through the variation of injection timing. Further benefits are achievable when multiple injections are used. Spray-guided combustion systems can be used with both solenoid multi-hole and outward opening piezo injectors. Thus, the knock limit at full load can be improved, and it is also an enabler to run stratified lean.

As can be seen in ②, packaging both the injector and the spark plug in the centre of the combustion chamber will influence the combustion chamber layout significantly. The critical requirement is, specifically for high downsizing, to maintain sufficient cooling around both the spark plug and the injector tip. The Mahle 1.2 l downsizing engine (with spray-guided GDI) was therefore designed with a nearly complete coolant flow in this critical area [2].



❶ Compression ratio of pressure charged engines against specific power output

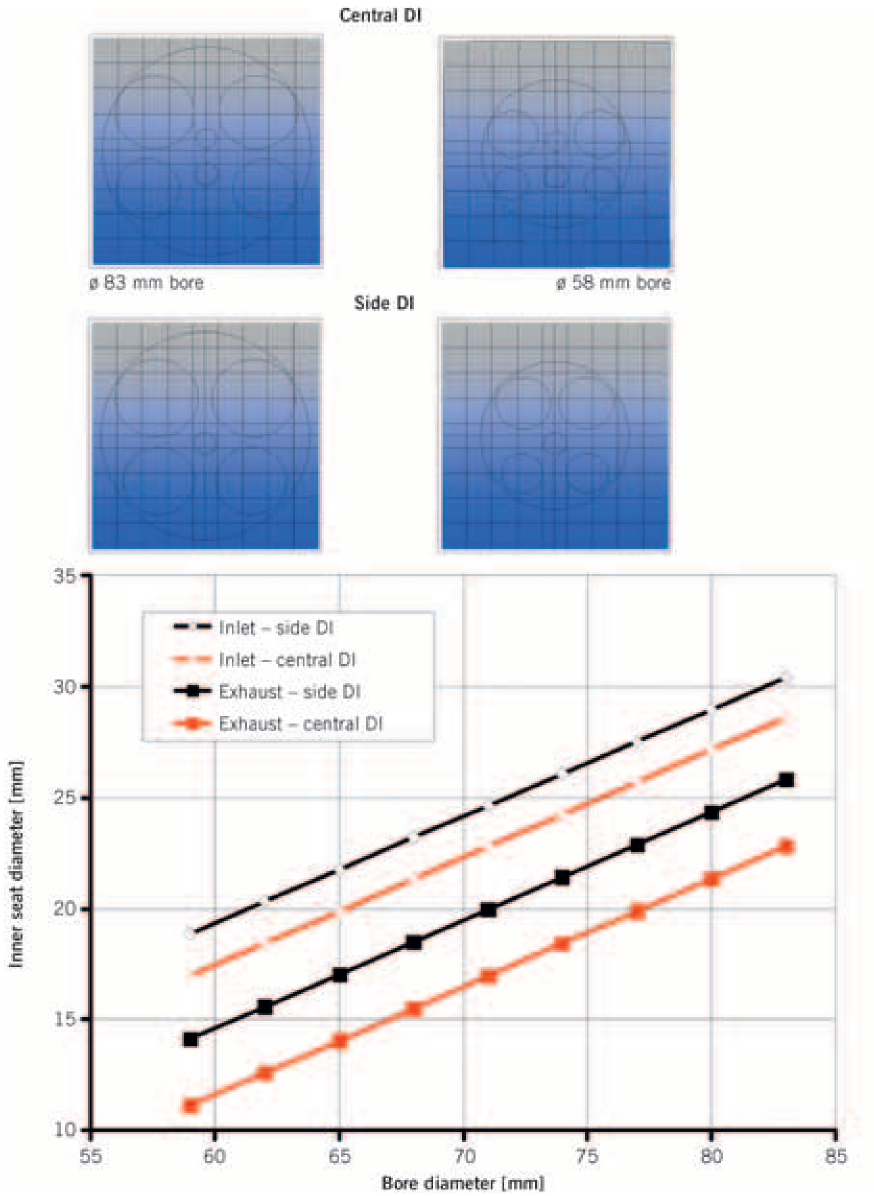


❷ Typical side and central GDI combustion system layout

The injector layout also influences the possible valve sizes (and thus the achievable specific power output), depending on the bore size (and thus the cylinder displacement). In order to investigate the potential of reducing bore size an analytical study by using 1D-cycle simulation has been carried out for a turbocharged three-cylinder engine with a displacement of V_H = 0.8 l. Target performance was set to achieve in excess of 21 bar BMEP, with a specific power output of 85 kW/l. Typical side and central injection combustion systems were designed, with valve sizes optimised for the given bore size. The central injection system was designed to ensure sufficient cooling around both the injector and spark plug. Bore diameter was swept from 58 mm to 83 mm. ❸ shows the differences in valve size with bore diameter and demonstrates clearly the offset between the combustion chamber configurations.

1D-cycle simulation with GT Power was carried out at the peak power engine speed (6000 rpm) using the same combustion chamber geometries. At each of the speed sites a sweep of BMEP was carried out. ❹ shows the plots of specific power and compressor pressure ratio for both injector locations.

With the central direct injection system the point at which the full load performance can no longer be achieved occurs earlier, limiting the absolute specific output of the engine. This is also reflected in the higher compressor pressure ratio required in order to overcome the increased restriction, which is also manifested in an increase in exhaust residuals, having a potential knock-on effect of increasing the knock sensitivity. However, even in this scenario it can be seen that it is possible to achieve 85 kW/l with the central GDI combustion system down to 66 mm bore diameter (which would give a stroke of 78 mm and an under-square bore/stroke ratio of 0.85). It highlights the need to carefully consider the combustion chamber design, particularly with small bore diameters, if the advantages of a central GDI combustion system are to be realised.



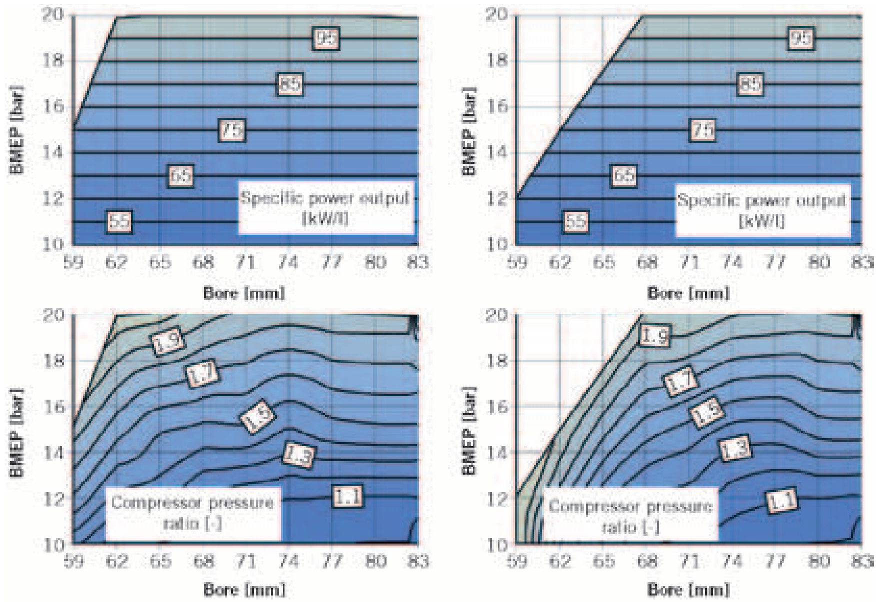
❸ Valve size diameter against bore diameter

VARIABLE VALVE TRAIN

Downsizing concepts normally use intake or dual Variable Valve Timing (VVT) to provide further improvements when combined with pressure charging and GDI. These technologies allow large valve overlaps to be run in the low speed high load region, allowing scavenging of fresh air from the intake to the exhaust, improving low speed torque. This is not possible on port fuel injection engines as it results in unburnt air/fuel mixture being drawn into the exhaust, increasing fuel consumption, HC emissions and exhaust temperatures and potentially damaging the catalysts. GDI engines do not have this restriction as the fuel can be injected after exhaust valve closing, allowing increased overlaps to be run. This results in increased scavenging, which improves the knock threshold of the engine, enabling a higher compression ratio and a higher BMEP at low engine speeds.

The Mahle Cam-In-Cam technology [3, 4] offers further options for VVT. The relative movement of two cam lobes of the same camshaft can be controlled by a cam phaser (event phasing), moving the inner part of the camshaft against the outer shaft. When this is combined with a second phaser both position and effective valve opening duration can be varied.

The presented engine results were measured with a pressure charged four-cylinder engine with GDI (displacement V_H = 1.4 l). Compared to the series production layout with supercharger and turbocharger the



❹ Effect of bore diameter on engine performance at 6000 rpm with side and central GDI

engine was operated without the supercharger. It has already been demonstrated [4] that turbo charging and Cam-In-Cam on the exhaust side provides similar low-end torque compared to turbocharger and supercharger. This increase in low-end torque is realised through a shorter exhaust valve opening, compared with series production, resulting in increased exhaust pulse separation.

Investigations have been carried out with Cam-In-Cam on the intake side in order to achieve lower fuel consumption; together with the series production Compression Ratio of (CR) = 10.0 also an increased CR of 10.5.

Late Intake Valve Closing (LIVC) operation provides a number of possibilities for enhancing engine efficiency through the decoupling of the compression and expansion ratio of the engine; they are as follows:

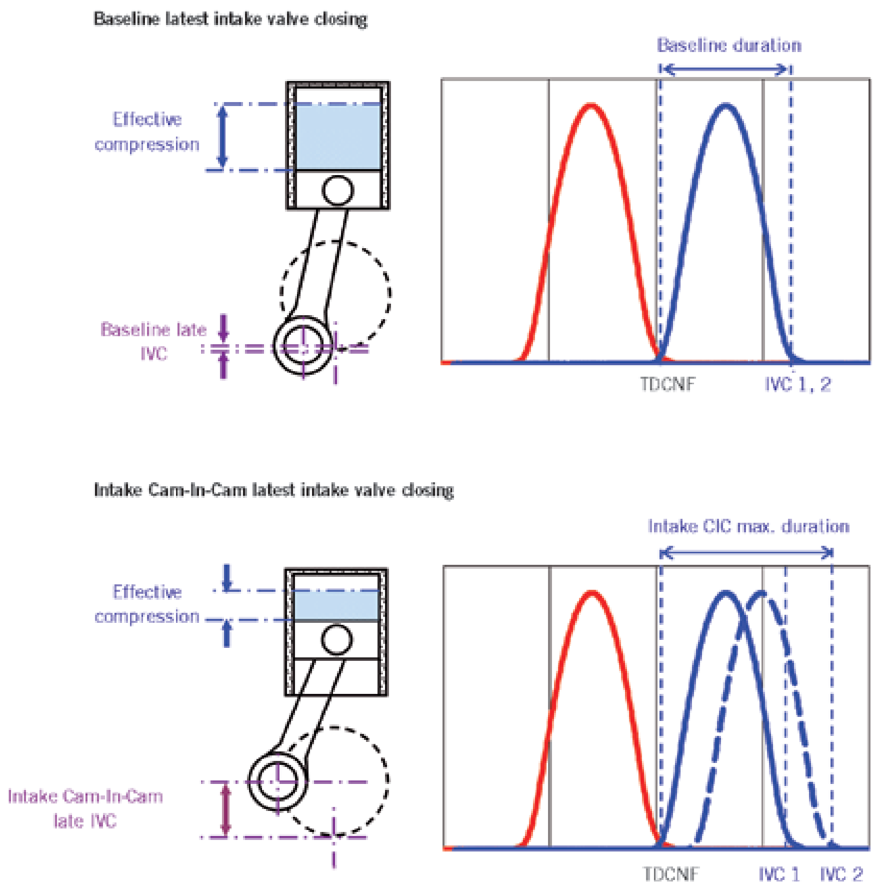
- Low Loads: The intake valve is closed late in the cycle, after BDC and excess charge is returned to the intake manifold, thus de-throttling the engine and reducing pumping work.
- High Loads: LIVC operation allows a reduction in effective compression ratio, thus mitigating knocking combustion and allowing combustion phasing to be advanced. A further benefit is that the compression work is transferred from the cylinder to the turbocharger compressor and charge cooled as an intermediate stage. This results in a reduction in cylinder temperature at the point of ignition, providing further knock mitigation.

❺ shows the function of the intake CamInCam to reduce the effective compression ratio.

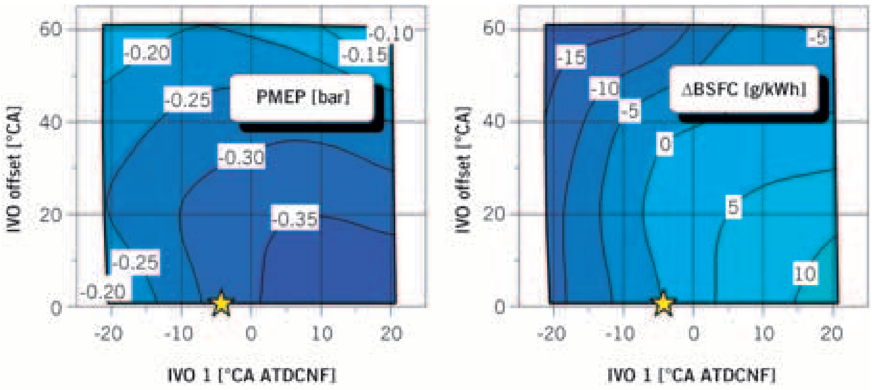
PART LOAD RESULTS

A benefit of gasoline downsizing is that it uses load point shifting to reduce pumping work at a given engine power output, therefore improving fuel consumption. However, despite this effect pumping work is still a significant portion of the indicated work at lighter loads and at 1000 rpm 4 bar accounts for 6.5 % of the Gross indicated mean effective pressure (IMEPg).

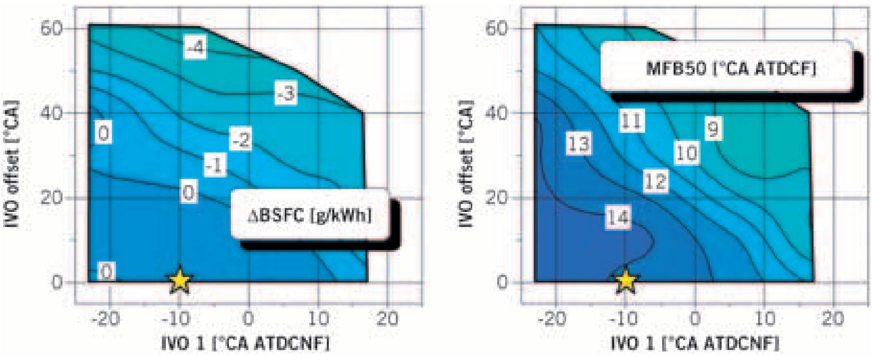
Intake Cam-In-Cam offers a further opportunity to reduce the pumping losses by LIVC; pushing cylinder charge back into the intake after bottom dead centre, resulting in an increase in intake manifold pressure and thus de-throttling of the engine. ❻ shows for the operating point 1000 rpm 4 bar BMEP and the reduction in pumping losses that is achievable with LIVC operation, with losses reduced from -0.31 bar PMEP to -0.10 bar, ⑥ (left). Through



❺ Mahle intake Cam-In-Cam: reduction in effective compression ratio through “Late Intake Valve Closing” (LIVC)



❻ 1000 rpm 4 bar BMEP: pumping work (left) and fuel consumption variation (right)



❼ 2500 rpm 10 bar BMEP: fuel consumption variation (left) and combustion phasing (right)

this BSFC can be reduced by 5 g/kWh, ⑥ (right). De-throttling can also be achieved with an increase of the valve overlap, resulting in a BSFC reduction of 15 g/kWh. If intake valve closing happens too late then the lower effective compression ratio results in reduced charge temperatures at the point of ignition, reducing combustion efficiency and working against the fuel consumption reduction from de-throttling.

KNOCKING COMBUSTION AND FUEL CONSUMPTION AT HIGHER LOADS

Intake Cam-In-Cam allows a reduction in effective compression through LIVC operation. This results in an improvement in the engine-knocking threshold at higher engine loads. ❼ shows operation of the engine at 2500 rpm 10 bar BMEP for a geometric compression ratio of 10.5 with varying intake cam timing and offset between the two intake cam valves. The yellow mark shows the series production operating point. As can be seen in the picture intake Cam-In-Cam allows a fuel consumption reduction of 4 g/kWh, ⑦ (left), which is a result of the improved combustion phasing, ⑦ (right) at the lower effective compression ratio.

❽ shows the intake Cam-In-Cam knock limited combustion phasing regions for two geometric CR of 10.0 and 10.5 at 2500 rpm 10 bar BMEP. The increase in geometric CR results in larger region of knock limited combustion. However, through LIVC operation, enabled by intake Cam-In-Cam, the effective compression ratio can be reduced to allow the engine to be run at optimum combustion phasing.

Overall a 3 % reduction in fuel consumption was achieved through the use of intake Cam-In-Cam at this real world fuel economy site.

PRESSURE CHARGING

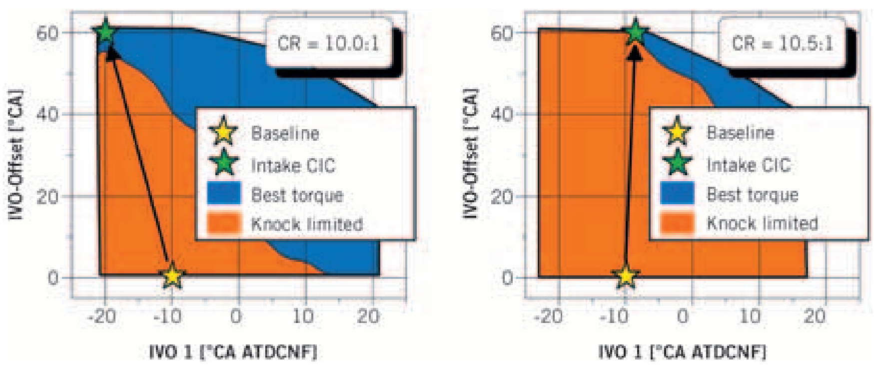
Pressure charging is, along with GDI and variable valve timing, the key technology of advanced downsizing. ❾ shows schematics of four different engine boosting systems:

: single turbocharger
: two-stage turbocharger
: supercharger & turbocharger
: electric supercharger & turbocharger.

Low end torque and transient performance are key for the acceptance of advanced downsizing. Therefore these four concepts were assessed in detail, based on the Mahle 1.2 l downsizing engine.

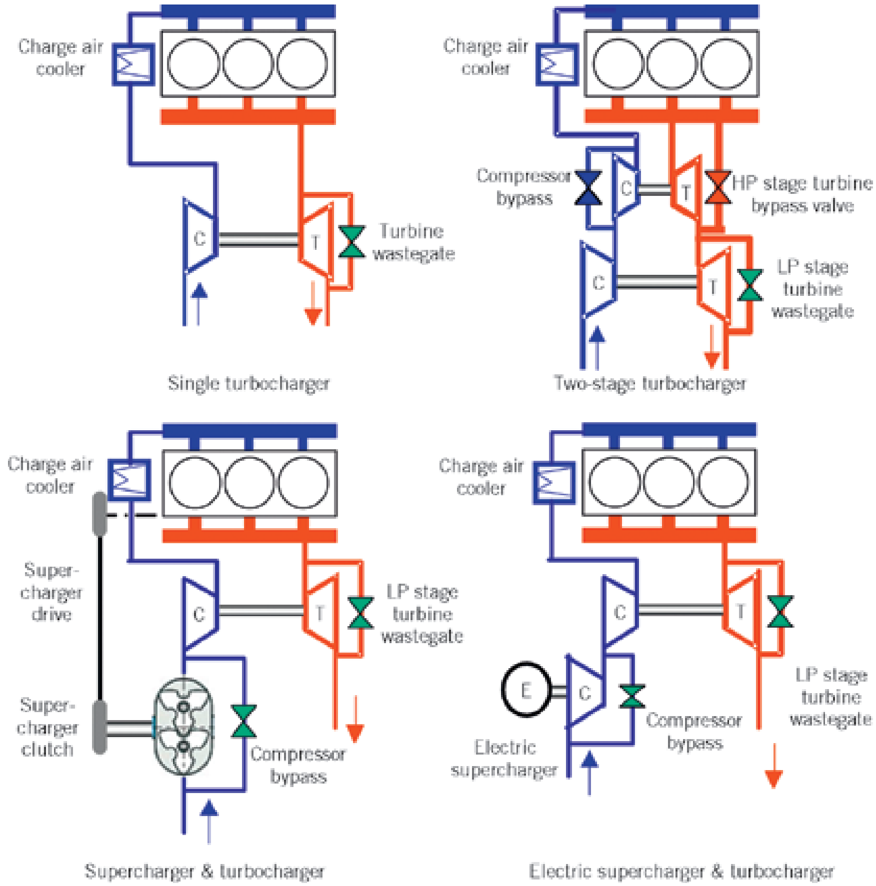
In ❿ the full load BMEP for both single and two-stage turbocharger configurations are compared. The single turbocharger variant is showing excellent full load performance (30 bar max. BMEP and 100 kW/l specific power output; [1]). Nevertheless, the results with the two-stage turbo demonstrate the potential to increase both low end torque (in the range between 1000 and 1500 rpm) and specific power output (from 100 kW/l to 120 kW/l).

To assess the transient performance load steps were carried out at a range of different speed sites. The engine load was set to 2 bar BMEP, the throttle opened to Wide Open Throttle (WOT), and the response of the engine was measured. ⓫ shows the response of indicated Mean Effective Pressure (IMEP) for the transient load step at 1250 rpm for different



❽ 2500 rpm 10 bar BMEP: intake Cam-In-Cam knock limited combustion phasing region for geometric compression ratio of 10.0 (left) and 10.5 (right) configurations (two-stage turbocharger, low pressure turbocharger only of the two-stage system, low pressure turbocharger with electric supercharger and, as a reference, a supercharged/turbocharged engine).

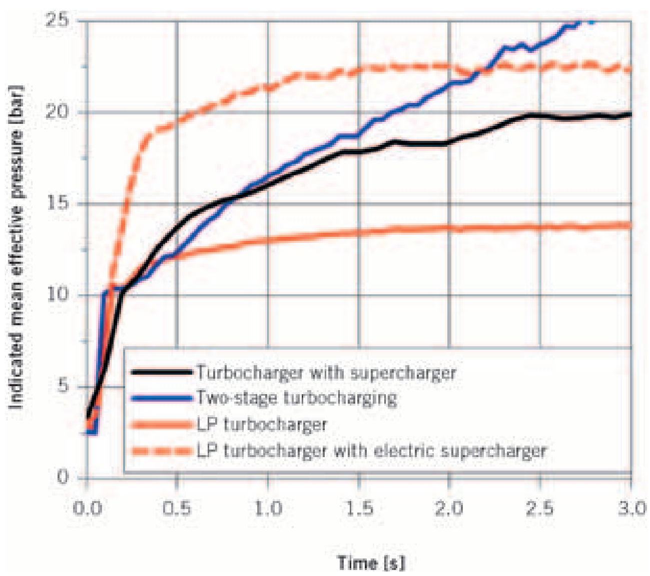
The two-stage turbocharger system achieves a similar transient response as the reference configuration of a mechanical supercharger. If only a low pressure turbocharger is used then the transient response is not acceptable. However, this can be significantly improved by the addition of an electrical supercharger, resulting in the best transient response behaviour of the variants investigated.



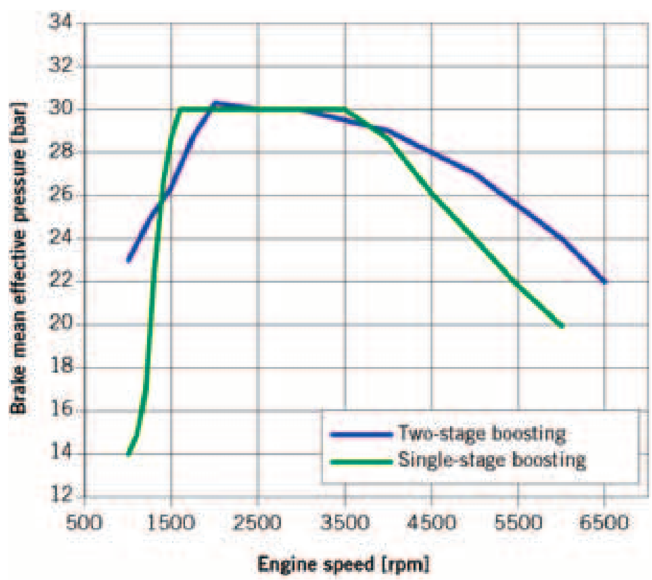
❾ Boosting system schematics

EXHAUST GAS COOLING AND EXHAUST GAS RECIRCULATION

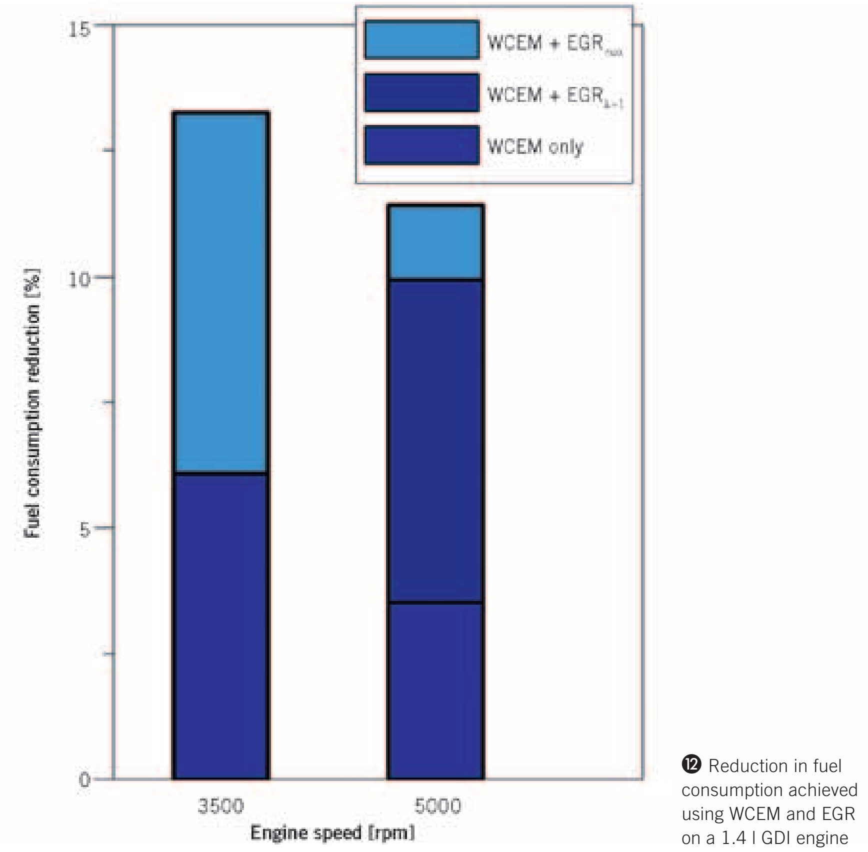
Turbocharged gasoline engines normally require over fuelling at high speeds and loads for a thermal protection for the components on the exhaust side. This results in a deterioration of fuel consumption, engine-out emissions and the efficiency of the three-way catalyst. Exhaust gas cooling using a Water-cooled Exhaust Manifold (WCEM) or the implementation of cooled external Exhaust Gas Recirculation (EGR) are potential technologies to avoid or reduce this over fuelling. A WCEM is a direct approach to reduce over fuelling by cooling the exhaust gas before entering the turbine of the turbocharger. With an EGR system cooled exhaust gas is fed back to pre or post compressor, increasing the specific heat of the cylinder charge. This results in a reduction of peak cylinder pressure and temperature, and thus in a reduction of the gas temperature when the exhaust valve is opened. ⓬ shows the reduction in fuel consumption by using WCEM and EGR on a 1.4 l GDI engine.



⓫ Transient load step comparison at 1250 rpm (IMEP against time)



❿ BMEP at WOT with single and two-stage turbocharger (Mahle 1.2 l engine)



⓬ Reduction in fuel consumption achieved using WCEM and EGR on a 1.4 l GDI engine

At 3500 rpm WOT WCEM results in a 6 % improvement in fuel consumption, and when it is combined with EGR it shows a 13 % fuel consumption improvement. At 5000 rpm WOT the fuel consumption improvement is 3 % for WCEM and an additional 7 % for EGR. Increased EGR rate could allow a further 1 % improvement.

CONCLUSION

Gasoline engine downsizing is firmly established as one of the main technologies for achieving fuel consumption and CO_2 reduction targets, with increasing degrees of downsizing being applied in the market place. With advanced downsizing concepts a fuel consumption reduction of 30 % can be achieved, as shown by the Mahle downsizing demonstrator engine with 1.2 l displacement for a 50 % downsizing [1, 2].

The challenges to realise a high degree of downsizing together with the best possible efficiency for both NEDC and real world fuel economy are:

- high low end torque and good transient response at low speeds
- optimisation of knocking combustion at high loads to realise a high compression ratio and a good combustion, as a prerequisite for high efficiency and high low end torque
- further reduction of pumping losses at part load
- minimisation of the over-fuelling area at high loads and speeds (for exhaust component protection).

Gasoline Direct Injection (GDI), compared to port fuel injection, allows an increase of the compression ratio by approximately one unit. A central injector location in comparison to a side location allows more accurate air fuel mixture control within the combustion chamber and at the spark plug. It is also beneficial for reducing the oil dilution with fuel.

Variable valve timing options can be increased by the use of Mahle Cam-In-Cam. Late Intake Valve Closing enables fuel consumption reduction in the part load range by further de-throttling. A reduction of the effective compression ratio at higher loads also allows an improvement of the knocking combustion.

The Mahle 1.2 l downsizing engine demonstrates that a high low end speed torque can be achieved with a single turbocharger. Further improvements in low end speed torque and transient response can be enabled by a two-stage turbocharger system or by the addition of an electric supercharger.

At high speed and loads a minimisation of the over-fuelling area (for exhaust component protection), thus a reduction in the real world fuel consumption, can be achieved by a water-cooled exhaust manifold or the implementation of cooled external exhaust gas recirculation.

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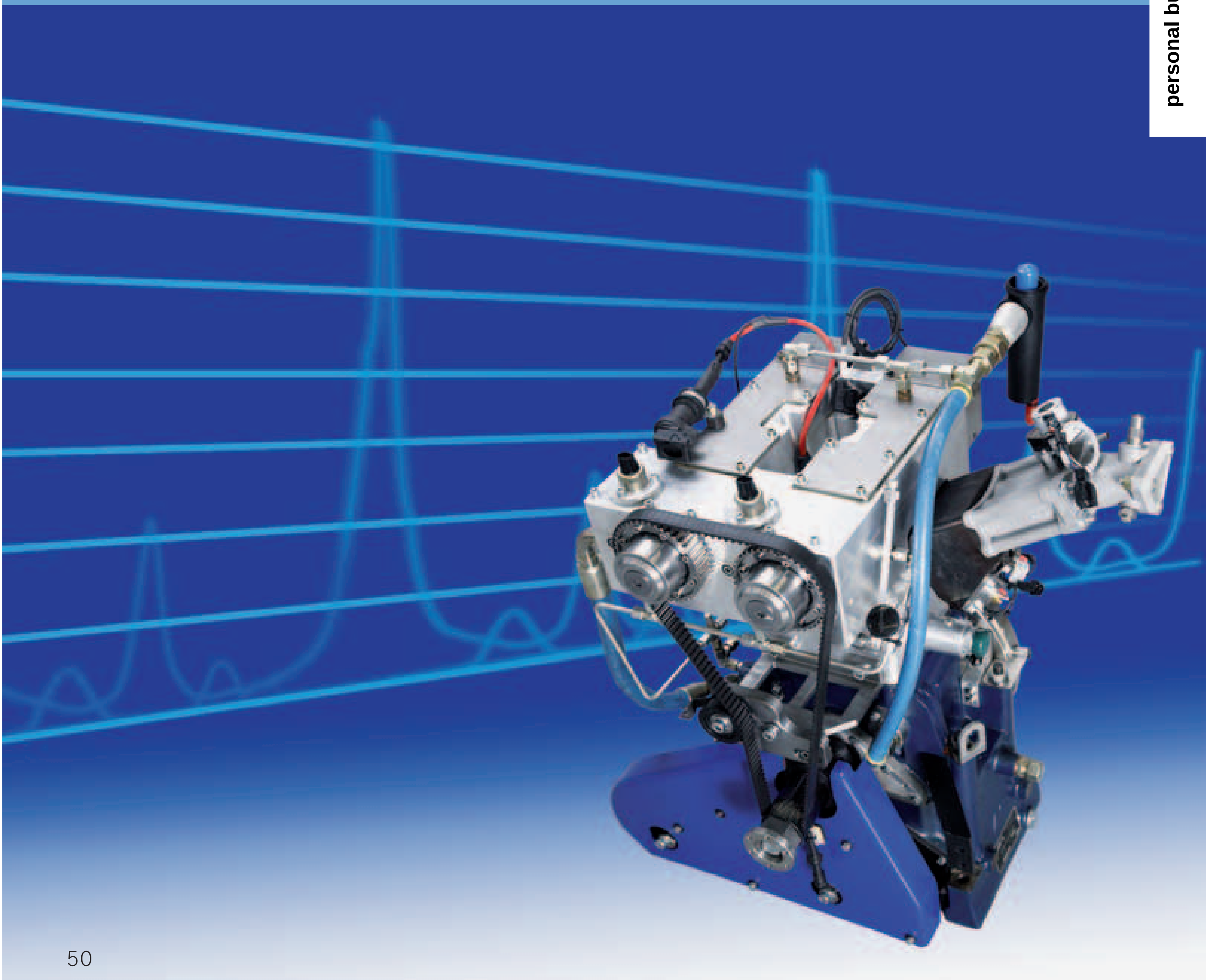
MIXTURE FORMATION WITH A GASOLINE HCCI ENGINE

Efficient Dynamics is the mission statement of Powertrain Development at BMW Group. The development work currently taking place in the field of combustion engines is concentrating on increasing the efficiency of the combustion process combined with systematic upgrading of the powertrain. For future combustion engines, HCCI combustion could provide the opportunity for simultaneous reduction of both fuel consumption and exhaust emissions.



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THE HCCI COMBUSTION

The HCCI combustion aims to combine the advantages of modern diesel and gasoline combustion processes – lowest emissions and high cycle efficiency. For this purpose, a typically highly homogenized mixture of air, fuel and combustion products from the previous cycle is caused to auto-ignite by means of compression. As a result, combustion starts simultaneously at various points in the combustion chamber. However, in contrast to knocking combustion, HCCI combustion proceeds in a controlled way and without excessive mechanical or thermal load to the engine.

In order to achieve the temperature required for auto-ignition at the end of the compression stroke, the valve control strategy "Negative Valve Overlap" (NVO) was employed for the current investigation. This gas-exchange strategy is characterized by small valve lift and short opening duration on both the inside and the exhaust side. The low valve lift, in combination with an early closing time of the exhaust valve, represents a comparatively precise method to control the residual gas fraction retained within the combustion chamber [1]. These residual gases are compressed until gas exchange top dead centre "recompression", and subsequently expanded until the intake valves open. The cylinder pressure which results from this valve control strategy, as well as the intake and the exhaust valve lift, are illustrated in ❶.

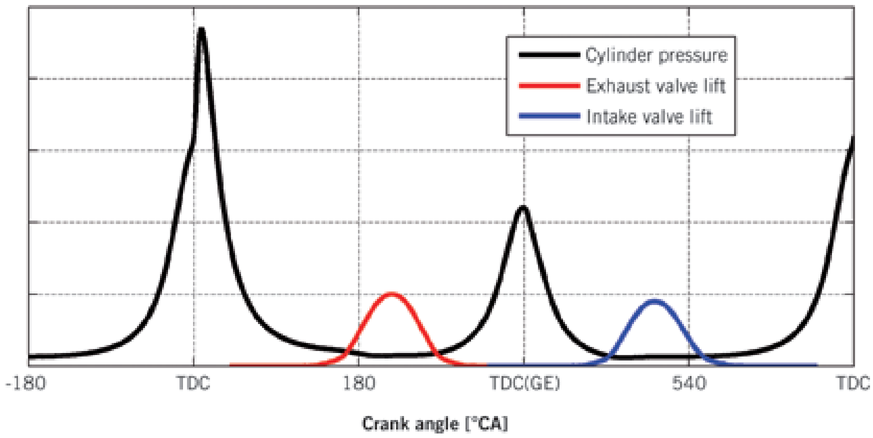
Mixture formation also has great influence on the HCCI combustion process. Earlier investigations have shown that with direct injection (DI), on the one hand the load limit for operation with auto-ignition can be lowered significantly [2], and on the other hand, it is possible to control the combustion timing by varying the injection timing [3]. In this context, however, it has to be considered that at low loads the fuel has to be injected during recompression. Although this leads to improved homogenization, it also results in increased pumping losses. Port fuel injection (PFI), on the contrary, has the advantage of providing excellent homogenization without any impact on the gas-exchange cycle. However, its disadvantage lies in the fact that there is no connection between the timing of fuel injection and the combustion phasing.

RESEARCH ENGINE

The experiments were carried out on a four-stroke single-cylinder research engine. Its geometry is based on the current BMW six-cylinder engine generation. ❷ shows the key data of the engine. For realisation of the flexibility required in the valve train, the engine has been equipped with variable valve timing (Vanos) and continuously variable valve lift (Valvetronic) on both the intake and the exhaust side. The engine is equipped with a multi-hole solenoid injector placed centrally in the roof of the combustion chamber (DI) and an injector in the intake manifold (PFI).

INFLUENCE ON VOLUMETRIC EFFICIENCY

In the case of direct injection, injection timing has a simultaneous influence on



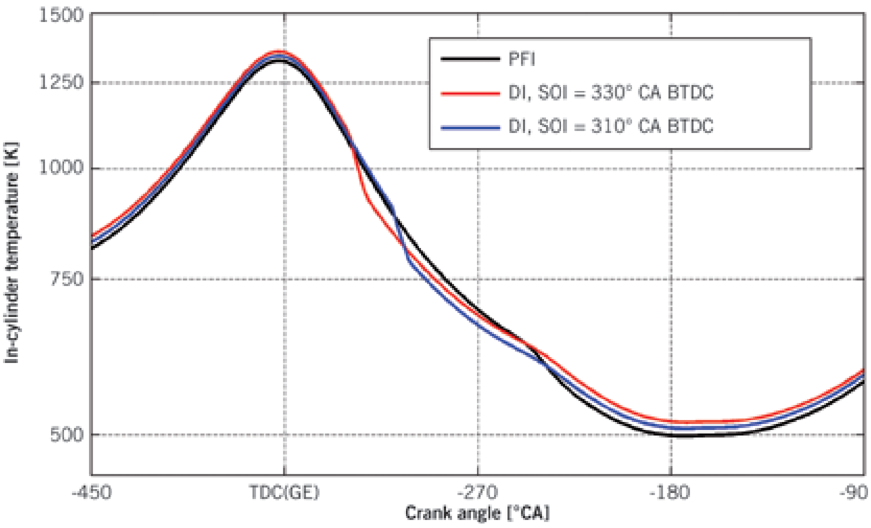
❶ Valve timing strategy and cylinder pressure several boundary conditions relevant to the combustion process. For example, injection timing influences the charging with fresh air, the in-cylinder temperature as well as the combustion phasing. In addition, fuel injection can be employed to generate stratification in the combustion chamber, which in turn, also influences the auto-ignition process [4].

❸ depicts the logarithmically applied trace of the in-cylinder temperature for a start of injection (SOI) of 330° CA BTDC and 310° CA BTDC with direct injection, respectively, as well as for port fuel injection. In the case of PFI, the fuel was injected into the intake manifold at 360° CA BTDC. With direct injection into the combustion chamber, engine charge is cooled down, since enthalpy of evaporation is absorbed from the area around the spray. At the same time, the injection of fuel results in a reduction of the polytropic exponent, with the temperature gradient turning out less steep during expansion than would be the case with pure residual gas. With advanced injection timing, this results in a higher in-cylinder temperature at intake valve opening (270° CA BTDC). In the subsequent induction phase, the higher temperature means that less fresh air is inducted, resulting in reduced dilution of the mixture. The following compression occurs at a higher temperature level. With port fuel injection, the mixture is cooled down more during the expansion phase of the gas-exchange cycle in relation to the measurements taken with direct injection. This results from the fact that, during expansion, the polytropic exponent is higher during the induction of the air-fuel mixture, which causes a steeper temperature gradient during this phase.

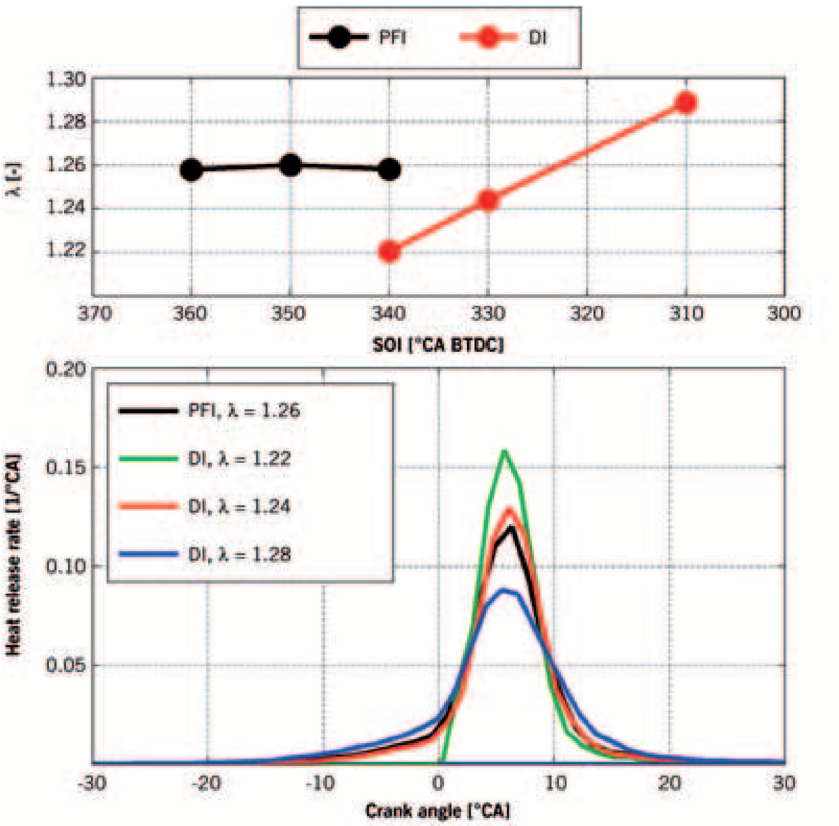
The significant influence of fuel injection timing on the air-fuel ratio in case of direct injection is illustrated in ❹. Advancing the start of injection leads to lower air-fuel ratio and reduced overall mass in the combustion chamber. As a result, the temperature is higher during compression and combustion than in the case of later injection timing. This, in turn, has considerable influence on whether the temperature required for auto-ignition is reached or not. The comparison of the heat release rates with identical timings of MFB50 (50 % mass fraction burnt), suggests that there is a strong correlation between peak conversion and air-fuel ratio. This

STROKE X BORE	90 mm x 84 mm
DISPLACEMENT	498 cm³
COMPRESSION RATIO	10.2:1
VALVE SPREAD INTAKE	80 – 160° CA ATDC(GE)
VALVE SPREAD EXHAUST	80 – 160° CA BTDC(GE)
VALVE LIFT INTAKE AND EXHAUST	0.3 – 9.7 mm
FUEL INJECTION SYSTEMS	Port fuel injection (p_{inj} = 5 bar) Direct injection (p_{inj} = 150 bar)
FUEL	Super plus gasoline RON 98

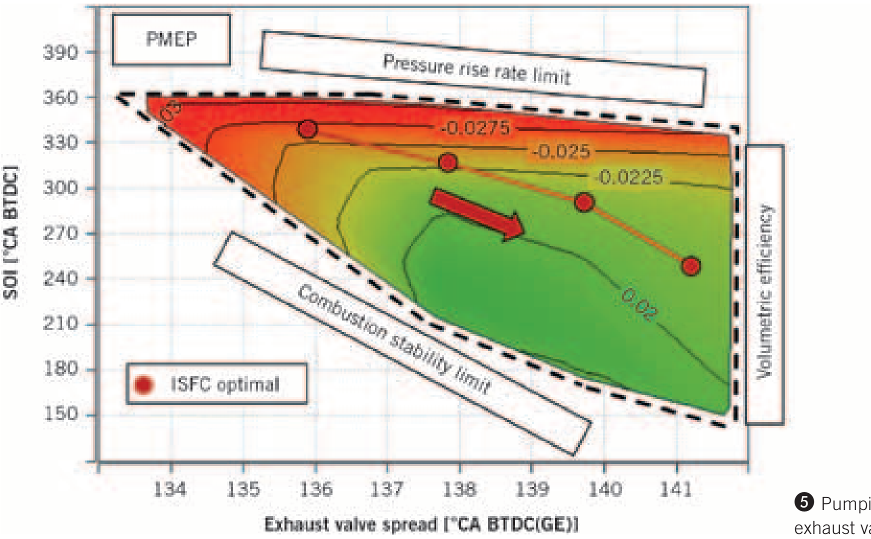
❷ Research engine specification



❸ In-cylinder temperature at 2000 rpm and 3 bar IMEP



❹ Air-fuel ratio and heat release rate for varying SOI at 2000 rpm and 3 bar IMEP



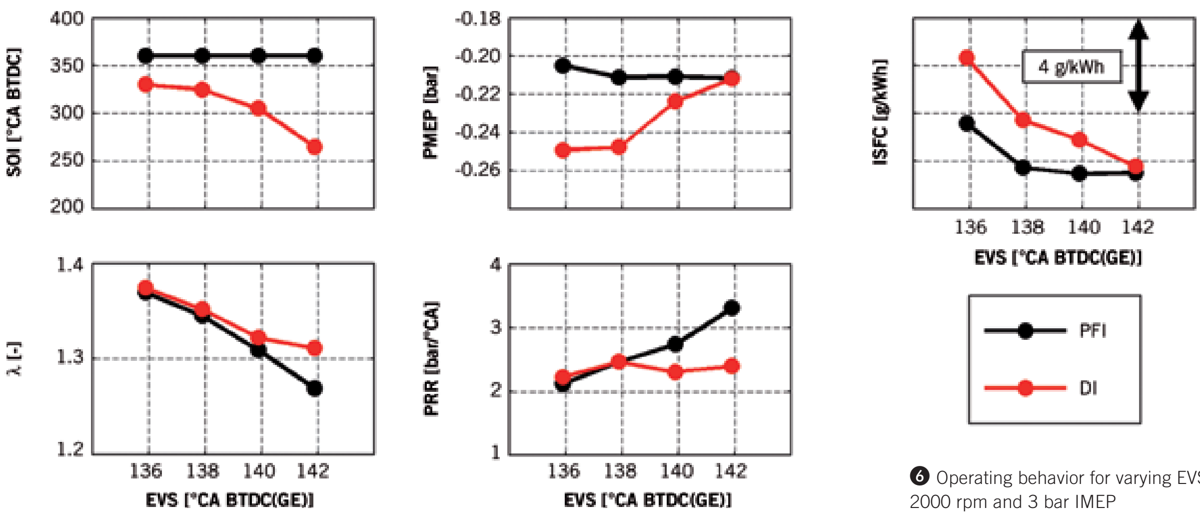
❺ Pumping losses for varying start of direct injection and exhaust valve spread at 2000 rpm and 3 bar IMEP

applies both to the operating points examined with direct injection and to the operating point with port fuel injection.

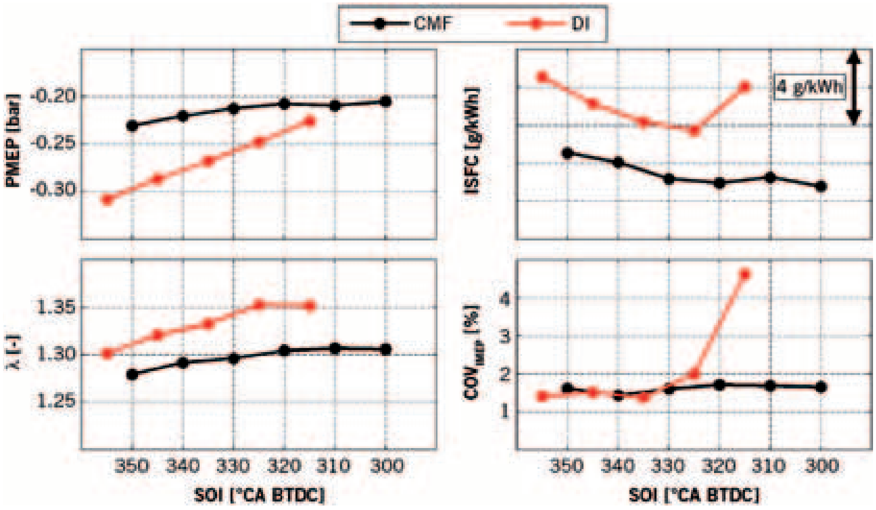
INFLUENCE ON PUMPING LOSSES

❺ shows the characteristic map of engine pumping losses in the case of direct injection for a variation of start of injection and exhaust valve spread (timing of maximum valve lift). The contour lines show that pumping mean effective pressure (PMEP) is decreased when valve spread is increased, which is due to the fact that fuel injection timing can be chosen significantly later. The earliest possible fuel injection timing is limited by the pressure rise rate (PRR). This limit is reached earlier when the exhaust valve timing is advanced. The latest possible fuel injection timing is determined by the decreasing stability of combustion (COVIMEP > 3 %). This limit is more sensitive to valve timing than the corresponding limit imposed by the pressure rise rate. Towards large valve spread, the characteristic map is limited by reduced volumetric efficiency and the resulting deterioration of overall efficiency.

❻ compares internal and external mixture formation. For direct injection, the operating points with optimum indicated specific fuel consumption (ISFC) from the characteristic map depicted in ⑤ are shown. As this evaluation shows, the direct injection timing of optimum indicated efficiency is delayed when the exhaust valve spread (EVS) is increased. In the case of port fuel injection, fuel consumption remains mostly constant despite the variation of valve timing. For both mixture formation processes, fuel consumption is equivalent at the largest exhaust valve spread shown. In the case of direct injection, this is due to the decreasing pumping losses which result from the retarded fuel injection timing. The different trend of air-fuel ratio clearly shows the influence of direct injection on charging with fresh air. With port fuel injection, the air-fuel ratio is only influenced by valve timing and thus the trapped mass of residuals, whereas in the case of direct injection, the injection timing has to be considered as an additional factor. The difference in air-fuel ratio between direct fuel injection and port fuel injection also significantly affects the pressure rise rate. The higher degree of dilution



❻ Operating behavior for varying EVS timing at 2000 rpm and 3 bar IMEP



❼ Operating behavior for varying SOI at 2000 rpm and 3 bar IMEP

with air leads to an increase of combustion duration and therefore lower peak combustion temperatures. The increased combustion duration causes that, in contrary to PFI, the value of the maximum rise in pressure rate remains almost unchanged with direct injection when the exhaust valve spread is increased.

COMBINED MIXTURE FORMATION

❼ shows a comparison between combined mixture formation (CMF) and direct injection at constant MFB50. For combined injection, the injected fuel mass was divided into two individual doses of 30 % and 70 % of the total mass. The smaller fraction was injected into the combustion chamber at the timing shown on the x-axis, while the remaining 70 % were injected into the intake manifold during the induction phase. For the measurements taken only with direct injection, the entire fuel mass was injected at the timing shown on the x-axis.

Considering the plot, it becomes evident that in comparison with direct injection, combined mixture formation results in reduced pumping losses and consequently reduced fuel consumption, despite the fact that injection occurs during recompression. Nevertheless, the air-fuel ratio can be influenced, which in turn allows direct control of combustion phasing. The strong effect of direct injection on volumetric efficiency also leads to significant cooling of the mixture, which causes COVIMEP to rise sharply in the case of retarded injection.

SUMMARY

Future engines with further reduced fuel consumption and ultra-low emissions require – among other measures – specific optimization of the engine process in order to minimize combustion and pumping losses. In this field, also the HCCI combustion process offers further opportunities for optimization. With the "Negative Valve Overlap" strategy, especially the timing of injection was identified as a significant influencing factor on pumping losses at medium and low load operating points. The aim of the experiments was to minimize these losses by injecting fuel into the intake manifold. It proved to be possible even when the valve timings remained unchanged. However, by adapting the valve spread as well as the injection timing, equivalently low pumping losses can be achieved for both direct injection and port fuel injection. The additional increase in charge mass with direct injection, however, has a positive effect on the pressure rise rate, which reduces the combustion noise and at the same time mechanical stress. In comparison with direct injection, the combined mixture formation method proved to be successful in combining the specific advantages of both direct injection and port fuel injection.

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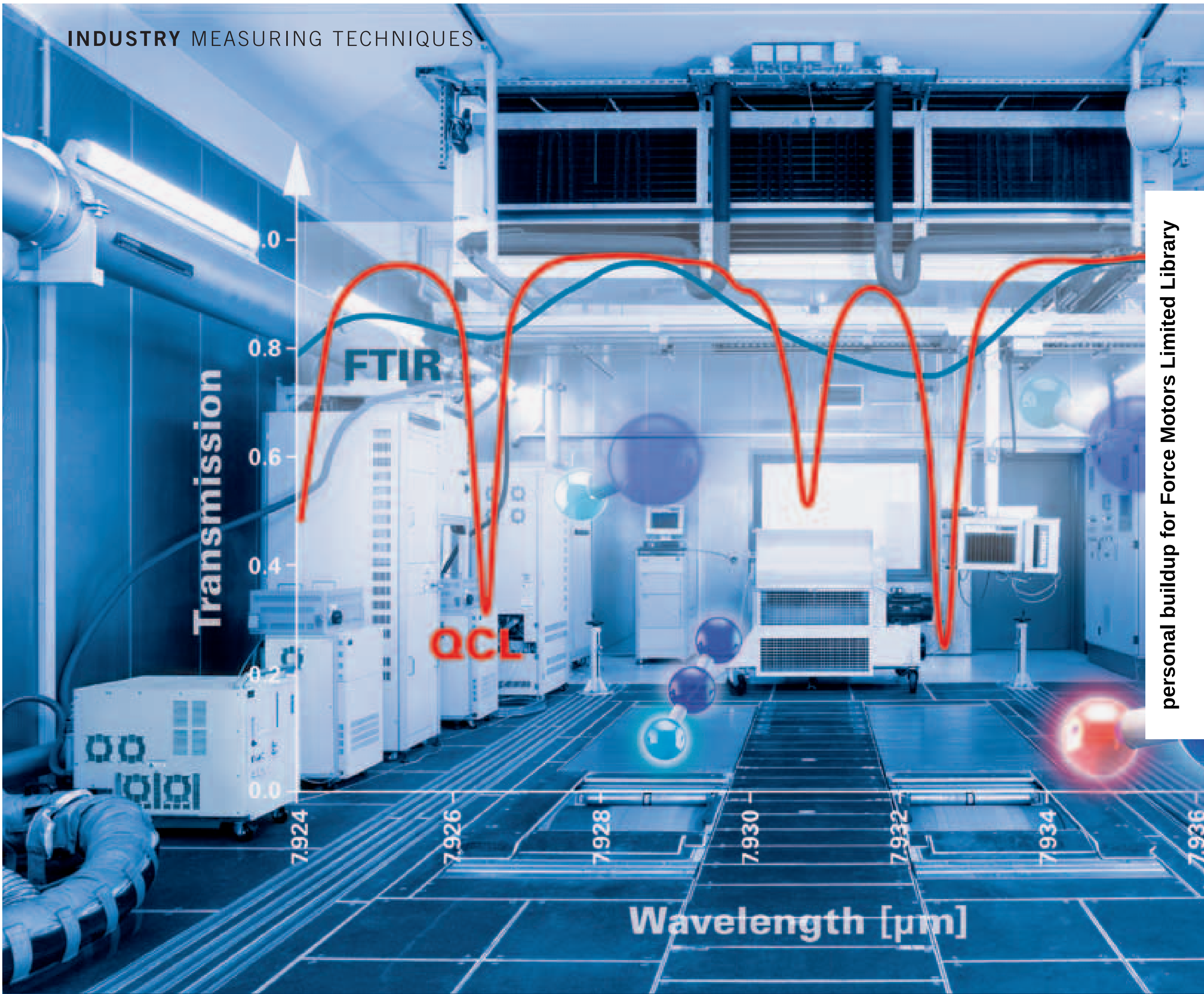

EMISSION MEASUREMENT WITH QUANTUM CASCADE LASER

With the Mexa-1400QL-NX Horiba offers a new exhaust gas measurement solution which is based on the Quantum Cascade Laser technology. The new method is able to test gasoline, diesel and alternative fuelled engines with regard to their NO, NO_2, N_2O and NH_3 emissions and is a tool for research and development of powertrains and exhaust aftertreatment devices.

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CONVENTIONAL MEASUREMENT TECHNOLOGIES REACH THEIR LIMITS

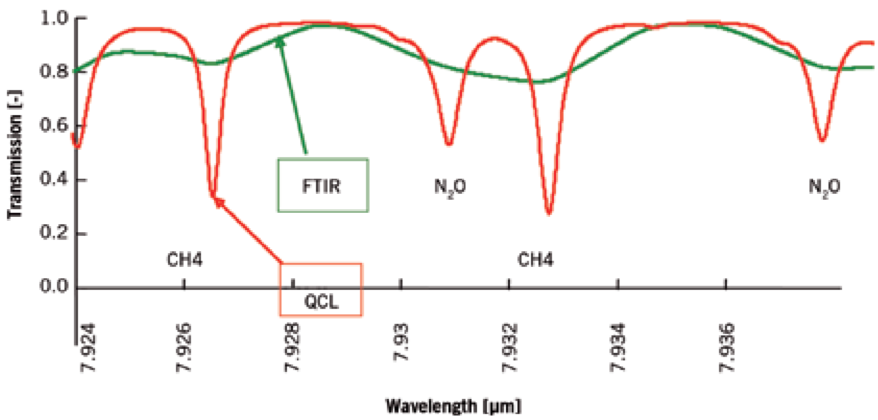
Automotive manufacturers are continually searching for improved analysers that will allow the optimization of engines and aftertreatment devices such as Lean NO_x Trap (LNT) catalysts and Selective Catalytic Reduction (SCR) including also those for alternative powertrains. As the concentration of nitrogen compounds (NO, NO_2, N_2O and NH_3) in the exhaust gas decreases significantly, conventional Chemiluminescence method (CLD), Non-dispersive Infrared absorption method (NDIR) and Fourier Transform Infrared spectroscopy (FTIR) reach their limits with regards to detection and response time. Thus new technologies have to be developed. In the last two decades, systems utilizing mid-infrared laser spectroscopy using Lead Salt Laser technology failed for the need of complex cooling systems which could only be provided by liquid nitrogen.

ADVANTAGES OF QUANTUM CASCADE LASER TECHNOLOGY

Against this background Horiba Automotive Test Systems (ATS) has developed an analyzer for gaseous NO, NO_2, N_2O and NH_3 components which is based on the Quantum Cascade Laser (QCL) technology. Theoretical modelling as well as experimental verification using different engine concepts and aftertreatment configurations played an important role during the R&D process. For each nitrogen compound to be traced in the exhaust gas, one specially calibrated QCL element is utilized. Thus, a total number of four QCL elements are used to carry out precise measurements of all four nitrogen compounds. In the analyzer the QCL technology provides accurate results at very low concentrations of 1.0 ppm (NO), 0.06 ppm (NO_2), 0.2 ppm (N_2O) and 0.3 ppm (NH_3) and an improved response time in comparison to other measurement technologies. Thanks to an intense research on the sampling tube and an optimized design of the whole sampling system, extractive NH_3 measurements can be carried out as well. A series of tests conducted in the course of the development proved the efficiency of the QCL technology for all three major exhaust gas testing methods. In diluted measurement, raw gas measurement and bag measurement, QCL is characterized by its precise and reproducible results. Its universal field of application makes it superior to the measurement techniques available at present, ❶.

MODE OF OPERATION

A QCL element generates laser beams by a pulsating electrical current in a constant interval. The laser element's temperature increases through the current pulse which is supplied to the QCL element. As the laser beam's wavelength depends on the temperature of the laser element it is possible to tune the wave-length easily by applying a current pulse and, by doing so, modify the temperature. The engineers use the ability of the QCL to emit lights in the mid-infrared region. Many molecules show very unique and strong absorption in the mid-infrared range, giving this wavelength region the name fingerprint area. This area is only practically accessible for gas analysis by use of QCL or FTIR technology. As Nitrogen compounds have strong absorption tendency in this region as well they can be detected with the laser beams at a limit of detection superior to conventional analysers.



❶ Low pressure in the sample cell greatly reduces the overlap of absorption lines; the super fine resolution of the QCL leads to very sharp peaks

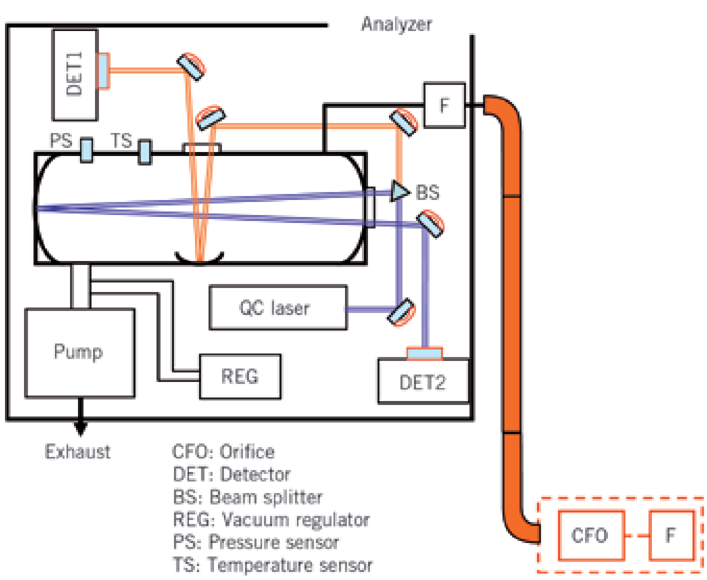
A basic problem is the co-existence of other gases whose compounds can also be detected in the mid-infrared region and thus falsify the measuring results of nitrogen compound measurements. One advantage of the QCL optics configuration is the fact that it can give a super fine resolution of 0.001 cm^{-1} of the mid-infrared spectrum. In combination with the greatly reduced pressure broadening effect, it allows reliable mathematical compensation of the interference caused by this spectral overlap of co-existing gases in engine exhaust.

The size of the optical cell is of utmost importance for the measurement results as well. A small cell results in a short path length of the laser which is advantageous for achieving quick response of the QCL element. On the other hand a very long path length of the cell makes very low detection limits possible but significantly extends the response time. Thus, an optimized balance of response time and detection limit needs to be assured in order to meet the demands and legislations for exhaust gas measurement.

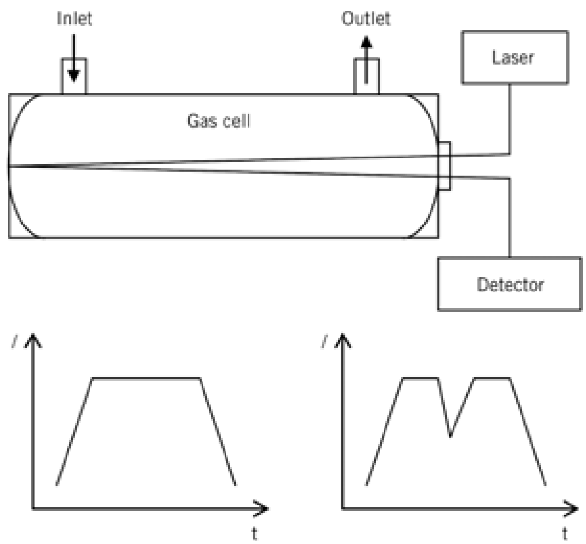
CONFIGURATION OF THE MEASUREMENT SYSTEM

❷ shows the schematic representation of the QCL measurement principle. A total of four laser elements are used. Each element is calibrated for measurement of one of the four nitrogen compounds NO, NO_2, N_2O and NH_3 in the device. Thanks to a sophisticated design of mirrors inside the analyzer two paths could be installed in one single cell. A short path of approximately 0.4 m length is realized with few light reflections whereas a high number of light reflections add up to an optical path length of approximately 40 m. The combination of these two path lengths in the analyzer cell allows the measurement of both, high and low concentration gases at the same time making the QCL measurements system usable for all three major emission tests.

The cell volume is 500 cc while the sampling flow rate is 10 l/min. To minimize the absorption spectrum's pressure broadening the sample gas is drawn into the gas cell by a critical flow orifice and a vacuum pump which maintains a constant cell pressure below ambient level. Sample gas is taken into the gas cell and a laser beam is reflected by mirrors inside the gas cell. A detector senses the laser beams and creates time-resolved spectra, ❸. On basis of these time-resolved spectra an absorption spectrum can be calculated. By application of the Beer-Lambert law the gas concentration can be obtained from the absorption spectrum with the help of library data.



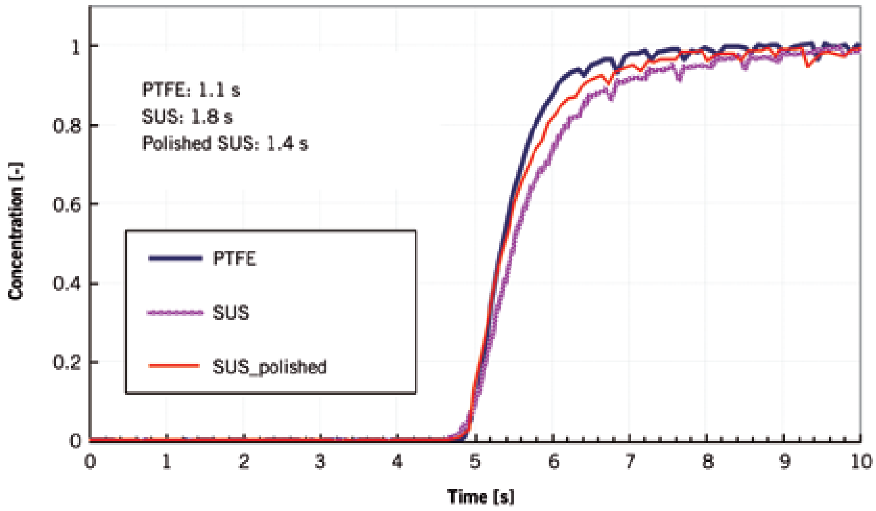
❷ Block diagram of the analyzer system



❸ Schematic setup of an optical analyzer based on absorption; the original puls shape of the laser (bottom left) shows an absorption peak if gas is present (bottom right)

EXTRACTIVE MEASUREMENT OF AMMONIA

The extractive measurement of Ammonia (NH_3) requires specialized technologies. A major problem which has to be tackled is caused by the fact that NH_3 is a sticky gas which has the nature to adhere to the surface of sample cell, transport line, etc. As a result, for a variety of conventional analysers the response time for NH_3 had to be chosen extremely long in the past. Thanks to comprehensive research in this field, the engineers managed to find a solution which enables a response time which lies below 5 s. In order to prevent solubilisation of NH_3 in condensed water, the sample inlet, filter, vacuum regulator and gas cell are heated at 190 °C. The sampling tube was designed as short and slender as possible and made from PTFE to achieve a very smooth surface of the walls, ❹. These means combined with a new mechanical design of filter and valve minimize the surface area and make extractive measurement possible as the surface exposure area is directly proportional to its NH_3 adsorption.



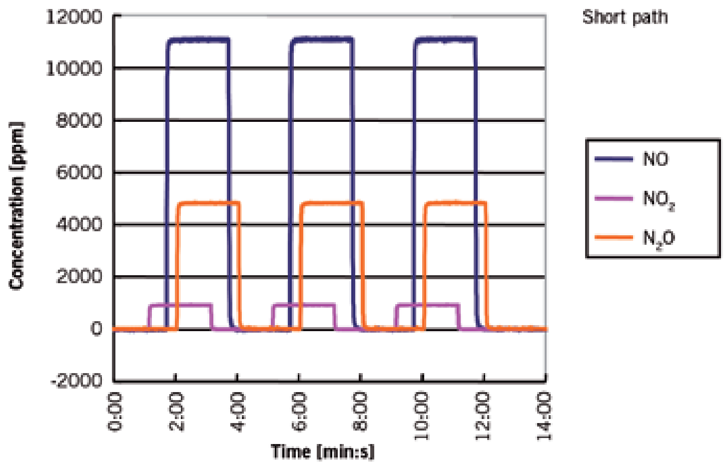
❹ It was found that a PTFE transfer line shows faster analyzer response time compared to stainless steel SUS even if it is polished

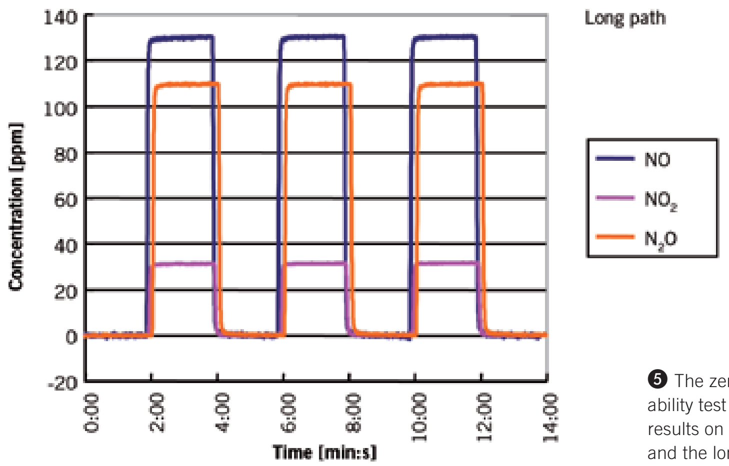
EXPERIMENTAL VALIDATION OF THE MEASUREMENT SYSTEM

In a fundamental experiment, tests for linearity, detection limit, response time, repeatability and Zero/span drift have been conducted with standard gases for all the four targeted compounds in order to evaluate the capability of the QCL technology.

LINEARITY, REPEATABILITY AND ZERO AND SPAN DRIFT

In the linearity test, sample gas was generated by mixing the span gas with N_2 gas at ten different dilution ratios. The reference concentration of each sample gas was calculated from these dilution ratios and the gas concentration labelled on the gas bottle. The results of the linearity test for low range (long path) measurement of all four gases showed that especially in the low concentration zone, good linearity could be reached for each gas. The test for repeatability was done by repeated supply of N_2 as zero gas and span gases of all four target species respectively. Both, high and low range span gases were considered for each of the four target gases. The readings of both ranges were very stable and the fluctuation of the readings was equal to or less than 0.5 % of the average value. As a result a high accuracy and repeatability of the whole measurement could be attested, ❺.





❺ The zero-span repeatability test shows constant results on both the short and the long path

To evaluate zero and span drift on the long run, zero (N_2) or span gas was supplied to the analyzer for 24 h. During zero gas measurement, the error was within ± 0.2 ppm, a value which is lower than 1 % full scale of low range. During span gas measurement, the error was within ± 0.4 ppm and this value is lower than 2 % of the reading value. Interference tests were conducted by repeated supply of N_2 as zero gas and interference gases such as CO, CO_2, H_2O and hydrocarbons. The measurements indicated that thanks to the ultra fine resolution of the lasers interferences by co-existing gases do not occur.

DETECTION LIMIT AND RESPONSE TIME

The detection limit of NO, NO_2, N_2O and NH_3 at both, high range (short path) and low range (long path) measurement was the double standard deviation equivalency. The concentration was averaged over 1000 data points. In the high and low range measurement, the detection limit was lower than 1 % of the full scale in all cases. With regards to response time, a reaction of the analyzer itself which is well below 2 s could be reached for all compounds. Thus, for the first time extractive NH_3 measurement with a response time well within Euro VI limits (maximum of 5 s) is possible by use of appropriate design of filter and sample handling.

VALIDATION IN FTP75 TEST CYCLE

After this first evaluation of the capabilities of the QCL technology with the help of different gases, a series of tests with real exhaust gas was carried out. Therefore, the analyzer was evaluated in a chassis dynamometer test cell using both, a gasoline and a diesel vehicle.

VEHICLE TEST

The basis of the evaluation was a part of the Federal Test Procedure (FTP) 75 drive cycle. The exhaust gas was taken directly from the tailpipe from both test vehicles without being diluted. The tests were validated by simultaneous measurements with QCL and established measuring methods such as CLD, NDIR and FTIR. Furthermore, a heated filter was used at the upstream of the sampling tube close to the tailpipe. The results of this comparative exhaust gas measurement showed agreement with those of FTIR, NDIR and CLD measurements. It was found out that test outputs of FTIR, the only measurement technology capable of multi-compound measurements were slightly higher than those of QCL measurements. This difference in test results can be explained by the spectral overlap of co-existing gases such as CO, CO_2, H_2O and hydrocarbons which can be further minimized with QCL. Furthermore, QCL shows an improved performance over a wider range of concentrations.

OUTLOOK

The result from the evaluation program has demonstrated the great potential of mid-infrared QCL spectroscopy in the automotive exhaust application. The very wide measurement range with the low detectable concentration and the fast response time are keys for this technology. The Mexa-1400QL-NX is the first product providing latest laser technology in form of an industrial, robust and easy to use system, which will be followed by single component devices for N_2O or NH_3 as well as application packages for alcohols and aldehydes.

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THE SEAL OF APPROVAL FOR SCIENTIFIC ARTICLES IN MTZ.
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INFLUENCE OF DIESEL-ETHANOL-WATER BLENDED FUELS ON EMISSIONS IN DIESEL ENGINES

Blended fuels represent one possibility to influence the engine-out soot and NO_x emissions of combustion engines. The work presented in this paper evaluates different fuel compositions in the form of diesel-ethanol emulsions and diesel-ethanol-water emulsions according to their potential to reduce emissions. By adding ethanol to diesel fuel, engine-out soot emissions can be lowered considerably. The addition of water has diminishing effects on the NO_x emissions. The goal of the current investigations is to reduce the engine-out emissions by using a blended fuel emulsion to fulfill the EU Stage IIIB limits, which will take effect in 2012 for rail motor coaches without exhaust gas aftertreatment.

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1 INTRODUCTION

Worldwide, increasingly stringent exhaust emission limits are major challenges for engine and exhaust aftertreatment systems manufacturers.

Soot and NO_x emissions are considered to be critical diesel exhaust gas components. In 2004 the nonroad mobile machinery directive EN 2004/26/EG was published. In this directive, emission levels for rail motor coaches and other off-road engine applications are included, ❶.

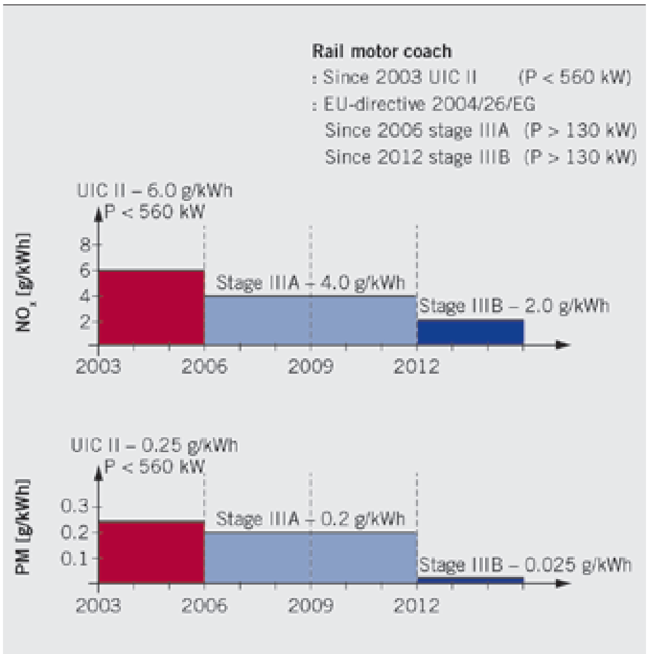
The stage IIIA limits could be achieved with minor modifications to heavy duty Diesel engine. The technical requirements needed to achieve the stage IIIB limits, which go into effect on January 1, 2012, will become more demanding. To meet these limits, SCR systems and EGR systems for NO_x emissions or soot particulate filters are required. These additional measures have significant effects on the railcar engine packaging, particularly on the cooling system. As a result, lifecycle and infrastructure costs are impacted to a significant extent.

As a result of these problems, Voith Turbo has begun to examine alternative solutions to fulfill future exhaust gas emissions limits for rail motor coaches. In cooperation with the Institut für Kolbenmaschinen (IFKM) at the Karlsruhe Institute of Technology (KIT), fuel blends were tested to investigate their potential for reducing engine-out raw emissions.

The tests were conducted on a single cylinder research engine. In a first series of measurements, various diesel-ethanol fuel emulsions were compared to standard diesel fuel in terms of their potential to reduce soot and NO_x emissions.

In the next step, the engine calibration was optimized for the one fuel emulsion that showed the strongest potential in reducing engine-out soot and NO_x emissions. This application was again compared to a separately optimized diesel calibration.

The emission limits shown here that are relevant for the results presented here correspond to the EU nonroad directive 2004/26/EG Stage IIIB (engines 130 ≤ P ≤ 560 kW). The load points tested in this work correspond to the C1 test cycle according to ISO 8178-4 [2].



❶ Emission limits for rail motor coaches [1]

2 ALCOHOL WATER EMULSIONS FUEL

At first glance, alcohols are not suitable for the use in Diesel engines because of their low heating values and their poor ignition qualities. However, with their addition to diesel fuel, several fuel properties of the obtained emulsion can be significantly influenced, ❷.

The heating values and stoichiometric air-fuel ratios of alcohols are lower when compared to pure hydrocarbons because of the oxygen-containing hydroxyl group.

Compared to diesel fuels, alcohols have a higher heat of vaporization, and therefore the combustion temperatures from diesel ethanol blended fuels are lower than of pure diesel [7, 8].

Furthermore, the cetane number of the emulsion decreases with the addition of alcohol, whereas the oxygen content is increased.

The lower cetane number increases the ignition delay and as a result the proportion of premixed combustion. This, combined with the additional oxygen provided by the alcohol blends acts to reduce soot formation.

These effects, on the other hand, enhance the formation of thermal nitrogen oxides [5, 6].

An increase of thermal NO_x, which occurs because of the locally higher oxygen concentrations that result from the presence of alcohol and the longer premixed combustion phase, can be counteracted by the addition of water to decrease the combustion temperatures [12]. This addition has two primary effects. On the one hand, water does not participate in the combustion, but rather absorbs heat from the combustion process due to its high heat of vaporization. On the other hand, based on its very high heat ca-

PARAMETER	DIESEL	ETHANOL	WATER
HEATING VALUE ACC. TO BOIE [MJ/KG]	43.16	27.46	0
STOICHIOMETRIC AIR REQUIREMENT [KG AIR/KG FUEL]	14.60	8.91	-
DENSITY AT 15 °C [KG/M³]	827.50	790	998
CETAN NUMBER [-]	61	5-15	-
VAPORIZATION HEAT [KJ/KG]	251	910	2258
BOILING RANGE/BOILING POINT [°C]	170-380	78,5	100
KINEM. VISCOSITY AT 40 °C [MM²/S]	2.87	1.52	1
OXYGEN CONTENT [%]	< 1	34.7	88.8

❷ Properties of diesel, ethanol and water [3, 4] pacity and its inert mass water can absorb a large amount of heat and lowers therefore the combustion temperature [12]. According to Velji et. al. [10], water must be added at the right location within the cylinder at the right time. As a conclusion of that water is added to the blended fuel emulsions to decrease combustion temperatures. Another effective way to reduce NO_x emissions is either to replace a portion of the fresh air with recirculated exhaust gas (substitutive EGR) or to add recirculated exhaust gas to the fresh air (additive EGR). Substitutive EGR, which was used in these investigations, substantially reduces the amount of oxygen available for combustion. This leads to a reduction of the local specific heat release and to a decrease of thermal NO_x formation due to the higher heat capacity of the recirculated inert exhaust gas. It is well known that above a certain increase in EGR rate, the available amount of oxygen for soot oxidation decreases to the extent that the engine-out soot emissions approach or exceed the acceptable limit.

This sooting limit can be shifted to some extent as a result of the higher amounts of oxygen in the blended fuels, which effectively increases the engine's EGR tolerance [9].

In general, hydrophilic substances (alcohols) mix poorly with lipophilic substances (diesel fuel); this is strongly dependent on the temperature, the water content and the molecular structure of the admixed fluid. For example, the higher the number of carbon atoms of alcohols is, the less they tend to separate from diesel.

Fuel emulsions with additives or micro emulsions provide better long-term stability, they are stabilized by an additional solubilizer (e. g. a surfactant), which counteracts the tendency to separate [3, 11].

The fuels tested in these investigations were all macroemulsions without any additives, mixed right before the common rail high pressure pump.

3 EXPERIMENTAL SETUP

The investigations were performed on a single cylinder research engine based on a Daimler OM 450 heavy duty diesel engine. The combustion chamber design and the cylinder size of this engine are representative of many railcar engines currently in operation. The technical specifications of the engine are shown in ❸.

The engine is set up with an externally-cooled EGR system, a water and oil conditioning system and an electrically-driven compressor. The common rail system was adapted to the engine and consists of a Bosch CP 3.1 high pressure pump and delivers up to 1600 bar rail pressure. A Bosch CRIN3-18 fuel injector with a seven-hole nozzle and 158° spray angel was used.

Every fuel emulsion tested was mixed directly upstream of the common rail high pressure pump using a rotor-stator emulsifying system. ❹ shows a schematic layout of the experimental setup.

PARAMETER	VALUE
DISPLACEMENT	1827 cm³
STROKE	142 mm
BORE	128 mm
NUMBER OF VALVES	4
COMPRESSION RATIO	17.7:1
SUPERCHARGING	Max. 2.2 bar absolute
INJECTION SYSTEM	Common rail
INJECTION PRESSURE	Max. 1600 bar
EXHAUST GAS RECIRCULATION	External via variable valve

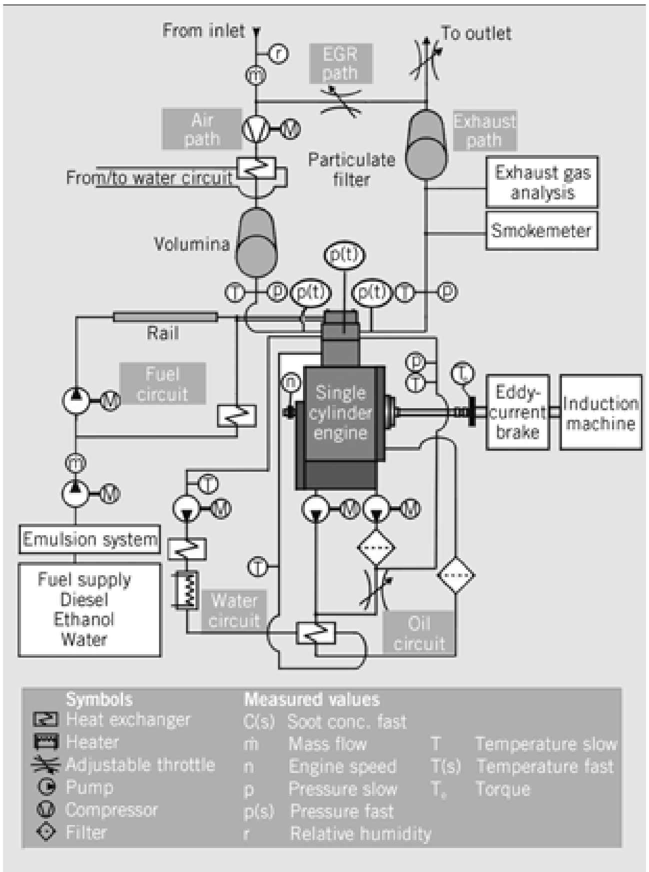
❸ Specifications of the engine OM 450

4 ENGINE TESTS AND RESULTS

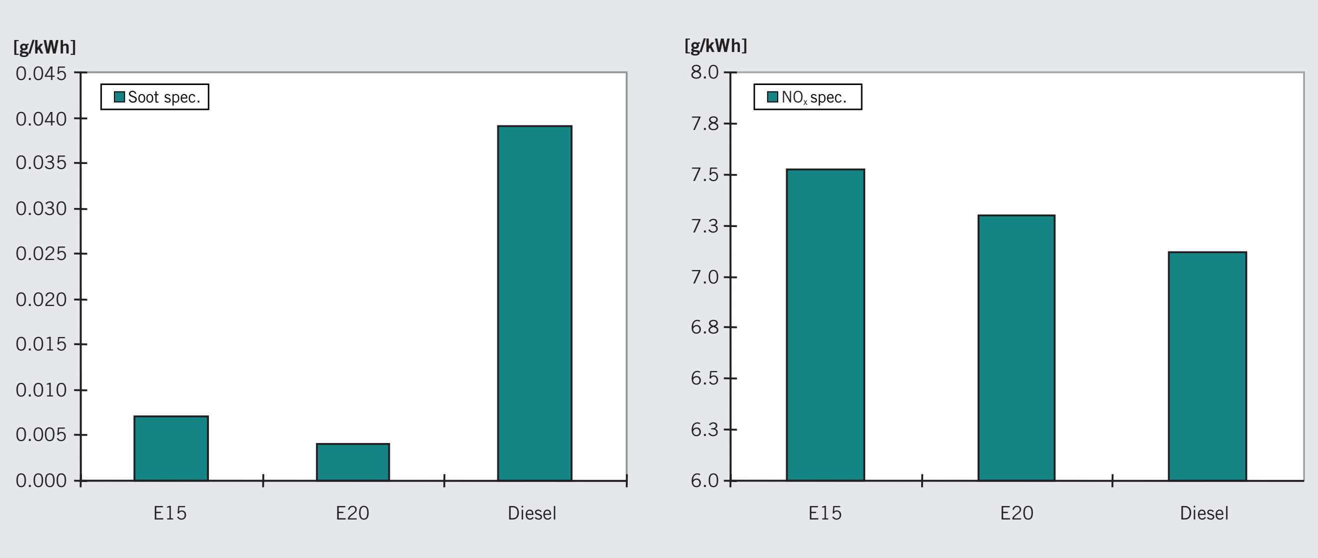
4.1 DIESEL-ETHANOL EMULSION

To examine the potential of various emulsions, the first series of measurements are consciously performed without EGR and without pilot injection. This shows the independent effects of the blended fuels on the combustion and the engine-out emissions. The test fuels are diesel-ethanol blends with 10, 15 and 20 Vol.-% ethanol proportions (named E10, E15 and E20). In the first step, investigations were performed on three characteristic load points at constant engine speed for each fuel blend.

As an example of the results obtained during these first investigations, the specific soot emissions of E15 and E20 are shown in ❺ in comparison to those achieved with diesel. The results show the explicit reduction potential of the engine-out soot emissions. In these data, a reduction in soot emissions of about 90 % was achieved with E20 when compared to diesel. The NO_x emissions measured with E20 are slightly higher than those measured with diesel.



❹ Testbed setup



❺ Potential evaluation of diesel, E15 and E20

PARAMETER	DIESEL	E10	E5W5	E10W10
COMPOSITION (VOLUMETRIC)	100 % Diesel	90 % Diesel 10 % Ethanol	90 % Diesel 5 % Ethanol 5 % Water	80 % Diesel 10 % Ethanol 10 % Water
CARBON MASS PROPORTION (%)	86.30	83.01	79.50	72.82
HYDROGEN MASS PROPORTION (%)	13.70	13.63	13.52	13.34
OXYGEN MASS PROPORTION (%)	0	3.36	6.99	13.85
STOICHIOMETRIC AIR REQUIREMENT [KG AIR/KG FUEL]	14.60	14.05	13.45	12.33
HEATING VALUE ACC. BOIE [MJ/KG]	43.16	41.74	40.08	36.95

❻ Characteristics of fuel emulsions

These first investigations provided the motivation for further measurements, which involved the addition of water to the emulsion to decrease the peak combustion temperatures and thus the amount of thermal NO_x formation.

In this further step, E10 and diesel-ethanol-water emulsions, ❻, with 5 and 10 Vol.-% ethanol and the same proportion of water (named E5W5 and E10W10) were tested. Again, pure diesel was taken as reference.

The following results were obtained with the eight C1 cycle operating points that correspond to the railcar emission certification procedure, ❼. The tests were performed without EGR and without modifications to the injection timing.

4.2 DIESEL-ETHANOL-WATER EMULSIONS

❽ shows the soot emissions as the measured Filter Smoke Number for each of the investigated fuels, normalized to the values of pure diesel.

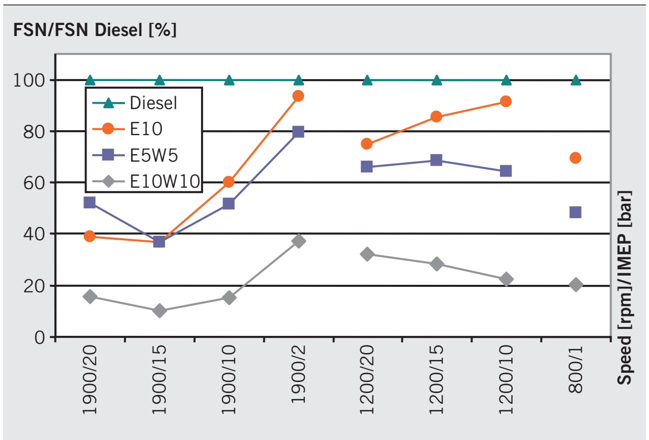
The soot emissions were decreased with all fuel emulsions. The E10 and E5W5 plots are somewhat similar, whereas E10W10 demonstrates the highest potential with soot reductions of up to 90 % at the 1900/15 load point.

Using E10, the decrease of soot emissions increases for every engine speed with higher load. Fuel emulsions with water show this behavior only at higher engine speeds. The effect of increased local oxygen availability is less important at intermediate engine speeds because the local oxygen content (lambda) is further away from the soot emission limit.

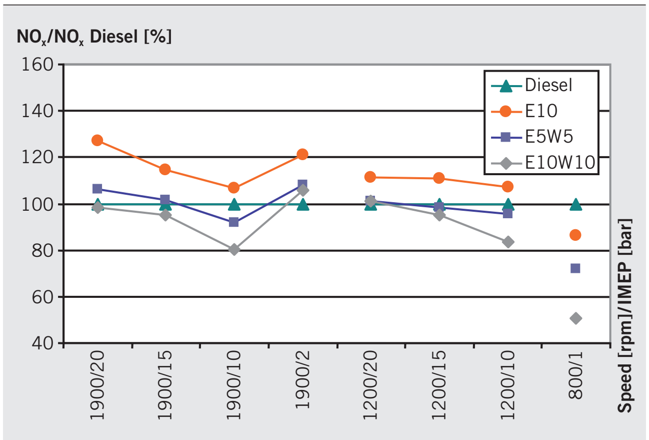
The increase of the soot emissions at 1900/20 with the use of E5W5 and E10W10 can likely be attributed to locally rich mixtures within the cylinder. Due to the addition of water, the lower

N°		SPEED/IMEP [RPM]/[BAR]	LOAD RATIO	WEIGHTING C1 [%]
1	NOMINAL SPEED	1900/20	1	15
2		1900/15	0.75	15
3		1900/10	0.5	15
4		1900/2	0.1	10
5	INTERMEDIATE SPEED	1200/20	1	10
6		1200/15	0.75	10
7		1200/10	0.5	10
8	IDLING	800/1	0	15

❼ Investigated load points C1 cycle according EN 2004-26-EG



❽ Comparison of soot emissions for different load points



❾ Comparison of NO_x emissions over load points

heating value leads to an extension of the injection time to achieve a given engine load. In combination with the shortened time for mixture preparation and oxidation, this helps explain the slight increase soot formation at 1900 rpm.

The specific nitrogen oxide emissions shown in ❾ are again shown relative to the specific nitrogen oxide emissions obtained by using diesel. The locally increased oxygen content and the increased premixed portion of the combustion when using E10 increases the formation of nitrogen oxides in every load point. With the exception of the idle operation point, the values are above the level of those measured with diesel fuel. In contrast to this, the NO_x emissions of both aqueous emulsion fuels are on the level of diesel fuel; the water decreases the combustion temperature and the increase of NO_x emissions caused by the ethanol is compensated.

To give precise evidence of the effect of the fuels to the maximum and mean combustion temperatures, an extended analysis of the combustion process by means of a thermodynamic heat release analysis is needed. For this, the models of caloric and evaporation must be adapted to the fuel mixtures.

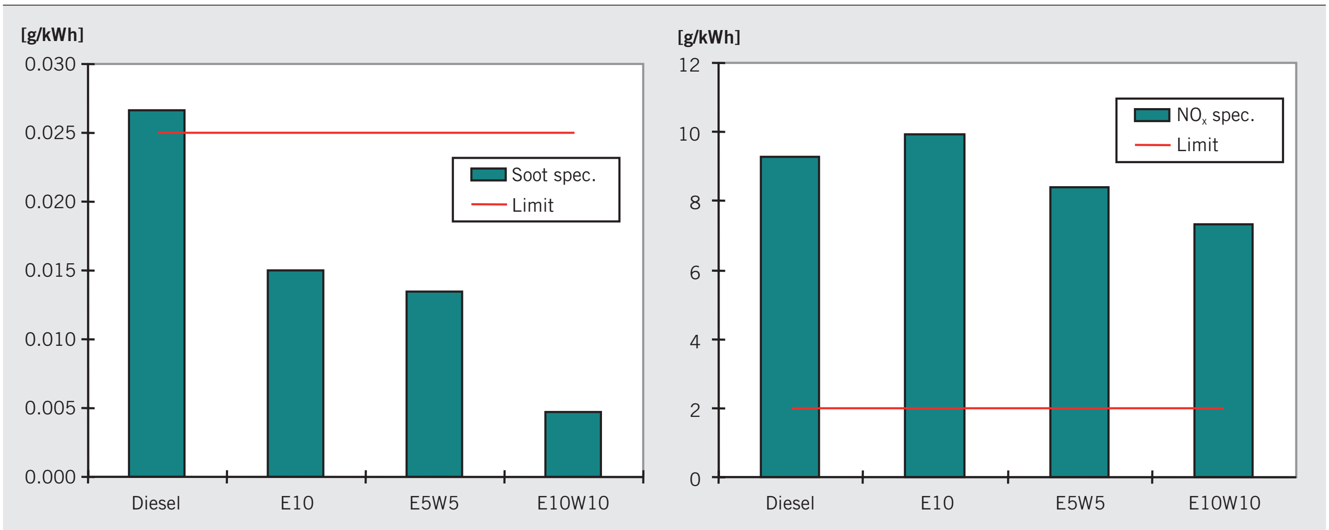
4.3 WEIGHTED SOOT AND NO_x EMISSIONS FOLLOWING THE C1 CYCLE

The specific soot and nitrogen oxide emissions resulting from weighted calculations according to the C1 cycle for rail motor coaches are presented in ❿. The legal cycle limit is indicated by a red line.

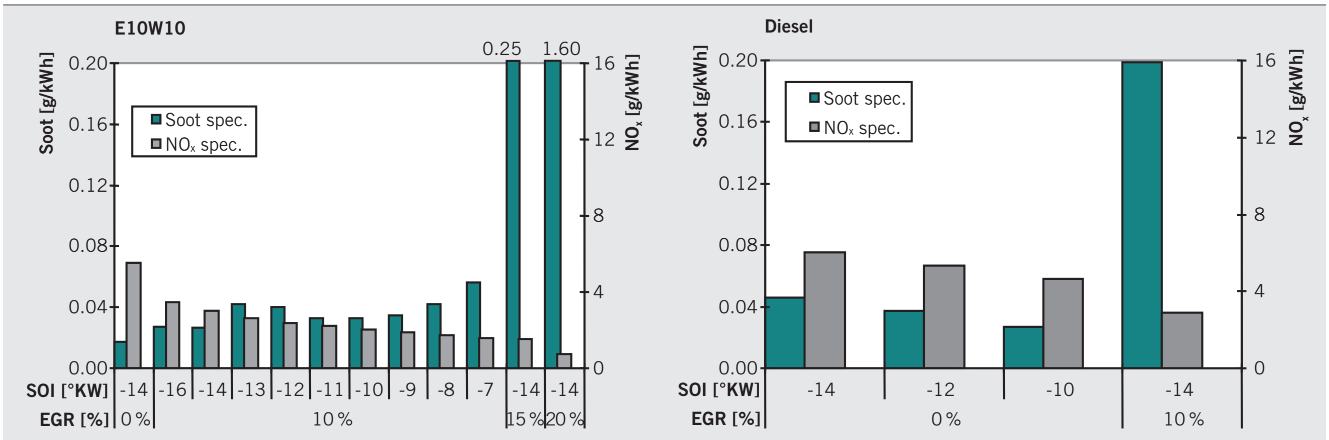
The specific soot emissions were reduced by about 44 % with E10 and by about 52 % with E5W5 compared to standard diesel. Using E10W10, the soot emissions were decreased by 81 %. Furthermore, the soot emissions of all fuel emulsions comply with the legislated cycle limit.

Altogether, the cycle values of nitrogen oxides for the fuel blends are four or five times above the cycle limit.

The use of E10 increases the NO_x emissions by 7 % compared to diesel. Adding water decreases the NO_x emissions by 5 % with E5W5 and by 21 % with E10W10. Since the nitrogen oxide limit could not be achieved throughout the entire test cycle, additional measures are necessary to reduce the soot and NO_x emissions without aftertreatment of the exhaust gas.



❿ Emissions weighted according to C1 cycle for rail motor coaches



⓫ Soot and NO_x emissions at variation of EGR and SOI, 1900 rpm and 15 bar IMEP

Without EGR and without optimization of the injection timing, E10W10 led to the lowest NO_x emissions. However, the amount of NO_x remained above the cycle limit. In combination with the lowest soot emissions, this emulsion fuel offers the largest EGR tolerance and therefore the biggest potential to meet the legal cycle limits only by engine internal measures.

The aim of the final step of this project was to decrease the NO_x emissions by optimization of EGR rate and injection timing without increasing the soot emissions above the legal cycle limit.

4.4 OPTIMIZATION OF E10W10 AND DIESEL IN C1 CYCLE

For every C1 cycle point, experiments were performed in which the EGR rate and injection timing were adjusted and optimized. In ⓫, the specific soot and NO_x emissions of E10W10 and diesel for the load point 1900/15 are shown as an example for these parameter variations.

Regarding these plots, the higher EGR tolerance afforded by the use of an E10W10 emulsion is demonstrated. All in all, varying the EGR rate showed a similar behaviour in decreasing the NO_x emissions for both fuels. Furthermore, the very low soot emissions of E10W10 enable higher EGR rates compared to pure diesel at high load points (1200/20 and 1900/15). With E10W10, compliance with the NO_x limit could therefore be achieved under optimized operation.

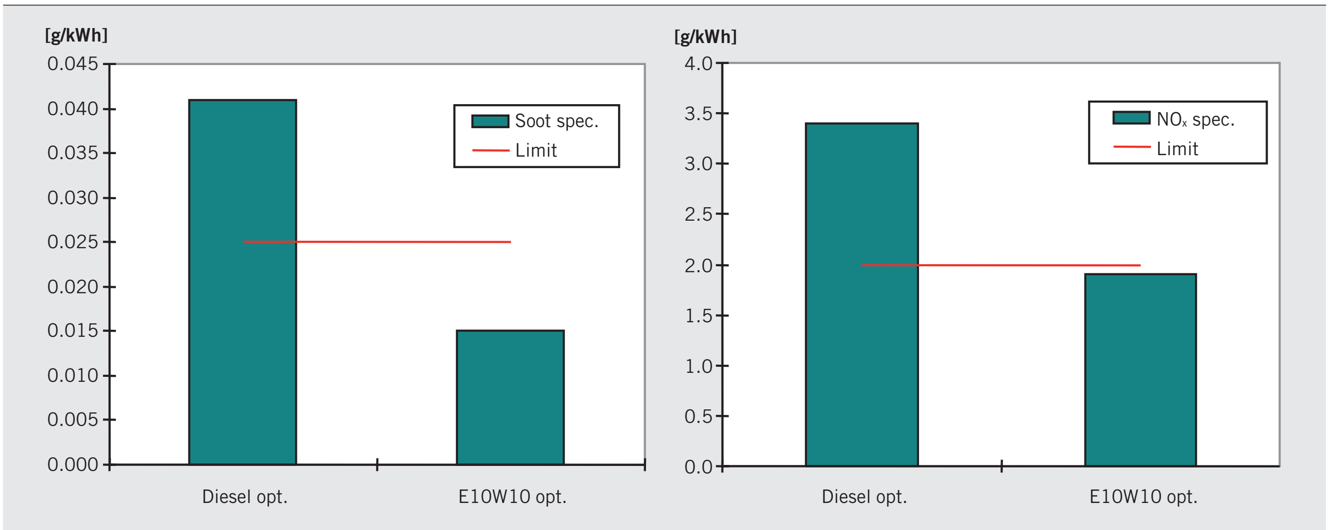
As a result of the described measures and because of the well-known soot NO_x trade-off, a decrease of the nitrogen oxides leads to an increase of the soot emissions.

Due to the higher EGR tolerance of E10W10, the complete cycle values are beneath the legal limits despite the increasing soot emissions, ⓬.

For low-load operation, significantly higher values of carbon monoxides and hydrocarbon emissions were measured in comparison to diesel. A possible explanation for this could be leaner conditions at the peripheral areas in the combustion chamber.

Due to the described additional drop in temperature and a prolongation of the ignition delay flame quenching appears [7].

After weighting these low-load operating points in the test cycle, the HC and CO emissions thresholds are exceeded.



⓬ NO_x emissions diesel and E10W10 after optimization

However, these can be effectively and reasonably reduced using standard aftertreatment components, for example a diesel oxidation catalyst (DOC).

4.5 CONCLUSIONS

When operating a diesel engine with E10W10, the raw soot and NO_x emissions could be lowered below the stage IIIB limits.

A complex aftertreatment system for the reduction of NO_x and soot emissions can be avoided by the use of E10W10.

4.6 OUTLOOK

Further investigations of the measurements using heat release analysis based on the cylinder pressure and the various fuel emulsions properties (evaporation, calorics) are necessary.

In particular, investigations to examine the mixture formation of the fuel emulsions (with various viscosities and surface tensions) should be performed to further adjust the injection nozzle with regard to the hydraulic flow rate, among other parameters [9]. Furthermore the positive results from the single cylinder research engine investigations shall be transferred and validated on a multi cylinder engine.

One target of further investigations is to prepare the optimal mixture ratio of diesel, ethanol and water in the fuel system based on the current operating point [12].

5 SUMMARY

Within these experiments, the potential of different fuel emulsions with ethanol and water was investigated with regard to the simultaneous reduction of soot and nitrogen oxide emissions. The studies were performed with a single cylinder research engine according to the legal emission legislation ISO 8178-4 C1 for rail motor coaches. Pure diesel fuel was compared with fuel emulsions consisting of diesel fuel and 10, 15 or 20 Vol.-% ethanol (E10, E15 and E20); diesel fuel and 5 Vol.-% ethanol and 5 Vol.-% water (E5W5); as well as diesel fuel and 10 Vol.-% ethanol and 10 Vol.-% water (E10W10).

The use of E10W10 produced the lowest soot emissions and therefore a higher EGR tolerance compared to diesel fuel. In a following step, the combustion was optimized for E10W10 with regards to EGR rate and injection timing. Thereby, the raw nitrogen oxide emissions could be decreased below the cycle limit value without exceeding the soot emission limit.

Carbon monoxide and hydrocarbons are to be reduced to the legal cycle limits by the use of a standard oxidation catalyst.

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DIE AND POWDER FORGING MATERIALS FOR AUTOMOTIVE CONNECTING RODS

The increasingly higher demands on gasoline and diesel engines require their component materials to possess higher strength. Two processes, powder and die forging, compete with each other during the development of advanced materials for connecting rods. During the last few years, the development of higher strength materials for fracture splitting has advanced for both manufacturing processes. The Fraunhofer Institute for Structural Durability and System Reliability LBF has now carried out fatigue tests for both manufacturing processes and contrasted the performance of these connecting rod materials.

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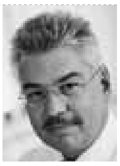

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1 STANDARD VALUATION OF CURRENT MATERIALS FOR CONNECTING RODS

The increasingly higher demands on modern gasoline and diesel engines also lead to higher loading of the engine's components such as, for example, connecting rods. At the same time, the demand for reducing the masses leads to the desire for higher strength materials for these components [1]. Currently, con-rods for fracture splitting are mainly employed in large-batch manufacturing and essentially two different methods compete for the manufacture of the blank material: die forged and powder-forged con-rods. Here, materials development for both manufacturing technologies aims at meeting the requirement for higher fatigue strength.

Within the scope of the project, these materials are uniformly assessed. A total of eight current materials, three powder-forging and five die forging steels, are investigated with respect to their fatigue strength. The fatigue tests were focused on the region of the con-rod shank using a loading ratio of R_F = -2.5, which closely resembles the operating conditions. The results are compared with each other and discussed with regard to their fatigue strength.

2 INVESTIGATED MATERIALS

In order to comprehensively compare the potential of die forged and powder-forged con-rods, the most common die-forging and powder-forging steels were investigated which are currently employed in manufacturing con-rods applying the fracture splitting method.

Whilst powder-forging technology gained an economic advantage over steel con-rods in the 1990's, by introducing the fracture splitting method [2], the forging steel C70S6 was already established as the first con-rod steel applying this method in the mid 1990's, too [3, 4]. The C70S6 steel is characterised by a pearlitic microstructure containing only a small percentage of ferrite. The low yield ratio ensures an acceptable fracture splitting [5]. Further developments of the materials lead to microalloyed steel grades investigating the effect of manganese, chromium, carbon and vanadium [6]. The steel 70MnVS4 was defined by alloying C70S6 steel with vanadium, where the yield point and the tensile strength can be elevated. For example, a tensile strength of over 1200 MPa is obtained with a vanadium content of 0.19 % (70MnVS4-high).

The steels 36MnVS4 and 46MnVS6mod developed later, exhibit a significantly different microstructure. Owing to the lower carbon content, a basically pearlitic microstructure is obtained with increased portion of ferrite compared to C70S6. Also these steel grades get increased strength by elevated manganese content and selecting suitable alloying elements for precipitation hardening.

In contrast to this, the strength of powder forging steels is mainly optimised by alloying with suitable fractions of copper and carbon [7, 8]. Alloying with further elements serves to improve the strength and the machinability [9, 10].

In ❶, the chemical compositions of the materials investigated here are listed. The microstructures of the individual materials are depicted in ❷. To determine the mechanical properties, tensile specimens were taken from the con-rod shank and tensile tests were performed for each material. Each hardness value was determined from five individual measurements, ❸.

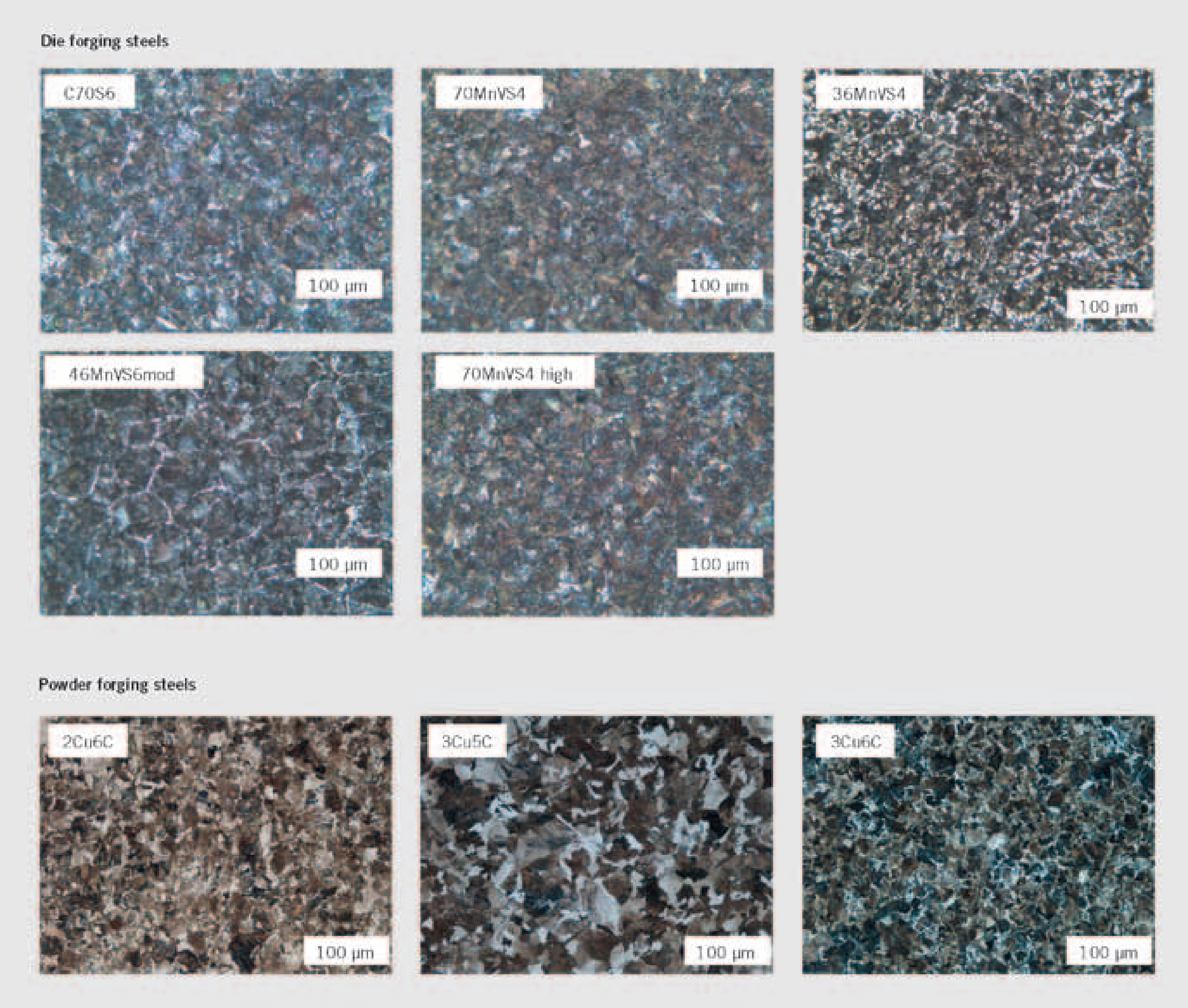
3 TEST PROCEDURE

To assess the behaviour of the individual component's materials and to take into account all the influences of the manufacturing conditions for the component's behaviour in the finished parts shank, all the tests were carried out concentrating on the con-rod shank. To investigate the forging materials, con-rod blanks were manufactured by Mahle, produced using the same die, ❹. To investigate the powder forging materials, con-rod blanks were also used or, as the case may be, finished (3Cu6C) con-rods for three different engines were taken from the serial production. The investigated con-rod shank cross-sections are arranged in ❺. These were also used for computing the nominal stress amplitudes in the shaft. The cross-sections were directly measured using the existing con-rod shanks.

The con-rod shanks were tested under uniaxial loading and a uniform load ratio of R_F = -2.5 to determine their fatigue strength using servo hydraulic and electromechanic testing machines. The

MATERIAL	C70S6	70MnVS4	36MnVS4	46MnVS6mod	70MnVS4-high	2Cu6C	3Cu5C	3Cu6C
	DIE FORGING STEELS [%]					POWDER FORGING STEELS [%]		
C	0.69	0.70	0.36	0.48	0.70	0.60	0.45	0.57
Si	0.22	0.18	0.73	0.63	0.20	<0.02	0.02	<0.02
Mn	0.59	0.83	1.02	1.02	0.83	0.35	0.33	0.33
Cu	0.09	0.07	0.12	0.14	0.17	1.88	3.05	3.12
P	0.005	0.010	0.018	0.010	0.014	0.007	0.006	0.008
S	0.065	0.061	0.075	0.063	0.082	0.13	0.12	0.12
Cr	0.15	0.11	0.15	0.26	0.15	0.042	0.043	0.051
Ni	0.08	0.10	0.11	0.20	0.14	0.049	0.054	0.074
V	0.04	0.11	0.27	0.13	0.19	<0.001	<0.001	<0.001

❶ Chemical composition of the investigated con-rod materials



❷ Microstructures of individual steels

testing frequency lay between 3 and 35 Hz, depending on the loading amplitude, in order to eliminate undue con-rod heating resulting from the loading. With regard to this, the widths of the con-rod's small and big ends were ground to a uniform size and gripped on their flanks, ❻. Since the powder forged con-rods were supplied as-finished, it was not always possible to grip them laterally and the force was introduced via inserted pins into the small and/or big ends. The test was finished by either rupture or achieving

MATERIAL	YIELD STRENGTH YS [MPa]	ULTIMATE TENSILE STRENGTH UTS [MPa]	FRACTURE STRAIN e [%]	REDUCTION OF AREA RA [%]	HARDNESS HB 2.5/187.5
C70S6	660	1058	9.5	31.0	306
70MnVS4	701	1077	9.1	33.7	307
36MnVS4	851	1077	10.3	42.1	321
46MnVS6mod	849	1157	10.0	32.2	324
70MnVS4 high	823	1217	9.0	19.2	345
2Cu6C	591	922	8.8	22.3	304
3Cu5C	630	877	6.7	23.0	321
3Cu6C	830	1149	7.7	9.6	352

❸ Mechanical properties of the investigated con-rod materials

❹ Die forged con-rod blank, shank cross-section 178 mm²

the limiting number of loading cycles at 10^7 cycles. Unbroken con-rods were retested at higher load amplitudes.

4 RESULTS OF THE FATIGUE TESTS

The results for all the materials were converted into the nominal stress amplitude in the shank using the respective cross-section of the shank, ⑤, and evaluated as S-N curves. As an example, the curves for the materials C70S6 and 2Cu6C are depicted in ❼. In doing so, the S-N curves were determined by using a line regression in the range of the finite fatigue strength and, following that of [11], were continued after the knee-point with a further drop of 2 % per decade until $N = 10^7$. Only those test results were evaluated for which the failure occurred in the shank.

Fatigue test results of all con-rods are listed in ❽. The endurable nominal stress amplitudes, which are related to the shank cross-section, at $N_G = 10^7$ loading cycles of all the investigated con-rods are summarised and depicted in ❾ as a bar chart. Here, the results confirm the fatigue strengths already obtained for the powder-forged steels in [7, 8, 9]. Whilst the endurable load amplitude of the forging steel C70S6 is comparable with that of the powder-forging steels and its endurable stress amplitude lies between those of 2Cu6C and 3Cu6C, the endurable load amplitude, particularly of the later developed forging steels, is significantly higher to those of the powder-forging steels.

Despite the different shank geometries and cross-sections, a direct comparison of the results from the component tests is permitted. According to [12, 13], no further reduction of the endurable loading is to be expected for forging steels with a maximum stressed volume larger than such as 30 mm³. All con-rods exhibit a higher maximum stressed volume and fracture occurs within the free region of the shank.

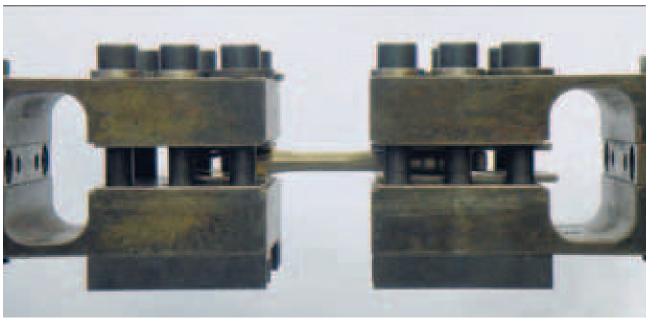
5 DISCUSSION OF THE RESULTS

The obtained results demonstrate that the endurable load amplitude of the forging steel C70S6 is comparable with that of the powder-forging steels. The endurable load amplitudes particularly of the later developed forging steels are significantly higher to those of the powder-forging steels. In addition, the scatter of test results for the die forged and powder-forged con-rods is comparable. For all investigated con-rods nearly the same scatter band was obtained. In this respect, the results of [7, 9, 14] could not be confirmed, in which a significantly larger scatter was measured for the C70S6 and the 36MnVS4.

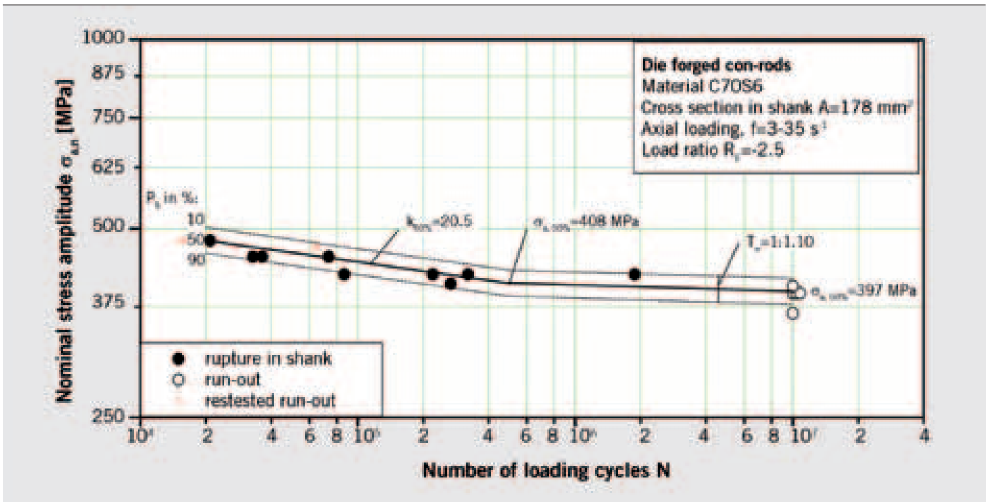
The forging steels possessing high nominal carbon contents of 0.7 % C demonstrate a higher fatigue strength with an increasing fraction of vanadium. For a vanadium content of 0.19 % (70MnVS4-high), a tensile strength of more than 1200 MPa and an endurable stress amplitude of $\sigma_{a,n} = 543$ MPa

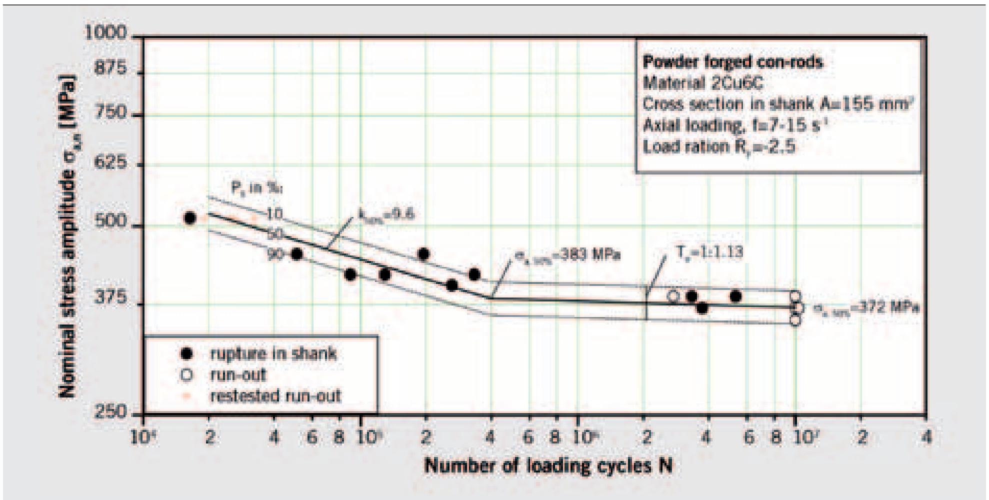
MATERIAL	CROSS-SECTION IN SHANK A [mm²]
DIE FORGED CON-RODS	178
POWDER FORGED CON-ROD 2Cu6C	155
POWDER FORGED CON-ROD 3Cu5C	422
POWDER FORGED CON-ROD 3Cu6C	107

❺ Cross-sections of the investigated con-rod shanks



❻ Lateral gripping of the con-rod for fatigue tests





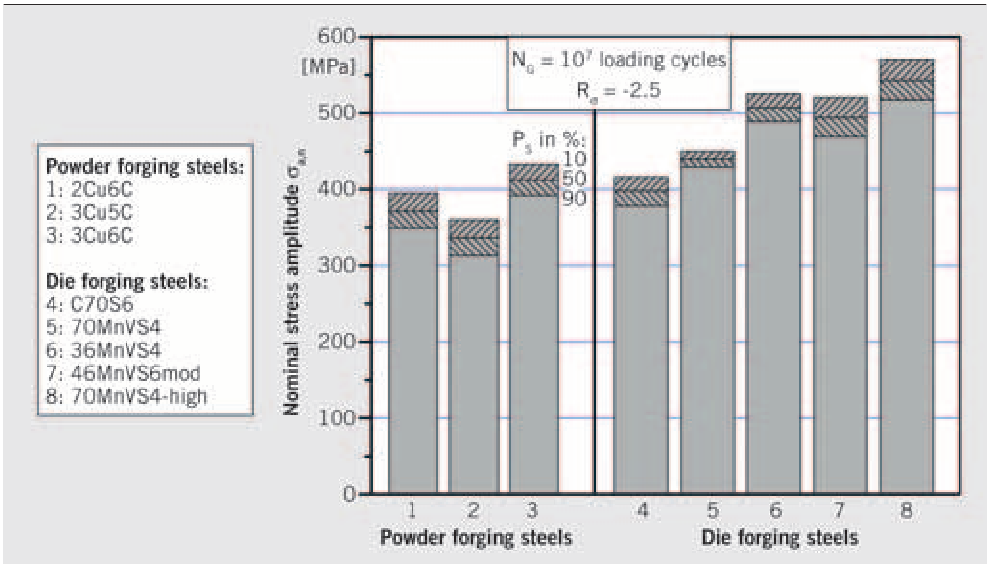
❼ S-N curves for the con-rod blanks of both the die forging steel C70S6 and the powder-forging steel 2Cu6C

is obtained at 10^7 loading cycles (P_S = 50 %) for the con-rod shank. The interaction between higher strength, the accompanying elevated fatigue strength and the resulting additional expenditure in the process chain requires a separate cost-benefit-analysis.

In contrast to the C70S6 and the 70MnVS4 steels, an even higher fatigue strength has been determined for the shanks at moderate tensile strengths for the investigated 36MnVS4 and 46MnVS6mod forging steels possessing low carbon contents.

MATERIAL	NOMINAL STRESSAMPLITUDE FOR P_S = 50 % AT $N_G = 10^7$ $\sigma_{a,50\,\%}$ [MPa]	KNEE-POINT AT N_k	NOMINAL STRESSAMPLITUDE FOR P_S = 50 % AT N_k $\sigma_{a,50\,\%}$ [MPa]	SLOPE OF THE WOEHLERCURVE $k_{50\,\%}$	SCATTER $T\sigma$
C70S6	397	5×10^5	408	20.5	1.10
70MnVS4	441	5×10^5	453	19.7	1.05
36MnVS4	507	1×10^6	517	20.5	1.07
46MnVS6mod	496	2×10^6	503	20.5	1.11
70MnVS4 high	543	5×10^5	557	18.0	1.10
2Cu6C	372	4×10^5	383	9.6	1.13
3Cu5C	335	1×10^6	342	12.1	1.13
3Cu6C	415	1×10^6	423	12.7	1.10

❽ Con-rod fatigue testing results



❾ Comparison of the fatigue strength and scatter of the investigated materials

The powder-forged con-rods fatigue strengths measured here are comparable with the results already published [7, 8, 9 and others], although in this investigation, the powder-forging steel 2Cu6C possessing 2 % copper has a higher fatigue strength to that of 3Cu5C possessing 3 % Cu. The reason for this could not be explained only by the carbon content which is with 0.6 % for the 2Cu6C considerable higher than that of the 3Cu5C with only 0.45 % C. Rather, the powder-forged con-rods from 2Cu5C are showing residual porosity in the surface area of the shank. The 3Cu6C, as expected, shows the highest fatigue strength for the powder-forged steels.

6 SUMMARY

In summary, the investigations show that, in contrast to the powder-forging steels, the currently available die forging steels exhibit a higher fatigue strength for the shank, whereby a greater potential for weight reduction or performance improvement is also gained due to the lower oscillating masses. Especially, the comparison of the fatigue behaviour between the 46MnVS6mod and the 3Cu6C showing nearly the same mechanical properties disclose a higher endurable fatigue strength of the shank of about 20 % for the 46MnVS6mod. The scatter in the results for the con-rods manufactured using both technologies is low and no significant differences between the powder-forged and die forged con-rods could be established.

However, the fatigue strength of the con-rod shank is not the only decisive factor for the component fatigue behaviour. Depending on its design, different stress concentrations exist in the con-rod; for example, in the transition between different cross-sections. For angle split con-rods in particular, the end of the blind bolt hole in the short leg of the con-rod is also highly loaded. In this region, high material strength as well as low notch sensitivity is also desired. Regarding material selection for a balanced economic and reliable design, additional knowledge of the cyclic material behaviour is also important for both the notched condition as well as the machinability of the particular materials.

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