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12 December 2011 | Volume 72

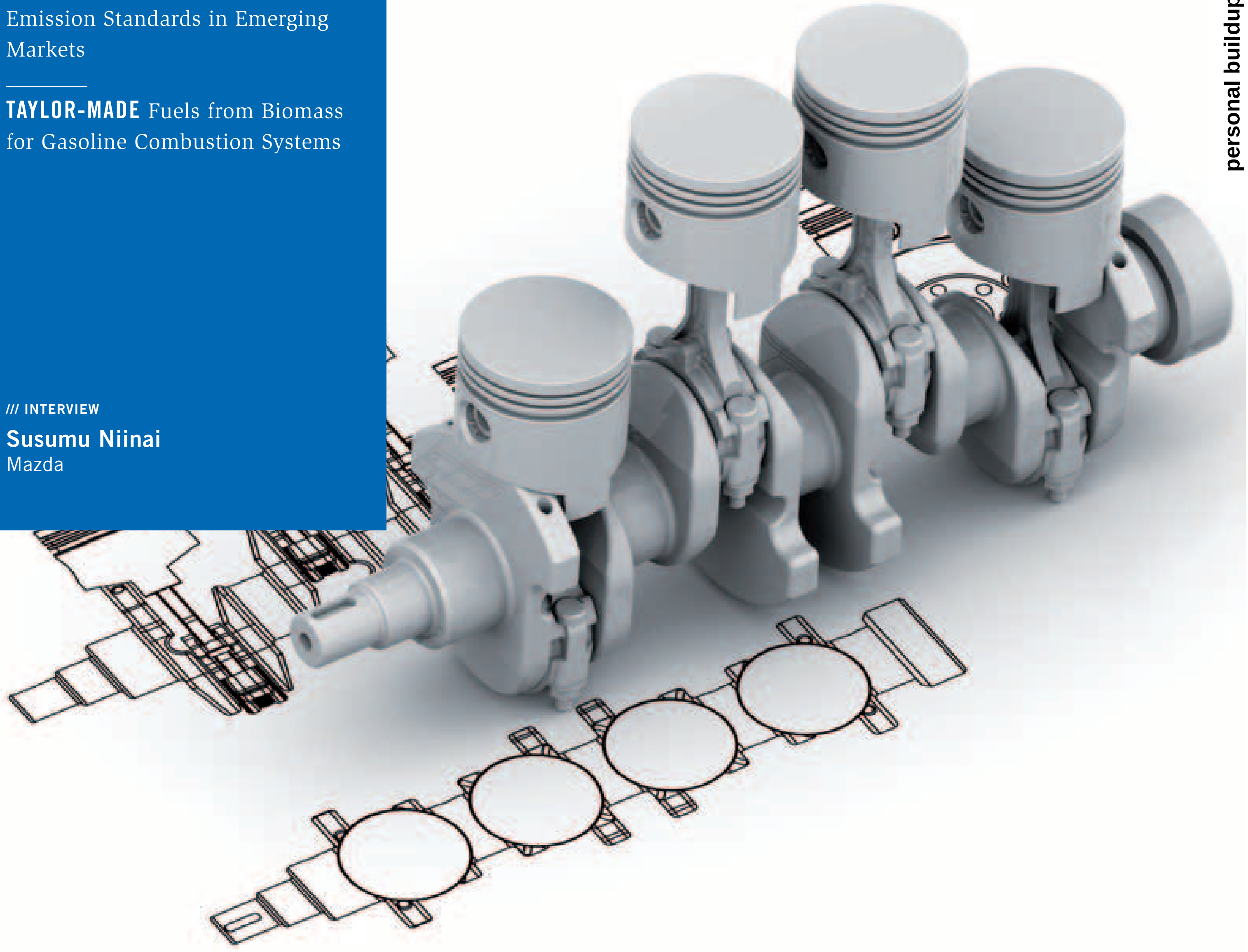
EFFICIENCY ENGINE as Cost-effective Alternative to Downsizing

DIESEL ENGINE Development for Emission Standards in Emerging Markets

TAYLOR-MADE Fuels from Biomass for Gasoline Combustion Systems

/// INTERVIEW

Susumu Niinai
Mazda



DESIGN OF FOUR-CYLINDER ENGINES

CONTENTS

COVER STORY

DESIGN OF FOUR-CYLINDER ENGINES

4, 12, 16 I Spark-ignition engines still offer a very high potential for reducing emissions and increasing specific power output. In most cases, the basic engineering approach is to apply various degrees of downsizing and supercharging, as shown by examples from BMW and Mercedes-Benz. The fact that not all manufacturers follow this strategy can be seen in this interview with Susumu Niinai, Head of Powertrain Development at Mazda.

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FIGURE ABOVE © [M] Vladislav Gajic I shutterstock

TURNING POINTS

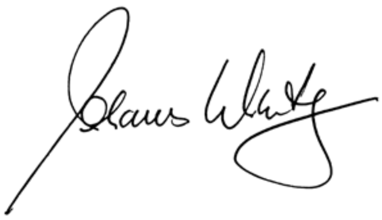
Dear Reader,

What a year this has been that is now drawing to a close! In North Africa, people have been liberating themselves from Arab despots and building new, hopefully democratic countries. This opens up new perspectives for our energy supply, not only due to the oil and gas reserves in that region. In North Africa, solar power can be generated at half the cost compared to Germany, as solar radiation is more than twice as high there. However, capital-intensive large-scale solar power plants can only be built if the political conditions are available.

Then came the nuclear accident in Fukushima, initiating what is likely to be the final phasing out of nuclear power in Germany. Who would have thought, just twelve months ago, that it could happen so quickly? But we can hardly talk of an energy revolution, as 80 percent of Germany's primary energy consumption is still based on fossil raw materials. The rapid phasing out of nuclear power will initially increase our dependency on natural gas and coal as sources of energy.

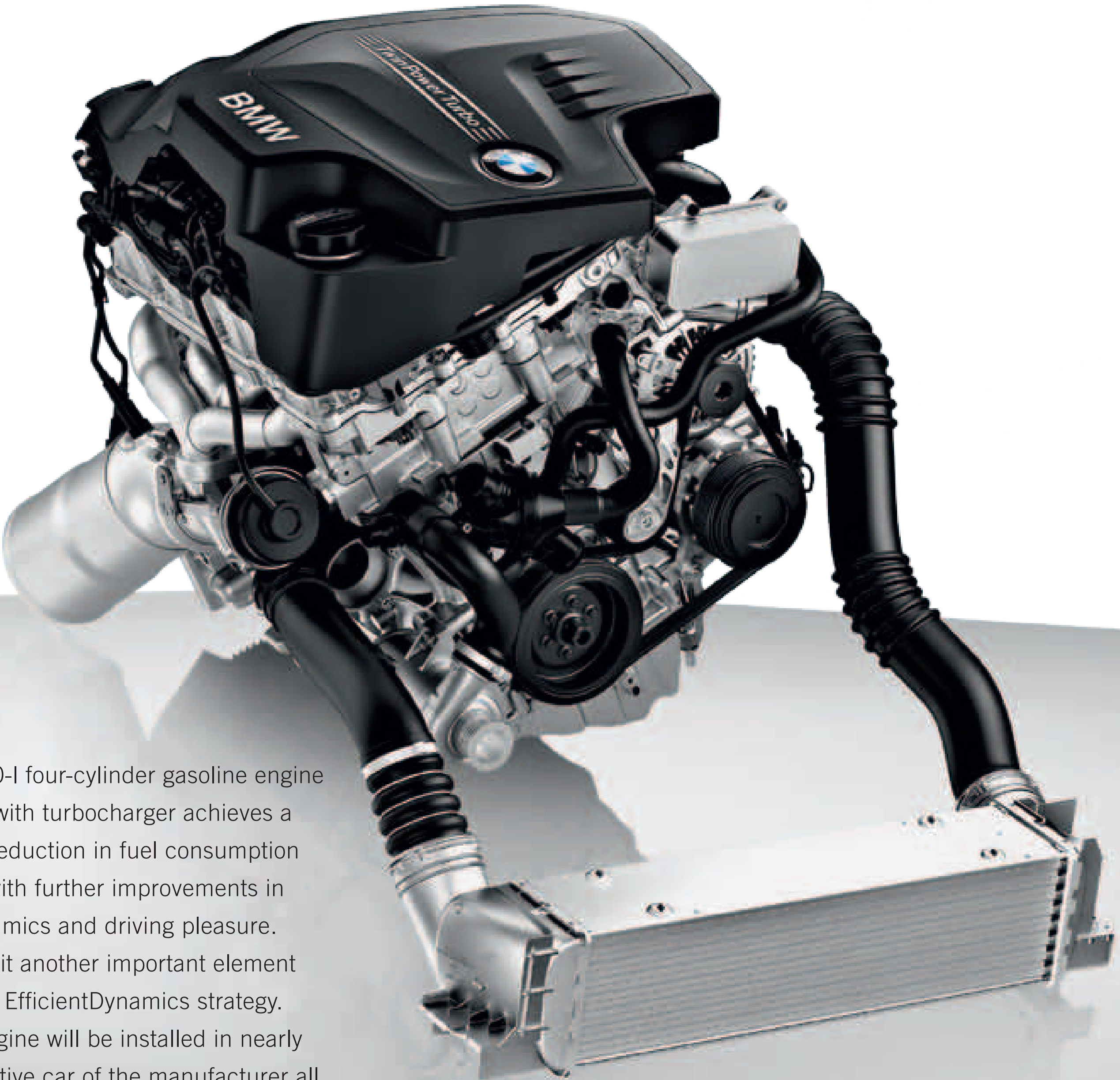
And what about you? You have no doubt been working all year on more fuel-efficient engines and have therefore contributed at least as much to protecting the climate as our politicians. One reason to be proud and to enjoy the final weeks of this year. Over Christmas and the New Year, take a little more time for your family and try not to work flat out until Christmas Eve. As we all know, rapid load changes are not good for long-term durability.

2012 may well bring further turning points that we cannot foresee. But one thing is certain: on 24 and 25 January, our MTZ Conference on the powertrain of tomorrow will be taking place once again in Wolfsburg. The focus this year: "In search of the perfect energy chain". After all, this will be a topic that will not go away. I look forward to seeing you at the conference.



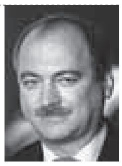
JOHANNES WINTERHAGEN, Editor-in-Chief
Frankfurt, 21 October 2011

THE NEW BMW 2.0-L FOUR-CYLINDER GASOLINE ENGINE WITH TURBOCHARGER



The new 2.0-l four-cylinder gasoline engine from BMW with turbocharger achieves a significant reduction in fuel consumption combined with further improvements in driving dynamics and driving pleasure. This makes it another important element in the BMW EfficientDynamics strategy. The new engine will be installed in nearly every derivative car of the manufacturer all over the world.

AUTHORS



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FOUR INSTEAD OF SIX CYLINDERS

In terms of engines installed, the 130 to 200 kW sector accounts for a significant proportion of global BMW vehicle sales. Up until now, as far as gasoline engines are concerned this sector has been the preserve of the brand's naturally aspirated inline sixes. This type of engine has a long history of success at BMW and indeed over many years the naturally aspirated straight six was the volume seller in the brand's range. This popularity was due in part to the much vaunted smoothness of these engines, and also to the very good consumption figures vis-à-vis the competition. However, BMW has developed six-cylinder gasoline engines, and small four-cylinder gasoline engines for the Mini brand, that boast direct injection, fully variable valve timing and turbocharging. Launched under the so-called Twin Power Turbo sobriquet, these engines are groundbreakers for the efficiency and performance capabilities of gasoline engines to come.

The new 2.0-l gasoline engine rolls out this technology for BMW's biggest-volume power sector of the global market. The new engine carries over the proven features of the turbocharged six, including the Twin Power Turbo (turbocharging Valvetronic direct injection) combustion process, twin-scroll turbocharging and all-aluminium design, combined with radically new technologies. A case in point: BMW is the world's first automaker to use arc spray wire deposition to surface the cylinder liners of an engine manufactured under large-scale production conditions.

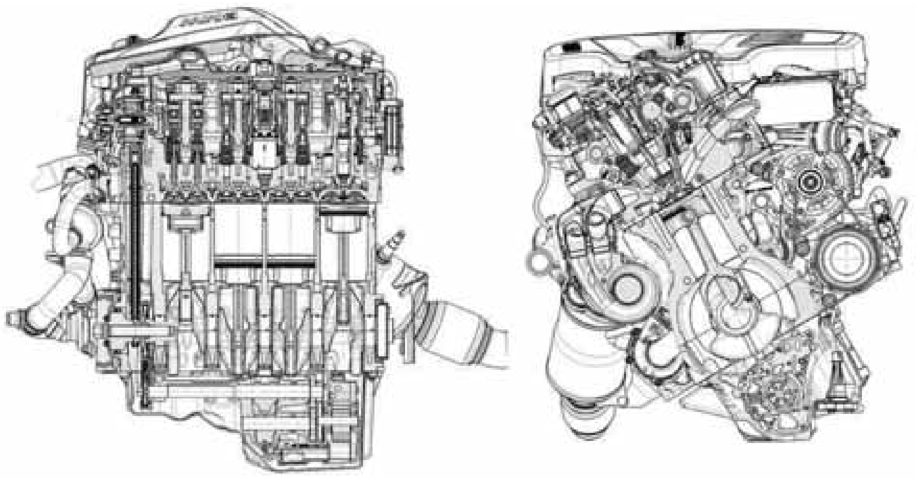
OBJECTIVES

More efficient, lighter, more powerful, more compact and designed to meet more stringent laws as they are enacted around the world: That briefly summarises the design brief for this new power plant. In the parlance of supplier specifications, the goals were set out as follows:

- high specific power output with up to 90 kW/l for the top-of-the-line version
- high low-end torque at just above engine idle
- response transient comparable with counterpart naturally aspirated engines
- low fuel-consumption figures both in the required test cycles and under real-life conditions
- potential to meet the highest emissions limits in the world, up to and including Sulev
- lightweight design with aluminium crankcase and new, innovative friction-face technology
- minimum running-gear friction on account of optimized basic engine design and optimum friction-partner detailing
- incorporation of consumption-control measures from the BMW EfficientDynamics package
- premium smoothness benefiting from equalizing masses with height offset and extra measures to reduce irregularities of rotation
- compatibility with all installation-package conditions across the entire range of BMW passenger cars
- all-wheel drive (AWD) capability in all vehicles
- compatibility with and integration into the BMW manufacturing structures.

ENGINE CONCEPT

The new four-cylinder gasoline engine with 2.0 l displacement will be available in a number of power ratings between 135 and 180 kW, with all versions sharing a common basic design. Characteristic of the engine are its extremely compact dimensions, its light weight and its very high static and dynamic rigidity. ❶ shows the longitudinal and cross-sectional views of the engine. The combustion chambers are of four-valve design with the injector and the spark plug centrally positioned. The charge cycle progresses across the engine, with air intake primarily on the left side of the engine and exhaust discharge on the right. All the auxiliaries except the electric water pump are on the left side of the engine, leaving as much latitude as possible in the space on the right side for optimum siting of the exhaust turbocharger and a catalytic-converter system in close proximity to the engine.



❶ Longitudinal and cross-sectional views of the engine

Fundamental engine sizing is footed on the 91-mm cylinder spacing that BMW has favoured for many years in its inline engines with a per-cylinder swept volume of 0.5 l. The engine was designed for a maximum combustion-chamber peak pressure of 130 bar, so there is ample reserve for further uprating in the course of the design lifespan. ❷ lists the main geometric dimensions of the new engine.

KEY SPECIFICATION	UNIT	MIN. POWER	MAX. POWER
MAXIMUM POWER (CORRESPONDING RPM)	kW at rpm	135 at 5000	180 at 5000
MAXIMUM TORQUE (CORRESPONDING RPM)	Nm at rpm	270 at 1250–4800	350 at 1250–4800
MAXIMUM ENGINE SPEED	rpm	7000	7000
SPECIFIC POWER	kW/l	67.7	90
SPECIFIC TORQUE	Nm/l	135.3	175.4
MAXIMUM SPECIFIC WORK	kJ/l	1.7	2.2
MASS DIN 70 020	kg	138.0	138.0

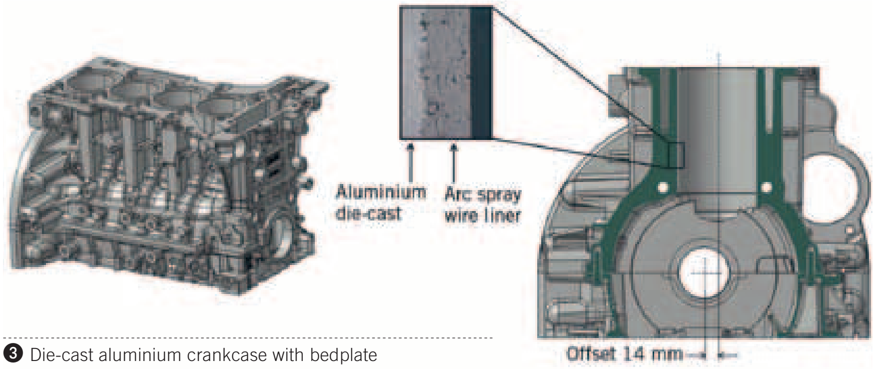
KEY SPECIFICATION	UNIT	
BASIC MEASUREMENTS		
PISTON DISPLACEMENT	cm^3	1995
BORE	mm	84
STROKE	mm	90
STROKE/BORE RATIO	–	1.07
VOLUME PER CYLINDER	cm^3	499
CONNECTING ROD LENGTH	mm	144.35
CONNECTING ROD RATIO	–	0.312
BLOCK HIGHT	mm	221.35
CYLINDER DISTANCE	mm	91
BIG END BEARING		
DIAMETER	mm	50
WIDTH	mm	17
CONROD BEARING		
DIAMETER	mm	50
WIDTH (PINTLE)	mm	17.55
PISTON		
COMPRESSION HEIGHT	mm	32.4
TOP RING LAND	mm	7.9
PISTON PIN		
DIAMETER	mm	22
LENGTH	mm	52
VALVE		
DIAMETER, INLET/EXHAUST	mm	32/28
VALVE LIFT, INLET/EXHAUST	mm	9.9/9.3
STEM DIAMETER, INLET/EXHAUST	mm	5.0/6.0
COMPRESSION RATIO MIN./MAX.	–	10.0/11.0

❷ Technical data and main dimensions of the engines

ENGINE BLOCK

The all-aluminium crankcase, ❸, is a bedplate design with the split-line along the centreline of the crankshaft. Both parts are die-cast in AlSi9Cu3 alloy. To make it even more rigid, cast steel bearing inserts are cast into the bedplate. The crankshaft bearings are offset 14 mm from the axis of the cylinders. This offset of the crankshaft drive reduces piston shear force during the combustion cycle. Each inter-cylinder land has a double-V coolant-gallery bore for intensive cooling.

Arc spray wire deposition for coating the friction faces of the cylinder liners is a world's first for usage of this technology in a series-production engine. The technology employs electric arcing to a pretreated friction face to deposit an iron alloy in a coating that is very thin (approximately 0.3 mm) in comparison to grey cast iron cylinder liners. Pretreatment involves high-pressure emulsion blasting to roughen the friction face and create a multiplicity of undercuts in which the spray-deposited material forms a mechanical key. Lower weight, better heat removal and more leeway for selective cooling of the lands than are achievable with grey cast iron cylinder liners are the principal advantages of this technology.



❸ Die-cast aluminium crankcase with bedplate

CYLINDER HEAD, VALVE GEAR AND INJECTION SYSTEM

The cylinder head, ❹, is gravity die-cast AlSi7MgCu0.5 alloy heat-treated for added strength. Very good reduction of thermal strain on the tops of the combustion chambers is the result of positioning the water jacket low and computer-modelling coolant flow in the critical areas for optimization. The casting incorporates the bearing supports for the camshaft and eccentric-shaft bearings.

The configuration and most of the parts of the chain drive, the valve-drive components including VVT servomotor, injectors and spark plugs, were lifted straight from the well-established and proven six-cylinder turbocharged engine [1].

The camshafts are multipart constructs and they also drive the high-pressure fuel pump and the vacuum pump. The phase adjusters for the camshafts are of the hydraulic sliding-vane type with a range of 70° for the inlet camshaft and 55° for the exhaust camshaft. The 4/3-way solenoid control valve is integrated into the camshafts' central threaded fastener.

The main components in the high-pressure fuel system are the high-pressure pump, the fuel rail, the high-pressure lines and the injectors. System pressure is map-controlled and peak pressure is 200 bar. The high-pressure pump is of single-ram design and is driven directly by a triple-lobe cam on the exhaust camshaft. The injectors have solenoid-valve actuators. The injection nozzles use a six-hole pattern with a hole diameter of 0.2 mm and are characterized by very good minimum-injection capability and constancy.

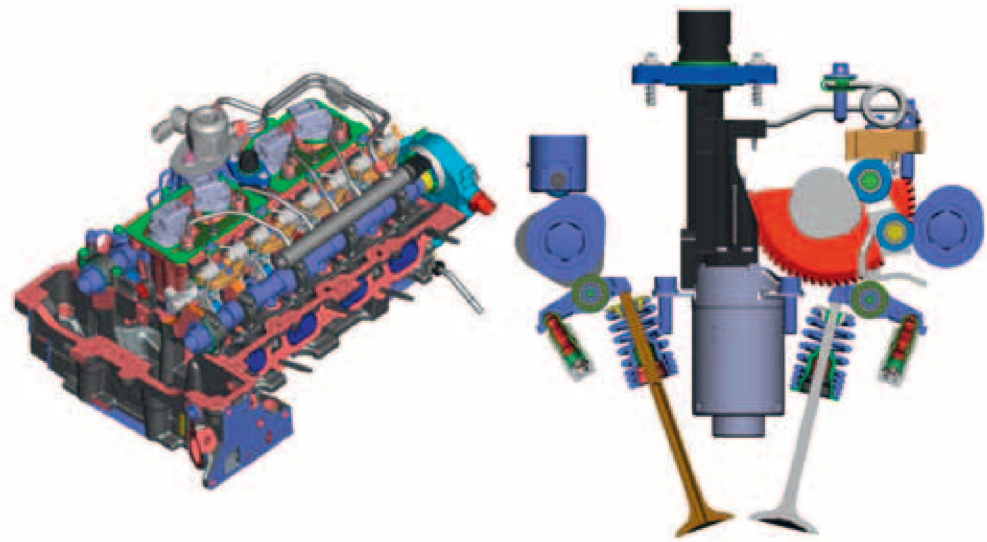
CRANKSHAFT DRIVE

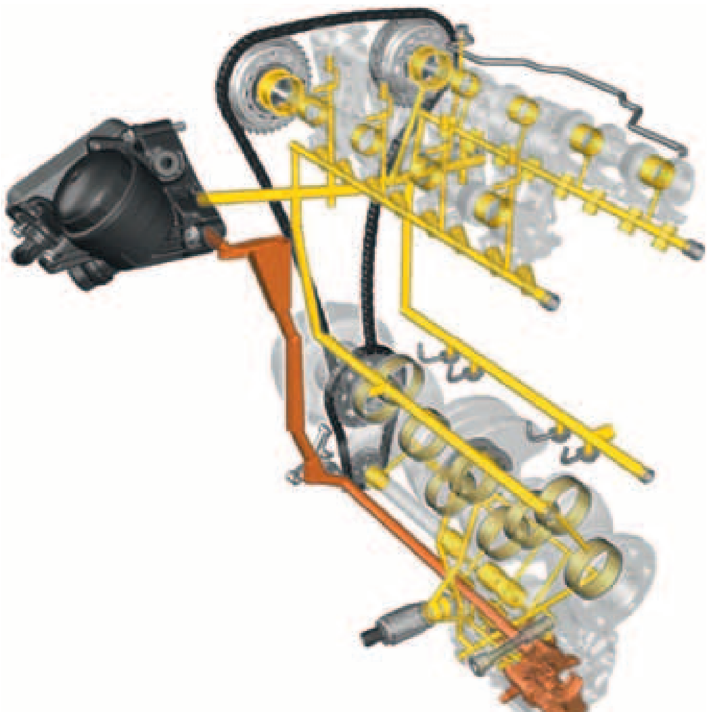
The five-bearing crankshaft is a C38modBY forging with four counterweights. 50 mm is the diameter of main bearings and crankpin bearings alike. All bearing journals are induction-hardened. The thrust bearing is between the second and the third cylinder and is a 180° bearing. All the bearing journals are finished to minimize friction and bearing wear. The main bearings are binary-alloy bearings, but dependable operation throughout the service life of the engine in combination with the automatic engine start/stop functionality dictates the use of a five-layer build-up for the conrod bearings. The conrods are also forged and the small-end bores have wound solid-bronze bushings. The bushings have a ground shaped bore and are cold-pressed into the conrods.

The cast aluminium pistons are hard-anodized in the first ring groove. The ring set consists of a rectangular-section ring in the first groove, a taper-faced Napier ring in the second and a three-part oil scraper ring in the third groove. In order to minimize friction losses in the running gear, ring preload was kept as low as possible and the piston skirts have a friction-reducing coating. Two compression-ratio variants are used, 10.0 for the top-of-range versions and 11.0 for the entry-level engines.

OIL PUMP AND MASS EQUALIZATION

The engine receives its oil supply from a volumetrically controlled pendulum-slider oil pump sited together with the mass-equalization module as a unit in the sump, driven off the front end of the crankshaft by an extremely low-noise sprocket chain. The design and functionality of the mass-equalization unit are described in more detail in the acoustics' chapter. Engine-oil pressure is adjusted to demand for each operating point by a solenoid valve, an arrangement that considerably reduces drive power.


❹ Cylinder head complete with valve gear and fuel system

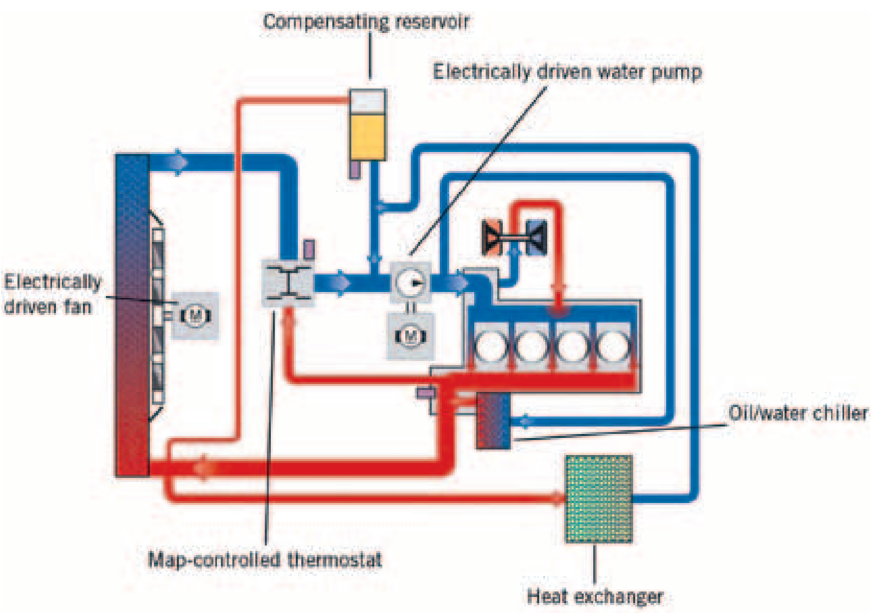


❺ Engine-oil system

CYLINDER-HEAD COVER, CRANKCASE BREATHER AND OIL SEPARATION

The cylinder-head cover is plastic and as well as sealing the top end it integrates the crankcase breather with oil separation. As in most turbocharged engines, the breather gases are returned along two paths, namely directly into the intake ducts of the cylinder head behind the throttle valve in part-load operation, or at higher loads and the consequent higher charge-air pressures to the air intake in front of the turbocharger compressor wheel.

The engine features crankcase air induction in order to ensure ideal oil quality over the entire time between oil changes regardless of engine-operation profile and in markets with critical fuel qualities or extreme climatic conditions. The directed, permanent flow of fresh air through the crank chamber ensures that condensate cannot collect in the engine oil, effectively preventing sludge from forming.



❻ Cooling system

ENGINE-OIL SYSTEM

The engine-oil system, shown schematically in ❺, receives pressurized oil from the controlled pendulum-slider oil pump situated inside the sump, toward the rear. From the oil pump the engine oil passes through a plate-type oil-to-water heat exchanger and then through the oil filter situated directly in front of the main oil gallery, flowing on from there to lubricate the engine. Oil spray nozzles direct lube oil from the oil system to the pistons, the chain drive, the moving elements of the Valvetronic including the teeth of the servomotor and the exhaust camshaft.

A combination oil-pressure and oil-temperature sensor is seated in the main oil gallery. The pressure signal is used for map-specific control of the engine-oil pump, and the temperature signal is an input for the fully integral engine thermo-management coordinator integrated into the electronic engine management system. A thermal oil-level sensor in the sump is used for permanent monitoring of the oil level so that the driver can check the current oil level at any time in the dashboard. A warning is issued if the oil level drops too low.

COOLING SYSTEM

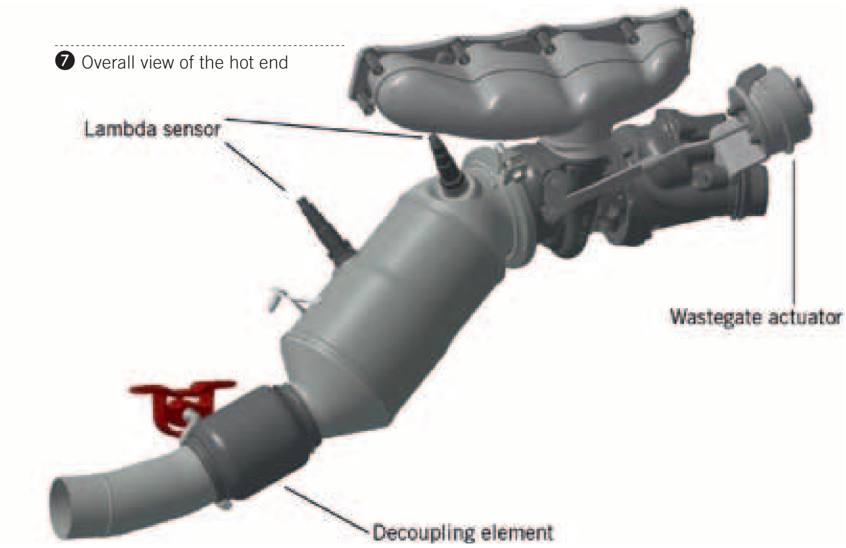
The coolant system, ❻, is also controlled on demand by the engine thermo-management coordinator in the ECU. The actuators are the electrically driven water pump, the map-controlled thermostat and the electric fan of the cooling system. These possibilities allow the engine to settle to the optimum operating temperature in any operating state. The map-controlled thermostat is positioned at the engine inlet, so reactions to spontaneous load changes are rapid. On one hand the electric water pump can be shut down completely for warm-up, while on the other hand it runs on to remove waste heat from the turbocharger, for example, when the engine is switched off after full-load operation.

TURBOCHARGING AND EXHAUST GAS TREATMENT

The design of the turbocharger and the exhaust manifold are crucial in terms of achieving best possible transient response on a par with that of a naturally aspirated engine. Standard BMW practice is to separate inflows from mutually interfering cylinders in the charge cycle, and the approach is carried over into this design. The twin-scroll turbine housing maintains the effectiveness of flow separation through to the turbine wheel. The exhaust manifold is jacketed sheet metal with air-gap insulation to minimize energy loss from the exhaust gas on its passage from exhaust valve to turbine.

The wastegate duct in the turbine housing is extremely short. In the catalytic converter warm-up phase the heat in the exhaust gas is crucial for effective emission control, and this short duct very efficiently retains this heat en route to the catcon monolith in close proximity to the engine. The wastegate is actuated pneumatically by an amply dimensioned vacuum unit. An overrun bypass valve is integrated on the compressor side to avoid undesirable pumping effects on spontaneous load rejection. The catalytic converter is located in close proximity to the engine and is of cascaded design. First in the cascade is a high cell density monolith with 600 cpsi; a substrate with 400 cpsi is used for the second monolith. The first oxygen sensor for control and OBD monitoring is in the catalytic converter's intake funnel, while the second is in the catcon cartridge between the two monoliths. The geometry of the entire hot end is shown in ❼.

❼ Overall view of the hot end



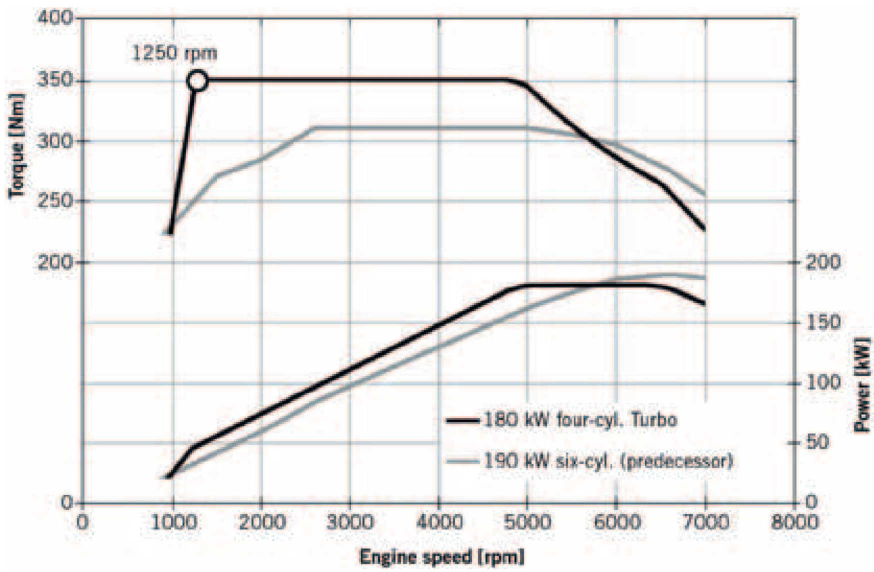
POWER OUTPUT

The new four-cylinder gasoline engine will be available in a number of versions to cover the 135 to 180 kW range, so it will largely supersede the successful six-cylinder naturally aspirated engines. This clearly indicates that the new engine unequivocally continues the downsizing approach espoused by BMW. Flow separation is again essential as a means of achieving direct throttle response in a four-cylinder engine. As with the inline six, the new gasoline engine will have a twin-scroll mono turbocharger – the ideal choice when all aspects are factored in, including low-end torque, throttle response, maximum power, emissions and costs [2].

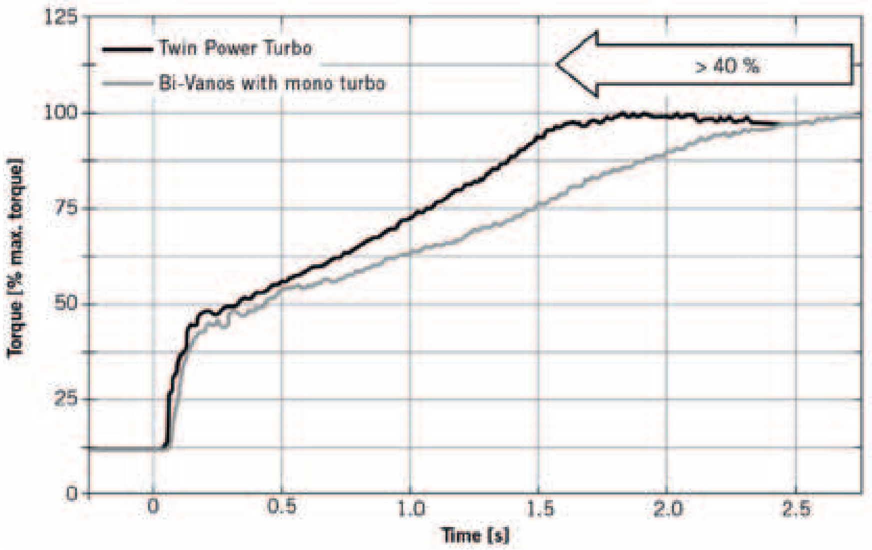
The Twin Power Turbo Technology is the secret to the extremely well-filled torque transient shown in ❽, and it is of particular note that the maximum torque of 350 Nm is available at the very low engine speed of 1250 rpm. Comparison with the predecessor six-cylinder naturally aspirated engine underscores the huge advantage with regard to available torque – particularly at low engine speeds. Maximum power is 180 kW, available at 5000 rpm.

Twin Power Turbo Technology virtually eliminates the turbo lag that was a common feature of past designs. The engine's very direct throttle response is the result of a combination of measures. Valvetronic eliminates the lag commonly experienced for intake manifold filling in engines without fully variable valve drive. Load control is managed by adjustment of the intake valve opening – so there is no delay in cylinder fill increase and increased fill is immediately usable for an increase in torque.

Close to full load for naturally aspirated engines, moreover, Valvetronic makes it possible to reduce inlet valve lift to minimize residual gas proportions and simultaneously maximize volumetric efficiency – the measure for the fresh charge remaining in the cylinder on completion of the charge cycle. In turbocharged engines at low rpm the camshaft adjuster is set to



❽ Torque and power in comparison with the predecessor engine



❾ Advantage of torque build of a Twin Power Turbo engine compared to Bi-Vanos mono-scroll turbocharged engine (at 1500 rpm)

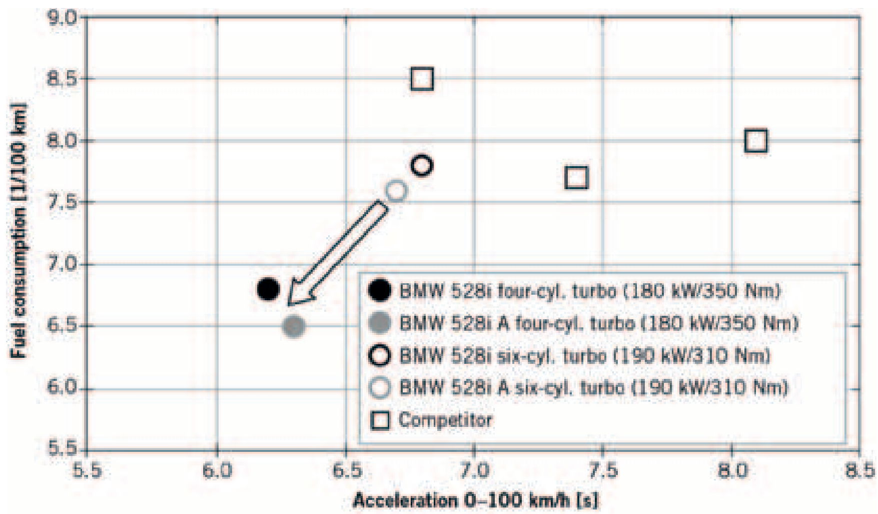
utilize the positive purging delta – induction pressure is higher than exhaust backpressure – to purge residual gas from the cylinder. The increased exhaust mass flow, in turn, produces a correspondingly higher energy supply for the exhaust-driven turbine, increasing the drive power available for the compressor accordingly [3].

The load step shown in ❾ clearly illustrates the advantages of Twin Power Turbo Technology vis-à-vis a conventional engine without Valvetronic and with a conventional mono-scroll turbocharger. After a load step the Twin Power Turbo engine achieves maximum torque more than 40 % sooner than the conventional engine. The combination of outstanding low-end torque and very direct throttle response enables the new four-cylinder gasoline engine to operate with supreme smoothness at exceptionally low speeds.

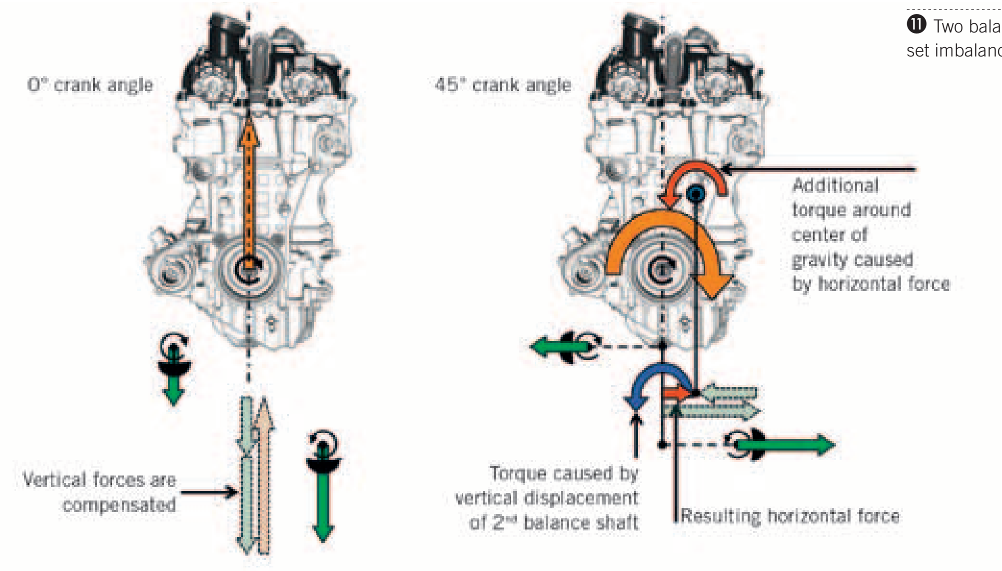
FUEL CONSUMPTION

Despite the exceptionally high degrees of supercharging with 175 Nm/l and 90 kW/l the compression ratio is high at 10:1, establishing the basis for low fuel consumption. In fact the reduced-power variants have an 11:1 epsilon. Another contributor is stepless inlet-valve lift variation by Valvetronic, permitting throttle-free load control and reduction of friction work on account of reduced part-load inlet-valve lift. Combined with other innovative engine-design features – including map-controlled oil pump, electric water pump and a multiplicity of friction-reducing measures – the measures dedicated to the combustion process lead to a very low specific fuel consumption across the entire characteristic map [4].

Moreover, with high torque potential at low rpm and direct throttle response, the Twin Power Turbo Technology package enables driving at low engine speeds. Particularly in combination with the eight-speed automatic transmission and a long rear-differential ratio, the customer drives in characteristic-map points where specific consumption is very low. The combination of specific improvement of consumption in the characteristic map and load-point repositioning meant that



❿ Fuel consumption and driving dynamics



⓫ Two balancing shafts with asymmetric, height-offset imbalance masses: method of operation

depending on vehicle model and type of transmission, consumption could be reduced by approximately 15 % vis-à-vis the predecessor. ❿ shows that the EfficientDynamics goals have been achieved: in terms of economy and performance capability, the new Twin Power Turbo four-cylinder engine is considerably better than its normally aspirated inline-six predecessor.

ACOUSTICS

The new four-cylinder unit will largely replace the inline-six naturally aspirated engines that have a reputation for smoothness, so vibration and acoustics merited special attention. Along with a raft of design details – including for example increased rigidity of the bedplate, decoupled fuel rail and Stop-Choc insulators at the injectors – this is the first engine to feature two balancing shafts with asymmetric, height-offset imbalance masses. Inline fours are challenging particularly on account of the free mass forces and moments of the second order. As ⓫ shows, the two balancing shafts fully compensate the free mass forces. Equalization of the mass moments can be influenced by the height offset of the balancing shafts. The height offset produces a moment that counters the free mass moment of the crankshaft drive.

The new four-cylinder gasoline engine optimizes moment compensation in an innovative way: the physically viable height offset is limited, so the effect of the counterweights on moment compensation was enhanced by making the weights of the two balancing shaft asymmetric. In addition to the moment balanced by the height offset (blue arrows in ⑪), another balanced moment derives from the difference in weight of the imbalance masses (red arrows in ⑪). The resulting moment is larger, corresponding in effect to a bigger height offset. Compensation of mass forces is retained in full. Since mass moments are more effectively compensated, the result – particularly in low-load operation – is a much reduced level of vibration. The customer in the vehicle experiences soft and effortless engine acceleration right through to the top end of the rpm range [5].

SUMMARY

The new 2.0-l four-cylinder turbo engine combines superior torque delivery with high specific power to achieve extremely low fuel consumption with excellent noise and vibration behaviour. Its foundation is due to the very compact and firm construction of the base engine, the Twin Power Turbo Technology with a fully variable valve control system, direct fuel injection and twin-scroll turbocharger. With the aid of this new engine, BMW will further reduce the overall fleet consumption of its vehicles and ensure worldwide compliance with the latest emission regulations despite the ever increasing legislation.

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"WE DO NOT HAVE TO FOLLOW THE TREND FOR DOWNSIZING"

Skyactiv is Mazda's answer to the global downsizing trend. Whereas most OEMs work on the downsizing of engines, Mazda engineers focus on the optimization of the combustion process, for example by adjusting the compression ratio to 14:1 for both diesel and gasoline engines. MTZ interviewed Susumu Niinai, Programme Manager, Powertrain Development Division, Mazda Motor Corporation, at the IAA 2011 in Frankfurt about Mazda's reasons for moving away from the mainstream.

Susumu Niinai has been Programme Manager of the Skyactiv G & D Programme in the Powertrain Development Division at Mazda Motor Corporation since October 2009. After graduating from Hiroshima University in 1986, Niinai started his career at Mazda in the Engine Testing and Research Division. In 1999, he joined the Powertrain Development Division where he has held several management positions. With his extensive expertise in powertrain technologies, Susumu Niinai has been heavily involved in the development of Mazda's new Skyactiv technology.

MTZ _ The Skyactiv concept involves a different approach to the majority of Mazda's competitors. Can you explain the philosophy behind the Skyactiv technologies?

SUSUMU NIINAI _ In the beginning, we were not really focusing on the compression ratio. Instead, we tried to implement additional systems in order to achieve better engine performance, and this was similar to the packaging or downsizing concept. Then we started to think in a different way. It is more or less like climbing a mountain. Our goal, which is to obtain the most out of the combustion process, is at the top of the mountain. Some people start climbing from the north side of the mountain and others from the east side. There is more than one way of reaching the summit. Other manufacturers have chosen a different route to ours. We focused on our existing technologies and came to the conclusion that we have to concentrate on the combustion process to reach the top. This is the philosophy that lies behind the Skyactiv concept.

Most OEMs have opted for a downsizing strategy involving turbocharging. Mazda has chosen a very high compression ratio and variable valve timing. Why do you think none of the other engine manufacturers has adopted this approach?

Because they felt that it was impossible to achieve and other engineers had already given up on increasing the compression ratio before we tried it. Basically, the ideal gasoline internal combustion engine is based on a high compression ratio. However, this also leads to some problems, such as less knock resistance. So, our

"We have to concentrate on the combustion process to reach the top."

engineers began thinking about what would happen if we were to start increasing the compression ratio from 10 or 11 to 14, 15 or even 16. A higher compression ratio also leads to a greater sensitivity to pre-ignition, due to the hot environment. Nevertheless, our engineers continued their research and further optimised the combustion process itself. For more than ten years, we have been producing forecasts and analysing the combustion process. We have accumulated the necessary data and have finally found the solutions which will resolve the problem of knocking.

The compression ratio of your diesel and gasoline engines is exactly the same. How did you manage the resulting problems, such as knocking or starting, at below-zero temperatures?

It is true that the compression ratio is exactly the same in both the gasoline and diesel engines at the moment, but this was not our original intention. Our engineers worked on the overall performance of the engine, which means eliminating knocking, as well as reducing NO_x and soot. And within this optimisation process, the engineers also adjusted the compression ratio. So, it was not our primary target to change the compression ratio for both gasoline and diesel engines to 14:1, it actually happened by accident as part of the overall optimisation process.

Susumu Niinai believes in focusing on the combustion process to produce environmentally friendly and cost-effective engines

Do the diesel and gasoline engines have a lot of parts in common as a result of having the same compression ratio?
The engine blocks are different and we do not use exactly the same parts for both engines. But the underlying architecture of the engines is the same. For example, in a previous version we used a ductile cast iron block for the diesel, but now we use a high pressure die cast method to create an aluminium block for the Skyactiv-D.

"Downsizing pushes up the costs."

Why did you not follow the current trend for downsizing?
Mazda engineers have been focusing on the combustion process and the key technologies of internal combustion engines. As I already mentioned, we have been continuously developing the combustion process for more than ten years. We experimented with different solutions and also used computer simulations to analyse the fuel mixture until we reached the final break-through in terms of knocking. And the other reason why we haven't opted for the downsizing concept is because downsizing means that additional systems are required, for instance in terms of turbocharging and intercooling. And these additional systems push up the costs. So, we do not have to follow the downsizing trend as long as our route leads to the same goal.

I am sure you have compared the costs of the downsizing strategy with those of the Skyactiv strategy. What cost benefits are there for a 129 kW diesel engine and a 121 kW gasoline engine?
Our Skyactiv technology does not require any additional systems for gasoline engines, for example for intercooling, which is a major advantage compared to the downsizing approach. Furthermore, downsizing means that you have to change the whole design, as the engine has to become smaller. And because of that, the crankshaft and con rods also have to be redesigned to maintain the necessary stability and rigidity and this ultimately results in much higher costs.

Mazda, as a smaller OEM, has developed a whole new generation of powertrains as part of its Skyactiv programme, including a six-speed automatic transmission. Why did you not buy this transmission from a supplier such as Jatco, ZF or Getrag?
Throughout Mazda's history, we have always produced our transmissions ourselves. In fact, we did buy in an automatic transmission from an external company in the past, but there were some issues, for example in terms of costs, which made it somewhat problematic. Furthermore, our engineers are more confident about producing the appropriate manual or automatic transmission than they would be with a product of an external company. And finally, we want the expertise to remain inside our company.

Why a six-speed and not, as recently announced by a supplier, a nine-speed automatic transmission?
I am not a transmission engineer, but my personal opinion is that with regard to gear ranges, a six-speed transmission represents a mature technology. The engine and the transmission form a system which has to be perfectly balanced. Undoubtedly, a nine-speed transmission offers several advantages. But I think that a six-speed transmission is currently the best choice for our engines.

"The dual-clutch transmission is not yet technically mature."

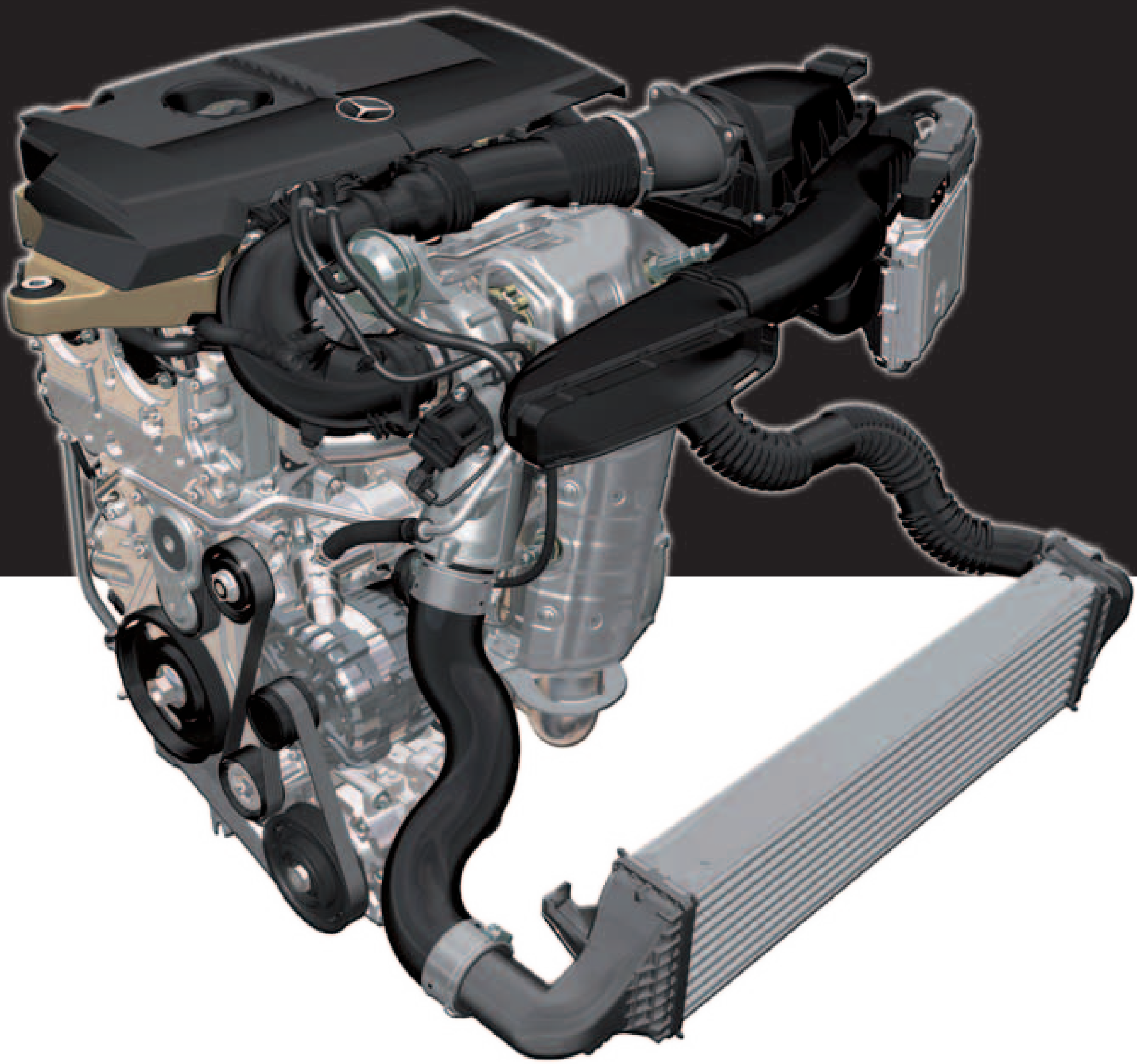
Has this automatic transmission been developed specifically for the new generation of gasoline and diesel engines?
Yes. This transmission is a completely new design and has been especially developed for the whole Skyactiv system to provide outstanding driving performance. The transmission is equipped with a full range lock-up clutch. It stays in lock-up condition for about 90 % of the time, except in idle and launch condition. The broader lock-up range increases the efficiency of the torque converter and improves the torque transfer efficiency. It also provides a direct driving feel that is comparable to a manual transmission.

Why did you not develop a dual-clutch transmission, because this type of transmission seems to be increasingly popular?
In my opinion, the dual-clutch transmission is not yet technically mature. There are still some problems, especially in terms of shocks or at very low speeds. Furthermore, especially on the European market, most customers prefer a manual transmission, which means that they are not particularly concerned about the minor comfort disadvantages of a dual-clutch transmission (DCT) compared with a conventional torque-converter automatic transmission. However, the Japanese and the US markets require a smoother launch and shifting. But, since we sell our products on the worldwide market, we believe that our Skyactiv-Drive automatic transmission, which combines the direct feel of a DCT with the smoothness of a torque-converter automatic transmission, seems to be the best choice for our global approach.

Mr. Niinai, many thanks for this conversation.

INTERVIEW: Roland Schedel
PHOTOS: Mazda

THE NEW TURBOCHARGED FOUR-CYLINDER GASOLINE ENGINE BY MERCEDES-BENZ



At the end of 2011, a new 1.6 l turbocharged four-cylinder engine will enter series production in the new Mercedes-Benz B-Class. With this engine a completely new four-cylinder in-line gasoline engine family including the so called Mercedes-Benz combustion system will be started. The combination of direct injection with piezo injectors with an optimum turbocharger design and a consistently friction-reduced basic engine complies with the high demands in terms of agility, fuel consumption and comfort.

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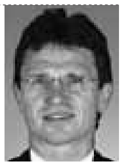

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CONCEPT DEVELOPMENT AND TECHNOLOGY TRANSFER

From autumn 2011, alongside the new Mercedes-Benz B-Class, a new turbocharged four-cylinder in-line gasoline engine generation will also be launched, starting with a displacement of 1.6 l and an output of 90 to 115 kW. This new engine was designed based on systematic downsizing, modularization and advanced technologies and shall be used in the entire Mercedes-Benz passenger car and van product portfolios. That is why the different requirements such as longitudinal and transverse installation, passenger car and van applications as well as all-wheel drive in all the vehicle model series covered, had to be considered to find a uniform platform for the new engine family in terms of product design.

Therefore, for the first time within an engine family that is installed in both a longitudinal and a transverse manner, a so called front exhaust design is being implemented for transverse installation. The cooling concept implemented in each case was specifically adapted to the respective application. Future variants with larger displacements that will cover the top performances in the four-cylinder segment have a modular Lanchester balance shaft design.

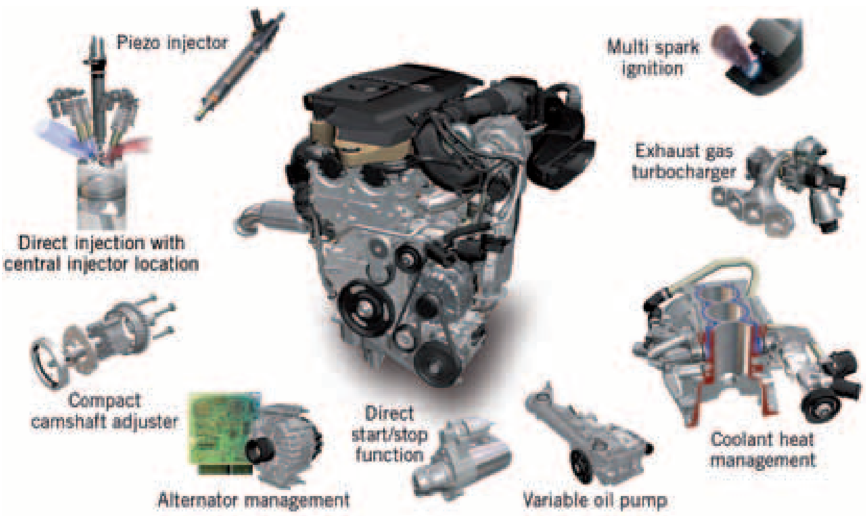
In 2010, for the new V6 and V8 engines [2, 3] with exhaust gas turbocharging and direct injection, an enhanced downsizing strategy was launched across the market with a technology portfolio involving spray-guided combustion with third-generation direct injection, multiple-spark ignition and an ECO direct start/stop function. This technology portfolio and the Mercedes-Benz combustion system developed from it were transferred to the new future four-cylinder in-line gasoline engine generation and further optimized for the four-cylinder-specific requirements, ❶. These include the use of a newly developed double pipe turbocharger exhaust manifold with turbocharger with 1050 °C maximum temperature, a water pump optimized for the installation space with a mixed-flow impeller, a pressure-loss-optimized ball thermostat, a two-stage regulated vane-type oil pump, integrated ancillary component and coolant heat management and the reduction of the friction power in the powertrain and chain drive.

ENGINE DESIGN AND MECHANICS

The development specifications for the die-cast aluminium cylinder crankcase and the powertrain involved a significant reduction of the weight, an additional reduction in powertrain friction and the introduction of a new cross-flow cooling.

The crankcase concept for the product design is based on the integration of the production systems for a transverse installation variant and a longitudinal installation variant. As a central measure for weight reduction, the crankcase was designed, as for the previous engine, with a cylinder spacing of 90 mm – a so called deep skirt and open-deck design – and rendered in die-cast aluminium (aluminium alloy AlSi9Cu3). The crankcase installation spaces are reduced such that the engines in future vehicle model series will all be all-wheel-capable. Later installation of an add-on Lanchester balance shaft system for a larger displacement variant is possible. As the result of extensive structure optimizations, a weight saving of approximately 3.5 kg compared to the previous engine was achieved at the crankcase.

The cylinder barrels with wear-resistant and millionfold-proven roughcast cylinder liners are cooled by bores in the wall between the cylinders and are cast on the external sides to produce pure cross-flow cooling. The surface precision geometry for the cylinder liner known from the previous engine was developed further to what is called precision honing. The characteristic structural height of the honed

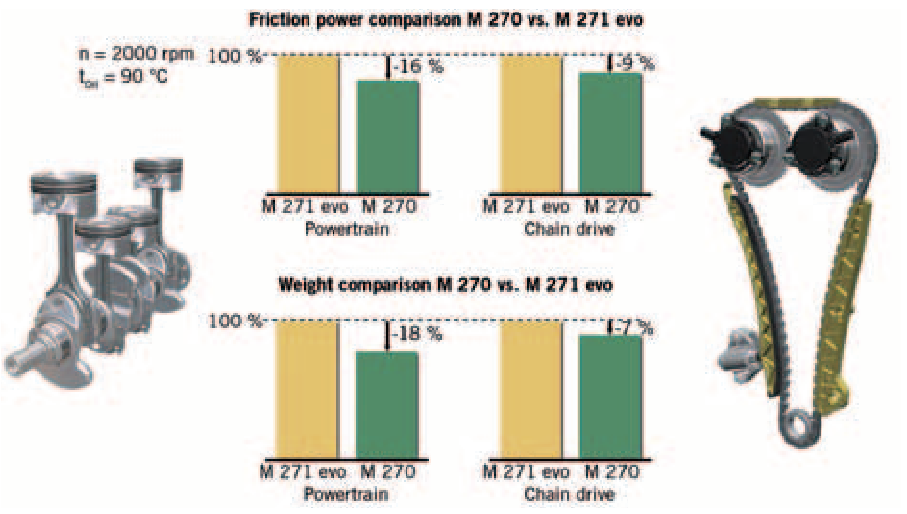


❶ Technology portfolio of the M 270 engine surface was reduced by approximately 50 % compared to the previous surface structure, and the oil-retaining volume was reduced by a third at the same time. Together with a carefully coordinated friction package on the piston group, this measure leads to a reduction in engine oil consumption of approximately 30 % and a power loss reduction of 16 %.

POWERTRAIN

The crankshaft made of spheroidal graphite cast iron was a particular focal point of development. Casting made it possible to carry out light-weight construction consistently and economically by hollow casting crank pins. Four instead of eight, and in addition weight-reduced counterweights as a knock-on effect of the reduced oscillating and rotating powertrain masses make a crankshaft weight advantage of 27 % possible compared to the previous engine.

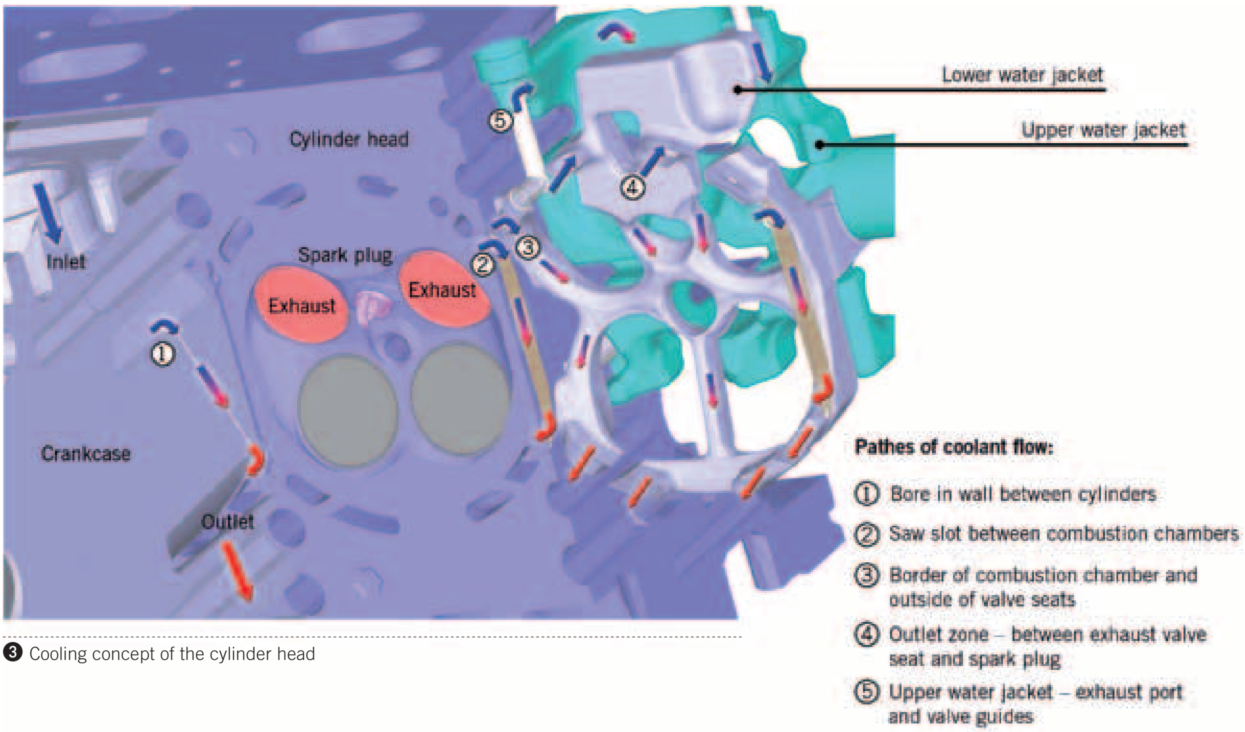
The consistent use of modern simulation and development methods made it possible to use a cast piston in a turbocharged engine with a specific output of 72 kW/l. Despite a bore increased in size by 1 mm compared to the previous engine, a weight reduction of 9 % was achieved as a result of the lightweight design of the cast piston made of an ultra-heat-resistant Al alloy.



❷ Friction and weight optimization

The piston rings used are as follows: symmetrically crowned rectangular section rings in the hard-anodized first groove, taper face rings in the second groove and three-part coated oil control rings in the third groove. In extensive series of tests, the tangential forces of the piston rings in conjunction with the further optimized cylinder liner with precision honing were optimized with respect to friction power, oil consumption and blow-by volumes on the piston to achieve minimum values. In addition to the reduced piston ring forces, the significantly reduced asymmetrical piston shaft bearing surfaces with optimized carbon coating and precision adjustments to piston clearance and the oil sprayer cooling contributed significantly to friction reduction.

The sophisticated forged connecting rod is designed for minimum weight. The small eye is designed as a trapezoidal connecting rod with an upper guide and is equipped with a bronze bushing with thin walls.



❸ Cooling concept of the cylinder head

The large connecting rod eye is cracked and screwed on using M7 screws. The length of the connecting rod is 152.2 mm. With a connecting rod ratio of 0.24 and a stroke/bore ratio of 0.89, an outstanding basis for advantageous engine acoustics and low friction was achieved. In conjunction with the significant reduction of the oscillating masses mentioned above, the NVH-relevant second-order inertia forces on the crankshaft could be reduced by 31 %.

CHAIN DRIVE

The newly developed engine is equipped with a new silent chain design, which is characterized by 30 % higher durability while simultaneously having an overall width of 20 % less than standard silent chains. This is achieved, among other things, by a crimp connection of the pin in the centre chain link plate, leading to a balanced load, minimal deformations and thus ideal load distribution in the joints. The rigidity of the chain also increases by about 50 % as a result. The elongation caused by wear for the chain remains below 0.2 %, even under extreme conditions. The precision-blanked chain plates with high proportions of straight cuts result in extremely smooth function surfaces, and friction measurements show a significant friction advantage.

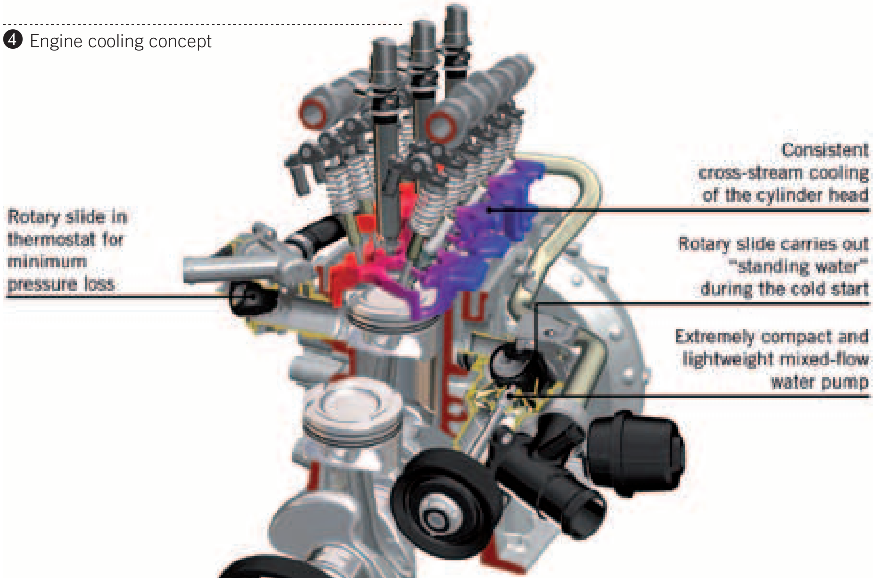
The sum of the measures mentioned leads to a friction power reduction in the powertrain and chain drive of 16 % and 9 % respectively compared with the previous engine. It was possible to reduce the weight of the powertrain by 18 % and the weight of the chain drive by 7 %, ❷. Furthermore, the chain drive is designed for minimum NVH, in order to prevent high-frequency noise disturbance.

CYLINDER HEAD, COOLING AND HEAT MANAGEMENT

The high degree of engine supercharging and the resulting high heat input make intensive cooling of the combustion chamber necessary. The optimizations were done mainly in the so called digital assembly stage, when iterative loops take place in close cooperation with Design in order to reach the highest possible maturity level before testing.

To cool the combustion chamber, a two-part water jacket design was developed to achieve completely cross-flow cooling.

❹ Engine cooling concept



The coolant is fed into the centre of the exhaust side of the crankcase and after this is divided into five main coolant paths per combustion chamber, ❸. The respective coolant flows are guided over the relevant hot areas just once and thereby, in combination with an optimized volume flow distribution, achieve a highly effective cooling efficiency with simultaneously low pressure loss. The comparatively low and, due to the cylinders, almost constant temperature of the combustion chamber roof provided the necessary prerequisites for stable combustion with low consumption. At the same time as the coolant routing was designed, the structural stiffness and durability were optimized. Despite a significantly more compact type of construction and reduced weight, the cylinder head achieves the required high stiffness, and thus guarantees, as the proven previous model series did, a very high mileage for the customer.

Supplying the cylinder head with a sufficient quantity of coolant, while simultaneously having smallest structural dimensions and an distinctively light-weight design, was achieved by the development of a new racing water pump with a mixed-flow design, ❹. It was possible to limit the total weight of the water pump including water guide, locking rotary slide and vacuum-operated actuator to just 1200 g. The basic concept of this water pump can also be implemented in modular form for the longitudinal versions of this engine design. In order to achieve optimum fast heating of the coolant, the water pump was equipped with a ball rotary slide, which, when starting from cold, enables what is known as the standing water function. Sequential activation of this rotary slide heats the thermal masses of the engine and the engine oil more quickly, but above all the actual cold-start phase of the engine is reduced significantly, leading to a faster activation of energy-saving measures such as start/stop or lean-burn operation and a CO_2 saving of around 1 %. The cooling of the exhaust gas turbocharger is also ensured in the event of standing water, in order to prevent overheating and coolant damage.

In order to minimize pressure loss and thereby friction power, the thermostat was also fitted with a ball rotary slide, identical to the water pump's rotary slide. When open (coolant operation), this rotary slide exhibits just minimum pressure losses. For a volume flow of 120 l/min, this is approximately 70 mbar compared to over 500 mbar for a corresponding three-disc thermostat. A pressure differential valve ensures that the customer's comfort requirements (heating system and climate control) are given priority in short-circuit mode.

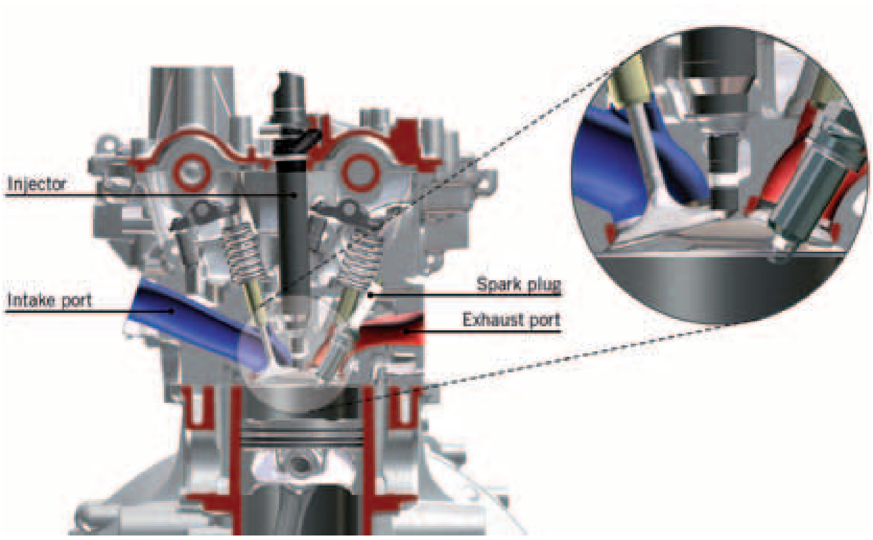
ENGINE OIL CIRCUIT

The engine is supplied with oil via a regulated vane-type oil pump of the latest generation. The oil flow and thereby also the operating energy input are regulated via the oil pressure in the main oil duct. By means of a characteristic map appropriate to the engine load and speed requirements, the target oil pressure is reduced additionally via a solenoid valve on the oil pump to further reduce the operating energy input.

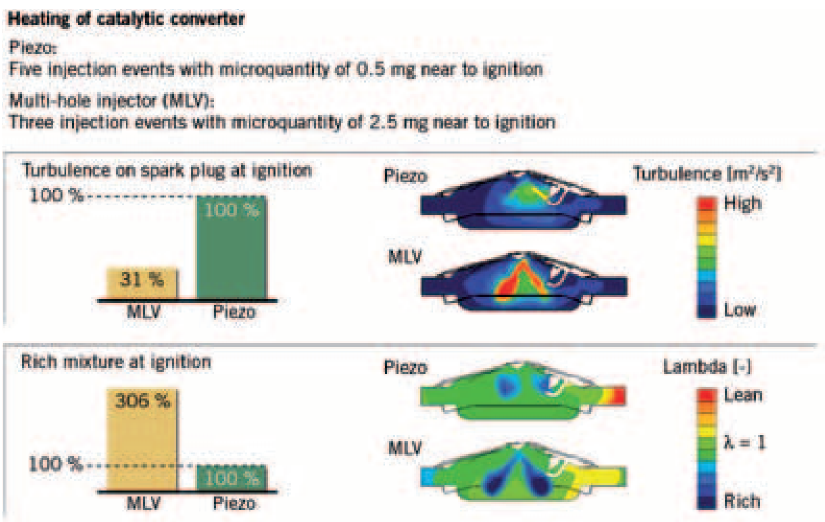
COMBUSTION SYSTEM

At Mercedes-Benz, the course has been set within gasoline engine development over the last few years for an innovative and sustainable technology portfolio that is the optimum base for the implementation of maximum fuel efficiency in all vehicle segments without sacrificing comfort or superior drive performance [4, 5]. Starting with the new V6 and V8 engine generation with the designation BlueDirect, the so called Mercedes-Benz combustion system, comprising the technology portfolio with third-generation direct injection, spray-guided combustion system, multiple-spark ignition (MSI) as well as integrated ancillary component and heat management, was launched [2, 3], ❺.

The centrally arranged injector represents the design basis of the spray-guided combustion system. This is an injector that opens outwards with a piezo actuator. Two characteristic benefits distinguish it primarily. Firstly, the nozzle opening outwards facilitates very good mixture formation properties in conjunction with simultaneously high spray stability and the lowest possible fuel wetting of the combustion chamber surface. Secondly, the injection system is very flexible as far as the minimum and maximum possible injection quantities are concerned thanks to the directly controlled outward-opening nozzle. Therefore a very large quantity spread from under 1 mg to over 150 mg can be realized. As a result, both the areas of supercharged wide open throttle and the microquantity requirements of the deceleration-like characteristic map area are covered without problems for the most varied single-cylinder volumes and all standard fuels up to E100. In this way, one standard component can be used for the entire Mercedes-Benz gasoline engine portfolio, from the 5.5 l eight-cylinder engine to 1.6 l four-cylinder engine [6]. The high stability of the mixture formation together with the ability to inject well prepared, very small quantities of fuel into the combustion chamber results in a very stable combustion system with appropriate local mixture composition. This is the basis for rapid and complete combustion, which means a low tendency to knock towards the end of combustion and low emissions.



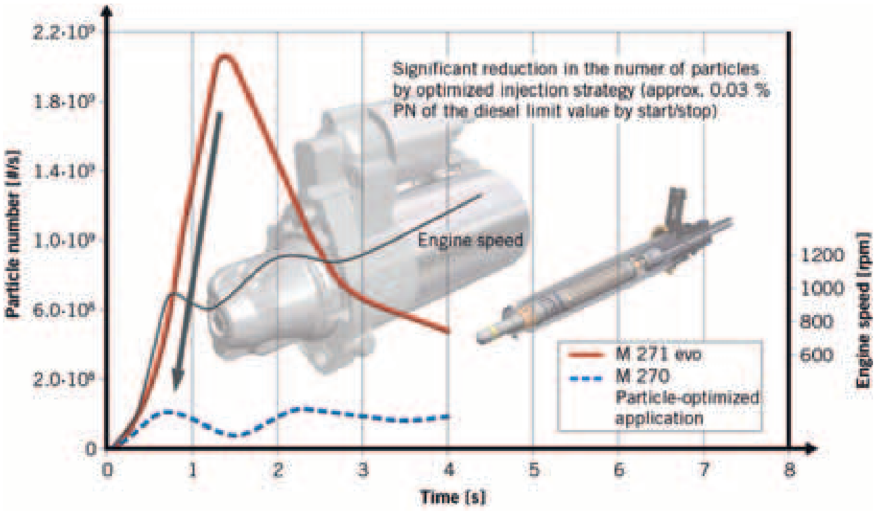
❺ Combustion system with centrally arranged piezo injection



❻ Mixture formation and turbulence distribution in catalyst heating operation (idling)

COLD START/WARMING UP

The reduction of fuel consumption and emissions are of great importance when developing an engine design. Particular attention is dedicated to the cold start and warming up behaviour. Given the need to comply with the emissions limits, the first part of the test cycle has already been of essential significance for many years. Until the conversion capability of the catalytic converter is reached, the resulting deterioration in efficiency causes an increase in the exhaust gas heat flow and the required fuel mass flow in this operating mode. In the case of single injection this leads to huge wall and piston wetting and poor mixture homogenization. The combustion system with its central injector location and spark plug located close by has the ideal prerequisites to be able to operate the engine with the required running smoothness, even in this operating state. The piezo injector used makes it possible to inject the smallest quantity of fuel for local mixture enrichment just before ignition, combined with a significant turbulence increase at the spark plug. This ensures a fast and repeatable ignition of the mixture. ❻ shows the turbulence and mixture distribution at the ignition point using the example of an idle point as the result of a CFD (Computational Fluid Dynamics) simulation in comparison to a current multi-hole injector (MLV). Thanks to the piezo injection valve used, the turbulence intensity achieved at the spark plug can be over three times higher despite a quantity of injected fuel that is five times lower. An important parameter with respect to the particle number limit designated in Euro 6 is the homogenization of the mixture. The piezo injector also exhibits significant advantages here thanks to the significant reduction of the sub-stoichiometric mixture areas at the ignition point. An adapted multiple injection can be used to significantly reduce the particulate emission. Based on three-fold injection, the relative particle number can be reduced by over 60 % using the piezo injector compared with the multi-hole nozzle. The five-fold injection optimized for particles is even 90 % below the level of the multi-hole valve. Particle reduction is based on an adaptation of the injection quantities and times as well as a reduction in the partial injection quantities, in particular the ignition injections to under 1 mg. Due to the exceptional microquantity capability, it is possible to increase the number of injections further and thereby to reduce the number of particles yet again.



❼ Particulate emissions in the NEDC by start/stop

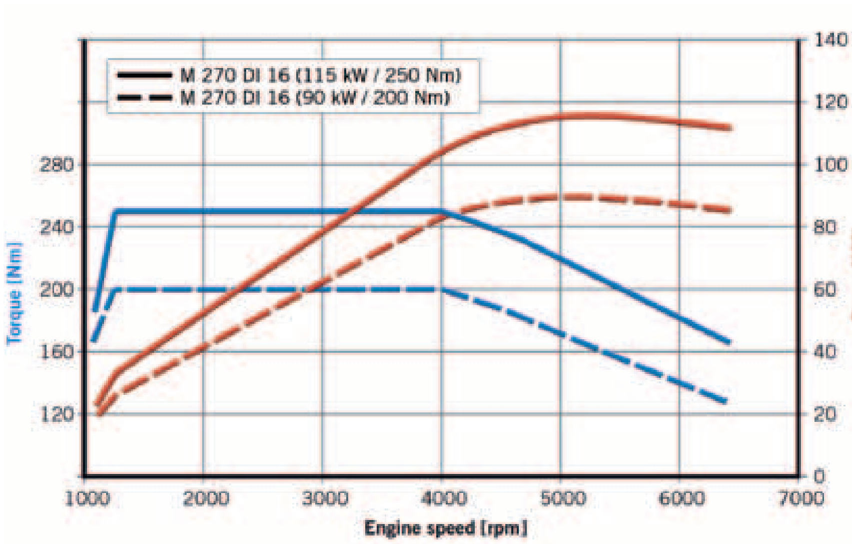
START/STOP

As a result of the high potential to decrease fuel consumption, start/stop systems are the basis for all new engine developments at Mercedes-Benz. In addition to the advantages of the starter-supported direct start/stop function with piezo injection, in light of the limiting of particle numbers in the NEDC, there is a further very important aspect [2]. As known, the start phases, in particular when the engine is not yet completely at operating temperature, are critical with regard to the contribution to the total particulate emissions in the test cycle. A start/stop system amplifies the emissions relevance of the start noticeably. The high precision, the good fuel preparation and the excellent microquantity capability of piezo injection provide significant potential for optimizing by means of application. Adapting injection quantities and times can thus reduce the particulate emissions in the start/stop phases to almost zero, ❼. In this way, the number of particles emitted by repeat starts in the NEDC could be reduced by over 98 % compared to the previous model. For the engine optimized in this way, all the start phases within an NEDC merely produce a negligible contribution of under 0.03 % in terms of particulate emissions, based on the very strict particle number limit for diesel engines.

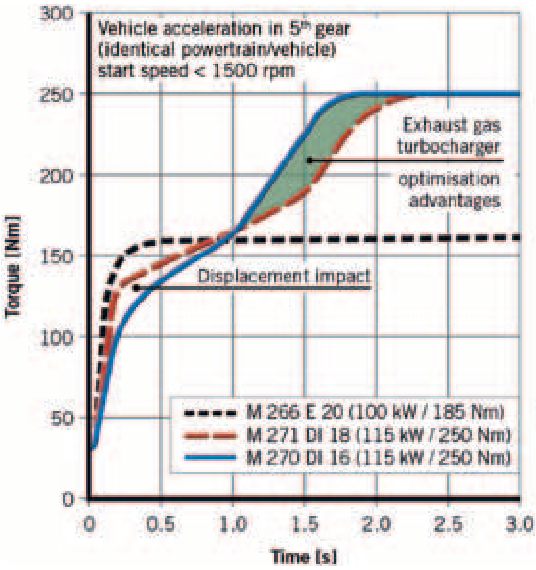
CHARGING AND GAS EXCHANGE

With the maximum torque already applied at 1250 rpm, the engines enable a wide speed range with high tractive power capability for both performance variants, ❽. This ensures a driving manner with reduced consumption at the same time as a high potential of agility.

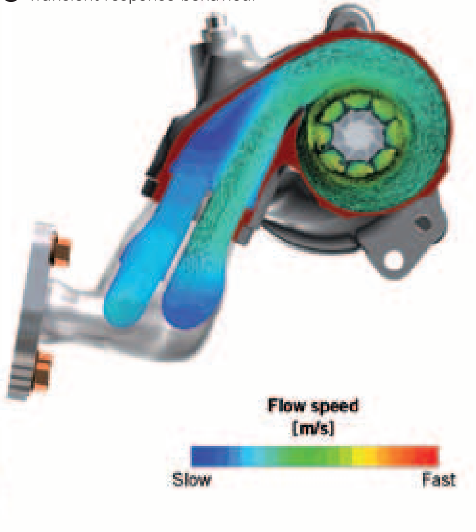
For the optimization of the gas exchange, minimizing the residual gas is very important due to knocking sensitivity. 1D and 3D simulations were used to investigate and valuate various measures. To do this, the width of the exhaust valve lift is shortened as much as possible to reduce the influence of cylinders and to get the residual gas advantage involved. The minor consumption disadvantages at nominal power are overcompensated by the reduction of the enrichment using a turbocharger with 1050 °C maximum temperature. Up to a vehicle speed of 200 km/h, the engine is operated stoichiometrically. By optimization of the twin-pipe exhaust gas manifold it was possible to get no



❽ Stationary power and torque values



❾ Transient response behaviour



return flow into the respectively other pipe and the flow through the spiral is thus uniform. In this way, the impulse turbocharge effect is completely retained and the entire spiral is available for each pipe at high flow rates. Target of all measures is the best possible boost pressure rise from partial load. The narrow exhaust camshaft with corresponding timing enables creating high purge air volumes. In addition, the consistent decrease in size of the wheel diameters both at the compressor and at the turbine made it possible to improve the response behaviour comparison with the larger-displacement predecessor naturally aspirated engine as well as with the larger-displacement turbocharged predecessor engine, ❾.

DRIVING CHARACTERISTICS AND FUEL CONSUMPTION

The B-Class technical data shown in ❿ reflects the result of the optimizations to the new engine described above. Compared to the previous vehicle, it was possible to reduce the NEDC consumption by 13 % with significantly improved driving performance. The outstanding response behaviour in conjunction with the very good low end torque not only results in excellent vehicle agility but also enables a very low engine speed level. As a result, it was possible to reduce the actual customer consumption significantly.

		B-CLASS (BR 245) M 266 I 20	NEW B-CLASS (BR 246) M 270 DI 16
Cylinder arrangement/number		I4	I4
Displacement	cm^3	2034	1595
Power output at speed	kW rpm	100 5500	115 5300
Nominal torque at speed	Nm rpm	185 3500-4000	250 1250-4000
Compression ratio		11:1	10,5:1
Acceleration 0-100 km/h	s	10.1	8.6
Maximum speed	km/h	196	220
Combined fuel consumption	l/100 km	6.7	5.9
Combined CO_2 emissions	g/km	158	138
Emissions class		Euro 5	Euro 5*
Inertia weight class	kg	1360	1360

*Meets already the Euro 6 emission limits still in discussion

❿ Driving characteristics and fuel consumption data in the B-Class

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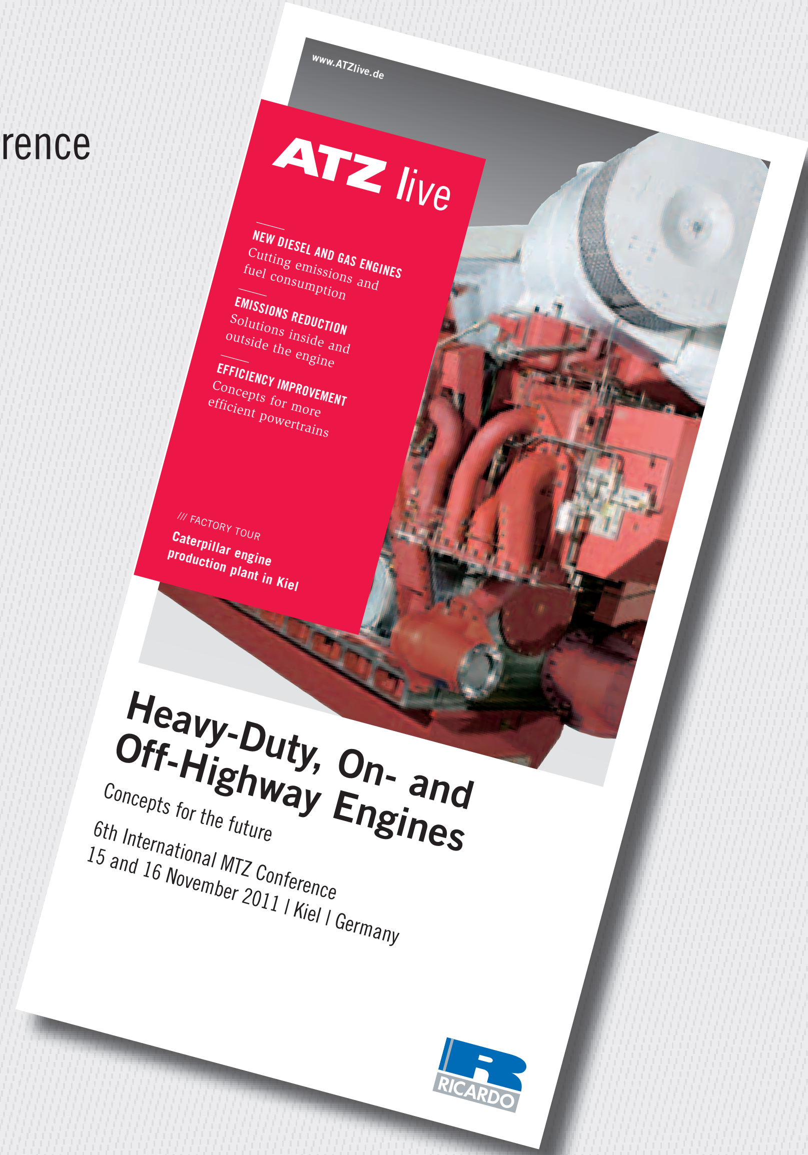

NEW DIESEL AND GAS ENGINES
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Solutions inside and
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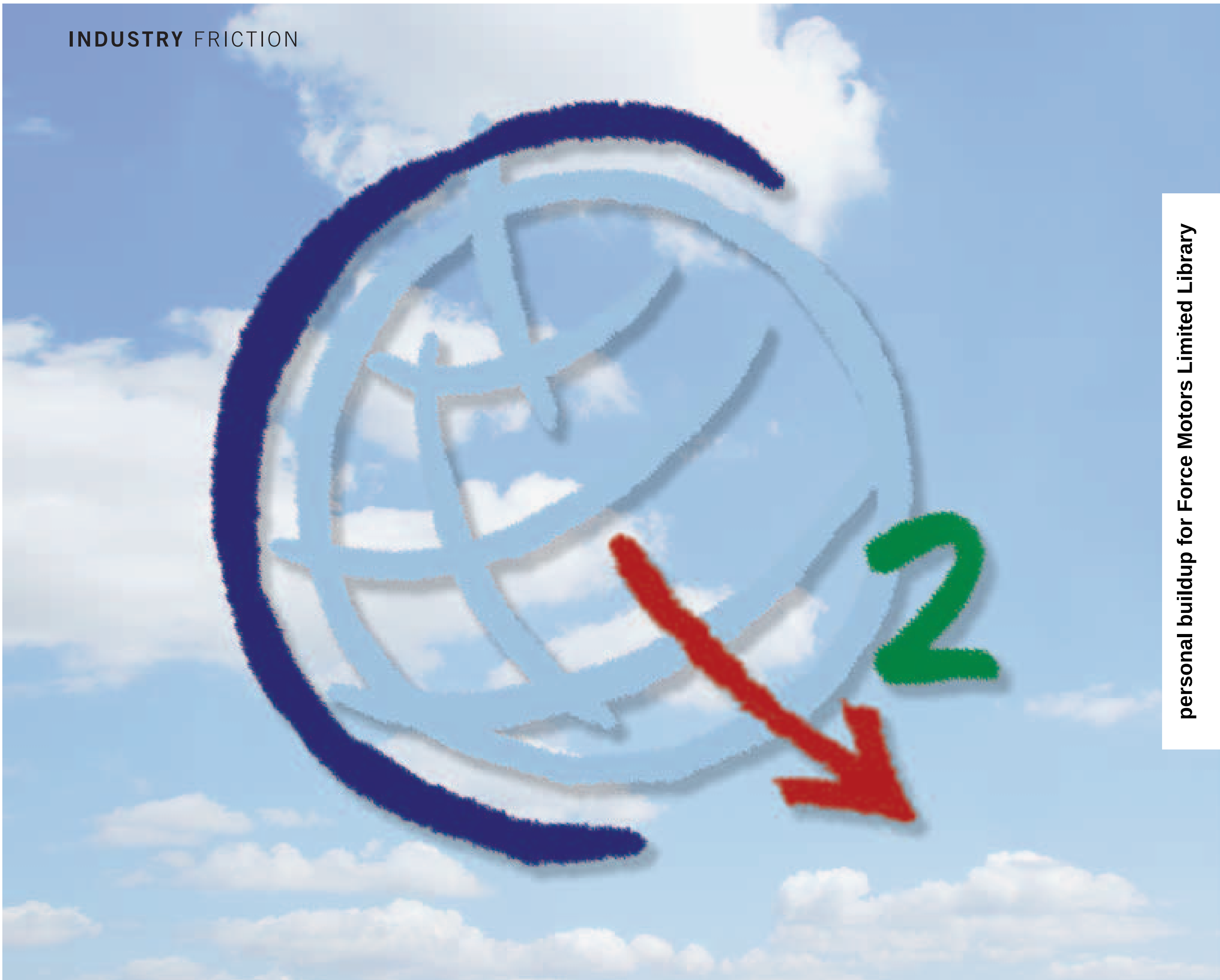

FRICTION POWER MEASUREMENTS OF A FIRED DIESEL ENGINE
CYCLE-RELEVANT CO_2 SAVING

Systematic tests performed at Mahle measured the effects of changes in individual design parameters of the piston group on frictional losses, providing conclusions about the reduction in friction mean effective pressure over the entire engine operating map. Using a cycle simulation program, together with additional vehicle data, the friction power differences can be converted into fuel savings in a driving cycle. The corresponding CO_2 savings potential can then be derived. The developer can thus design the engine mechanics for optimal friction loss according to criteria relevant to the cycle.

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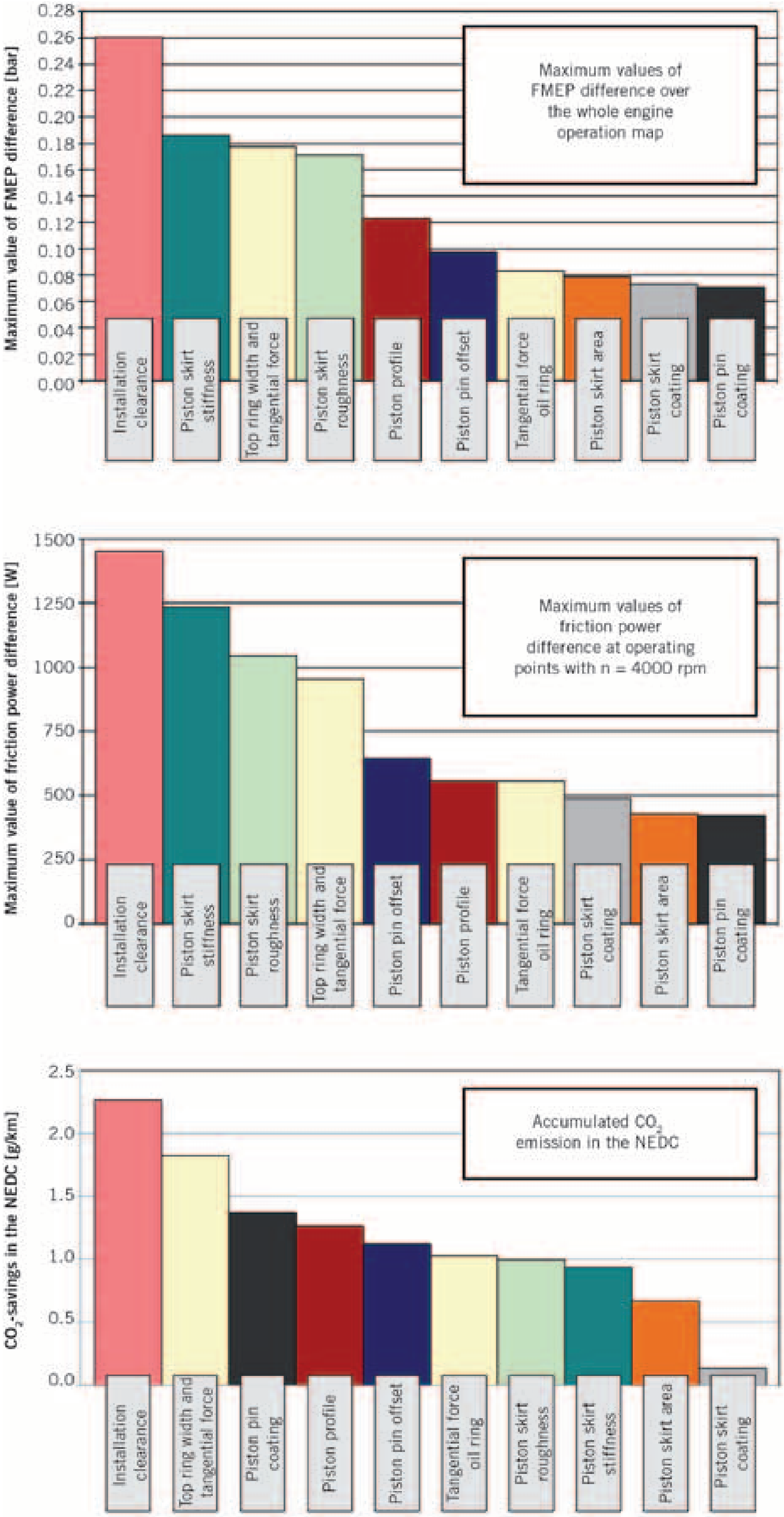
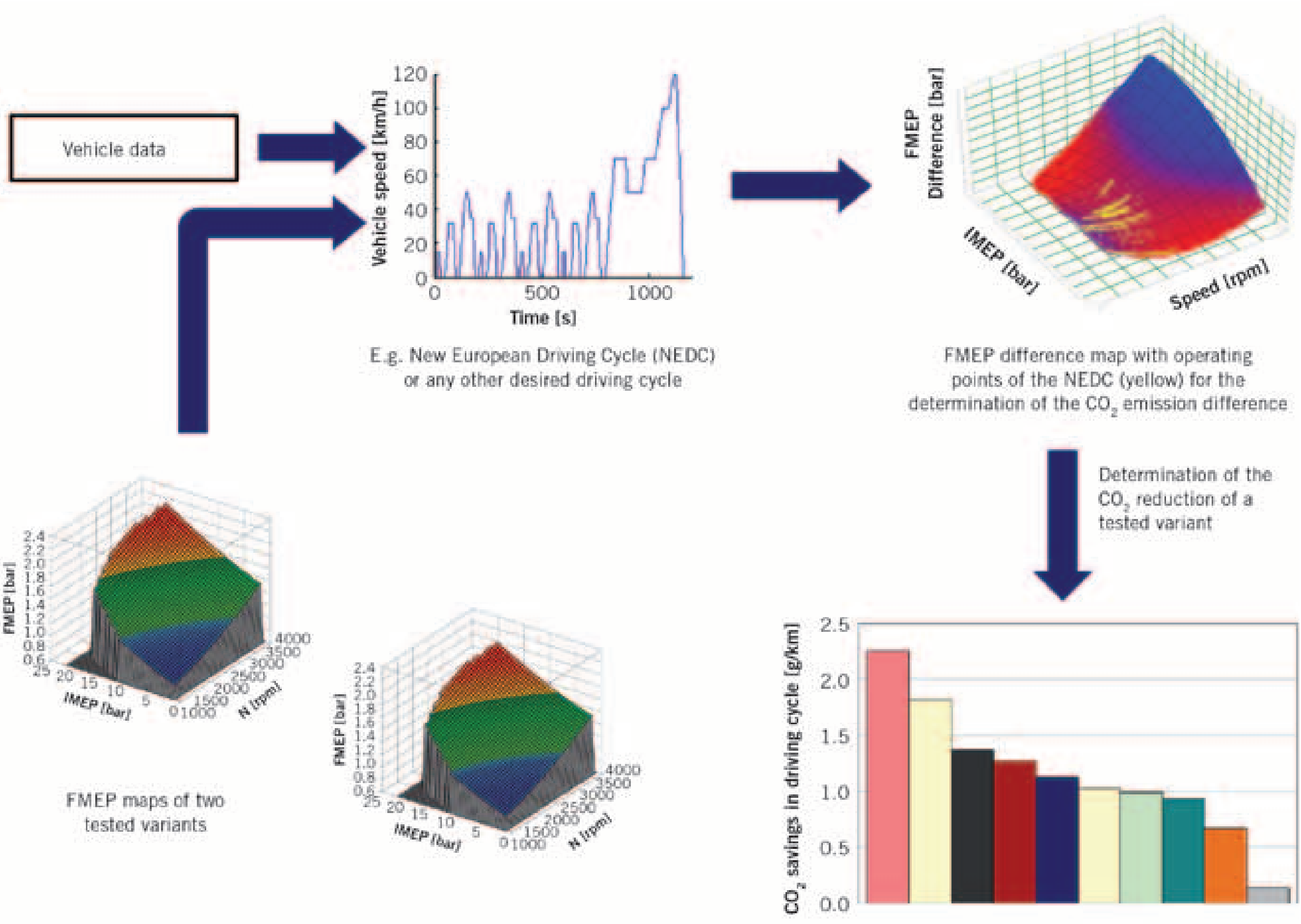
FRICTION POWER TEST BENCH AND DIFFERENCE MAPS

Friction mean effective pressure (FMEP) maps are determined in individual parameter studies on the friction power test bench, as described in detail in [1]. The difference between two such FMEP maps yields a FMEP difference map. This shows the effects of changes of individual design parameters on engine friction over the entire engine operating map [2]. Extensive parameter studies can then be performed to show the effects of the numerous individual parameters.



❶ Savings potential of various design parameters of the piston group; top: maximum values from FMEP difference maps; centre: maximum values from FMEP difference maps at 4000 rpm; bottom: accumulated CO_2 savings in NEDC



❷ Derived cycle-relevant CO_2 savings using FMEP difference maps

The results from the engine test bench can be directly displayed as the FMEP in bar. ❶ (top) shows a ranking of the maximum FMEP difference in the operating map for a number of experimental parameters. Superposition of the individual savings potential should not be performed because they were measured at different operating points. The friction power loss can be calculated from the combination of FMEP and the engine speed, so the maximum potential power savings can be displayed in a corresponding ranking for a particular speed, ① (centre).

Typically the focus of optimisation is not on the individual operating map points, but on the reduction in fuel consumption, and thus the CO_2 emissions, in a driving cycle.

It is very difficult to directly determine the fuel savings related to frictional losses for changes of individual parameters on an engine test bench, because many different efficiencies affect fuel consumption. These cannot be held sufficiently constant or be reproduced for all measurements. The variance of these effects can overshadow the expected result of the changes of individual design parameters many times over.

DRIVING CYCLE SIMULATION

The driving cycle simulation is a powerful tool for estimating fuel consumption and CO_2 emissions in a cycle. It provides a way to investigate changes in individual design parameters of the piston group with good accuracy and reproducibility, without affecting other efficiencies. This allows a friction specific optimisation of a component in a driving cycle to be represented using the CO_2 savings per driven kilometre.

The software package GT Drive was used for these analyses. The New European Driving Cycle (NEDC) was used as the baseline driving cycle for the analysis. The NEDC is, as is generally known, made up of a city phase, which consists of four equal speed profiles, and an extra-urban phase.

The cycle simulation programme uses a kinematic calculation method, where a vehicle speed profile and gear ratio are provided to the model, depending on the driving cycle being investigated. The angular velocities, efficiencies, and torques of the various vehicle components are then calculated backward along the powertrain, until the corresponding engine operating points have been determined. In addition to the FMEP maps derived from engine testing, geometric vehicle data and experimentally derived driving resistances are required, ❷.

The driving resistances (tyre rolling and aerodynamic drag resistance) are determined for the vehicle using coast down tests, conducted in accordance with the procedure in the European Directive of the Council 70/220/EEC. Additional data required for the vehicle and engine specifications have been taken from the manufacturer's data.

Using the GT Drive Model, the effects of various design parameters on fuel consumption and CO_2 emissions are evaluated. A flexible approach is needed for determining the fuel consumption from the FMEP maps of the variants tested. The corresponding procedure is described below, ❸.

Instead of deriving the instantaneous fuel mass flow directly from a BMEP versus engine speed map, the instantaneous brake mean effective pressure (BMEP) required during a driving cycle is first converted into an indicated mean effective pressure (IMEP). The FMEP maps measured on the engine test bench are used for this purpose. The IMEP is plotted as a function of the engine speed and the BMEP. At constant BMEP, a change in FMEP thus leads to a change in IMEP.

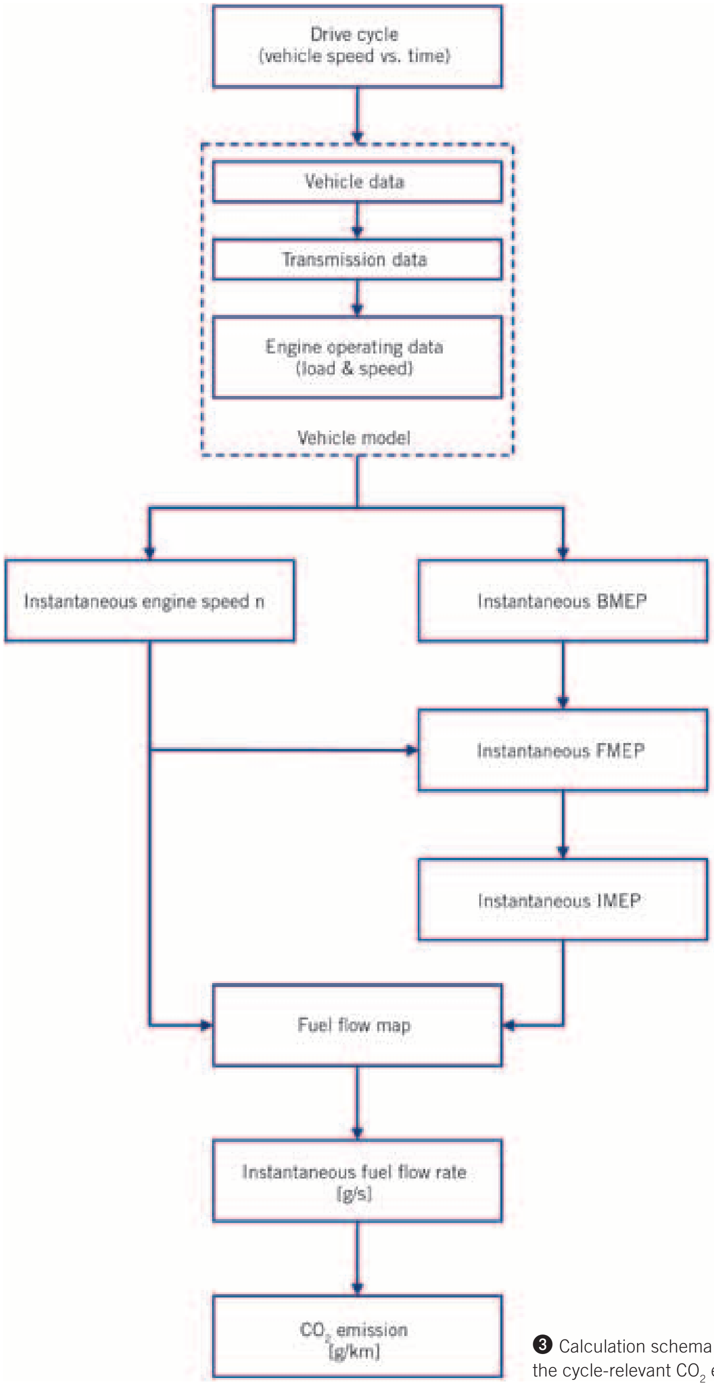
The corresponding instantaneous fuel mass flow results from a fuel consumption map, in which the fuel mass flow is plotted against the IMEP and the engine speed. This operating map is also derived from test bench measurements; however, it remains unchanged for all of the FMEP maps under test. This makes the strict assumption that combustion and internal engine heat losses are not affected by friction.

Because engine warm-up must be considered in the NEDC, the effect of a cold start on engine friction and the associated internal engine heat losses are simulated in the model. A simple correction factor is applied to the fuel mass flow for the warm-up period. This fuel correction factor was derived on the chassis dynamometer at Mahle Powertrain from the speed profiles that are repeated four times during the urban phase in the NEDC. The fuel mass flow is thereby measured individually for each speed profile, and then divided by the fourth profile. In the fourth profile, the temperature of the coolant and oil can be considered to be static. A linear equation (fuel correction factor) is then created, in which the elapsed cycle time is the main variable.

A test on the chassis dynamometer demonstrated the validity of the entire vehicle model. This results in a good correlation of the engine speed calculated from the model and the fuel mass flow, ❹. The good correlation of the engine speed confirms the correct selection of the transmission ratios and effective tyre rolling radius in the model. ④ (bottom left) shows the consistency in the dynamic fuel mass flow. The calculated accumulated fuel consumption of the model, ④ (bottom right) shows only a slight deviation of 0.85 % in the city phase. In the extraurban phase, the resulting deviation is only 0.04 %.

The accumulated CO_2 emissions can be derived from the accumulated fuel consumption. The difference between two variants then yields the corresponding CO_2 savings in the driving cycle being investigated.

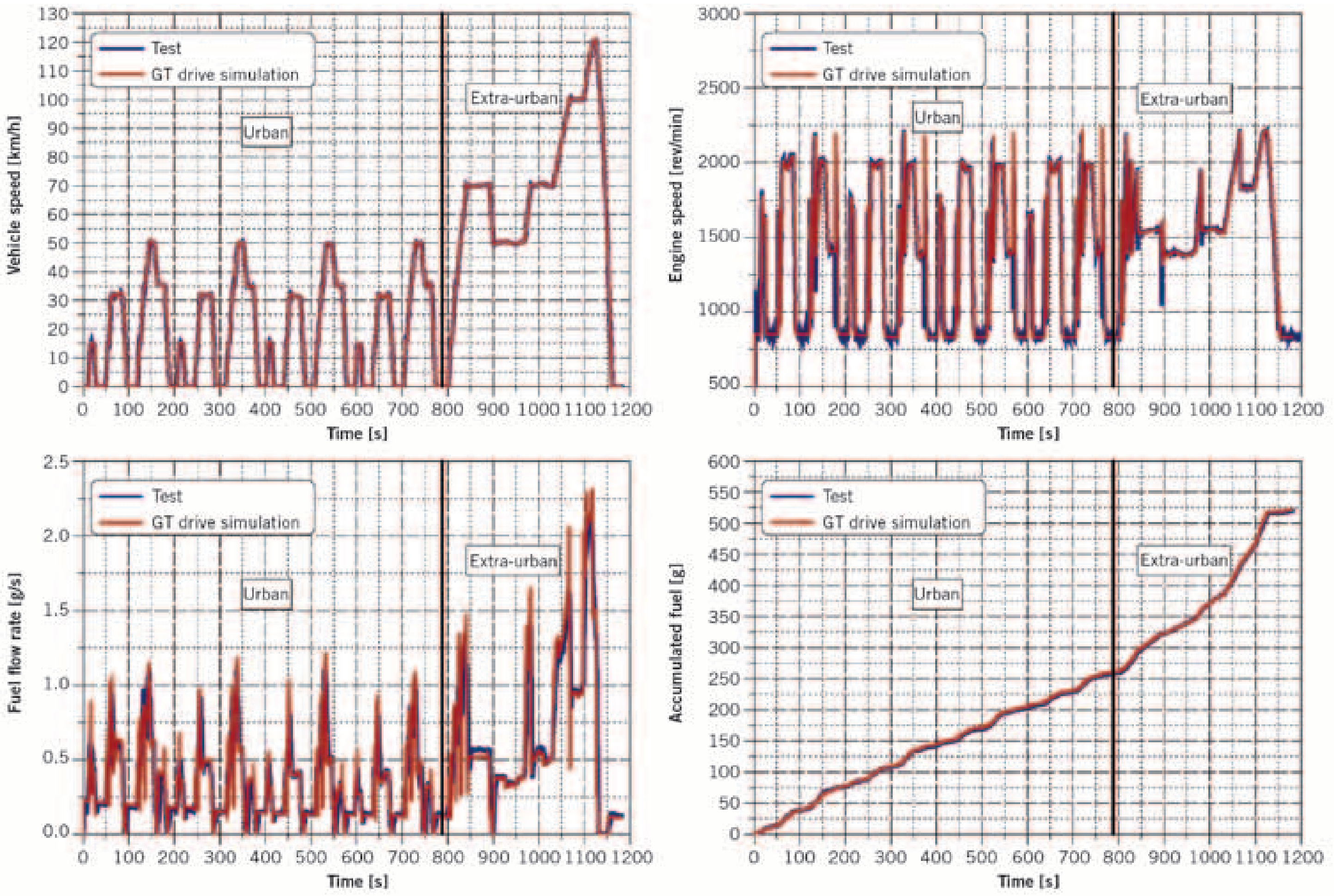
Using the existing vehicle model, additional driving cycles can be investigated, such as the ARTEMIS cycle, or customer-specific running profiles. In addition, the model can be used to demonstrate the effect of a stop-start system on fuel consumption and CO_2 emissions in a driving cycle.



❸ Calculation schema for determining the cycle-relevant CO_2 emissions

RESULTS

The ranking in ① (bottom) shows the results of the driving cycle simulation. The



❹ Validation of the GT Drive model in the New European Driving Cycle (NEDC)

CO_2 savings in the NEDC are shown in grams per driven kilometre. A comparison with the corresponding rankings in bar and Watt illustrates that the significance of the individual parameters varies. The installation clearance is the parameter with the greatest effect in all three display types. The parameters piston profile and pin offset do not significantly change position in the rankings. Also the changes to the top ring and oil ring show the same potential in all three rankings. The parameters of skirt roughness, skirt stiffness, and pin coating, in contrast, are particularly notable as they react very severely to the collective load due to their varying effects in the FMEP difference map.

Superposition of the CO_2 savings in the NEDC, ① (bottom), is possible, at least with some conditions, as each parameter tested was based on the same operation points in the NEDC. Any possible interactions between the individual design parameters must be noted. It is therefore recommended that the set of changes derived should then be tested as a variant in an engine.

CONCLUSION

Using the procedure presented, a developer can align the mechanical design of the engine with the criteria of real driving operation. Widespread investigation has allowed the targeted combination of many individual design parameters to reach the CO_2 reduction for different engine concepts. The fuel savings thus achieved can be attributed exclusively to improving mechanical efficiency, i.e., minimising mechanical friction, and not to changes in combustion conditions.

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THE EFFICIENCY ENGINE – COST-EFFECTIVE ALTERNATIVE TO DOWNSIZING

As an alternative to the familiar option of downsizing diesel engines, AVL has developed the so-called efficiency engine in collaboration with Renault. Because of the engine's moderate power density, its peak pressure requirements are lower in some areas than those of turbocharged gasoline engines. Consequently, its mechanical friction and fuel consumption can be significantly reduced, as the comparison with a conventional and a downsized diesel engine demonstrates.

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REDUCED FRICTION AND FUEL CONSUMPTION

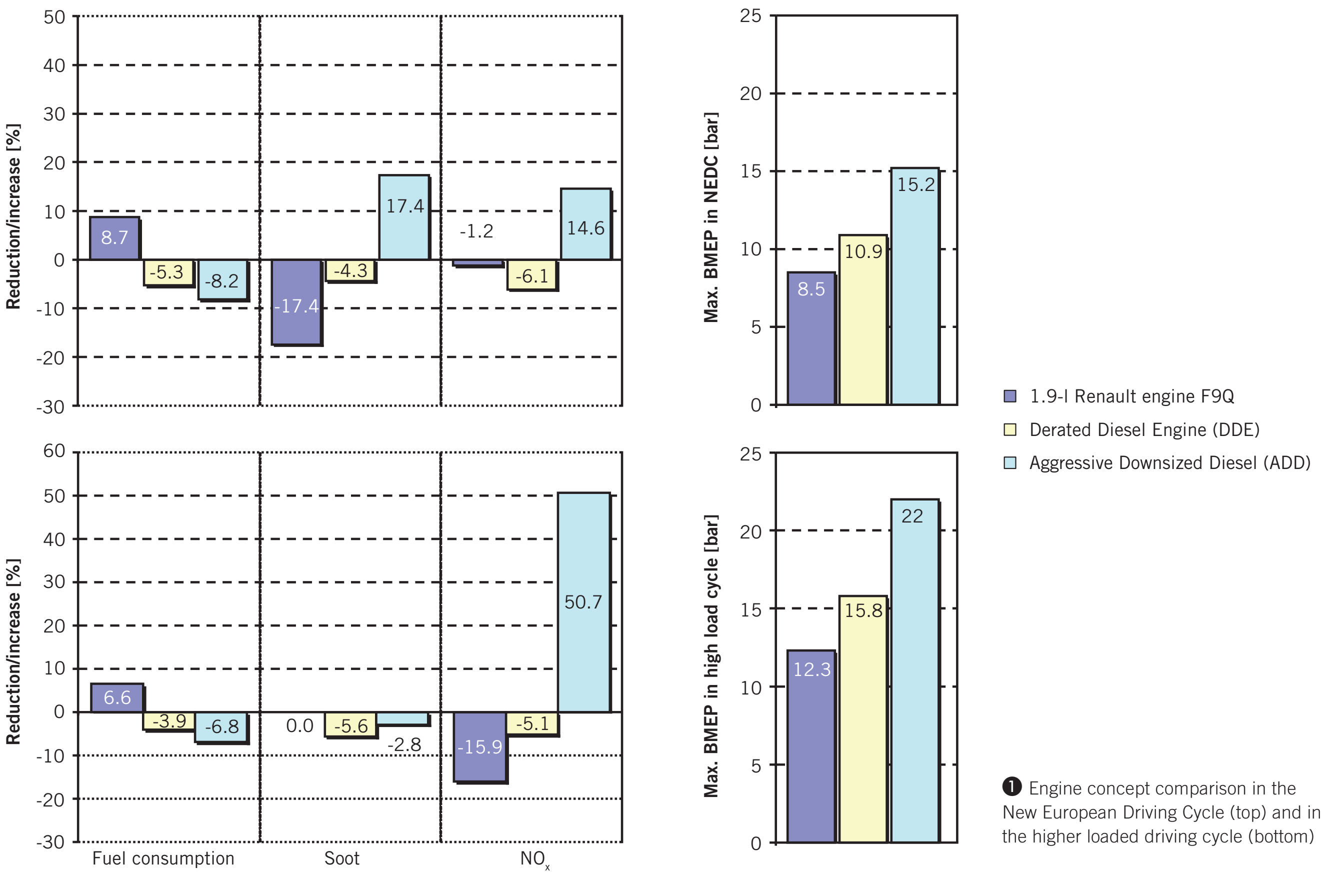
Reduction of fuel consumption and costs dominate the future market requirements in drive train development. Alongside electrification, downsizing concepts currently play a major role in CO_2 reduction. However, this route means a further cost increase for the diesel engine in order to offer acceptable performance in heavy vehicles using engines with reduced displacement. Main cost drivers are the modified fuel injection and boosting-system for increased specific power density as well as the higher efficiency requirement for the NO_x aftertreatment system.

An attractive alternative to a classical downsizing concept is an efficiency-engine concept, optimized for a more moderate power density and minimized frictional losses. In this case, instead of reducing swept volume, the specific power density is reduced to 45 kW/l in the first generation, thus covering the needs of the high volume market. For this moderate power density the required peak firing pressure is reduced to a value even lower than those of some turbocharged gasoline engines. With a consequent modular redefinition of the crank train the friction can be significantly reduced, achieving a CO_2 reduction corresponding to a swept volume reduction by 20 to 25 %. In this paper the details of such an efficiency engine concept (DDE: Derated Diesel Engine) will be discussed in comparison to a typical downsizing concept (ADD: Aggressive Downsized Diesel) [1].

SELECTION OF ENGINE SWEPT VOLUME VERSUS POWER DENSITY

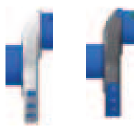
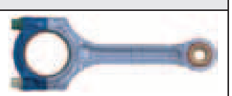
In this article the selection of engine size for a given vehicle definition is made considering different driving cycles since the NEDC (New European Driving Cycle) will be replaced by the WLTP (Worldwide Harmonized Light Vehicles Test Procedure) cycle. The final definition of the new cycle was not available at the time this paper was written, so a test procedure with more focus on higher load and considering more transients than the NEDC was defined for the simulations, thus covering the expected trend for the new cycle definition. The comparison starts with the internally named F9Q 1.9 l engine which was quite typical for this vehicle class (Renault Laguna) in the recent past. The DDE efficiency engine is based on a Renault K9K engine (1.46 l) as described in detail in the next chapter, while the downsizing concept (ADD) is represented by a 1.05 l three-cylinder engine with 80 kW. This downsizing concept was also defined for Renault and has already been presented in detail in a publication [2].

The results of this comparative vehicle simulation can be seen in ❶. The already existing K9K with 1.46 l swept volume serves as base for this relative comparison. All listed percentage changes relate to this engine size. In ① (top left) the NEDC results are listed. Applying the efficiency engine with reduced friction, the fuel consumption can be reduced by 5.3 % while soot and NO_x emissions are reduced by 4.3 or 6.1 % respectively while keeping the swept volume constant. With the 1.05 l engine the fuel consumption decreases by 8.2 % while NO_x and soot emissions increase significantly due to the shifting of the operation area. An emission optimized combustion strategy in the higher load area considering low pressure EGR [3] is already considered in this concept. In



❶ Engine concept comparison in the New European Driving Cycle (top) and in the higher loaded driving cycle (bottom)

① (bottom) the simulation results for the higher loaded driving cycle are plotted. A significant increase in NO_x emissions and deterioration of the NO_x/soot trade-off can be seen at first glance while showing otherwise quite similar trends. This is due to the different characteristics of the driving cycles. The average power required for the given vehicle model amounts to 4.24 kW in the NEDC, while the value for the assumed future driving cycle goes up to 9.26 kW, around double the value. The maximum engine load in the NEDC reaches 15.2 bar BMEP, while up to 22 bar are required in the high loaded cycle. These results collectively clearly indicate that aggressive downsizing concepts have to be seen critically in view of new, higher loaded driving profiles. At the same time it is seen that the efficiency engine can be an attractive approach not only in view of system costs.

		Basis	DDE	
Crankshaft				
Main journal diameter	mm	48	41	
Main journal width	mm	25	25	
Crank pin diameter	mm	44	40	
Crank pin width	mm	22.2	24	
Weight	kg	10.2	9.2	
Piston assembly				
Piston pin diameter	mm	26	20	
Compression height (KH)	mm	41.75	36.5	
Ratio KH/D	%	54.9	48	
Piston cooling	–	Gallery	Undercrown	
First piston ring height	mm	2	1.5	
Second piston ring height	mm	2	1.5	
Third piston ring height	mm	2.5	2	
Weight	g	603	457	
Connecting rod assembly				
Conrod ratio λ	–	0.301	0.289	
Manufacturing process	–	Forged	Sinter forged	
Weight	g	578	442	

❷ Dimensions of cranktrain components

PROTOTYPE IMPLEMENTATION OF EFFICIENCY ENGINE

Having defined the system requirements, the next step towards engine hardware was to define the basic engine architecture and dimensions. Here a systematic approach, supported by experience (benchmarking) and simulation tools is applied to ensure that the system level targets are fulfilled. For the efficiency engine the focus was on minimizing the frictional losses from the cranktrain and the parasitic losses from oil and water pumps.

The defined peak firing pressure for the DDE concept is only 110 bar; the gas forces are hence 30 to 40 % lower than a typical state of the art Euro 5 diesel engine. A careful redesign of the complete cranktrain according to the lower loads was performed, ❷. Significant benefits are realized in terms of reduced mass and friction.

At the pistons, the lower gas forces allow a reduction of piston pin diameter from 26 to 20 mm. A reduction in wall thickness below the combustion bowl and the elimination of the piston cooling gallery due to the moderate specific power together permit a reduction in piston compression height to only 48 % of the cylinder bore without requiring bearing bushes in the piston.

The lower gas forces also allow a reduction in heights and tensions of the piston rings to a level more familiar from gasoline engines with corresponding reduction of friction.

The small end of the connecting rod is redesigned to the reduced pin diameter and the shank cross-section adapted to the lower gas load. The big end is also reduced to suit the optimized crankpin diameter. The lower piston height allows an increase in conrod length of 5.25 mm which gives an additional friction benefit due to the reduction of side forces at the piston-liner interface. Nevertheless the weight of the conrod assembly was reduced by nearly 25 %. The application of cost-intensive sputter bearings at the big end could be avoided.

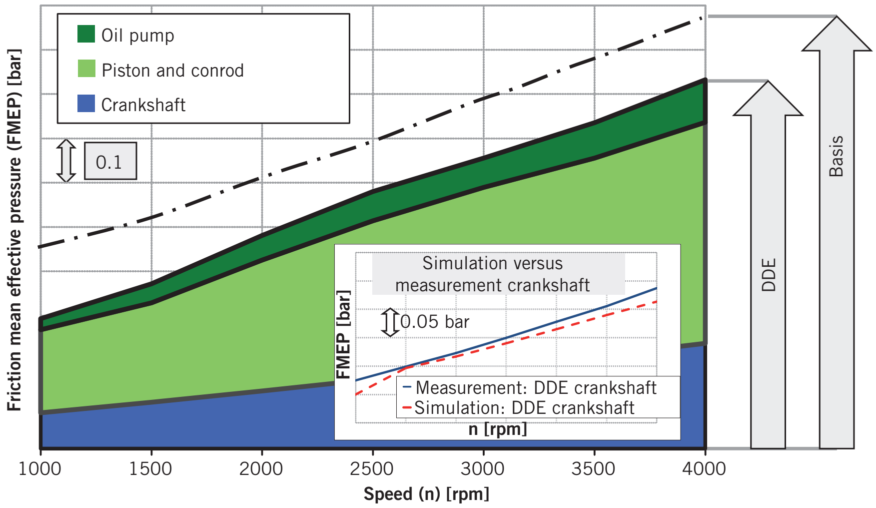
At the crankshaft, the focus is on minimization of diameters of the main and crankpin bearings. In the first step, parameter variation was performed using FEM analysis of a single crankweb, based on a metric of bending deflection under gas load. The new layout was compared to a wide range of proven designs based on this metric. The resulting layout was then validated using the full 3D Multi-Body-Simulation and Finite Element capabilities of AVL Excite, including elasto-hydrodynamic simulation of the bearings. It is noted that the crankshaft design is here limited by its stiffness – and hence the edge-loading of the main bearings – and not directly by the material fatigue strength as was the case for the ADD concept [2]. Including the proportional reduction of the counterweights, an overall saving in cranktrain mass of around 10 % was achieved.

The friction benefit of each of the measures described above was assessed individually with the help of simulation tools, ❸. The predicted reduction of 0.15 bar at 2000 rpm compares very well with the strip-down measurements which have been performed with modified engine hardware based on the K9K engine. For further refinement of the friction prediction capabilities in future, AVL has developed a new one-cylinder research engine which can measure frictional forces in the piston-liner interface under operating conditions up to 140 bar cylinder pressure and 4000 rpm engine speed, ❹.

Since the mechanical loads on the DDE are comparable to a high performance turbocharged GDI engine, the concept fits well to a modular engine family line-up with common components and production facilities for diesel and gasoline variants [4].

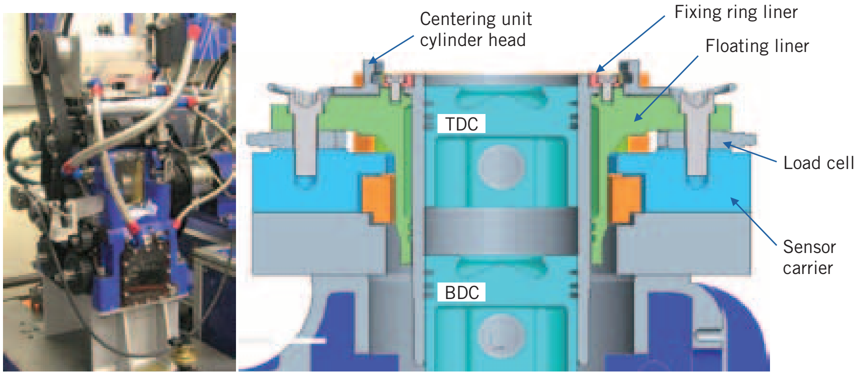
MEASUREMENT RESULTS

After completion of the friction strip-down measurements, the efficiency engine was installed on the development testbed. In ❺ the part-load traces for 2000 rpm are plotted. In the reference points at 2 bar BMEP an excellent brake specific fuel consumption below 290 g/kWh in combination with attractive NO_x figures has already been achieved after a very short development period. At higher load areas the benefit in specific fuel consumption is reduced because the internal engine friction is not as dominant anymore. Compared to the scatter bands in grey color the excellent trade-off between fuel efficiency and NO_x emission is evident. For future developments, a further fuel consumption potential is seen by shooting for somewhat higher NO_x emissions, assuming that NO_x aftertreatment will be the main route for the Euro 6 fleet. For future planned vehicle application it can be assumed based on these early results that similarly attractive fuel efficiency figures as with a downsiz-



❸ Friction measurement and friction simulation

❹ Research engine for friction development

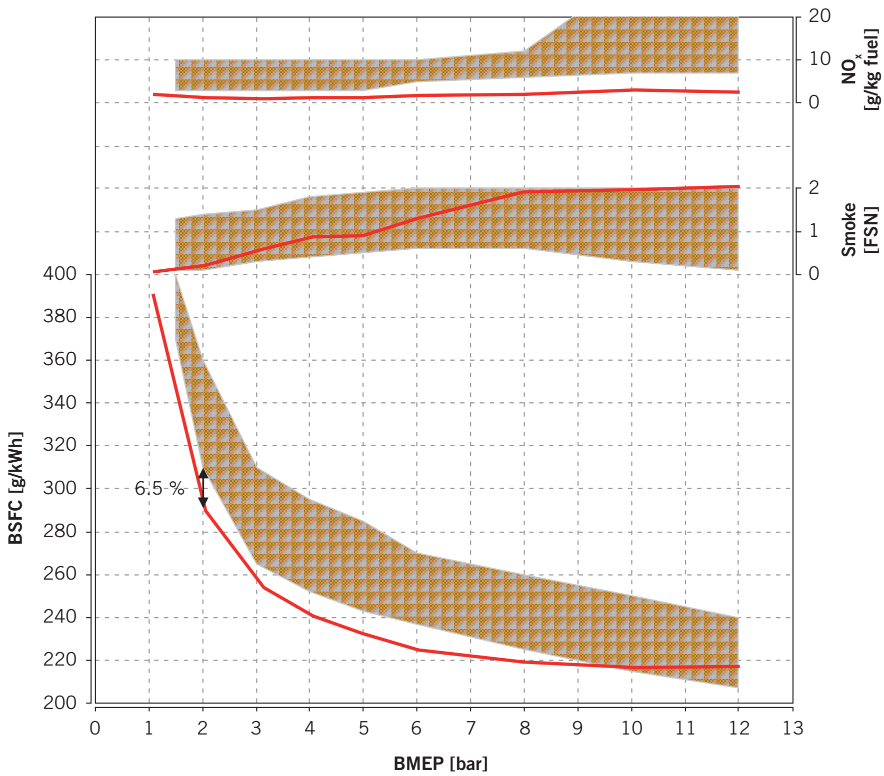


ing concept can be achieved; however with clearly reduced system costs.

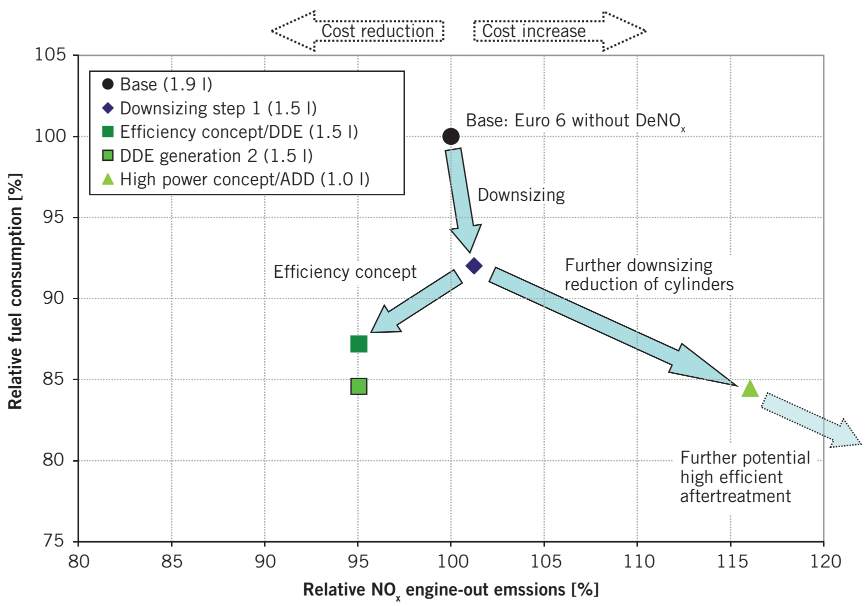
SUMMARY AND OUTLOOK

With the derating approach (efficiency engine) also benefits with respect to system costs and complexity can be seen. The efforts for boosting, fuel injection system and NO_x aftertreatment are significantly reduced. This cost effective engine has the potential for highest production figures due to global market chances, thus offering additional benefits in the cost structure. The customer will enjoy reduced maintenance costs and improved comfort compared to a three-cylinder downsizing concept. This derating program has been performed in close cooperation with Renault, taking the existing 1.46 l K9K engine as base.

In the derating program production boundaries of the existing facilities had to be considered; additional potential for the efficiency engine could be found if more freedom is available when defining a new engine family from scratch. ❻ summarizes the results collected. Downsizing promises lowest CO_2 values when combined with a high efficiency NO_x aftertreatment system. Current development is focussed on independent systems providing attractive reduction potential in an extended operation area. The biggest challenges at the moment are the system costs and further increased complexity. The efficiency/derating concept currently offers an attractive solution for the cost sensitive high volume market combining improved fuel economy with system simplification and potential cost reduction, at the same time offering an attractive base for hybridisation.



❺ Part load traces efficiency engine at 2000 rpm



❻ Concept comparison with respect to NO_x and fuel efficiency

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DIESEL ENGINE DEVELOPMENT FOR EMISSION STANDARDS IN EMERGING MARKETS

Following continuously strengthened fuel consumption regulations all over the world, the share of diesel-powered passenger cars increases as function of time. While approaching in parallel also challenging emission standards in emerging markets, additional emission control technologies like particle oxidation catalysts or diesel particulate filters have to be introduced. FEV GmbH describes the various challenges which arise therefrom for emission compliance and delivers potential solutions for implementation of capable exhaust aftertreatment systems under the given local conditions.

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MOTIVATION

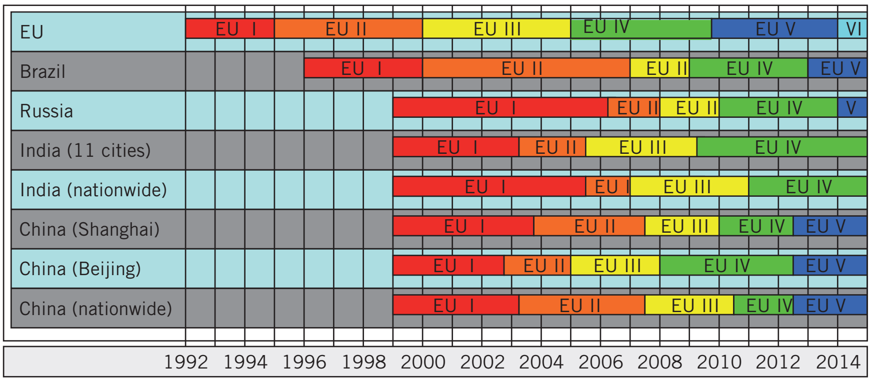
The global distribution of the passenger car sales has changed in recent years shifting from the classic western markets Europe and North America to the emerging markets. Among the latter in particular to the so called BRIC states (Brazil, Russia, India, China) which are the ones with significant vehicle market growth. With initially primarily vehicles with gasoline engines being sold, sales of vehicles with diesel engines are increasing. The present publication is intended to illustrate the challenges and efforts that are linked with sales of passenger car and light-duty truck vehicles with diesel engines in these countries. Based upon the explanation of the legal and other state specific differences relative to Europe, in a first step the raw emission potential for compliance of future emission legislation is discussed. In a second step the aftertreatment concept scenarios for the mentioned markets are explained, focussing on the aspect of local fuel quality.

SPECIFIC BOUNDARY CONDITIONS OF THE BRIC STATES

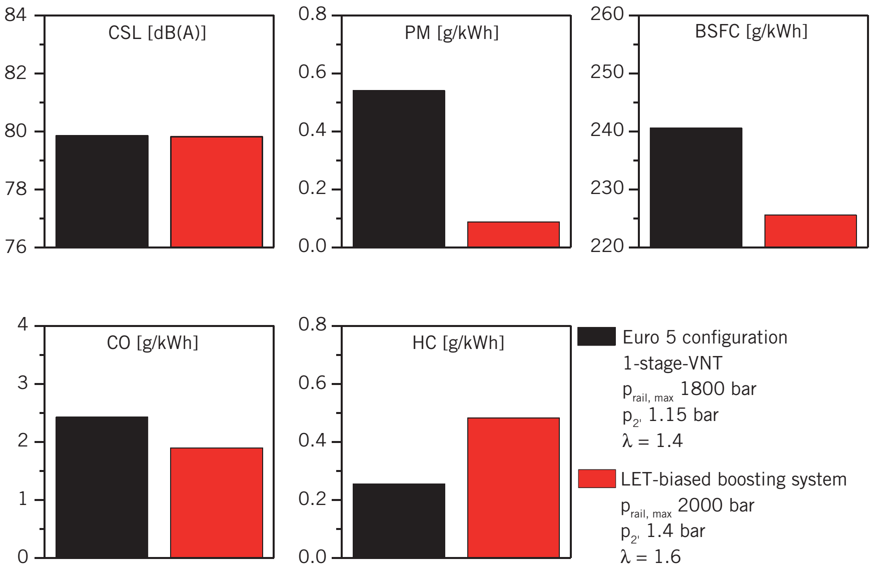
❶ shows the time schedule for the introduction of emission legislation levels in the BRIC states. The following conclusions can be drawn based on this: Firstly, all displayed states are dragging behind at least one emission level relative to European legislation. For today's reference meaning that while having Euro 5 active in Europe the majority of the other states still features Euro 4 legislation. In addition it is obvious that at least in India and China legislation differs between urban areas (India: 11 metropolitan centers, China: Beijing and Shanghai) and the rest of the state. As expected, in these congested urban areas legislation is stricter since it requires earlier introduction of a specific legislation level. Regarding Euro 5 emission legislation, China does not distinguish any more between regions and introduces nationwide validity in 2012. Thus for the Chinese market the demand for a Euro-5-compatible emission concept for vehicles with diesel engines is rather urgent. Possible solutions to this question will be presented in this publication while considering associated boundary conditions.

In addition to the emission levels, legislation in the BRIC states also differs regarding the test cycles. While Russia and China use the regular New European Driving Cycle (NEDC), emission levels in Brazil for light duty vehicles have to be met in the FTP75 cycle and in India in a NEDC cycle with a maximum vehicle speed of 90 km/h. Besides that, state-specific tests for emission approval exist, e. g. opacity measurements during engine rev-up at certain sea levels.

A second fundamental difference to the European car market is the climatic boundary conditions which need to be considered during the vehicle application phase. Typically, the European car market is covered with environmental temperatures of -25 °C to +40 °C and altitudes up to 2500 m. Opposed to that, in the Russian market significantly lower temperatures than -25 °C occur influencing both cold start application and real life driving behaviour. Icing issues of certain engine or vehicle components can be associated hardware challenges. Regarding altitude protection, in particular the South American market requires approximately 4000 m of maximum sea level coverage. Exemplary topics as turbocharger protection, high altitude cold



❶ Implementation road map of European emission legislation in BRIC states



❷ Potential of increased boost and injection pressure at n = 1500 rpm, BMEP = 6 bar

start behaviour, regeneration and overrun operation need to be considered and worked on accordingly.

A third important aspect being different to the European market is scattering fuel quality with various critical components. In this context water content and the content of catalyst poisons – here in particular sulphur – have to be seen. While higher water content can be tackled with dedicated water separators, the sulphur content is exceptionally critical. In the BRIC states a sulphur content of up to 5000 ppm can still be fueled, which in contrast to the European limit of 10 ppm has a dramatic effect on the after treatment system components. These effects will be described in more detail in the next paragraphs.

POTENTIAL FOR REDUCTION OF ENGINE-OUT EMISSIONS

The technical definition for typical Euro 5 diesel engines targeting the western European market does not vary significantly from OEM to OEM. Typical Euro 5 engines are equipped with a cooled High Pressure Exhaust Gas Recirculation (HP-EGR) in combination with high-performance fuel injection systems. This technology package can typically achieve raw particulate emissions, depending on vehicle category and requirements for vehicle acoustics between 20 and 40 mg/km in the NEDC. The use of a wall flow DPF allows secure compliance with the Euro 5 PM (Particulate Matter) emission limit of 5 mg/km with reasonable DPF regeneration intervals. This approach is subject to certain risks and disadvantages in emerging markets, since varying fuel qualities and mostly low-load operation in urban areas complicate a safe DPF regeneration. Therefore, a reduction of particulate emissions in NEDC below 10 mg/km is desirable, which would make an engine concept with an open DPF feasible. An increase of engine-out PM emissions would in principle be acceptable, if the efficiency of an open DPF is increased, e. g. by an adapted design. However, such developments can often not be implemented due to packaging constraints. In particular, there is a risk of filter overload, which increases by use of low-quality fuel which is not uncommon in emerging markets.

Starting from a standard Euro 5 concept with wall-flow DPF the potential to reduce engine-out particulate emissions is subsequently evaluated. To assess the potential to reach such low engine-out PM emissions, the operating conditions contributing the most to the overall cycle PM emissions have to be identified at first. An analysis of emission testing in the certification cycle (NEDC) for different vehicle classes shows that PM emissions originate mainly from the operation at medium and high engine loads between 5 and 15 bar BMEP. For a reduction of PM emissions in these areas the following measures are considered as promising:

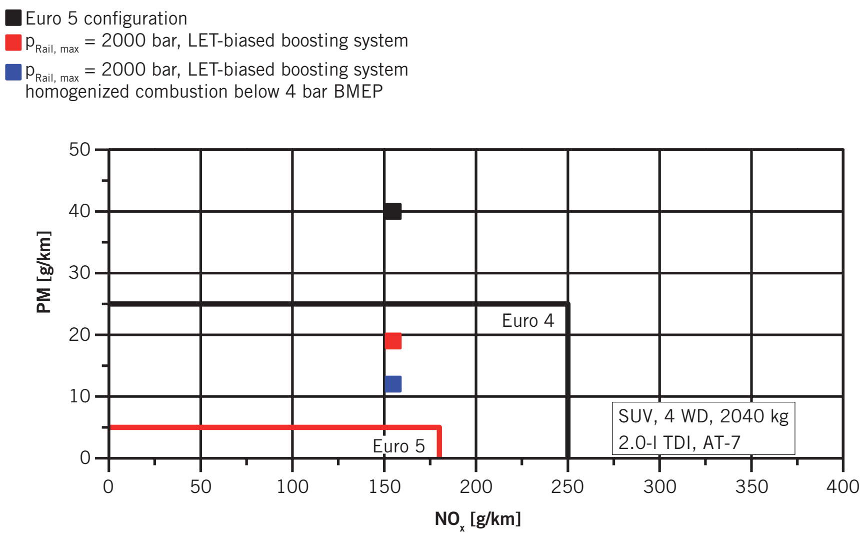
: further development of fuel injection system (increase of injection pressure)
: increase of boosting level
: homogenized combustion
: low-Pressure EGR
: increase of HP-EGR cooling capacity.

Robustness issues prevent the implementation of a low pressure EGR without a wall-flow DPF. Furthermore, a further increase of HP-EGR cooling capacity does not seem be a reliable solution, as the engine configuration used as baseline for this study is already equipped with a powerful EGR cooler, with a maximum EGR cooler outlet temperature only slightly above 120 °C under fully warm operating conditions. A further reduction of EGR cooler outlet temperature would increase the risk for EGR system fouling significantly. Therefore major improvement of engine-out emissions can mainly be expected from increased boosting levels and usage of fuel injection equipment with higher pressure capability. The potential of these measures is exemplarily shown in ❷ by comparing a conventional single stage boosted engine equipped with a 1800 bar injection system against an engine with a further developed boosting system biased towards high levels of low end torque and a fuel injection equipment with 2000 bar pressure capability at 1500/min and 6 bar BMEP.

The operation with leaner rel. air/fuel ratio and the improved mixture formation by usage of nozzles with smaller spray holes lead to a massive reduction of engine-out PM emissions, which result in halving the cycle PM emissions if this technology package is calibrated for the complete engine map. NVH performance is not adversely affected by these measures.

A reassessment of PM-critical map areas now indicate that low load operation below 4 bar BMEP contributes significantly to the overall cycle PM emission result. Under these operating conditions, the implementation of a homogenised combustion system to realise a more premixed combustion characteristics is advantageous. However, it needs to be taken into account that the realisation of a more premixed combustion characteristic poses huge challenges especially to the boundary condition of varying fuel qualities common in emerging markets. From today's perspective, a homogenised combustion system is most likely only feasible with a cylinder pressure based engine control unit [1]. Taken into account the discussed measures, a PM emission level of approximately 12 mg/km for a SUV-type vehicle equipped with a 2.0 l engine seems feasible. This shows the enormous potential to reduce engine-out PM emissions with advanced technology. Nevertheless, that target window to rely on an open DPF for PM emission post treatment is narrowly missed, ❸.

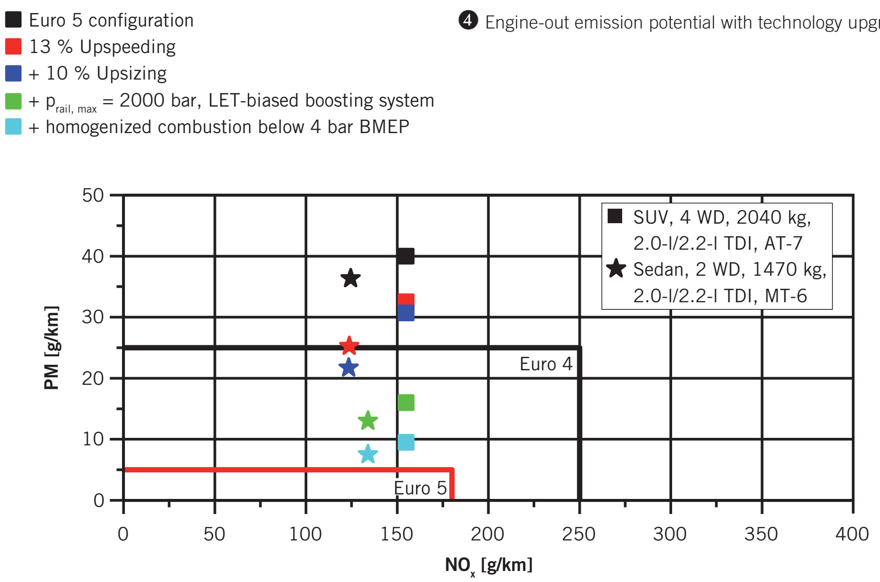
Besides the previously discussed engine internal measures, a shift of the engine operating points towards lower loads during the certification cycle can also be used to reduce PM emissions and NO_x emissions at the same time. This shifting of operating points can be accomplished by a shorter gearing (upspeeding) or by using a bigger displacement engine (upsizing). Both approaches however require additional technologies to avoid a compromised vehicle fuel consumption, which can be disadvantagous especially for the Indian market. Implementation of a start-stop system in combination with alternator management can compensate drawbacks due to shorter gearing or usage of an engine with larger displacement. Further emission potential however is not to be expected from these measures. A detailed friction optimisation of the base engine delivers similar benefits: Significant fuel economy benefits are accompanied by only small PM emission improvements.



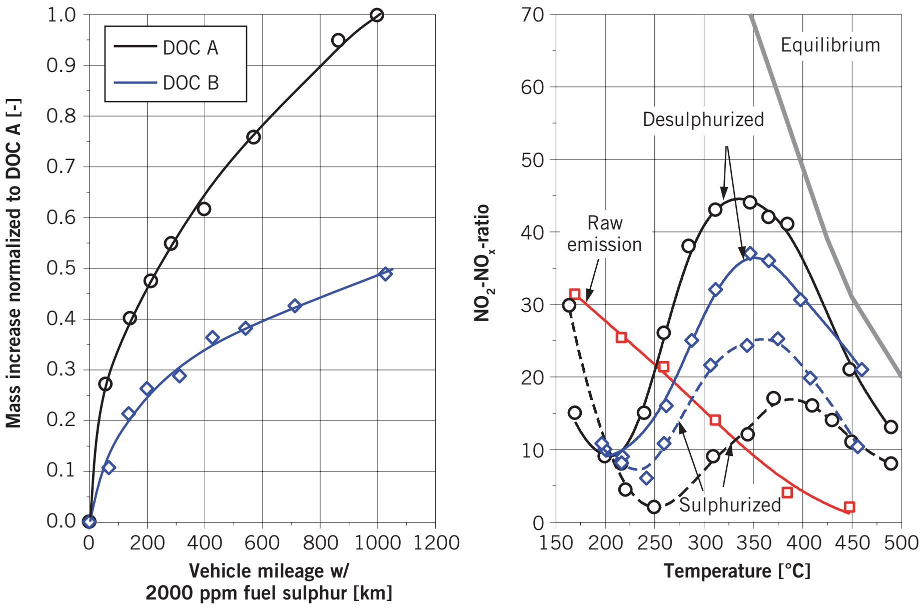
❸ Engine-out emission potential for a mid-size SUV

An evaluation of the measures upspeeding and upsizing including the previously discussed measures to reduce engine-out emissions indicate a significant PM reduction potential depicted in ❹. A middle class vehicle can achieve with this layout an engine-out PM level of 7 to 8 mg/km which allows meeting Euro 5 emission legislation with an open DPF. A heavier vehicle of the SUV-class can achieve an engine-out PM level of approximately 10 mg/km, which makes relying solely on an open DPF not directly impossible but risky.

A further emission reduction enabling the usage of an open DPF across the entire vehicle range is only possible by further technological development. The fuel injection equipment is the element with highest development towards further increased maximum fuel injection pressures [2] and increased flexibility to shape the injection rate to the needs of the combustion process [3]. Such a technology package results in high costs for the base engine, which is opposed to the requirements of diesel powertrains in the BRIC states. Additionally, the robustness of the further developed fuel injection system has to be thoroughly validated also taking into account varying fuel qualities. An Euro 5 engine concept not relying on a wall-flow DPF also for big



❹ Engine-out emission potential with technology upgrade and altered engine operating area



❺ Sulphur storage and influence on NO_2 formation

reduces the sulphur storage efficiency in the catalyst. As a consequence, the drop of NO_2 formation activity can be reduced. However, the increase of sulphur resistance typically harms the general catalyst activity so that a optimum compromise between sulphur resistance and base catalyst activity has to be found.

A reduced NO_2 concentration results in decreased passive soot conversion rates. For closed DPF application this has to be compensated by shorter regeneration intervals. In open DPF systems the balance point will move towards higher soot mass. However, this increase is not only based on a worse NO_2/soot emission ratio but also a consequence of a possible increase of filtration efficiency. The latter effect can be caused by an increased gravimetrical soot density that is typical for high fuel sulphur contents. If the balance point exceeds the soot mass limit for the specific filter an active regeneration strategy has to be used – even for the open filter.

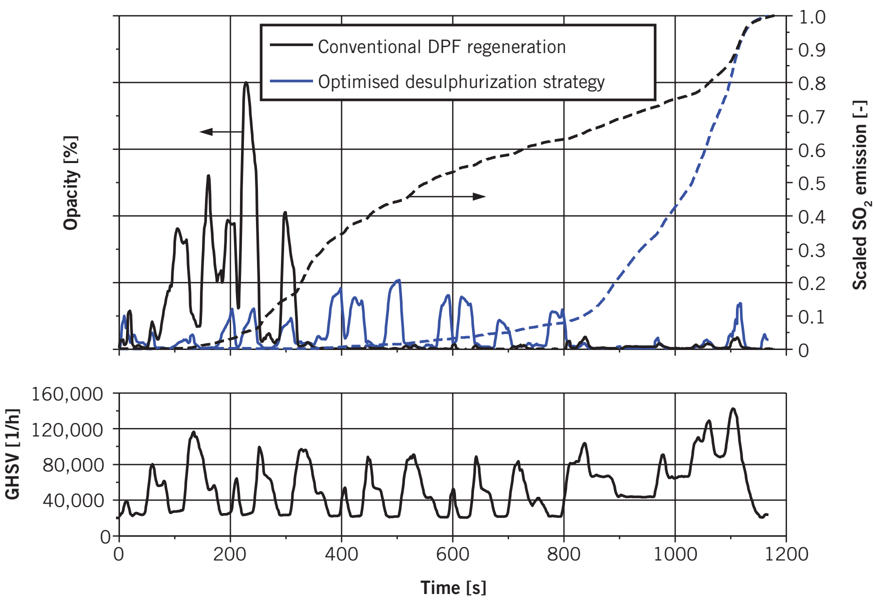
heavy vehicles will not be a mainstream solution. Therefore, the challenge to safely regenerate a wall-flow DPF also under the specific boundary conditions of passenger cars with diesel engines in BRIC states has to be taken on.

EXHAUST AFTERTREATMENT

As discussed above, exhaust aftertreatment applications for BRIC markets are challenging due to low load driving cycles as well as the risk of bad fuel quality. Especially a significant risk of high fuel sulphur contents is given which can harm the performance of the aftertreatment system. Fuel sulphur is emitted from the engine almost entirely in the form of SO_2, which can be adsorbed within the catalytic converters directly or after oxidation to SO_3. The effective adsorption efficiency is a function of temperature, space velocity and exhaust gas sulphur concentration. A part of the adsorbed sulphur is blocking active sites of the catalyst and reduces the reaction rates of HC and CO conversion as well as NO oxidation to NO_2 [4]. NO_2 formation happens mainly after completion of the HC and CO conversion, thus it is most sensitive to sulphur poisoning among these reactions.

❺ presents the NO_2 formation for two different Diesel Oxidation Catalyst (DOC) technologies as a function of temperature for sulphurised and desulphurised state. In the area of HC light-off engine out NO_2 will be reduced and a net NO_2 production only occurs at higher temperature. The sulphur originated catalyst deactivation does not only suppress the NO oxidation at elevated temperatures but also the NO_2 reduction in the low temperature window. An optimisation of the coating with respect to an increased sulphur resistance significantly

Except for the consequences of catalyst deactivation and the negative impact on passive regeneration, the DOC sulphur load itself is of interest. At high temperature levels, e. g. during an active DPF regeneration or engine operation close to full load, the stored sulphur can be released for the catalyst at very high rates. Depending on the operation conditions high amounts of SO_3 can be formed. This leads to emission



❻ White smoke emission during catalyst desulphurization of sulphates in cold parts of the exhaust system, which are strongly visible as white smoke.

❻ presents such a white smoke emission that occurs during DPF regeneration in a NEDC (black lines). The white smoke is quantified by an opacity measurement in this test. It is obvious that the highest emissions occur at small space velocities and behave reciprocal to the SO_2 release. In order to avoid such high opacity values a special desulphurisation strategy is used (blue lines). This strategy is based on an optimised temperature management for a constant sulphur release at a low rate. The desulphurisation is combined with the DPF regeneration in order to minimise the fuel penalty.

Besides the effects of elevated fuel sulphur content the emerging markets are often characterised by low speed driving cycles. In order to ensure a safe DPF operation the regeneration strategies have to be optimised for low loads and idle. Depending on the boundary conditions given by the entire propulsion system this can cause increased oil dilution. Alternative regeneration measures such as the use of fuel borne catalyst or external fuel dosing can mitigate this problem. In case of engine internal post injection the oil dilution can effectively be reduced by splitting of early and late post injection into multiple quantities each. However, this requires injection systems with improved flexibilities.

CONCLUSION

For the selection of the optimum raw emission and exhaust aftertreatment concept for emerging markets both minimum system cost and maximum robustness have to be considered. As a consequence Euro 4 applications should avoid a closed DPF. Even if further development of engine technology – especially further increased boosting concepts and injection pressures as well as further advanced flexibility of the injection shapes – offers significant potential to reduce particle emissions, the raw emission level of < 10 mg/km that would be required for the application of an open DPF will most likely be not achieved. Therefore, short term Euro 5 applications will require a closed DPF. Nevertheless, the particulate raw emissions should be limited as far as possible in order to lengthen the necessary regeneration intervals, especially under the boundary condition of low load operation and bad fuel qualities. In order to avoid the negative consequences of high fuel sulphur contents an optimisation of the catalyst coating technology towards higher sulphur resistance can be accompanied by an active desulphurisation strategy. Depending on the combination of vehicle and engine the application of alternative regeneration measures, such as post injection splitting or external fuel dosing, can be favourable to effectively reduce oil dilution.

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SIMULATION-BASED DEVELOPMENT AND OPTIMIZATION OF SCR SYSTEMS

2014 new emission regulations for commercial vehicles with diesel engines of will come into effect which will cause a further reduction of the already quite ambitious limits. Without an effective exhaust gas aftertreatment those limits cannot be achieved. Many engine manufacturers consider the Selective Catalytic Reduction as the most promising technology to meet these requirements. Multi-disciplinary simulation tools are used to utilize the entire potential of the aftertreatment system under all operating conditions. In the following, Ansys shows the opportunities of virtual development tools.

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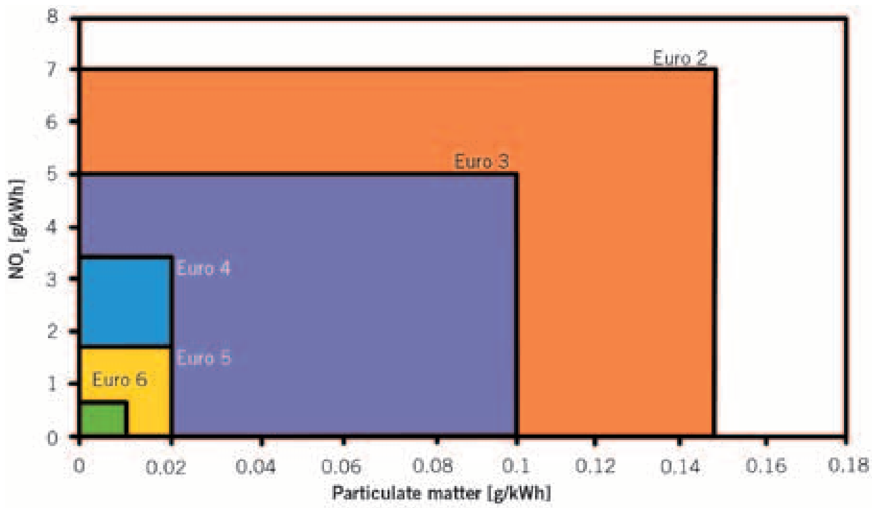
SELECTIVE CATALYTIC REDUCTION

The manufacturers of diesel engines for commercial on-highway and off-highway vehicles are confronted with increasingly more stringent emission regulations. Especially the drastic reduction of the limits for CO_2, NO_x and particle emission coming into effect in 2014 causes headache for most engine manufacturers, ❶ and ❷. Different measures were already taken in the past such as the introduction of ultra-low sulfur diesel fuel and advanced technologies for internal combustion optimization or cooled Exhaust Gas Recirculation (EGR). But these measures will not be sufficient on their own and therefore engine manufacturers have to focus on additional measures for an exhaust gas aftertreatment whereas the Selective Catalytic Reduction (SCR) represents the favored technology.

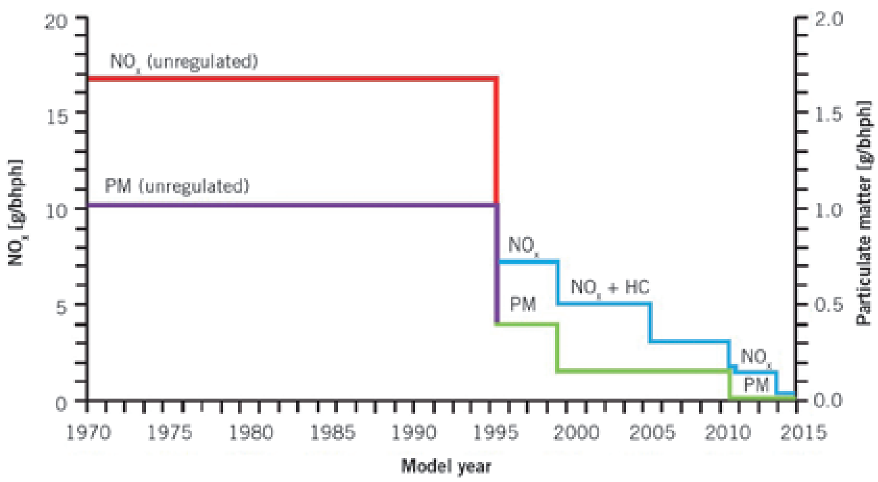
In such SCR systems aqueous urea is injected just upstream of the catalyst and decomposes into ammonia and isocynic acid. The ammonia and the isocynic acid react with NO_x over the catalyst, producing nitrogen and water, ❸. The extraordinary high conversion efficiency of 65 % (open-loop systems) respectively more than 95 % (closed-loop systems) for SCR systems can be used to tune the internal combustion process for a better fuel consumption and to achieve an up to 5 % better fuel economy.

CHALLENGES ON THE DEVELOPMENT

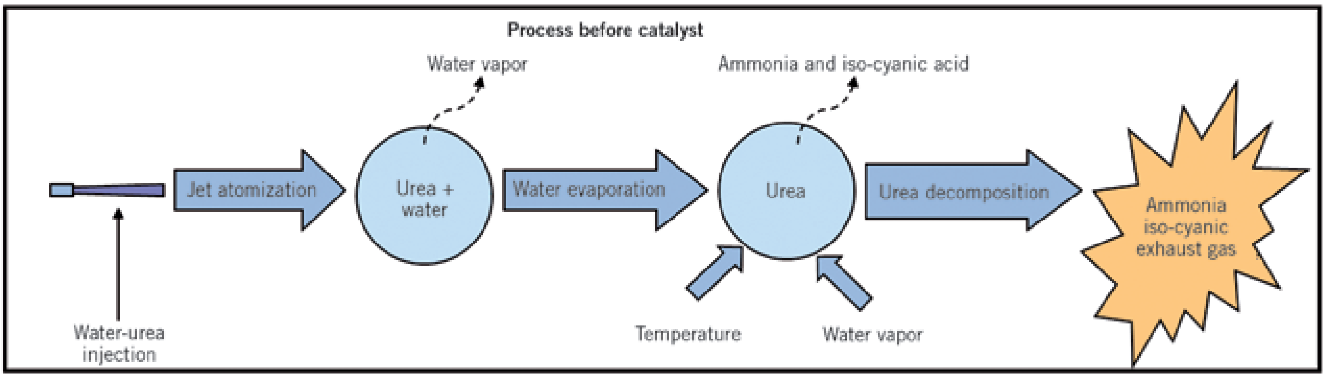
The reliable compliance with the required NO_x and particle limits over the entire operating range of the engine has to be in the focus of all development activities for

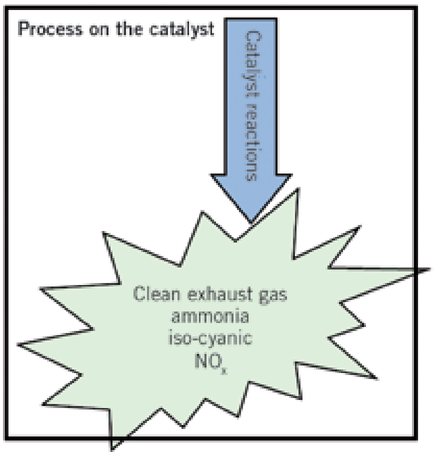


❶ European on-highway emissions standards (source: U. S. EPA and European Union emission standards)



❷ Off-highway EPA Standards (1 bhp = 745.7 W) (Source: U.S. EPA standard)





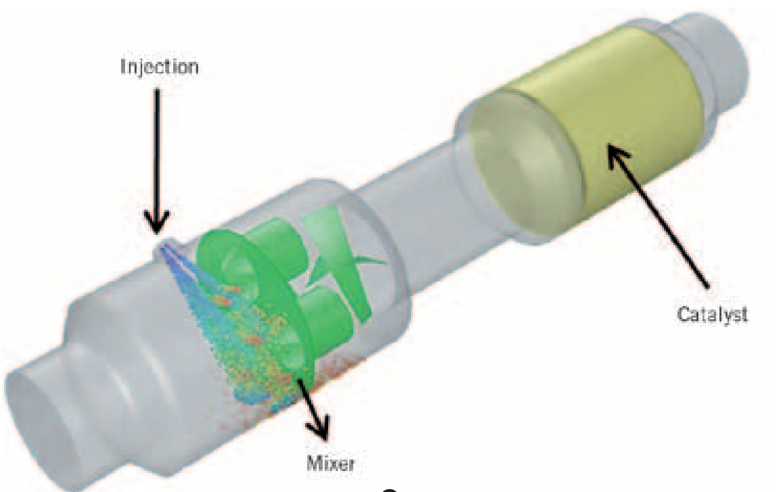
❸ SCR process

SCR systems. But the utilization of the whole potential of SCR systems may be interfered by a number of different problems. For example, a so called ammonia slip can occur when unreacted ammonia is released. Such an effect can be observed when excess ammonia is injected into the exhaust gas, when there is insufficient catalyst surface area, or an uneven exhaust-ammonia mixture occurs. Another often noticed malfunction is the crystallization of urea which occurs when excess urea injected into the process accumulates on surfaces. This phenomenon can cause a lower engine performance or an increased fuel consumption. The reason for this effect are recirculation zones upstream the injector where urea particles become entrapped und can accumulate on surfaces.

Furthermore, in particular thermal stresses can have an impact on SCR systems. Typically, catalytic conversions occur at high temperatures above 500 K. Changing engine load conditions additionally lead to fluctuating exhaust temperatures that can cause high thermal stresses and the potential for thermal fatigue in exhaust components.

SIMULATION HAS A CENTRAL IMPORTANCE

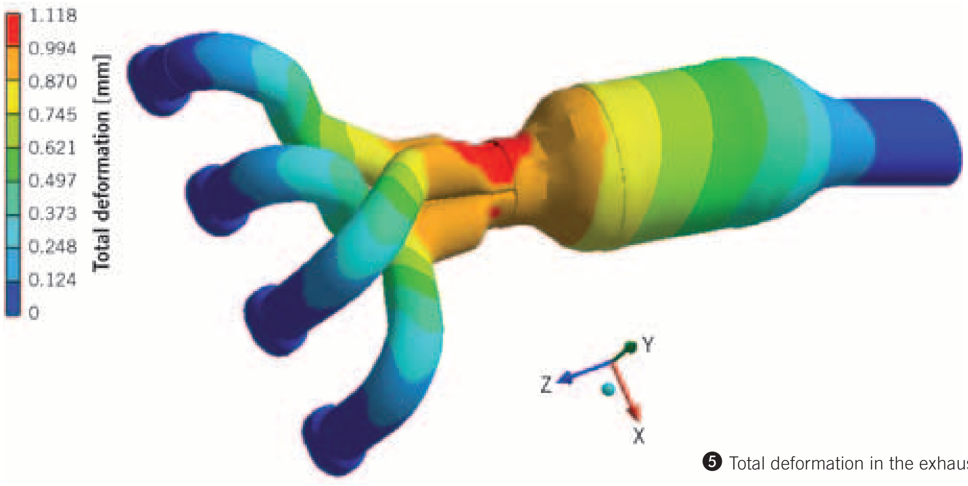
For the avoidance or minimization of these and other effects comprehensive evaluations are inevitable, whereby physical tests are still popular but having a number of significant disadvantages. To find an optimal SCR solution that covers all engine operating conditions, hundreds of tests with different design variants would be necessary. Besides a strong dependency from external factors (availability of design documentation for prototypes, engines with suitable specifications) the extremely high financial and time efforts would make such a proceeding nearly impracticable under industrial boundary conditions. Numerical simulation methods offer a number of important advantages. Simulation-based evaluations can be started already at an early process stage – closely after the definition of the engine specifications. Further on a comprehensive system optimization under consideration of the variety of dependencies from different physical disciplines can be processed with moderate effort. But one of the most convincing arguments is the fact that engineers can develop a much deeper understanding of procedures on the component as well as on the system level. Simulation results can be presented in nearly every graphical or animated form what significantly increases the information density.



❹ Urea injection upstream of mixer

REQUIREMENTS ON THE SIMULATION

For each involved discipline – chemistry, fluid flow, heat transfer, structural mechanics – exist powerful stand-alone simulation programs. But for the processing of a realistic system optimization a multi-disciplinary approach is necessary which combines all aspects in only one



❺ Total deformation in the exhaust system

simulation. Modern simulation platforms like Ansys Workbench incorporate all needed solvers/solutions and make it much easier to control the unavoidable higher effort by an easy and clear workflow. Central properties of such a solution are:

: bidirectional parametric CAD connectivity for easy data exchange, native geometry cleanup and meshing tools
: unified environment and same platform for seamless data exchange, e.g. for Fluid-Structur Interaction (FSI) and thermal simulations
: tools for design exploration, optimization, Design of Experiments (DoE).

For the optimization of SCR systems, fluid dynamics and thermal analysis (Computational Fluid Dynamics – CFD) and Finite Element Method (FEM) have a central importance. While CFD is used to understand effects like the mixing of urea with exhaust gas, its evaporation and decomposition, to ensure chemical processes and reactions and the resultant behavior of the exhaust gases and mechanical components, the FEM is necessary for the modeling and analysis of the structural behavior of the components under stress and vibrations.

TASK-SPECIFIC CUSTOMIZATION

Without doubt, already the standard environment of such multi-disciplinary simulation platforms makes it much easier to process multi-disciplinary tasks, but a task-specific customization or enhancement can cause a further increase of efficiency to find a design that meets specific objectives. In this context the parameterization of the model has a central importance. Selected design parameters can be quickly varied and different simulation models can be generated to cover the entire design space.

All input data for the process setup are gathered in an additional panel. After the import of the geometry the effort to collect all necessary data will be reduced to some few minutes. Such a process automation additionally offers the advantage that a consistent process workflow will be secured. Definable presets can be modified by the user and the consistency of the input data is checked to avoid input errors.

SIMULATION PROCESS

Based on the corresponding model geometry the flow region will be automatically modeled, beginning upstream of the urea doser and progressing to the outlet of the catalyst, ❹. The simulation itself starts with the injection of urea and water into the exhaust gases. The jet interacts with high temperature gases and starts evaporating water. At higher temperatures, urea reacts with exhaust gases and decomposes into ammonia and isocynic acid. The decomposition is modeled using a two-step reaction:

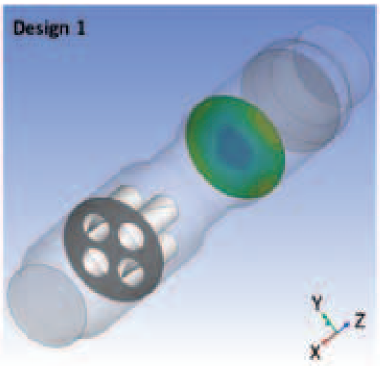
: $NH_2 - CO - NH_2 \rightarrow NH_3 + HNCO$
: $NH_2 - CO - NH_2 + H_2O \rightarrow 2NH_3 + CO_2$.

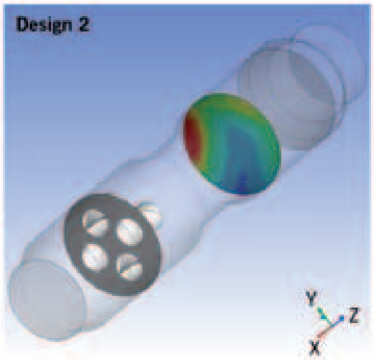
Special interest lies on an uniform exhaust-urea mixture. For the simulation model a sophisticated turbulence model is used where the urea injection is simulated as a flow of discrete particles that are tracked in the so-called Lagrangian frame. Advanced models are even used for the description of the wall-film formation, breakup and evaporation of aqueous urea droplet particles, and decomposition into ammonia and isocynic acid. Features such as In Situ Adaptive Tabulation (ISAT) algorithms and chemistry agglomeration can be used to speed up the calculations.

MODELING THERMAL STRESSES

An important technology for the prediction of thermal stresses and deformations on a SCR structure is the FSI. This technique enables a Finite Element Analysis on the basis of the wall temperatures calculated with CFD, ❺. The results are passed across a dissimilar mesh interface, allowing the fluid and structural meshes to be optimized for their specific applications. The robust convergence behavior of implicit coupling ensures accuracy and minimizes the engineering time needed to achieve valid simulation results.

For the prediction of NO_x conversion ratios detailed $DeNO_x$ reactions can be modeled. CFD simulations allow the calculation of flow velocity and direction, temperature, pressure and material concentrations. Based on these results a prediction can be made about the uniformity of the fluid flow and about the mixture of ammonia and isocynic at the catalyst entry. With animations that plot the flow of particles through the SCR system, dead zones can be identified as well as the geometrical features that are responsible for their generation. Even the concentra-

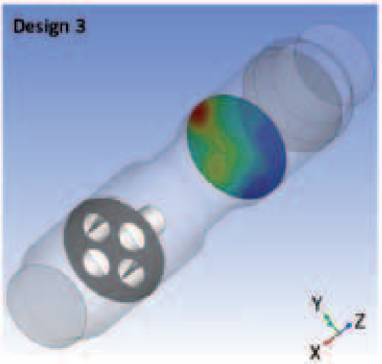


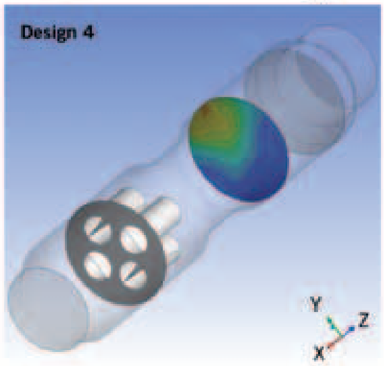


Mixer design	NH_3 uniformity [-]	Pressure drop [KPa]
1	0.84	10.7
2	0.73	10.8
3	0.69	10.8
4	0.57	10.6

$$\gamma = 1 - \frac{1}{2}\,\frac{\Sigma(|X - \bar{X}|)}{\bar{X}}$$

$$\bar{X}_{R,i} = \frac{\Sigma \rho V_n X dA}{\Sigma \rho V_n dA}$$





❻ Ammonia mass fraction at the catalyst inlet, NH_3 uniformity, and pressure drop for different mixer designs

tion of different species can be plotted on model cross sections or surfaces, such as on the catalyst itself to help understand the mixing processes.

ITERATION TO AN OPTIMIZED DESIGN

The initial simulation model of a SCR system is typically based on an existing prototype. Calculation results of this model can be validated on the base of measurement data of physical tests. After such a validation and calibration the user can evaluate a vast number of variants for a design optimization in a short time, ❻. All parameters used for the analysis are managed via a project window. In this project window design points can be built up in tabular form and executed to process a alternative design in a single operation. For example the user is able to evaluate the influence of injection parameters such as the start of injection or injection orientation.

STATISTICAL METHODS FOR DESIGN OPTIMIZATION

With a traditional optimization approach only the effects based on the variation of a single design parameter will be evaluated. Interactions between design factors and second order effects can lead to a result that only a locally optimized design will be found. Modern simulation tools offer the capability to control the optimization process via statistical methods of DoE and Response Surface Methods (RSM). With these techniques tests can be developed to investigate first-order, second-order and multiple factor effects with relatively few simulation runs. The global optimum can be found with a much higher level of certainty and in much less time than with a traditional approach.

SUMMARY

Modern simulation solutions are able to calculate with high accuracy complex multi-disciplinary correlations as they can be found at the analysis of SCR systems with high accuracy. Taking into account that with a simulation-based optimization of SCR systems much more design parameters can be considered and a much higher coverage of the design space is possible than with a conventional approach it is obvious that not only the product quality will be increased but even significant savings of time and costs can be achieved.

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THE SEAL OF APPROVAL FOR SCIENTIFIC ARTICLES IN MTZ.
MTZ
PEER REVIEW
REVIEWED BY EXPERTS FROM RESEARCH AND INDUSTRY.

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EFFICIENCY-OPTIMIZED SPARK-IGNITION ENGINE

The spark-ignition engine will still play an essential role in future mobility. However, a precondition is the further optimisation of its efficiency, which in most cases also includes downsizing. The University of Stuttgart has examined potentials simulatively under real world driving conditions.

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1 INTRODUCTION

In order to make different engine concepts comparable it is necessary to evaluate fuel consumption and emissions in a driving cycle. Appropriate test cycles have been developed for this purpose. At present in Europe evaluation is conducted almost exclusively in the certification-relevant New European Driving Cycle (NEDC). However, this driving cycle covers only a relatively small proportion of the entire engine operating map, whereby engine concepts are mostly evaluated only in a limited operating range. This paper presents an integrated simulative approach which takes into account real-world driving situations and with them the entire engine operating range in the evaluation of engine concepts.

2 SPARK-IGNITION ENGINE SIMULATION

Suitable predictive models are required in order to facilitate simulations in the entire engine operating range. These models must be able to provide a sufficiently accurate representation of both the internal-combustion engine including gas dynamics and the vehicle. The models and their parameterization are described in [9]. Homogeneous spark-ignition engine combustion is simulated in accordance with [4], stratified combustion is simulated in accordance with [7]. Both models respond to the corresponding boundary combustion conditions in the form of physical descriptions. Modeling of the knock tendency is of crucial importance when it comes to taking the full operating range of the spark-ignition engine into account. This is calculated using a new approach which was developed in the course of the FVV project and is set out in [8, 9]. Here – in addition to the conventional consideration of the preliminary-reaction states in the non-combusted zone – “hot spots“ in the combustion chamber and combustion-chamber turbulence are taken into consideration. A comprehensive verification of this model was conducted with reference to a wealth of measuring data obtained from analyses of turbocharged engines with manifold and direct injection.

3 DRIVING CYCLE ANALYSIS

It should be possible by using driving cycles to provide the vehicle user, the vehicle manufacturer and the legislators with a basis for evaluating the fuel consumption and emissions of a vehicle in traffic. Four selected driving cycles (ARTEMIS [1], FKFS [3], NEDC [5], US06 [6]) were analyzed in the course of the work undertaken. Detailed documentation is provided in [9]. It turned out that real-world driving cycles too show a marked focusing in the range of medium engine speeds and engine loads. However, in contrast to the NEDC, the main emphasis has shifted to higher engine speeds and loads. In real-world driving cycles – if only for brief periods – very high power demands are encountered which cannot be ignored in an engine-concept evaluation. In view of the marked focus of the US06 driving cycle on high power demand, this cycle – in combination with the NEDC – is used as an integrated evaluation criterion.

4 STATIONARY SIMULATION RESULTS

An identical, verified set of parameters for combustion and knock modeling was used for all engine concepts (identical combustion system) in order to establish comparability of the engine concepts. The flow models for the analyzed engine concepts were, where possible, structured to match and are based in principle on verified models. The characteristic maps used for the exhaust-gas turbochargers were scaled for comparability reasons. Engine friction was considered using a matched model in accordance with [2]. In the case of displacement variations only the friction of the piston unit was adapted and the further friction-relevant components were left identical. With this procedure it is possible to show the fuel-consumption effects obtained in an engine series where the displacement is reduced (“downsizing“).

Three different engine displacements with similar power potential were analyzed:
- four-cylinder naturally aspirated engine with 2.2 l displacement and direct injection
- four-cylinder engine with single-stage turbocharging, 1.6 l displacement and direct injection
- three-cylinder engine with two-stage turbocharging, 1.2 l displacement and direct injection.

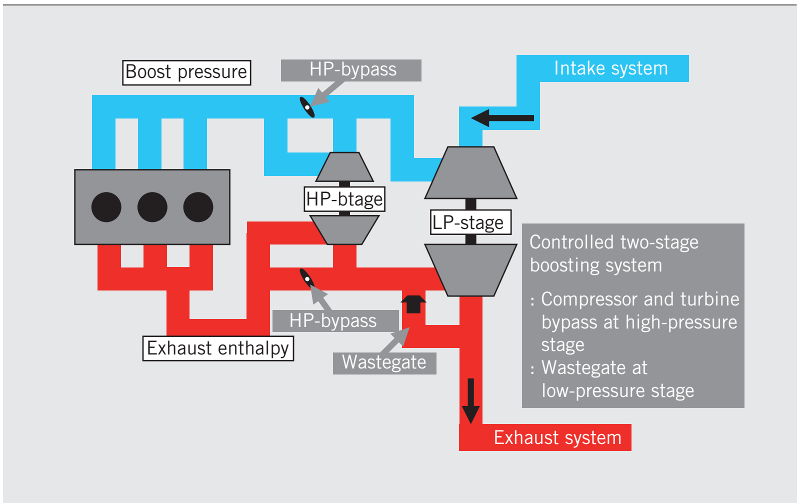
4.1 2.2-L-FOUR-CYLINDER NATURALLY ASPIRATED ENGINE WITH DIRECT INJECTION

On the naturally aspirated engine the essential boundary conditions of stroke/bore ratio and compression ratio were analyzed. Variation calculations show that long-stroke designs show greater efficiency nearly in the entire operating range [9]. A fuel-consumption benefit is obtained for long-stroke bores even when the cumulative fuel consumptions in the NEDC and in the US06 are considered. However, the overall influence of the stroke/bore ratio is very slight at approx. ± 1 % in the driving cycles in a wide range about the square-stroke bore.

In order to optimize the compression ratio simulation calculations were made taking into account the knock tendency in accordance to [8] with a compression ratio of between 10 and 16. A fuel-consumption benefit of 4.3 % in the NEDC and 3.3 % in the US06 could be demonstrated for a variable compression ratio. In view of the knock tendency there is however only an extremely limited possibility – especially when considering real-world driving cycles – of generally increasing the compression ratio (reference: 12:1) in the engine under consideration.

4.2 1.6-L-FOUR-CYLINDER DOWNSIZING CONCEPT WITH SINGLE-STAGE TURBOCHARGING – FIRST DOWNSIZING STAGE

Variation calculations were made, on the basis of a verified model, to show the fuel-consumption benefit through downsizing. At part load the engine was optimized with regard to minimum fuel consumption through adjustment of the intake and exhaust tim-



❶ Two-stage boosting system (symbolic)

ing. At full load the valve timings were optimized to demonstrate "scavenging". Scavenging is possible here at speeds of between 1500 and 2500 rpm in the configuration used with a mono-scroll turbine, the present exhaust-manifold geometry, and a rigid, given camshaft profile [10]. The valve timings at full load interact closely with the boosting system. The choice of geometry for the exhaust-gas turbine and the compressor in particular are crucial to optimization. Understoichiometric engine operation in the combustion chamber, which is caused by component protection and by scavenging, produces at full load, in combination with retarded ignition timings to avoid knocking, a significant increase in the specific fuel consumption compared with the naturally aspirated engine.

4.3 1.2-L-THREE-CYLINDER DOWNSIZING CONCEPT WITH TWO-STAGE TURBOCHARGING – SECOND DOWNSIZING STAGE

A second downsizing stage is considered in order to determine the further potential through downsizing. A 1.2-l three-cylinder engine with two-stage turbocharging was chosen for this purpose. The friction of this concept was calculated in accordance with [2] on the assumption of a constant, cumulative main-bearing width with the omission of a cylinder. In this concept in particular the function of controlling the boosting system in stationary and transient engine operation poses a huge challenge. The boosting system is shown in ❶.

The valve timings were optimized with the same methods and goals analogously to the first downsizing stage. The high efficiency over a wide range of the complex boosting system facilitates scavenging up to speeds of approximately 4000 rpm. While the knock limit is maintained, very high brake specific mean pressures can be achieved with this concept. Proportionally poor efficiency levels are encountered at full load because of the identical coherences as for single-stage turbocharging.

5 CROSS-COMPARISON OF ENGINE CONCEPTS

The essential engine parameters of the three compared engine concepts are initially set out in ❷. Increased maximum power was permitted in the turbocharged engines on account of the possibility of scaling the desired power against the boost pressure and to show the power potential. ❸ initially shows the fuel-consumption benefit delivered by the first downsizing stage in the engine operation map.

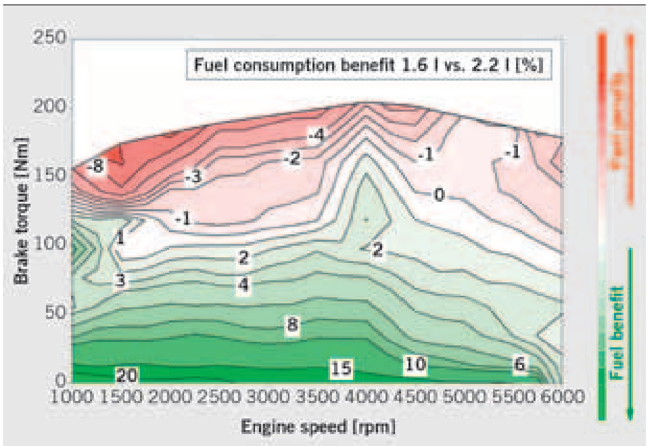
The fuel-consumption benefits in the low-load range delivered by the downsizing concept through dethrottling are clear. In the mid-load range, a balanced fuel-consumption performance is obtained with no benefits for one of the concepts. At high load clear fuel-consumption penalties are obtained with the downsizing variant – on account of enrichment and the knock tendency. Efficiency decreases even further with the downsizing variant at higher loads, which cannot however be shown in this diagram, since the naturally aspi-

DESIGNATION	–	2.2-l Reference engine	1.6-l Downsizing engine	1.2-l Downsizing engine
NUMBER OF CYLINDERS	–	4	4	3
MIXTURE PREPARATION	–	Direct injection	Direct injection	Direct injection
ENGINE DISPLACEMENT	cm^3	2198	1596	1198
STROKE/BORE RATIO	–	1.1	1.25	1.25
EXHAUST-GAS RECIRCULATION	–	External	Internal	Internal
COMPRESSION RATIO	–	12:1	10:1	9:1
MAX. TORQUE	Nm	211	305	286
MAX. POWER OUTPUT AT ENGINE SPEED	kW/rpm	112/5600	140/5000	133/5000
BOOSTING SYSTEM	–	–	One-stage turbocharging	Two-stage turbocharging

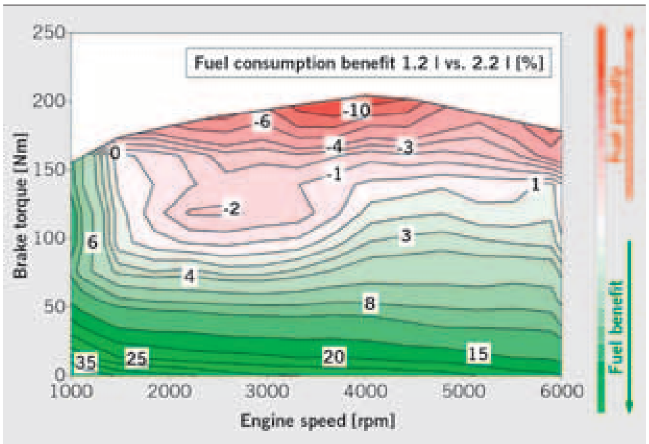
❷ Technical data of engine concepts

rated engine does not attain corresponding torque ranges (without turbocharging).

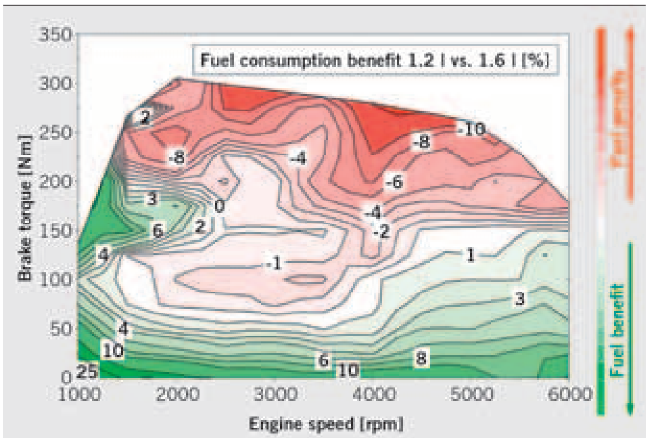
The fuel-consumption benefit delivered by the second downsizing stage compared with the naturally aspirated engine is shown in ❹. The fuel-consumption benefit at low loads is even more pronounced than with the first downsizing stage. Further fuel-consumption benefits are registered at very low engine speeds and high loads due to the lower residual gas fractions on account of the improved charge cycle of the three-cylinder engine. However, at high engine speeds and loads – despite the excellent scavenging for reducing the knock tendency – a relevant fuel-consumption penalty is registered with the second downsizing stage.



❸ Differences in fuel consumption between the naturally aspirated engine and the first downsizing stage



❹ Differences in fuel consumption between the naturally aspirated engine and the second downsizing stage

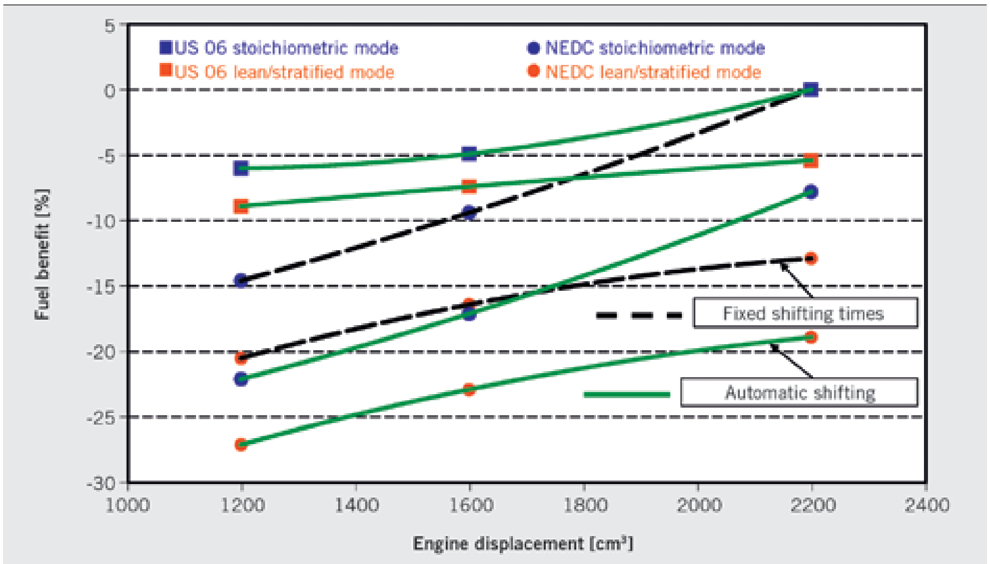


❺ Differences in fuel consumption between the first and second downsizing stages

❺ shows the differences in fuel consumption between the first and second downsizing stages. Interesting effects are discernible here. Firstly, it can be seen that at very low loads fuel-consumption benefits through dethrottling delivered by the second downsizing stage are obtained. The fuel-consumption benefits at low engine speeds and high loads of the second downsizing stage can be put down to the improved scavenging. However, because of the typical NVH (Noise, Vibration and Harshness) problems in a three-cylinder engine, this range may not be useful. In a wide mid-speed and load range no clear benefits are discernible for one of the concepts. It must be noted that this result could only be assured by the particularly good scavenging effects of the costly two-stage turbocharging system. At full load penalties are nevertheless obtained for the second downsizing stage. With a higher knock tendency due to reduced scavenging clearer penalties would be visible. If a (mild) hybridization is taken as the basis, it is questionable whether fuel-consumption benefits can be achieved by such a highly boosted concept.

Because of the low loads required in the NEDC, a really high saving potential can nevertheless by achieved by downsizing. This is shown in ❻. The vehicle used in this case was a mid-size sedan weighing 1540 kg with c_w = 0.27 and without a start/stop function. Here the NEDC was driven with a longitudinal-dynamics simulation based on characteristic maps with fixed shifting points as per the shift input in accordance with [5] and alternatively with an automatic, optimum-consumption shifting strategy [9]. The gear and final drive ratios were chosen in respect to the delayed response behavior of the boosted engines, which has a negative impact on the consumption benefits because of shorter gear ratios [9]. In addition, simulations with a combined stratified/lean-operation mode were conducted which cover the effect of quality control in the spark-ignition engine. These simulations however do not take into account the effect of denoxing, which must be considered in the evaluation of the results.

For the purpose of evaluating the fuel consumption in real-world driving cycles the benefits of the engine concepts in the US06 (simulated with an automatic shifting strategy) are additionally recorded in ⑥. The saving potentials delivered by displacement reduction are in this case much lower than in the NEDC. Particularly evident is the fact that in this cycle the concept with a 1.2 l displacement delivers hardly any saving potential compared with the concept with a 1.6 l displacement. The virtually constant saving potential by the combined stratified/lean-operation mode in the real-world driving cycle plotted against the displacement is of interest. This can be put down above all to the homogeneous lean-operation mode at higher loads (freeway speeds) and not to the stratified-operation mode. Further information on simulation of the combined lean stratified- and lean homogeneous-operation mode can be found in [9].



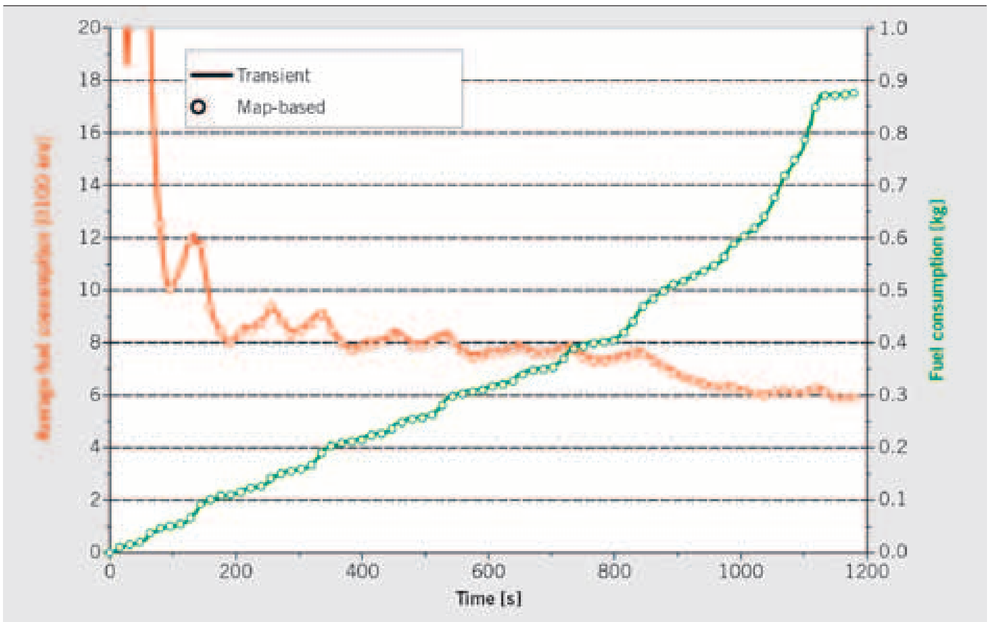
❻ Fuel-consumption benefits of downsizing and stratified/lean-operation mode in the NEDC cycle and in the US06 cycle (reference: 2.2-l naturally aspirated engine with direct injection)

6 TRANSIENT SIMULATION RESULTS

On account of the calculation times, a map-based simulation was used in Section 5 for driving cycle calculation, i. e. first simulations of stationary operating points were conducted with gas dynamics and combustion, and then fuel consumption in the driving cycle was calculated with these stationary characteristic maps. Transient simulations based on gas dynamics which do not use any stationary engine maps will now be used in this section: The internal-combustion engine is thus directly simulated in the driving cycle and torque output is regulated by adapting the air mass in the flow model in accordance with the vehicle's instantaneous demand. In this way, the full transient gas dynamics during the charge cycle are taken into account in crank-angle resolution. Combustion is a direct result of the boundary conditions (exhaust-residual content, air ratio, boost pressure, etc.) of the flow simulation. In particular, it is possible with this type of simulation to take into account the transient response of the boosting system with scavenging effects.

6.1 TRANSIENT SIMULATION OF THE NEDC AND THE US06 ON THE EXAMPLE OF THE 1.2 L DOWNSIZING CONCEPT

The results of transient simulations of the entire NEDC and US06 with full gas dynamics and knock control are presented in the following. The response of the exhaust-gas turbocharger is also depicted in the transient simulations. An automatic shifting strategy was used. The controllers used in the simulation are to the

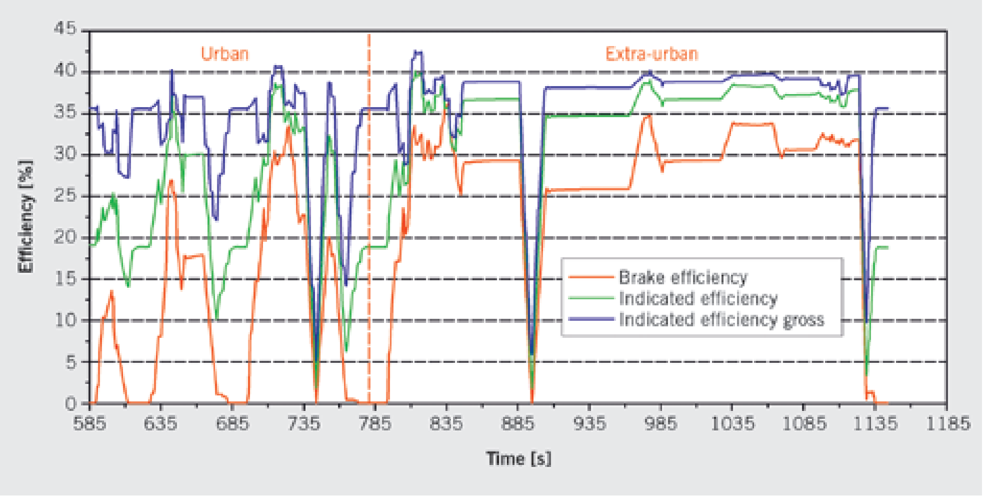


❼ Cumulative and averaged fuel consumption in the NEDC: gas-dynamics- and map-based simulation greatest possible extent identical to the map based simulation (see Section 5). However, it was necessary to develop additional transient capable idle, lambda, boost-pressure and ignition-timing controllers, as well as transient phase adjusters. In ❼ the cumulative fuel consumption in the NEDC is compared for the gas-dynamics- and the map-based simulation. It can be seen that only minimal differences in fuel consumption occur on account of the transient effects, and thus the fuel consumption is reliably depicted by map simulations, if identical engine operation modes (heat-up strategy, Ignition timings etc.) are used.

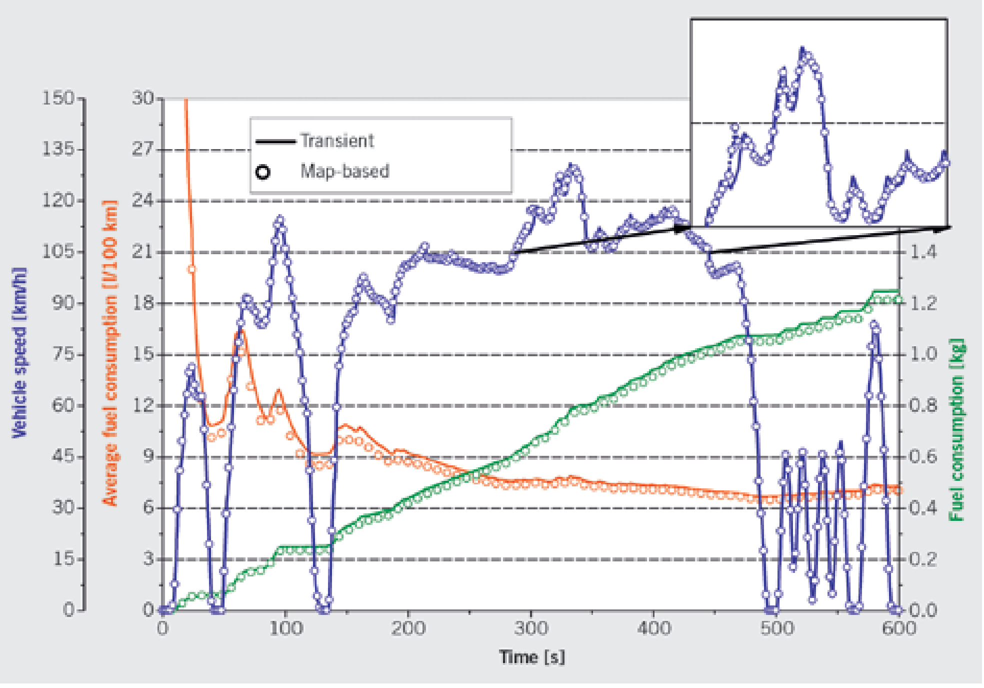
❽ shows the instantaneous effective, indicated and high-pressure efficiencies. It becomes clear that with this downsizing concept clear throttling losses are apparent only in the urban range, during the idle phases and at very low vehicle speeds, and therefore only minimal potential can be achieved by further downsizing. The throttling losses can be seen by reference to the difference between the indicated efficiency level and the gross indicated efficiency.

The instantaneous vehicle speed of the gas-dynamics- and map-based simulation in the US06 is plotted in ❾. There are with regard to the vehicle speed slight deviations between map calculation and gas-dynamics-based simulation, which can be explained by the control response of the boosting system and the scavenging used.

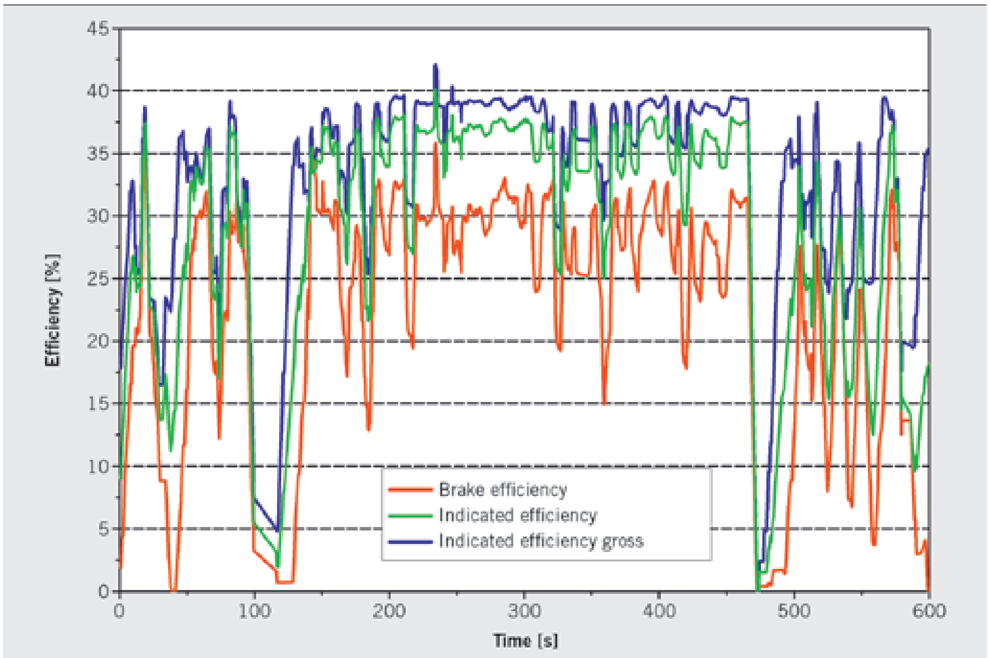
The cumulative total fuel consumption is also shown in ⑨. It clearly shows deviations which can be put down to an altered control response of the engine and the transmission which are caused by the response of the boosting system. The quantity of the deviation is dependent on the concrete engine control strategies, the engine response behavior and the transmission control, and cannot be generalized. The gas-dynamics-based simulation is how-



❽ Instantaneous effective, indicated and gross indicated efficiencies in the NEDC



❾ Vehicle speed and fuel consumption in the US06



⑩ Instantaneous effective, indicated and gross indicated efficiencies in the US06

ever able to analyze realistically the influence of different control strategies and engine applications in real-world driving cycles, making transient optimization strategies possible. The efficiency levels in the US06 are shown in ⑩. It is clear that in practical terms a reduction of the gas-exchange losses is not possible by further downsizing.

7 SUMMARY

A procedure for integrated optimization and evaluation of spark-ignition engines has been developed [9] and presented using the example of downsizing engines. Here, it was possible to quantitatively demonstrate that fuel consumption benefits by downsizing mainly at low engine loads, whereas, at high loads generally fuel penalties have to be noted. Generally in the NEDC considerable fuel consumption benefits can be achieved by means of downsizing for the homogeneous, stochiometrical mode. Further potential is accessible by an additional combined lean/stratified mode at low loads. However, if real world driving cycles like the US06 are considered, clearly less fuel saving potential is available for both modes. In addition, the limits of map based longitudinal dynamic simulations were depicted. It is shown, that for a relatively static cycle, like the NEDC, the fuel consumption can be calculated reliably by map based simulations. In contrast for the dynamic US06 cycle a dependable representation of the fuel consumption is not possible with map based simulations.

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TAILOR-MADE FUELS FROM BIOMASS FOR GASOLINE COMBUSTION SYSTEMS

Fuels made from biomass offer an enormous potential for use in vehicle applications to significantly reduce carbon dioxide emissions and increase the thermal efficiency of gasoline combustion systems. Scientists in the Cluster of Excellence "Tailor-made Fuels from Biomass" are researching production as well as the combustion of innovative bio fuels. The long-term objective is to optimize the well-to-wheel balance without competing with the food chain to the extent possible. The potential and challenges for gasoline combustion systems are identified by the research of engine operation in spark-ignited as well as controlled auto-ignited combustion mode.

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1 INTRODUCTION

One major challenge for today's society is the sustainable satisfaction of its energy demand. Today, the entire transportation sector is primarily based on fossil fuels. Despite the recent improvements of electrical vehicles, a total independence of internal combustion engines cannot be foreseen for the upcoming years. Hence mobility free of greenhouse gas emissions can only be realized by a complete substitution of conventional fuels with bio fuels.

2 BIO FUELS

Requirements for new types of fuels to be used in future diesel combustion systems have been identified with oxygen content, ignition delay, volatility and aromatic content as the major influencing factors [1-3]. With Tetrahydro-2-methylfuran, it was possible to identify a first tailor-made fuel that meets most of the defined requirements. Particle emissions can be avoided almost entirely even at highest EGR rates due to the combination of longer ignition delays and the oxygen content in the fuel molecule. However, Tetrahydro-2-methylfuran has poor ignitability at low engine loads and has disadvantages in noise, HC and CO emissions [4]. Ethanol as a bio fuel for gasoline engines has been widely studied by several researchers with strong focus on direct injection engines in the past years [5-9]. Former work of the authors has revealed the potential of increased ethanol blending rates [10-11] and the potential of combustion systems optimized for operation on pure bio fuels like for example Ethanol or Butanol [12]. Controlled auto-ignited operation in a gasoline engines allows further significant part load efficiency improvements of up to 25 % compared to stoichiometric spark-ignited operation. It results from reduced gas exchange losses, lean air/fuel-ratio and faster combustion. Auto ignition is initiated by the recirculation of high amounts of hot exhaust gas from the previous combustion cycle. Thus, the combustion kinetics and other fuel properties have a decisive influence which has not been studied for bio fuels aside from Ethanol with port-fuel injection [13]. Consequently the presented research focused on the influence of further alcohol based bio fuels in combination with direct injection. Tetrahydro-2-methylfuran whose properties allow for a usage in gasoline engines as well was also studied together with conventional fuel. ❶ summarizes the most relevant fuel properties of the fuels investigated in this study.

	RON 95	ETHANOL	1-BUTANOL	2-BUTANOL	TETRAHYDRO-2-METHYLFURAN
CARBON MASS CONTENT [%]	84.47	52.14	64.82	64.82	69.72
HYDROGEN MASS CONTENT [%]	13.27	13.13	13.60	13.60	11.70
OXYGEN MASS CONTENT [%]	2.26	34.73	21.58	21.58	18.58
DENSITY [kg/m³]	737	786	806	803	849
VISCOSITY [mPa*s]	~ 0.42	1.08	2.53	3.00	0.48
SURFACE TENSION [mN/m]	–	22.10	24.37	23.11	24.56
BOILING TEMPERATURE [°C]	41.5 – 173.5	78	118	99	79
SPECIFIC ENTHALPY OF VAPORIZATION AT $\lambda = 1$ [kJ / kg_{air}]	~ 28	101.5	61.6	60.2	31.8
STOICHIOMETRIC AIR REQUIREMENT [–]	14.14	8.98	11.16	11.16	11.21
LOWER HEATING VALUE [MJ / kg]	42.13	26.84	33.12	32.92	34.12
LOWER HEATING VALUE [MJ / l]	31.05	21.09	26.69	26.44	28.97
MIXTURE HEATING VALUE (DIRECT INJECTION, $\lambda = 1$) [kJ/m³]	3446	3538	3512	3491	3604
RESEARCH OCTANE NUMBER [–]	96.3	108.6	99.2	105	88.2
MOTOR OCTANE NUMBER [–]	85	89.7	84.3	90.9	71.2

❶ Fuel properties

	ENGINE A	ENGINE B
BORE [mm]	75	84
STROKE [mm]	82.5	90
DISPLACEMENT [cm^3]	364	499
VALVES PER CYLINDER [–]	4	4
MAX. IMEP [bar]	40	15
COMPRESSION RATIO [–]	Possible range: 7 – 13.5 Used for this study: 12	Possible range: up to 12.9 Used for this study: 12
MAX. CYLINDER PEAK PRESSURE [bar]	170	80
FUEL INJECTOR [–]	Continental Piezo	Continental Piezo
MAX. FUEL PRESSURE [bar]	200	200

❷ Technical data of research engines

3 RESEARCH ENGINE AND BOUNDARY CONDITIONS

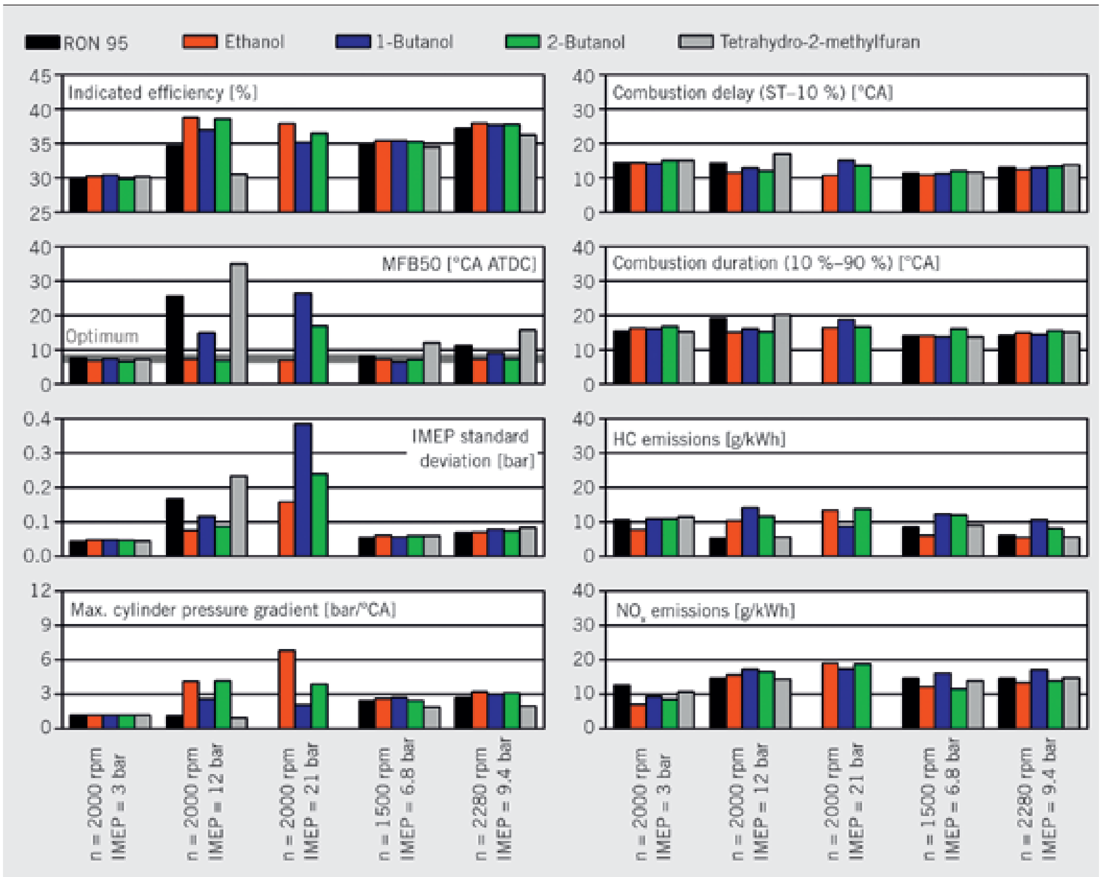
The two engines used for the investigations are direct injection single cylinder engines. Injector and spark plug are placed in central cross position in the combustion chamber roof with the spark plug installed between the exhaust valves and the injector being installed between the intake valves. Engine A has a very high charge motion and consequently a low flow coefficient of the intake port in order to enable IMEP levels of 40 bar. The cylinder peak pressure capability is adapted to the high possible power output, allowing a maximum of 170 bar. Engine A has cam phasers while engine B features a fully variable electro-mechanical valve train. Both engines use a piezo-actuated injector. The high flow rate and short minimal injection durations allow a high mass flow spread [14]. A compression ratio of 12 was chosen for the investigations on both engines as a compromise of the requirements for controlled auto-ignited operation, spark-ignited operation and the different knock resistance of the investigated fuels. ❷ gives a summary of the technical data of the test engines. All fuels were analyzed in homogeneous stoichiometric spark-ignited operation at a constant center of combustion or at the knock limit if required. Combustion chamber recirculation was chosen as strategy for controlled auto-ignited operation whereas the operation limits were defined to an IMEP standard deviation of 0.15 bar, a maximum cylinder pressure gradient of 5 bar/° CA or a stoichiometric air/fuel-ratio. The calibration parameters for each measurement are depicted in ❸.

4 RESULTS SPARK-IGNITED OPERATION

❹ depicts the results of the testing in spark ignition operation performed in [12]. Testing in the load point of IMEP=21 bar at an engine speed of n=2000 rpm was impossible with RON 95 and Tetra-

	SPARK-IGNITED			CONTROLLED AUTO-IGNITED		
ENGINE SPEED [rpm]	2000	1500	2280	1500	1500	1500
IMEP [bar]	3, 6, 12, 21	6.8	9.4	4	4	var.
MFB50 [°CA ATDC]	7.5 or at knock limit			var.	8	var.
INJECTION PRESSURE [bar]	200			100		
START OF 1^{ST} INJECTION [°CA BTDC]	300			var.		
END OF 1^{ST} INJECTION [°CA BTDC]	var.			280	410	410
END OF 2^{ND} INJECTION [°CA BTDC]	–			–	280	280
END OF 3^{RD} INJECTION [°CA BTDC]	–			–	–	40
INTAKE VALVE OPENING [°CA ATDC]	10			var.		
INTAKE VALVE CLOSING [°CA ABDC]	16			5		
EXHAUST VALVE OPENING [°CA BBDC]	16			15		
EXHAUST VALVE CLOSING [°CA BTDC]	10			var.		
RELATIVE AIR/FUEL-RATIO [–]	1			var.		
INTAKE AIR TEMPERATURE UPSTREAM THROTTLE FLAP [°C]	25			50		
EXHAUST GAS BACK PRESSURE [mbar]	Throttled operation: 1013 Charged operation: as intake manifold pressure			as intake manifold pressure		
COOLANT OUTLET & OIL TEMPERATURE [°C]	90					

❸ Engine calibration settings



❹ Results spark-ignited [12]

hydro-2-methylfuran at the chosen compression ratio. As can be seen, both fuels already suffer from knock limitation at n=2000 rpm, IMEP=12 bar and n=2280 rpm, IMEP=9.4 bar. For the first load point the center of combustion has to be retarded to 25.7° CA ATDC using RON 95 and even to 35° CA ATDC using Tetrahydro-2-methylfuran. For the last fuel, this results in an efficiency deficit of 21 % compared to Ethanol. Ethanol has the highest knock resistance allowing for an optimum combustion phasing even at IMEP=21 bar and consequently the highest indicated efficiencies of 38.8 % at n=2000 rpm, IMEP=12 bar and 37.9 % at n=2000 rpm, IMEP=21 bar. Combustion delay and combustion duration differ by only 2° CA in maximum as long as the center of combustion can be kept within the optimum range. Combustion stability, expressed in terms of the IMEP standard deviation and the maximum cylinder pressure rise are closely related to the combustion phasing and the engine load. Consequently differences in combustion stability are irrelevant among the fuels at the three operation points with the lowest load. The maximum cylinder pressure rise is more sensitive to the combustion phasing, hence Tetrahydro-2-methylfuran is generally the fuel with the lowest pressure gradients at every load point despite IMEP=3 bar at an engine speed of n=2000 rpm. Hydrocarbon emissions deliver the best indication on the quality of mixture formation. Compared to RON 95 it can be concluded that, by tendency, the usage of Ethanol improves the HC emission behavior at low engine loads while both Butanol fuels emit a higher amount of hydrocarbon emissions in every investigated load point. At higher engine loads mixture formation gets worse with all alcohol fuels resulting in an unusual HC emission increase. Tetrahydro-2-methylfuran emits similar HC emissions than RON 95 in every operation point despite the high surface tension.

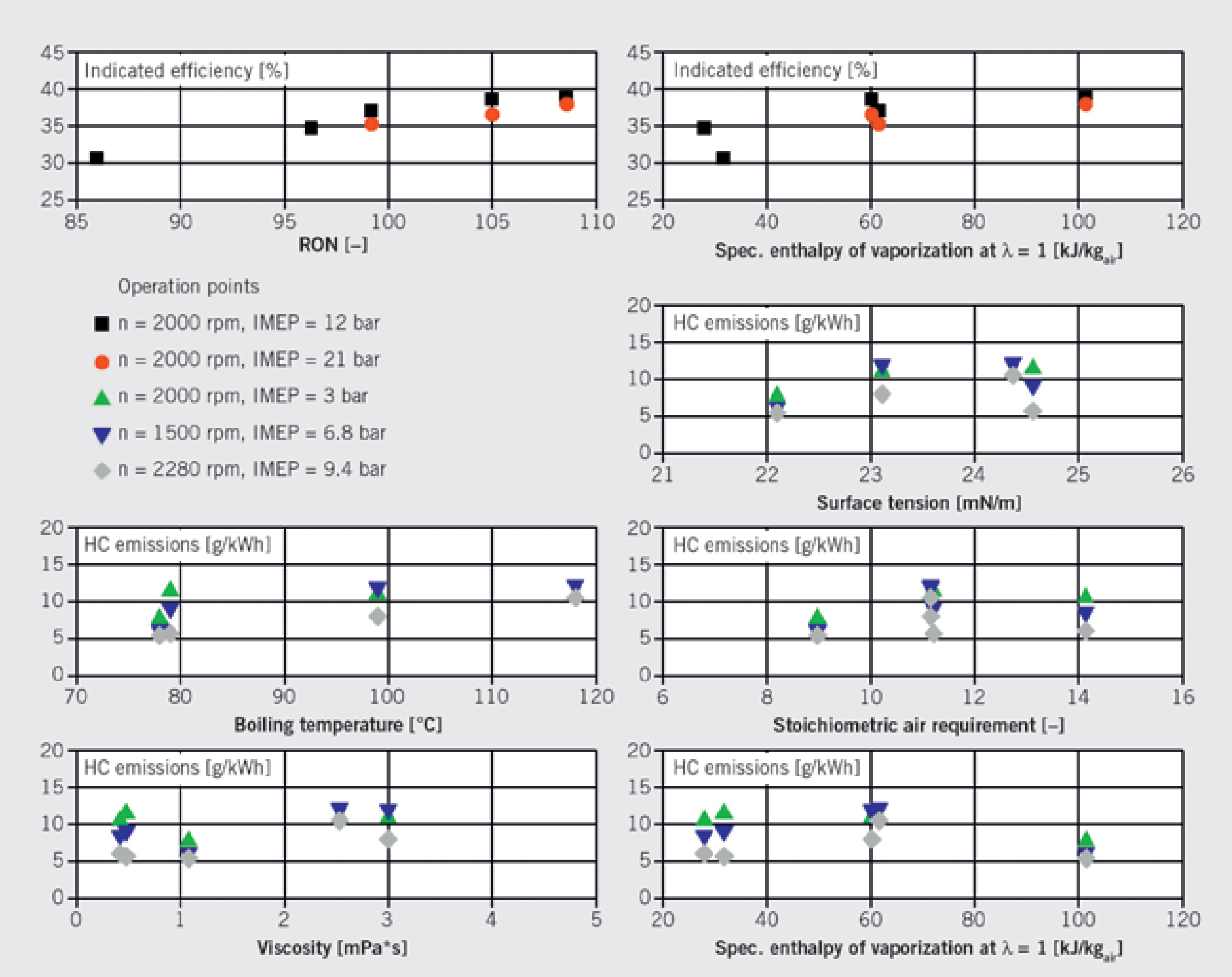
❺ illustrates the correlation between the indicated efficiency and the RON as well as the specific enthalpy of vaporization for the two operation points with the highest IMEP. Moreover HC emissions are correlated to the fuel properties (Boiling temperature, viscosity, surface tension, stoichiometric air requirement and specific enthalpy of vaporization) in the three operation points with the lowest IMEP in order to avoid influences of different knock resistance leading to better HC post oxidization because of disproportional exhaust gas temperature increase. The most distinct trend is observed regarding the boiling temperature but it can be concluded that the entity of all fuel properties together determines mixture formation and thus HC emissions.

5 RESULTS CONTROLLED AUTO-IGNITED OPERATION

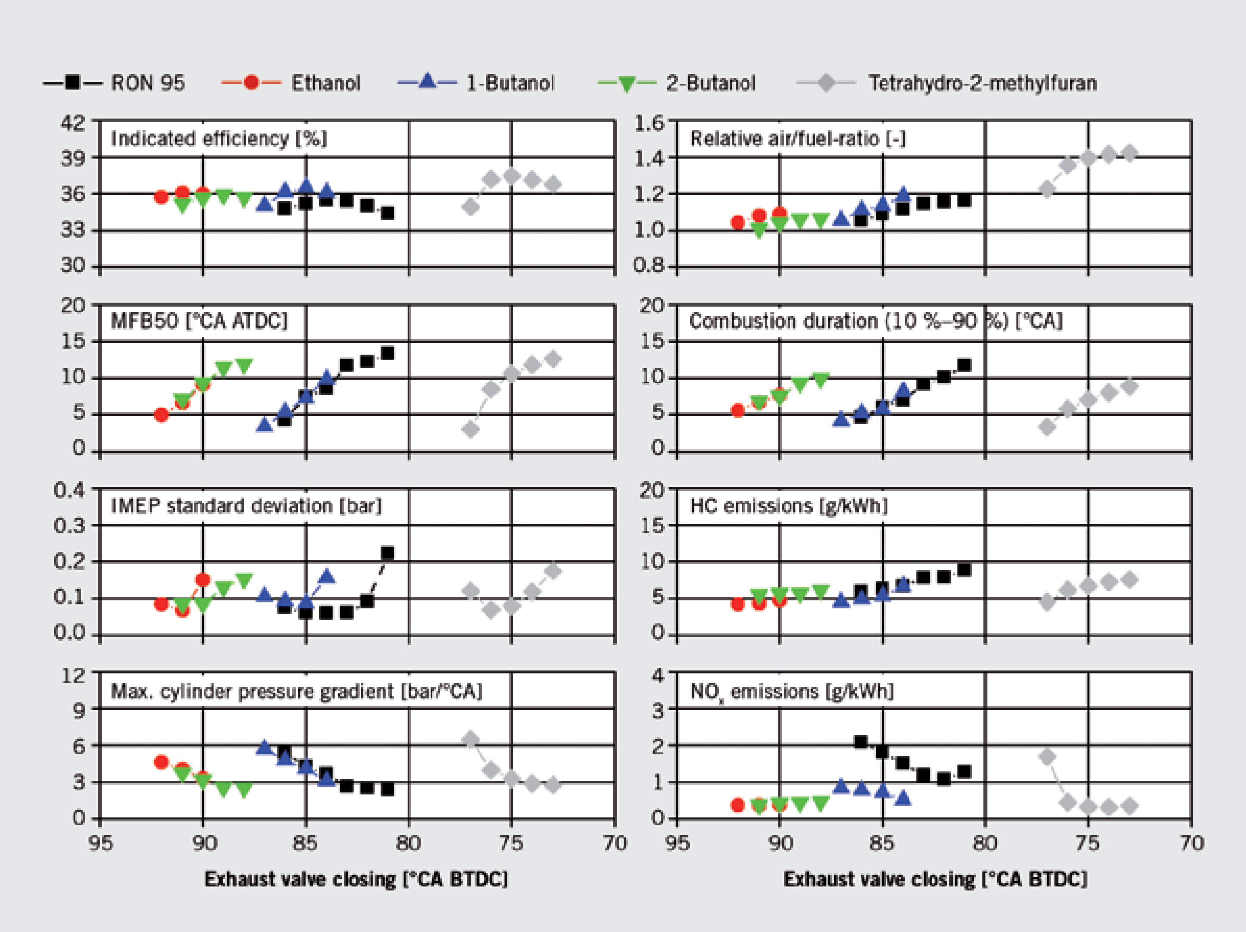
The auto ignition process of hydrocarbons is controlled by chemical reaction kinetics and is triggered by prompt decomposition of hydrogen peroxide during the compression stroke. Measures that can be taken to accelerate the accomplishment of this temperature of approximately 1000 K lead to earlier auto ignition and vice versa [15]. This combustion system can be influenced by global pressure and temperature level in the combustion chamber, local stratification of residuals, temperature and lambda [16] as well as the fuel's reaction kinetics. For most of the fuels considered in this study these reaction mechanisms are not known in the engine relevant area of low to medium temperatures and high pressures. Thus the auto ignition characteristics can only be studied experimentally.

❻ shows an internal EGR variation at an engine speed of n=1500 rpm and a load of IMEP=4 bar with single injection. This operation point enables a comparison of the fuels at a load with detectable NO_x emissions. The variation of the internal EGR amount is carried out by changing the exhaust valve closing with coincidental changing of the intake valve opening symmetrically to top death center. Starting at the maximal possible trapped internal EGR amount the negative valve overlap (NVO) is reduced. In general for a sufficiently lean air/fuel-ratio an increasing amount of residuals leads to an advanced and faster combustion with enhanced pressure gradients and higher NO_x emissions as well as a decreasing air/fuel-ratio. HC emissions rise at decreasing internal EGR amounts due to the low exhaust enthalpy. Finally standard deviation of IMEP increases till combustion failure. Significant differences between the fuels are noticeable for the necessary amounts of trapped residuals in order to auto ignite the mixture. Especially regarding the alcohol fuels the stable operation range is limited for a single injection strategy to a few ° CA. The necessary amount of residuals tends to correlate with the RON number of the fuels. Ethanol with the highest RON and the highest enthalpy of evaporation needs the major amount of trapped residuals (highest NVO) in order to reach auto ignition, which diminishes the possible relative air/fuel-ratio. 2-Butanol reveals similar behavior. 1-Butanol and conventional RON 95 which have lower research octane numbers need a reduced level of trapped residuals. Tetrahydro-2-methylfuran with the lowest RON auto ignites with the lowest amount of residuals, resulting in the highest efficiency due to lower gas exchange losses (lower recompression of residuals) and decreased wall heat losses in combination with the highest enleanment potential. All investigated oxygenated fuels reveal significant potential to cut down NO_x emissions in contrast to conventional RON 95. For the alcohol fuels this effect is caused by the high enthalpy of evaporation and lower adiabatic flame temperature resulting in reduced in-cylinder temperatures. For Tetrahydro-2-methylfuran the lowest level of NO_x emissions is provoked by higher enleanment potential. For the operating point with highest efficiency the relative air/fuel-ratio can be increased up to 1.4 with Tetrahydro-2-methylfuran leading to the lowest level of NO_x emissions of 0.33 g/kWh in case of single injection. In comparison, NO_x emissions are 1.5 g/kWh for RON 95 and 0.36 g/kWh for Ethanol.

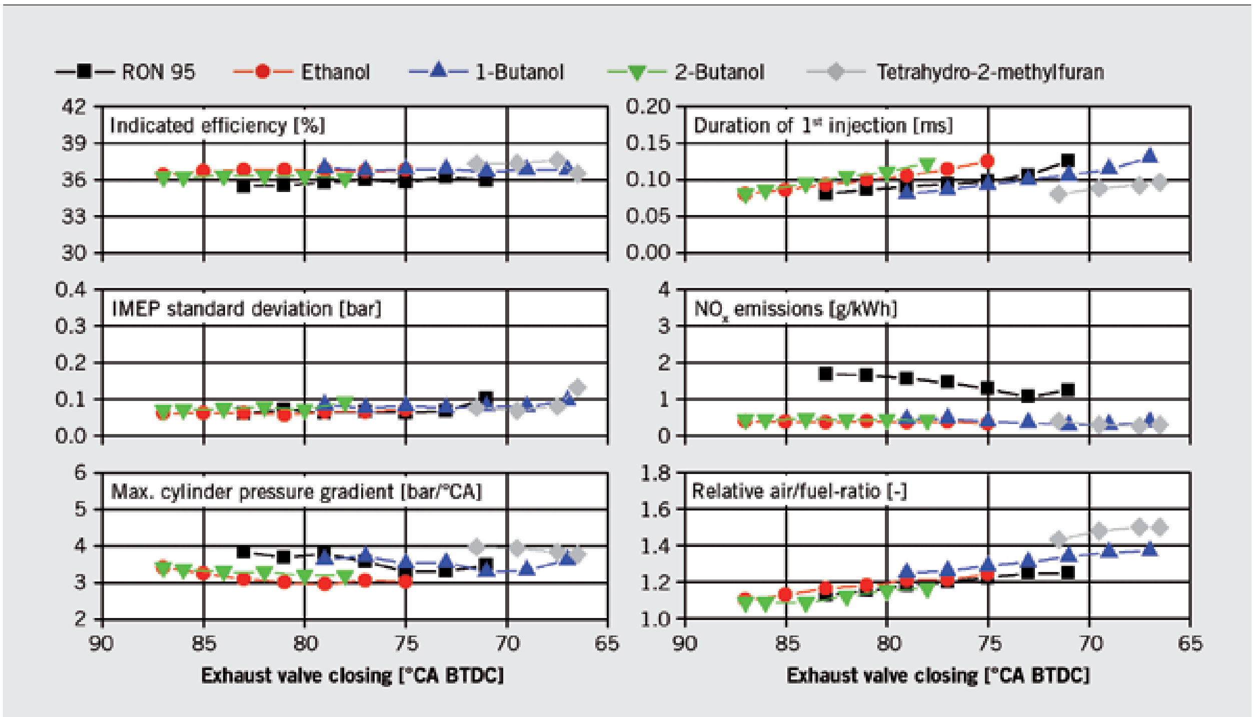
❼ shows an internal EGR variation at n=1500 rpm, IMEP=4 bar with a constant centre of combustion (50 % conversion point) at



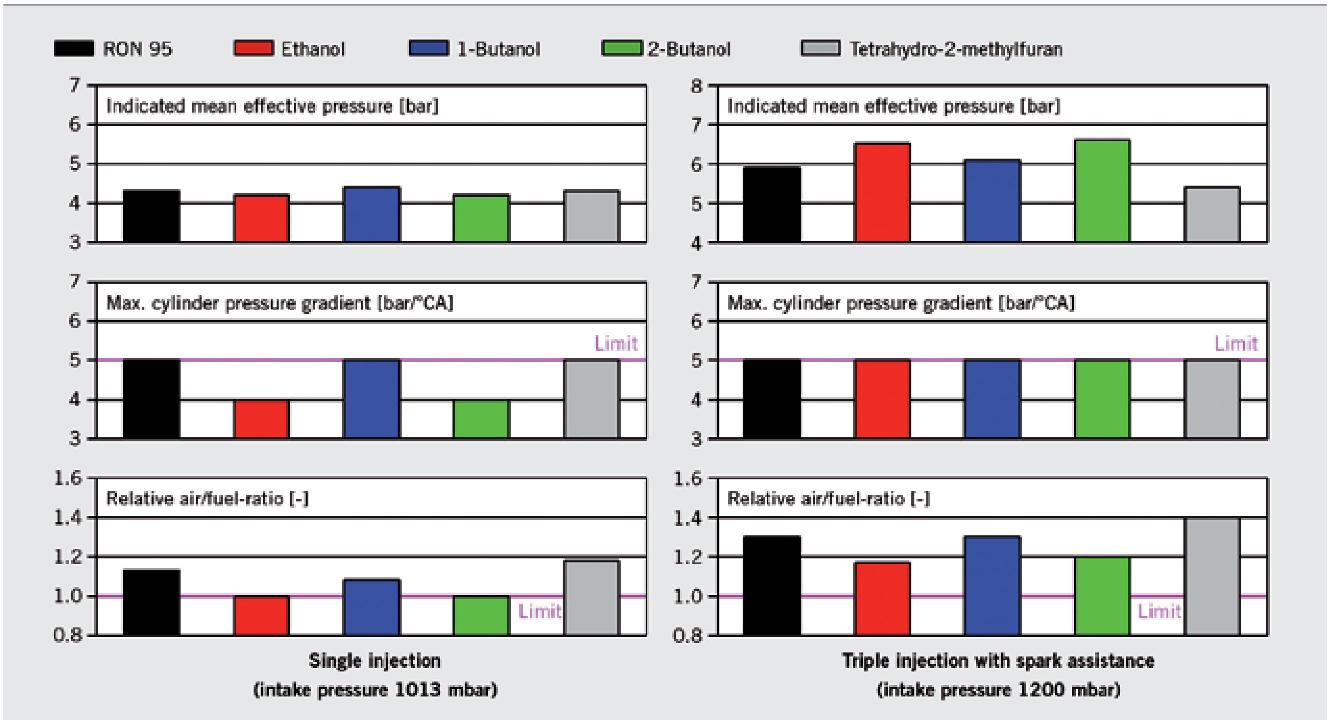
❺ Correlation between the results of spark-ignited engine operation and fuel characteristic numbers as well as fuel properties



❻ Results controlled auto-ignited – EGR variation at n = 1500 rpm, IMEP = 4 bar with single injection



❼ Results controlled auto-ignited – EGR variation at n = 1500 rpm,
IMEP = 4 bar with constant center of combustion of 8° CA ATDC and double injection



❽ Results controlled auto-ignited – Maximum possible load at n = 1500 rpm with single and triple injection

8° CA ATDC for a double injection strategy. In order to keep the conversion point constant for a reduced amount of trapped residuals an increased amount of fuel is injected 50° CA before TDC gas exchange starting with the minimal injector duration of 0.08 ms. In general an enhanced amount of fuel injected before top death centre at sufficient lean conditions and constant valve timing leads to an advanced combustion, due to the partial conversion and reformation of the fuel. The double injection enables an enlargement of the feasible operation range for all fuels at changing internal EGR amounts, which is especially pronounced for the alcohol fuels. Within the stability limits the mixture can be further enleaned by enhancing the amount of the first injection and decreasing the rate of trapped residuals, leading to slightly lower NO_x emissions. Efficiency remains almost constant as the benefits of leaner mixture is compensated by a reduction of the polytrophic exponent due to injection before gas exchange top death centre and higher wall heat losses due to the partial conversion of the injected first fuel mass.

The results of the maximum feasible loads at n = 1500 rpm are depicted in ❽. Beside the values for non charged operation with single injection, the results for charged operation with a manifold pressure of 1200 mbar, assisting spark ignition and triple injection (last injection coupled to the ignition) are presented. For non charged operation with a single injection Ethanol and 2-Butanol reach a stoichiometric air/fuel-ratio at approximately IMEP = 4 bar, as the amount of trapped residuals needed for auto ignition is relatively high. At charged operation with an optimized multiple injection strategy the maximum obtainable load can be increased for all fuels. Due to their relatively low pressure gradients Ethanol and 2-Butanol enable the highest loads of approximately IMEP = 6.5 bar corresponding to a load increase of 10 % compared to RON 95.

6 OUTLOOK

Future research will include 2-methylfuran and 2,5-dimethylfuran as further potential furan-based bio fuels which promise a higher knock resistance than Tetrahydro-2-methylfuran due to the furan ring structure with two carbon double bonds. Combustion has not yet been studied for both fuels at high engine loads in spark ignition and in controlled auto-ignited operation. Moreover, the investigations will focus on a more detailed characterization of the HC emission spectrum, especially with regard to potentially toxic hydrocarbon emissions.

7 SUMMARY

Tailor-made fuels made from biomass have the potential of significantly reducing carbon dioxide emissions due to the closed carbon circle. Moreover considerable efficiency improvements of gasoline combustion systems become possible. Compact fuel molecules with high knock resistance and high enthalpy of vaporization like for example Ethanol or 2-Butanol enable extensive efficiency benefits in spark-ignited operation at high engine loads. A significant further increase of the compression ratio is possible offering even more potential for efficiency improvements in the entire operation area. Results derived from controlled auto-ignited operation also tend to correlate with the research octane number. Fuels with higher knock resistance require increased EGR rates. Combined with higher enthalpy of vaporization and lower adiabatic flame temperature this has a positive influence on NO_x emissions. At the same time the operation range can be expanded to higher engine loads. Low-octane fuels benefit in controlled auto-ignited operation from a higher enleanment capability.

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