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FRICTION REDUCTION in Power Cylinder Systems of Gasoline Engines

WATER SEPARATION from Diesel Fuel

POTENTIALS of a Mechanical Fully Variable Valve Actuation at Boosted DI Gasoline Engines

THREE-CYLINDER ENGINES

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COVER STORY

THREE-CYLINDER ENGINES

4, 10 I The three-cylinder is increasingly finding its way into the gasoline engine range. In the small car segment it is almost established as a standard. Hyundai/Kia has developed a 1.0 l three-cylinder gasoline engine with variable valve train and intake manifold. As to the diesel engine there is restraint up to now. Compared to the gasoline engine, the effect on fuel saving as against the costs constitutes a more unfavorable ratio. That the three-cylinder engine still has potential shows an investigation by Bosch.

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COVER FIGURE © Volkswagen

SYMBIOTIC

Dear Reader,

Professor Hans-Peter Lenz chose a term from the field of biology when describing the electrification of the powertrain at the Vienna Symposium. The interaction of two different species to achieve a common objective. Actually, it is remarkable that no contradictory murmur passed through the room when Lenz talked about symbiosis in the powertrain. After all, the symposium is traditionally the terrain of combustion engine developers.

However, there was no murmur. Quite the opposite. The fact that the electric motor is able to sensibly support the reciprocating piston engine now appears to be a matter of consensus. The industry is now approaching the layout of future powertrains in a satisfyingly rational manner, without any hype-like ventures. Perhaps also because the rational route is still unable to bypass the combustion engine.

This is not only due to its extensive potential for increasing efficiency. It is only its advantages over and above a purely electric drive – particularly in terms of range – which will open up the doors to wider use in automobiles for the electric motor. Professor Christian Beidl of Darmstadt Technical University actually goes one step further: the combustion engine is leading to an increase in the number of kilometers driven purely electrically. This sounds paradoxical but is plausible when we bear in mind the greater flexibility offered by hybrid or range extender vehicles. In turn, it means that powertrains powered by electric batteries will initially remain a niche application, primarily for compact vehicles in urban settings.

Which brings us back to the necessary symbiosis. And the recognition that the change taking place within the powertrain field is more evolutionary than revolutionary. This does not make this development any less exciting, as the diversity of the vehicles' facets increases along with their efficiency. GM's Tom Stephens formulated it a little more enthusiastically: this is the best time in history to be working in our business.

Yours

R. Danisch

RUBEN DANISCH, Vice Editor-in-Chief
Wiesbaden, 3 June 2011

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POTENTIALS AND LIMITS OF DOWNSIZING A DIESEL ENGINE

In order to reach lowest CO_2 emissions, Bosch modified a three-cylinder diesel engine regarding its turbo-charging and fuel injection system and integrated it into a mid-size luxury car. So it is possible to determine both, the potentials concerning fuel consumption and emissions of this downsizing approach as well as its limits.

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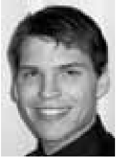

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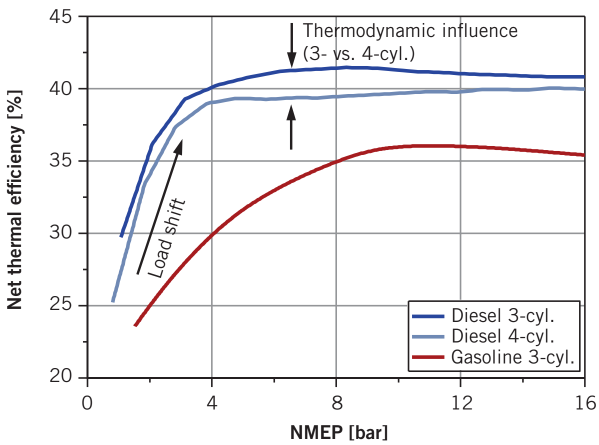
POTENTIALS OF COMBUSTION ENGINES

Caused by the CO_2 discussion and the shortage of resources a lot of new concepts for electrification of the powertrain are presented currently. This gives the impression that the internal combustion engine only will play an underpart in the future. Actually the combustion engine remains the most important component in the drivetrain in the foreseeable future and shows enormous potential by the usage of innovative developments. This is basically necessary since reaching the CO_2 fleet targets, in spite of the success of alternative concepts, is only possible, if strong efforts are also taken for internal combustion engines as the dominant drive source. An effective method therefore is to reduce the swept volume of the engine, which leads to a reduction of friction power. This goes along with a load shift in the engine map towards operating points with higher efficiency.

The effects of a load shift by downsizing on the net thermal efficiency are shown in ❶. Compared to a gasoline engine, which reaches its maximum efficiency very slowly due to throttling losses at part load, the diesel engine already maintains a constant high efficiency level from an NMEP of 4 bar. Furthermore it is obvious that the reduction of swept volume accompanies a reduction of the amount of cylinders for a consequent downsizing approach at a diesel engine. By this means more advantages arise from keeping the ideal single cylinder volume of approximately 0.5 l, which offers a higher net thermal efficiency due to lower heat losses compared to a four-cylinder engine with the same swept volume. Positive aspects on the gas exchange are additionally given by the ignition interval of 240° CA. The occurring free mass moments of 2nd order have negative effects on the oscillating behavior, as they cannot be diminished by a balance shaft. To figure out all the potentials and the limits of such a downsizing approach a demonstrator vehicle based on a midsize luxury car with the inertia mass of 1700 kg was built up with a three-cylinder diesel engine.

ENGINE CONCEPT

A 1.5 l three-cylinder engine with common rail injection is used in the presented downsizing approach. Compared to the four-cylinder engine, it equates to a downsizing level of approximately 30 % [1]. For the engine with the smaller swept volume a new compact serial two-stage turbocharger group was designed and assembled to realize the same driving performance. The charging system is one key component in this system and fulfils several tasks at the same time. On the one hand it allows to reach the targets in terms of specific power and specific torque, which is necessary to compensate the lower swept volume. On the other hand, the small high pressure turbine improves the dynamic behavior of the vehicle by a spontaneous acceleration and allows high EGR rates up to higher part load area caused by the back pressure. This is necessary to reach lowest possible engine-out NO_x emissions. As a matter of principle the turbocharger group achieves high efficiency in the whole engine map, which ensures low fuel consumption. In



❶ Comparison of the net thermal efficiency between a three- and four-cylinder diesel engine and a three-cylinder gasoline engine with the same displacement (V_{disp} = const. = 1.5 l)

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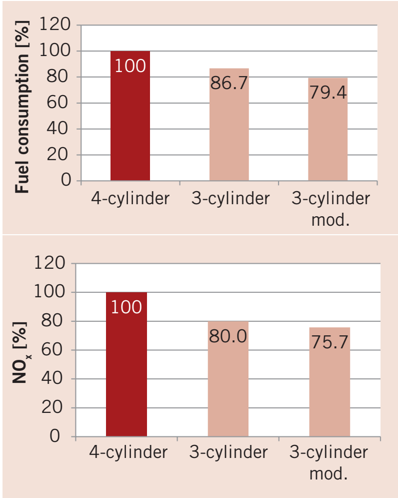
the exhaust system a PTC (Pre-Turbine Catalyst) is integrated, which enables a considerable faster conversion of HC/CO emissions, without any drawback on the dynamic behavior. The compression rate was also reduced to lower both the peak cylinder pressure at full load and the NO_x emissions at part load. The EGR cooler was built up with a separate cooling circuit. Thereby the cooling efficiency is heightened and the cooling can be used as needed.

The applied injection system uses Piezo valves and enables a maximum injection pressure of 2400 bar. This high pressure level permits a short injection timing at full load and allows a very good soot/NO_x trade-off at part load area [2].

	3-cylinder
Displacement	1.5 l
Compression ratio	16:1
Pressure charging	Serial 2-stage
Power output	108 kW
Peak cyl. pressure	180 bar
Torque	300 Nm
Injectors	CRI3 (2400 bar) Midi, 8-hole
High-pressure pump	CP4
Software	EDC17 with MBC/MCC
Charge air cooler	Water-cooled
EGR	Optimized EGR section with separate cooling circuit

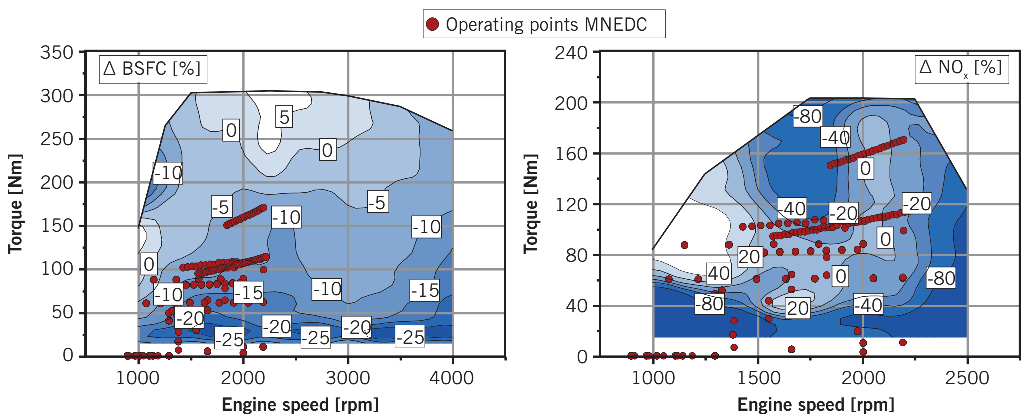


❷ Specifications of the modified downsizing engine; reduction of fuel consumption and nitrogen oxides in the MNEDC, measured on the chassis dynamometer

FUEL CONSUMPTION AND EMISSIONS

Measurements on the chassis dynamometer show a fuel consumption advantage of 13 % for the three-cylinder engine, ❷. Due to an optimized injection pattern and higher boost pressure in the part load area a reduction of 20 % NO_x emissions is feasible, while concurrently the desired limit of 25 mg/km for the particulate emissions before DPF can be undercut. Because of the raise of specific load the downsized diesel engine shows lower HC/CO emissions at equal torque. Additionally the exhaust temperature increases, whereby the temperature loss at the second turbine is compensated. By the usage of a PTC a pre conversion of HC/CO emissions takes place, so that the compliance of emission legislation limits is ensured without further heat-up strategy. It is possible to reach EURO 5 legislation values with the described measures. If a $DeNO_x$ system with an assumed conversion rate of 50 % will be used, the upcoming EURO 6 limits are also reachable. Adapting the rolling resistance parameters on the chassis dynamometer can simulate vehicle-side fuel economy improving measure. ② shows the influence of an improved aerodynamic drag and a lower rolling resistance.

For a detailed analysis of the advantages of the engine with the smaller swept volume in terms of fuel economy and emissions the deviation maps between conventional and downsized engine are shown in ❸. Especially at low torque, a difference up to 30 % is achieved. Therefore the steep gradient of the effective efficiency at low part load is responsible. Furthermore the three-cylinder engine has also advantages in terms of fuel economy at higher load and speed. This is caused by both an improved gas exchange due to higher turbocharger efficiency and the lower friction of the three-cylinder engine. Also the usage of a high pressure fuel pump with increased efficiency is an important factor. Merely in the overlap area between both turbochargers – approximately at an engine speed of 2400 rpm – the fuel consumption benefit is not so distinctive; respectively it shows a little disadvantage. Here the high pressure compressor works on its choke line. At the same time the exhaust gas flow is too little for gaining the complete boost pressure by the low pressure charger. Nevertheless a significant fuel consumption advantage is also maintained in the customer's relevant operation range. Comparing the NO_x emission maps, the two-stage turbocharged three-cylinder engine shows high reduction potential particularly at low torque. Due to the higher specific



❸ Deviation (BSFC and NO_x emissions) between three- and four-cylinder engine

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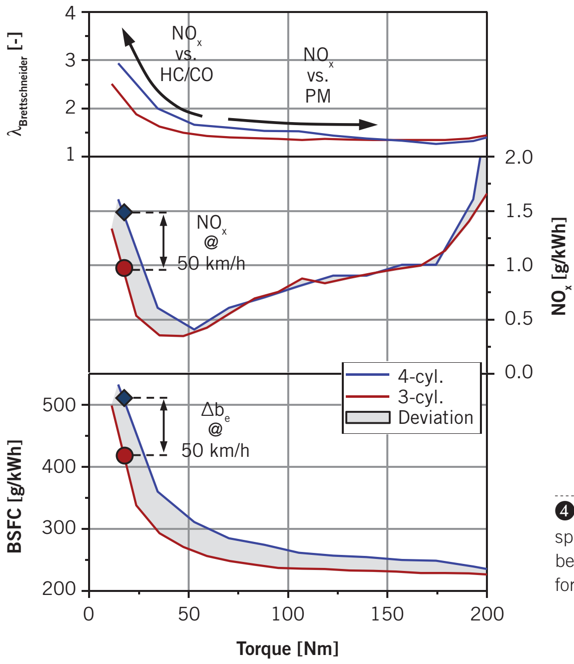
loads at this area, lower air masses can be calibrated without raising HC/CO emissions compared to four-cylinder engines with a bigger swept volume. For limiting the noise emission of the downsized engine, the injection pattern was adapted to the changed requirements. Therefore the enabler is a small hydraulic dwell time between the injections. At low speed and high load NO_x emissions are higher than the emissions of the four-cylinder engine. In terms of drivability it is necessary to reduce the EGR rate in this area to ensure an acceptable dynamic driveaway behavior.

❹ shows exemplarily a load sweep for both engines at 2000 rpm. the line of air-to-fuel ratio is presented in the upper graph. At low part load the limitation of HC/CO emissions requires a demand of sufficient air. At higher part load a lower air-to-fuel ratio leads to low NO_x emissions with respect to the particulate mass. As the graph in the middle shows, the three-cylinder and four-cylinder engines have almost the same NO_x emissions from 50 Nm on. The higher specific load even gives an advantage regarding HC/CO emissions below 50 Nm. Particularly at steady state points in the MNEDC (here 50 km/h) a significant NO_x reduction by downsizing is possible. The reduced friction that comes along with a smaller swept volume decreases fuel consumption through the entire load sweep. Additionally the higher specific load leads to a faster increase of the mechanic efficiency at equal torque. Therefore the greatest benefit in fuel consumption appears at low part load (steady state in MNEDC).

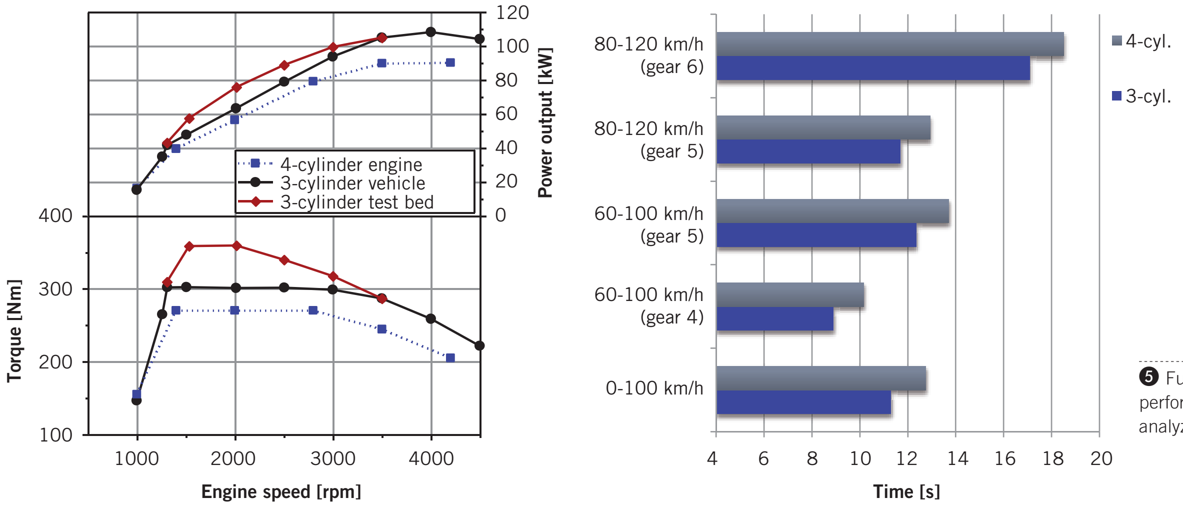


❹ Comparison of air-to-fuel ratio, specific NO_x emissions and BSFC between three- and four-cylinder engine for a load sweep at 2000 rpm

FULL LOAD AND DRIVEABILITY

The driveability demand for the demonstrator vehicle with the two-stage turbocharged downsizing engine is to accelerate from 0 to 100 km/h in less than twelve seconds. In addition a top speed of at least 200 km/h should be feasible. This corresponds to a rated power of about 90 kW. As shown in ❺, the maximum achievable torque is almost 100 Nm higher than the maximum torque of the four-cylinder engine. The rated torque of the vehicle is limited to 300 Nm, in order to avoid an overstressing of the clutch and transmission. An acceleration from 0 to 100 km/h in about eleven seconds can be reached with these specifications. This is an improvement of nearly two seconds compared to the four-cylinder engine. Simultaneously the vehicle with the downsized engine provides much better driving characteristics. The increased rated power up to 108 kW corresponds to a top speed of 215 km/h.

The key technology for an enhanced rated power is besides the two-stage turbo charging system a modified FIE with an increasedrail pressure. For this purpose Piezo injectors (CRI3) without leakage are used in the downsizing engine, which



❺ Full load and driving performance of the analyzed engines

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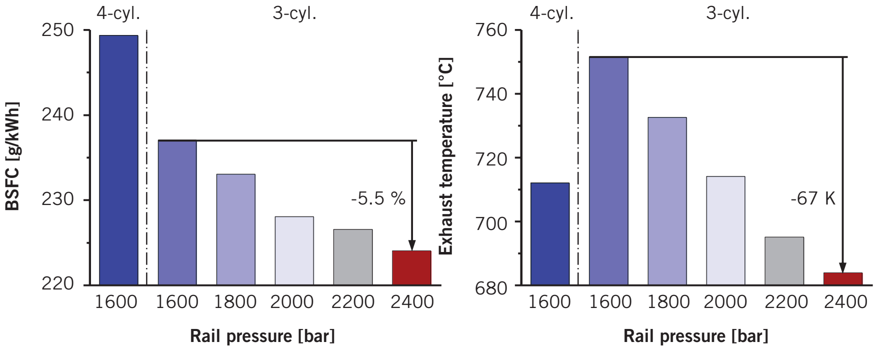
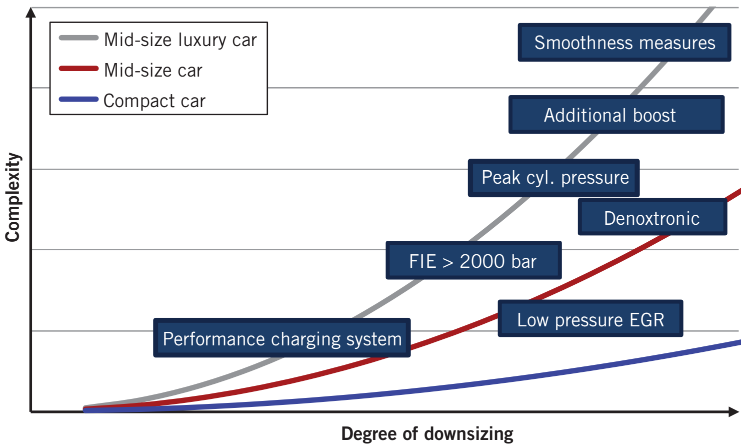
enable a rail pressure up to 2400 bar. Furthermore the high pressure pump CP4.1 is applied, to compress the fuel with a good hydraulic efficiency even at low volumetric flow rates [3].

In order to analyze the effects of the measures taken, a rail pressure variation at rated power of the four-cylinder engine was carried out and the results were compared to the specifications of the engine with the higher displacement. The downsizing engine features a lower specific fuel consumption at equal rail pressure, ❻, which is on the one hand a result of a better gas exchange efficiency for the two-stage turbocharged engine. On the other hand the 50 % mass fraction burnt point can be positioned near to its optimum due to the higher peak cylinder pressure allowed. The exhaust temperature for the engine with the smaller displacement increases, because of the higher brake mean effective pressure and as a result of the higher injected fuel mass,. Thereby an enhanced rail pressure is an effective countermeasure. This accelerates and improves the mixture formation, so that the exhaust temperature decreases below the four-cylinder level and a further reduction of the fuel consumption by nearly 6 % can be observed. Compared to the four-cylinder engine an improvement in fuel consumption up to 10 % can be registered.



❻ Rail pressure variation at rated power of the four-cylinder engine



❼ Components for the realisation of increasing downsizing demand

LIMITATIONS

Besides technical aspects there are also economic limitations for downsizing. The different measures are of more or less consequence depending on the vehicle category and the associated customer acceptance. ❼ shows exemplary several technologies, which serves as enabler for higher degrees of downsizing.

- performance charging system: increased boost pressure leads to enhanced dynamic behaviour and provides high EGR rates
- FIE > 2000 bar: raise of specific power output and limitation of pollutant emissions
- peak cylinder pressure: ideal 50 % mass fraction burnt point near rated torque for lower exhaust temperature and fuel consumption
- low pressure EGR: less NO_x emissions, better dynamic and lower fuel consumption
- denoxtronic: exhaust gas aftertreatment system for reduced NO_x emissions
- additional boost: overcoming of the critical area at low engine speed while start-up processes, for example by using a mechanical or electrical compressor, mild hybridisation, etc.
- smoothness measures: improvement of the vibration characteristics, for example by using a dual mass flywheel with an integrated centrifugal pendulum.

The choice of three instead of four cylinders in the 1.5 l engine displacement class has thermodynamic advantages but disadvantages regarding comfort. The number of cylinders determines the frequency of the engine order and the disturbances on the vehicle. ❽ shows the frequency bands of the main order for the whole engine speed spectrum for both engines. The three-cylinder induces lower frequencies than the four-cylinder. The left ordinate shows the weighting of the inconvenience level for these disturbances [4]. The operating point at an engine speed of 900 rpm and a torque of 25 Nm shows the consequences of downsizing with cylinder reduction. The smaller number of cylinders leads to both a lower frequency and a higher alternating engine torque. Hence the increasing engine speed amplitude (measured at the primary side of the two-mass flywheel) appears in combination with a higher weighting of the inconvenience level. Depending on the comfort requirements of the particular vehicle class additional measures for compensation are necessary.

OUTLOOK AND FURTHER MEASURES

Future legislative limits for carbon dioxide and pollutant emissions require further measures beyond downsizing. Possible ways are improvements in existing components, its integration in the complete system and the application of innovative drive concepts (e.g. mild-hybrid). The achievable potential can be estimated by simulation. The impacts of the described measures on the CO_2 and NO_x emissions are presented in ❾.

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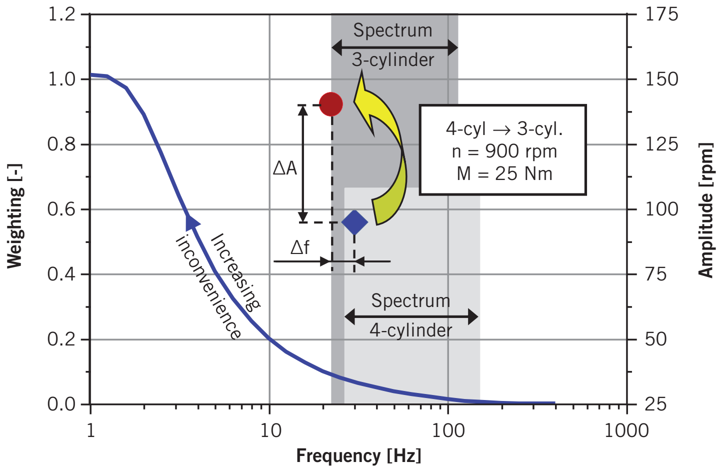
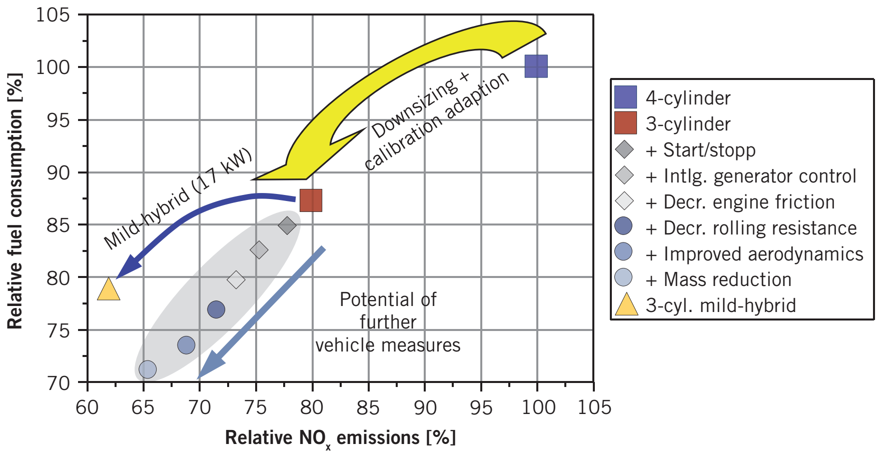


❽ Engine vibrations of three- and four-cylinder engines and their inconvenience level



❾ Potentials of vehicle measures and mild-hybrid for further decreasing of fuel consumption and nitrogen oxide emissions

Technologies like automatic engine start/stop and intelligent generator control are already feasible with a small effort [5]. Unnecessary engine idle operation is avoided at this and the usage of a part of the kinetic energy for supplying the vehicle electric system is enabled. Both the further development in materials and design and the application of thermal management grant a decrease of the appearing engine's friction by 15 %. These measures achieve an effective utilization of the internal combustion engine but it is also important to reduce the vehicle's driving resistances. The simulation shows the improvement for currently realisable values in this vehicle class (aerodynamic drag index 0.6 m²; rolling resistance coefficient 0.009) and a mass reduction of 100 kg. The little savings of every single measure summarize to a significant overall reduction of fuel consumption and pollutant emissions, ⑨.

Another possible approach is the extension to a mild-hybrid with 17 kW electric power output [6]. This form of hybridisation is especially suitable for downsized diesel engines. Without pure electrical driving there is no need to generally raise the engine loads further. High engine load demands can be decreased by an assisting torque of the electric motor. An increase of the engine torque at low load demands leads to a sustainment of the energy balance and an operation with improved combustion efficiency, ①, without disadvantages in nitrogen oxide emissions. The limited fuel saving potential of approximately 10 % requires a cost-effective system. However the additional features like transient phlegmatization or low-end torque boosting are more valuable for stronger degrees of downsizing.

CONCLUSIONS

By the displayed analysis is shown, that it is possible to achieve a considerable reduction of fuel consumption (over 13 %) by consequent downsizing of a diesel engine. Simultaneously future emission limitations (EURO VI) can be kept. Beside a performance charging system the key component for downsizing is an injection system, which provides a rail pressure of more than 2000 bar and which features a good hydraulic efficiency. Higher degrees of downsizing in combination with cylinder reduction are limited by the NVH behaviour, which is hard to quantify. The increased starting weakness can be compensated by combining the downsized engine with a mild-hybrid. In addition further benefits in fuel consumption and emissions can be achieved with this measure.

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THE NEW HYUNDAI-KIA 1.0 L THREE-CYLINDER GASOLINE ENGINE

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With the new 1.0 l Kappa (κ) gasoline engine, which can be found in the face-lifted Hyundai i10 and the new Kia Picanto, Hyundai-Kia combines a three-cylinder concept with modern technologies such as the initial application of variable valve timing within this displacement segment. The derivate with switchable intake manifold delivers 60 kW/82 PS and achieves a maximum torque of 94 Nm. Next to the gasoline engine a bivalent version for the use of gasoline and liquefied petroleum gas (LPG) is already available; an ethanol engine is under development.

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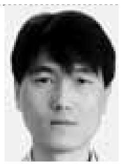

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CENTRAL ROLE IN THE COMPANY'S FUTURE SMALL CARS

Climate change and the influence of humans on it are the subjects of world-wide discussion. The transportation sector, including cars, trains, aircraft and ships, is responsible for more than one-fifth of all global CO_2 emissions. Hyundai-Kia is meeting the challenge to lower emissions as required by legislation in all markets. Such regulations, as well as rising energy prices, lead to an increased interest in compact vehicles. The new 1.0 l three-cylinder engine out of the Kappa family is part of the Hyundai-Kia sustainable product strategy and combines high achievement with fuel efficiency. The selected three-cylinder concept fulfils high acoustic requirements due to intensive detail optimization. From the beginning of development, the engine was designed with country-specific requirements in mind. The economic and eco-friendly Kappa engine family will – when flanked by additional derivates – take over a central role in the company's future small cars.

CONCEPT DECISION: NUMBER OF CYLINDERS

In the early phase of development, the four-cylinder engine was typical in the 1.0 l class. However, owing to increasing demands for improved fuel economy, three-cylinder engines were launched; the development of a two-cylinder engine was even reported. Friction loss and thermal efficiency are improved by reducing the number of cylinders. Therefore, determining the number of cylinders was crucial for development of the Kappa 1.0 l engine. Compared to a four-cylinder engine, a three-cylinder engine has better performance and fuel economy, ❶. Nevertheless, a NVH (noise vibration harshness) problem caused by increased unbalanced forces remained the weakness of the three-cylinder. However, through optimal design of the cranktrain, reinforcement of the engine structure and optimizing the match with a vehicle, NVH can be improved. Contrasting with a three-cylinder engine, a two-cylinder engine generally fitted to a motorcycle cannot meet NVH quality without a balance shaft because of excessive unbalanced forces caused by reciprocating mass. If the balance shaft is applied, the vibration of the C1 component will be decreased. However, fuel economy will become worse due to power loss. Also, applying a balance shaft increases cost and weight. Moreover, larger and longer intake and exhaust systems are necessary to reduce low-frequency combustion noise, the unique noise of a motorcycle. Even with the application of these technologies, there are limits to combustion noise reduction in a two-cylinder engine. Adding an alternator, air conditioner compressor and a starter motor on a two-cylinder engine body restricts design freedom. In conclusion, considering NVH quality, cost and other factors, the three-cylinder configuration was determined to best for the Kappa 1.0 l engine.

	4-CYLINDER	3-CYLINDER	2-CYLINDER
FUEL CONSUMPTION	Reference	+ (~ 3 % … 5 %)	+
PERFORMANCE		+	+
WEIGHT		+ (~ -10 %)	+
NVH		-	--
COSTS		+ (~ -9 %)	+
EFFECT OF SYNERGY*		+	o

*with κ 1.2 l four-cylinder

❶ Decision matrix: number of cylinders

VERSION	51 kW / 69 PS	60 kW / 82 PS	BIVALENT
NUMBER OF CYLINDERS [-]	3		
ARRRANGEMENT [-]	Inline		
DISPLACEMENT [cm³]	998		
BORE X STROKE [mm x mm]	71 x 84		
COMPRESSION RATIO	10.5		
DISTANCE OF CYLINDERS [mm]	78.5		
VALVE ARRANGEMENT [-]	4V – DOHC, Dual CVVT		
VALVE ACTUATION [-]	Tappet with mechanical lash		
TIMING DRIVE [-]	Roller chain		
INTAKE SYSTEM [-]	Fixed lenght	Variable length	
RATED POWER [kW]	50.7	60.3	
MAX. TORQUE [Nm]	95		
FUEL [-]	Gasoline		LPG / gasoline

❷ Engine specifications

SPECIFICATIONS

The new Kappa 1.0 l engine – DOHC with four valves per cylinder – achieves highest values for power, fuel efficiency and acoustics. The basic data of the aggregate are summarized in ❷. The engine design will be explained in the following paragraphs.

CRANKCASE

The aluminium alloy cylinder block is applied to reduce engine weight by 12 kg. Also, the cylinder block is designed as an open-deck type of high-pressure die-casting process. Meanwhile, to reduce the length and weight of the Kappa engine, the bore gap is designed to be 7.5 mm with a Siamese type. The cast-iron liner is applied to the cylinder bore to enhance the abrasion durability. With added 0.7 mm spine on the outer surface of the liner, adhesion between aluminium and the cast-iron liner is improved. Therefore, the deformation of the cylinder bore is reduced. Consequently, oil consumption and the amount of blow-by gas are decreased. The shape of the skirt is designed as a corrugated type to enhance stiffness. Also, for minimizing weight and improving NVH performance, the ribs and shape are optimized by FEM (Finite Element Method) analysis.

CRANKSHAFT AND PISTON GROUP

For reducing weight and manufacturing cost, the crankshaft is made of cast iron, FCD700C, and the shape of balance-weight is optimized by the cranktrain behaviour analysis to minimize three-cylinder engine vibration. In the case of the three-cylinder engine, the major design focus of the crankshaft is minimizing both the vertical pitching and longitudinal yawing. Both vibrations mainly depend upon the balance-weight and one is inversely proportionate to the other. Therefore, it is crucial to minimize the pitching and the yawing. By analyzing the crankshaft through dynamic simulations in the form of assembling pistons and connecting rods, the Kappa is designed to the optimal shape of its crankshaft balance-weight. The endurance of the crankshaft was ensured by computer-aided strength analysis and evaluating the physical part. To improve fuel efficiency, the offset crankshaft mechanism, ❸, is applied. The offset crankshaft mechanism is the fuel economy technology used to reduce the friction force between the piston thrust face and the cylinder bore inner face on the explosion stroke by optimizing the eccentricity e. But the contact force on a piston anti-thrust side becomes greater while a piston moves up. As a result of Computer-Aided Engineering (CAE) analysis, the eccentricity e is optimized at 11 mm, giving the Kappa improved fuel economy of 1 % at low engine speed. By using Design for Six-Sigma (DFSS) and FEM, the connecting rod is designed to be the lightest one in its capacity class, ❹, while improving fuel efficiency and ensuring endurance. In order to decrease the inertial force, the piston is optimized by minimizing the piston compression height (24.7 mm), pin-bosses distance and skirt length. As a result, piston weight is 161 g. Decreased weight of the piston and connecting rod enables the Kappa to improve fuel efficiency by about 0.5 %. Because the piston ring is coated with Physical Vapour Deposition (PVD), the tension of the piston oil ring is reduced by 33 %. MoS2-coated piston skirt and reduced piston ring tension provide 0.6 % better fuel efficiency to the Kappa. Two major technologies are applied on the bearings to improve fuel efficiency. First, the multi-boring bearing technology reduces oil leakage by eliminating the crush relief and optimizing the gap between crankshaft journals and bearings. Therefore, the optimized inner profile of the bearing decreases the amount of consumed oil. Second, the partially grooved bearing technology also reduces oil leakage by

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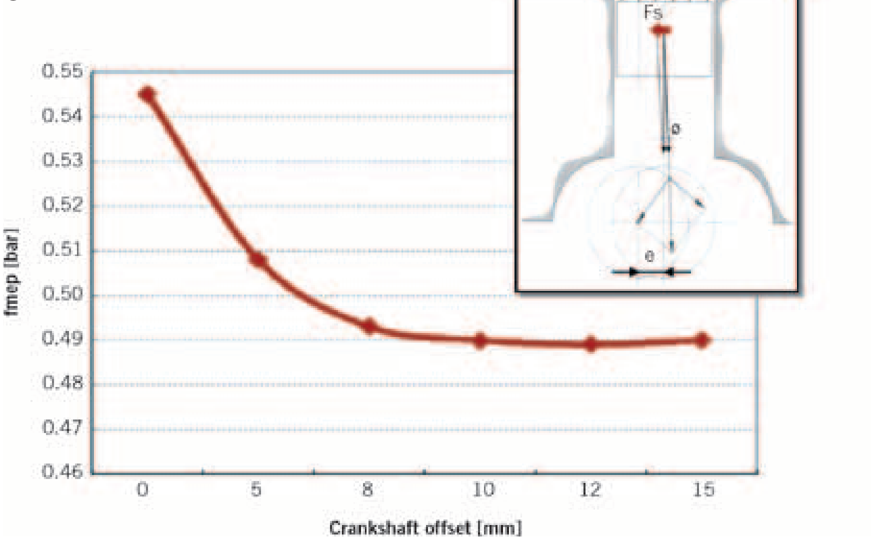
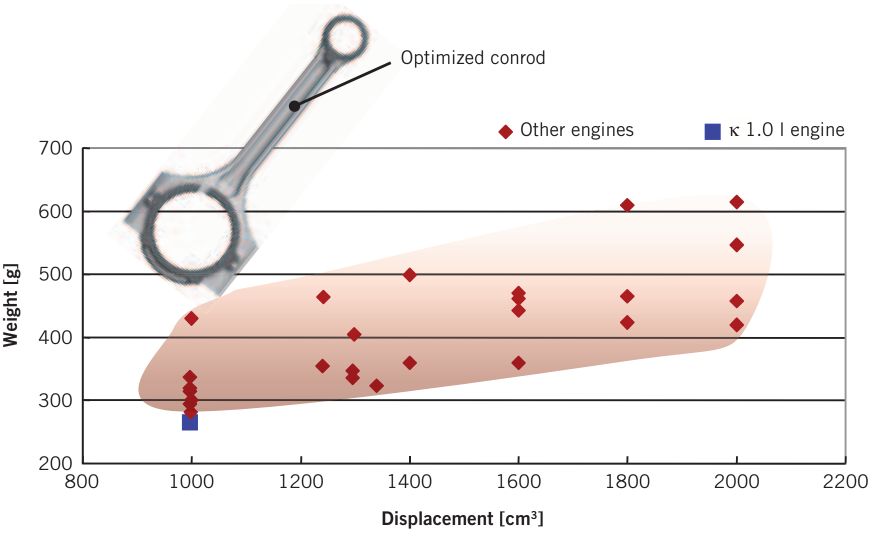


❸ Influence of crankshaft offset on friction



❹ Optimized κ 1.0 l conrod in the field of competition

decreasing the grooved area of both ends. With these two technologies the oil pump capacity is decreased by 13 %, increasing fuel economy by about 0.4 %.

CYLINDER HEAD

A pent-roof combustion chamber and a tumble inlet port, ❺, are applied to the cylinder head to reduce HC emission while improving the characteristics of combustion. Also, tumble flow, which was reinforced by 15.8 % than the initial design, was applied to improve combustion efficiency, therefore torque at low and middle speed (1500 to 3000 rpm) is improved by 1 %. For converging air-fuel mixture at the spark plug, the squish area takes 10 % of the cylinder bore area. The spark plug is placed in the centre to shorten flame paths thereby giving good combustion and reducing raw emission. The scissors angle of the valve was developed at 33.2 ° to minimize the surface of the combustion chamber, thereby improving combustion efficiency and minimizing the size of the cylinder head.

VALVETRAIN AND TIMING DRIVE

The Kappa adopts the world's first Dual Continuously Variable Valve Timing (Dual-CVVT) technology in its capacity class. Dual-CVVT technology maximizes fuel efficiency and performance by optimizing valve timing. It continuously alters inlet/outlet valve timing depending on driving conditions to reduce pumping loss and increase volumetric efficiency. With Dual-CVVT technology the Kappa improves fuel economy by up to 3 % and performance significantly compared to a non-CVVT engine. Also, it decreases emission gases such as NO_x and HC by the effect of the internal Exhaust Gas Recirculation (EGR). Moreover, the internal EGR helps to achieve cost reduction, because catalyst jewelry weight is reduced. The Kappa is developed with Mechanical Lash Adjuster (MLA) tappet of the direct acting type, ❻, for reducing inertial mass of the valve system and saving costs. The MLA tappet is coated with Diamond Like Carbon (DLC) to improve fuel efficiency. DLC coating overcomes the disadvantage of increasing friction due to sliding contact between camshaft and tappet. In comparison with nitrification coating, the DLC coating gets 0.3 % better fuel economy by reducing valvetrain friction. The friction of the DLC-coated tappet is improved relatively better at low-engine speed than at high-engine speed. For improving fuel economy by reducing valvetrain inertial mass, Kappa uses a beehive valve spring. Similar to the shape of a beehive, the top diameter of the beehive valve spring is designed to be smaller than the bottom diameter. This lowers the weight of the retainer and valve spring and reduces the inertial mass of the valvetrain. Valvetrain friction is lowered by 10 % at whole engine speeds compared to a conventional valve spring. To reduce inertial mass, the MLA tappet minimizes wall thickness. It is 20 % lighter than other replacements, creating the smallest valve spring load and reducing friction.

INTAKE AND EXHAUST MANIFOLD

The three-cylinder engine is alternatively equipped with an intake manifold of constant length and a variable counterpart in order to achieve two power variants. In both cases the plenum is made of plastic to reduce weight and costs. To ensure high engine torque at middle speeds, the static tube corresponds to the long position of the switchable runner. The manifold length was verified by simulation and experiment and specified to 451 mm. The shape of the surge tank is changed to a

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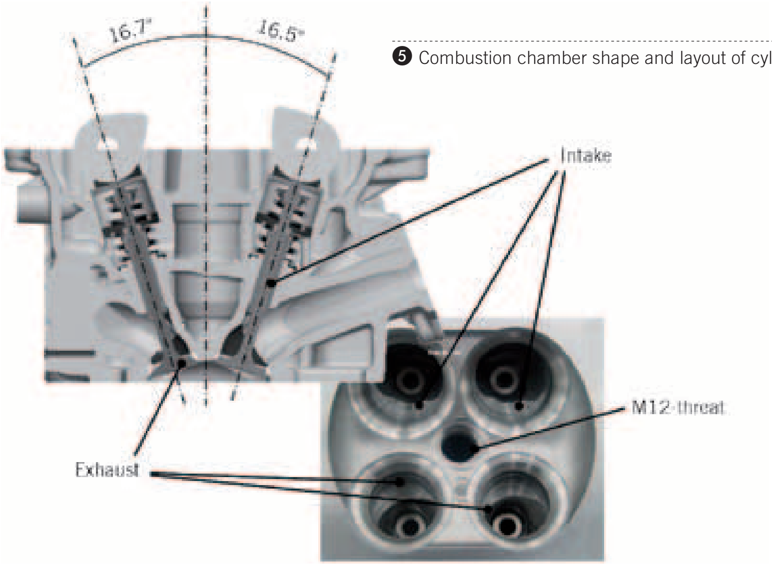
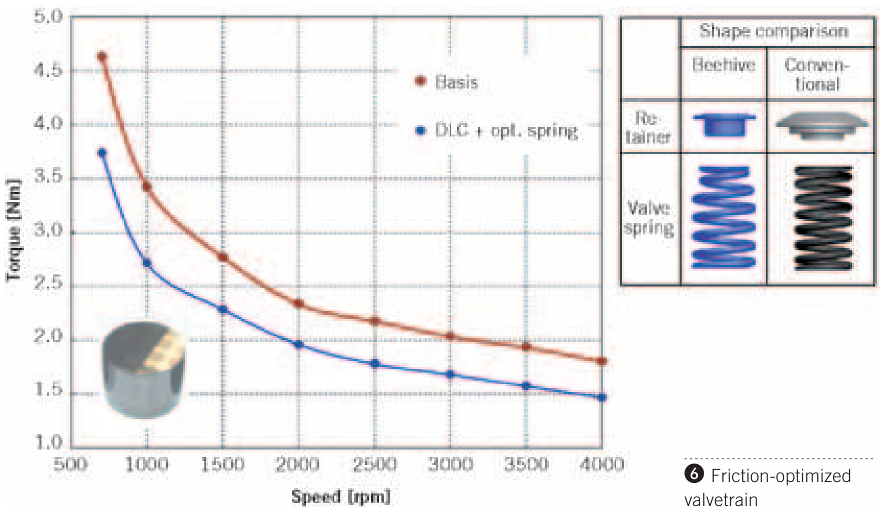


❺ Combustion chamber shape and layout of cylinder head



❻ Friction-optimized valvetrain

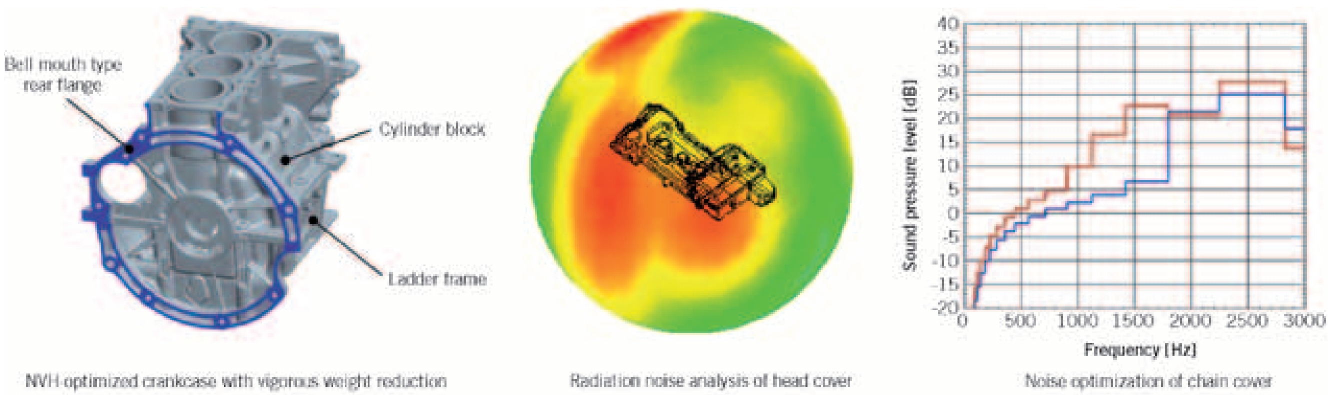
curved structure from the typical straight variety, which improves torque and maximum power by 1.0 Nm and 1.5 kW, respectively. The exhaust manifold is made of cast iron, thereby reducing cost by 30 % compared to a stainless steel exhaust manifold. The increased content of silicium enables the Kappa to resist oxidation under high-temperature conditions and to improve the catalyst durability. The new engine fulfils the latest Euro 5 standard.

OPERATION STRATEGY

All derivatives of the new three-cylinder engine are available in combination with an engine start-stop system which lowers fuel consumption by approximately 3 %. The decision to implement an engine start-stop system was taken after evaluating a number of single parameters, such as the clutch pedal position, the shift lever, vehicle speed, level of battery charging, the outside temperature and electrical consumption. In the algorithm, safety-relevant aspects get highest priority. Vehicles with a start-stop system have a more efficient starter as well as a battery with higher capacity. An Alternator Management System (AMS) controlling the alternator based on driving conditions is also used and increases fuel economy by about 1.5 %.

OPTIMIZATION OF ENGINE AND VEHICLE ACOUSTICS

For reducing noise when the engine is at idle, a ramp profile of the camshaft is optimized to eliminate vibration from valve action. The shape and volume of the delivery pipe are changed to decrease ticking noise of an injector, thereby minimizing the high-frequency noise component. In order to reduce radiated noise in Wide Open Throttle (WOT), the engine



❼ Improvement examples by NVH analysis

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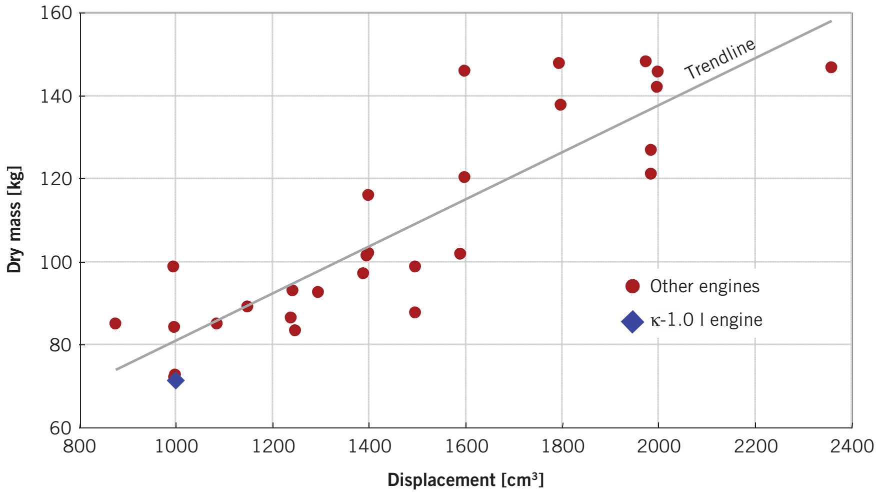


❽ Dry mass of new κ 1.0 l engine in comparison to competitor engines

structure is analyzed and modified by using extensive CAE. Also, to improve engine NVH, a high-strength aluminium engine block and ladder frame are used. The circular matching structure allows the powertrain to be stiffened. Additionally, compact and strong accessory packaging is applied by directly mounting both the alternator and air conditioning compressor on engine block. To reduce radiated noise, the vibration path from piston to ladder frame is optimized and radiation from radiation surfaces such as head cover, chain cover and in/exhaust is reduced.

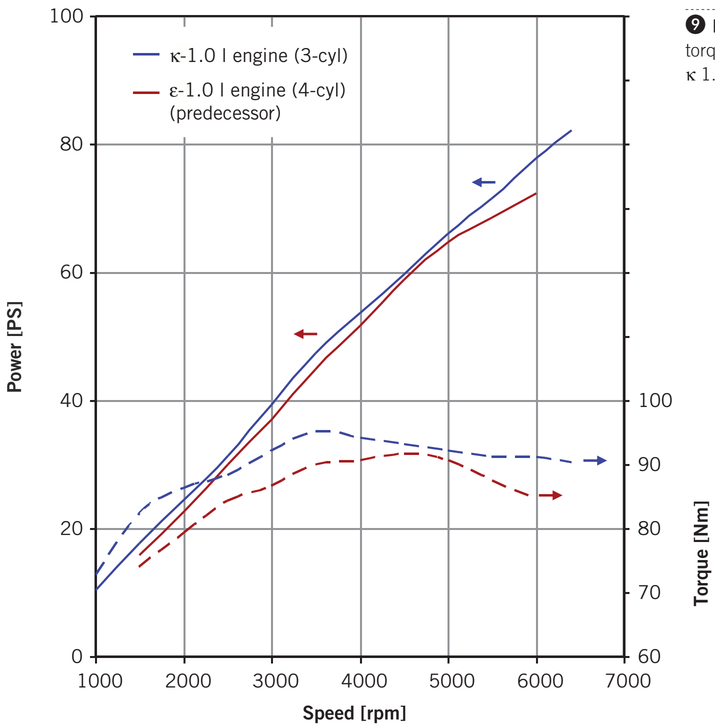
❼ shows the analytical results of reduction of vibration and radiated noise from a chain cover and a head cover by using CAE. To reduce rumble noise from the engine partial load operation conditions, ECU data, such as spark timing, are optimally matched. To decrease both whine and ticking noise of the chain drive a Pressure Regulation Valve (PRV) is applied to the chain tensioner. Also, cooling fan noise of the alternator at middle- and high-speed acceleration was reduced by applying a dual fan configuration. Vibration level in the vehicle interior is reduced by using a stiffer dash panel and a dense isolation pad.

BI-FUEL VARIANT FOR DRIVING WITH GASOLINE AND LPG

With tightened CO_2 regulations and oil price fluctuation, the need for developing an LPG engine is growing. However, because of the shortage of LPG infrastructure, demands on the development of Bi-Fuel engine – which consumes both gasoline and LPG fuel – are increasing. To meet these needs, Hyundai-Kia has developed the 1.0 l LPI Bi-Fuel version of the Kappa. Volumetric efficiency of the Kappa 1.0 l Bi-Fuel engine is improved by applying a Liquid Petroleum Injection (LPI) system. This injects LPG into each cylinder head port's entrance and controls the rate of fuel flow accurately. Therefore, CO_2 is reduced by 5 % while power is improved as much as in the gasoline variant. The Kappa LPI Bi-Fuel engine is equipped with both gasoline and LPG injector. And because of poor conditions in the combustion chamber caused by the dry characteristic of LPG, superior valve seats and valves in properties of abrasion, corrosion and heat conductivity are developed. Also, the piston top ring is PVD (Physical Vapour Deposition) coated to improve durability.

RESULTS

With the application of the latest technologies, such as Dual CVVT and the switchable intake manifold (VIS), the new Kappa 1.0 l engine achieves the best-in-class performance. The same technologies in combination with a carefully detailed optimization – particularly in the field of the engine mechanics – allow partial-load fuel consumption to represent a new optimum within the competitor engines ($\lambda = 1$ and Non-EGR) with a value of only 375 g/kWh at 2000/min and 2 bar. By using various technologies to decrease noise, the Kappa improves NVH quality about 2 to 3 dB over competitive engine at whole engine running zones. To reduce weight, the Kappa uses an aluminium



❾ Performance and torque curve of the new κ 1.0 l engine

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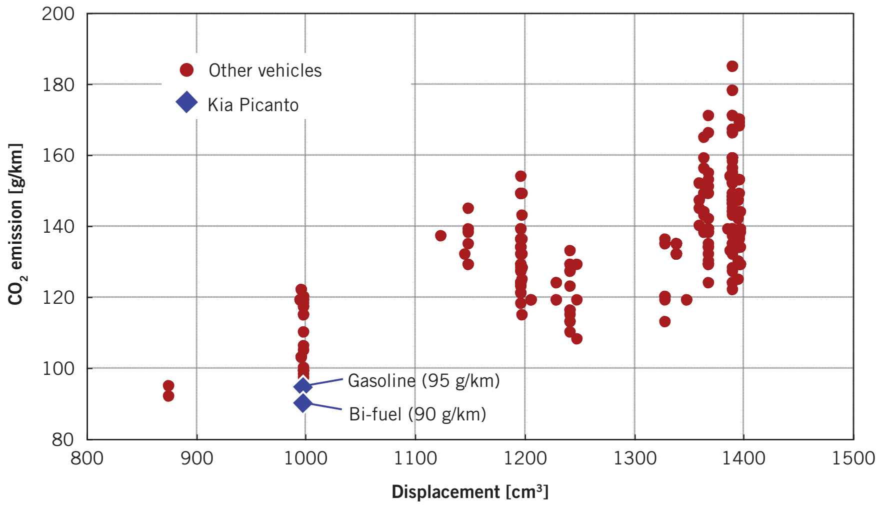

❿ CO_2 emission of new Kia Picanto in the field of competitors

cylinder block, plastic intake manifold and other technologies. Through strain and stress analysis and NVH development, the shape of rib is optimized and the thickness of the wall becomes thinner. As a result, the Kappa weighs only 71.4 kg, making it the lightest 1.0 l engine in comparison to competitor engines, ❽.

SUMMARY AND OUTLOOK

The interdisciplinary efforts in the course of the development of the new Kappa 1.0 l engine led to an aggregate with high power and efficiency. The derivate with switchable intake manifold delivers 60 kW/82 PS and achieves a maximum torque of 94 Nm, ❾. The specific advantages of the three-cylinder concept, the application of selected technologies – such as start-stop system, but also detailed optimizations of all components – contribute to an efficient vehicle engine. High requirements for comfort were fulfilled by consistent treatment of the concept-specific challenges. CO_2 emissions of 95 g/km for the new Kia Picanto with a gasoline engine and 90 g/km for the variant with Bi-Fuel engine identify a new benchmark in the 1.0 l class, ❿. As the further variant, the production of an ethanol-compatible engine (FFV) will start within 2011; a turbocharged version of the 1.0 l engine is under development and will mark a further, consistent step toward sustainable mobility in near future.

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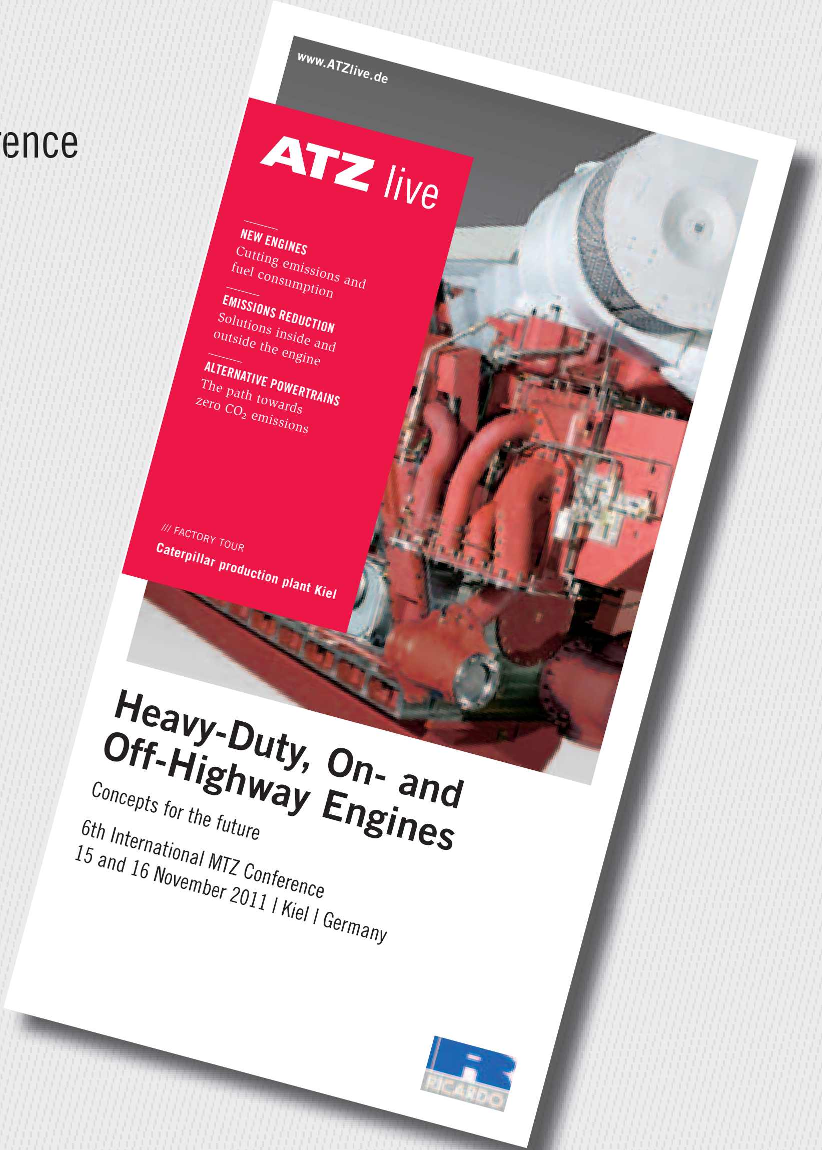

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EFFECT OF THE SKIRT STIFFNESS ON PISTON RELATED NOISE

The trend of increasing installation clearances in order to reduce frictional losses, being driven by the problem of CO_2 emissions, leads once again to acoustic optimisation having an increased significance. Mahle uses secondary piston motion measurements in a small, three-cylinder gasoline engine to demonstrate the effects of changes in the skirt rigidity, or rigidity distribution over the length of the skirt, on the piston motion, and therefore on the noise excitation.

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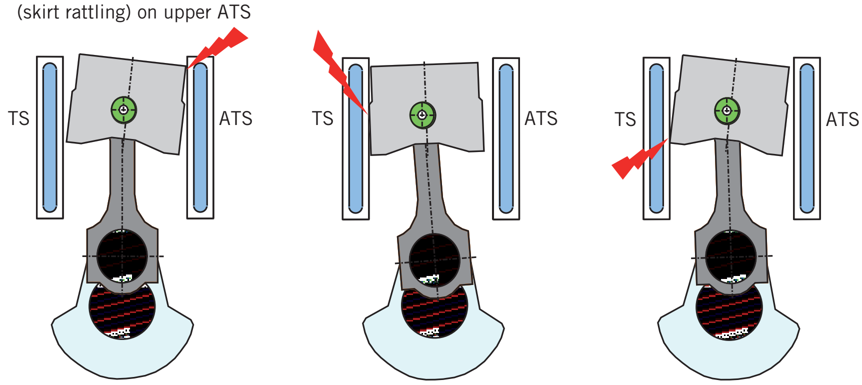
PISTON RELATED NOISES IN GASOLINE ENGINES

One of the mechanisms that generates noise in a reciprocating piston engine is the motion of the piston, due to its clearance perpendicular to its normal running direction. Depending on the load, the speed, and design parameters, such as piston pin offset, the piston performs very different motion sequences that are characteristic of particular gas and inertial force conditions. These motion mechanisms lead to structural excitation when the piston strikes the cylinder wall, which can lead to the most commonly occurring, unpleasant piston noises of "croaking" and "rattling", depending on the excitation intensity. The noise known as "croaking" is caused under conditions of gas pressure dominance, when the upper skirt region strikes the thrust side (TS) after the ignition top dead center (ITDC), while "rattling" occurs under inertial force dominance, when the top land or ring land (top land rattling) or the upper skirt region (skirt rattling) strikes against the antithrust side (ATS) around the ITDC, ❶ [1].

The test engine has a crankshaft offset in the TS direction, which is a multiple of the piston pin offset. Compared to an engine with no offset, this results in a different lateral piston force progression, and a different piston acceleration relative to the crank angle throughout the stroke. As a result, the contact alteration takes place unusually late at operating points that are dominated by gas pres-



❶ Structure-borne noise excitation locations of selected piston noises

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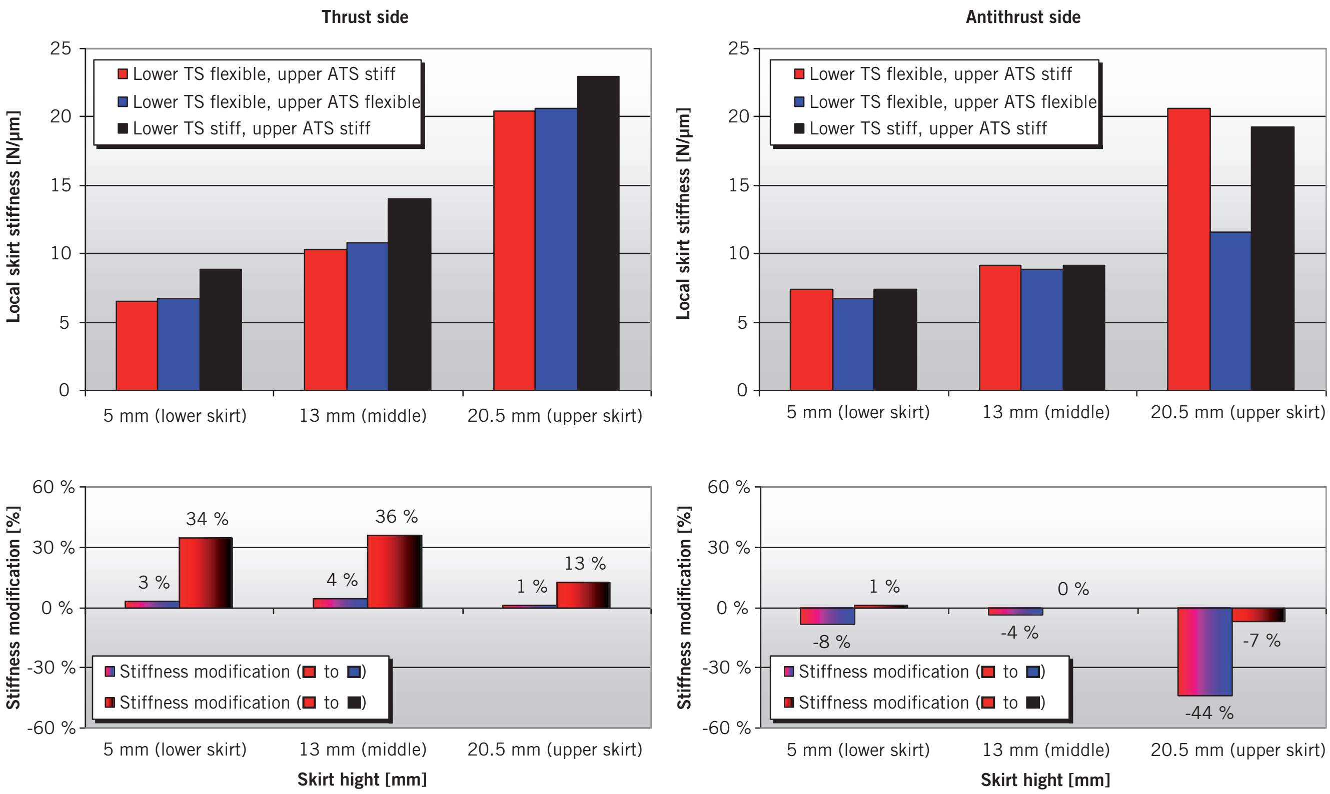

❷ Local stiffness of the skirt and percentage difference of the tested variants

sure, while the impact on the antithrust side occurs relatively early for operating points dominated by inertial force. For operating points dominated by gas pressure, another phenomenon can be observed in that the contact alteration of the bottom end of the skirt from the ATS to the TS does not occur until the area around the ITDC, and can therefore cause a noticeable structural excitation of the engine block (clatter noise).

MEASUREMENT OF PISTON MOTION

A measurement method using contactless displacement sensors mounted in the cylinder bore on the TS and ATS is used to determine the piston motion [2]. The advantage of this method is a quick and simple changing of the test variants, without requiring setup of the measurement equipment on the piston. The tilt angle curve for the piston axis and the translational displacement of the piston must be calculated from the usable individual segments of the measured lubricant gap curves. The signal range that can be interpreted is limited, in this case, a crank angle range from 33° before to 38° after top dead center (TDC). Within this range, the piston skirt surface is at the height of the measuring planes. The asymmetry of the range around the TDC that can be evaluated is due to the crankshaft offset in the TS direction in the test engine.

VARIATION OF SKIRT STIFFNESS

With increasing piston installation clearance, as is currently claimed more and more in order to reduce friction losses, it becomes more difficult to obtain a satisfactory noise profile over all operating ranges with their specific excitation mechanisms by varying the optimisation parameters of pin bore offset and piston shape. Another way to influence the piston motion and its associated structural excitation, as well as the intensity of the excitation, with potentially unchanged piston motion, is by varying the stiffness of the skirt. For the tests, the rigidity of the piston skirt was varied on both the TS and the ATS sides. For the thrust-side variation, the reinforcing collar in the transition between the skirt and the box wall in the lower skirt area was removed. In a further step, the flexibility in the upper skirt area on the ATS was increased by targeted removal of material. In this manner, the rigidity values were reduced by about 35 % at the bottom of the TS, and by about 44 % at the top of the ATS, ❷. These values were determined experimentally, by applying a defined static load and measuring the deformation.

EFFECT OF VARYING STIFFNESS ON PISTON MOTION AND NOISE EXCITATION

In order to visualise the effects of modifications to rigidity, two operating points are shown here. At a lower speed of 1600 rpm and partial load of 50 Nm, a typical motion sequence can be seen, which can lead to thrust-side skirt striking and the noise known as “croaking”, ❸. At the increased speed of 2400 rpm and very low load (5 Nm), the typical motion mechanism that may lead to “rattling” is evident, ❹.

In ③, the differences in structure-borne noise that are relevant to excitation can be observed on the TS at 393° CA. Due to the low level of

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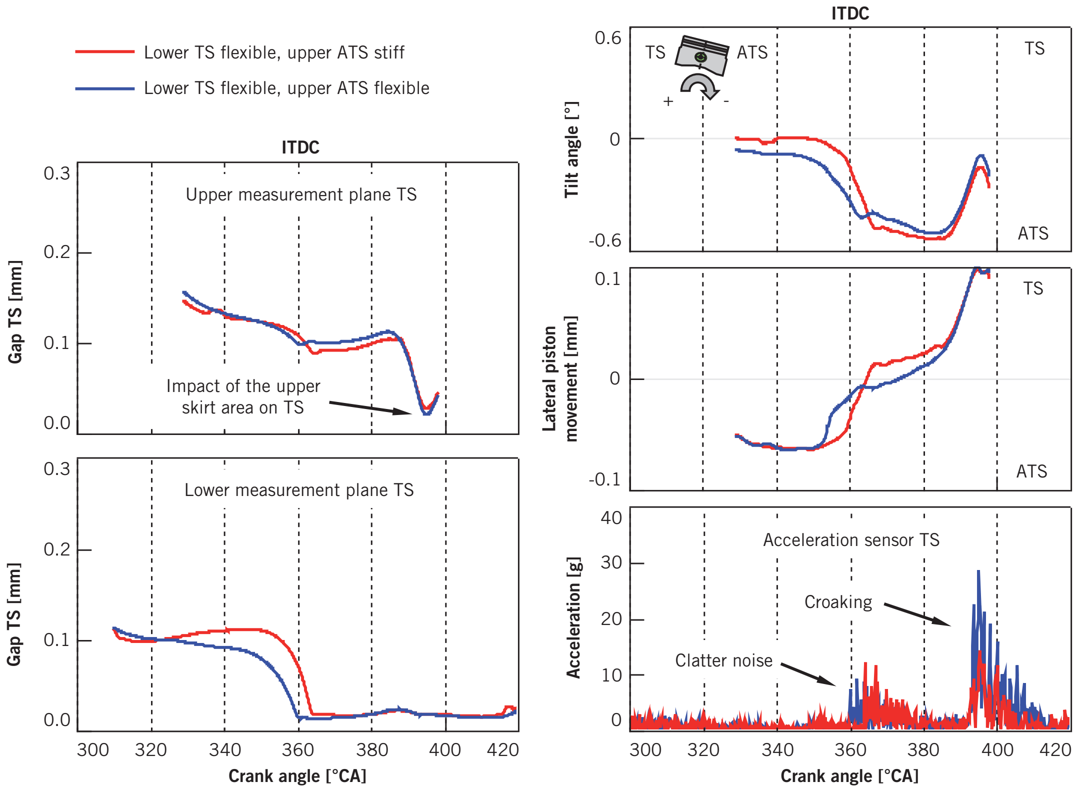
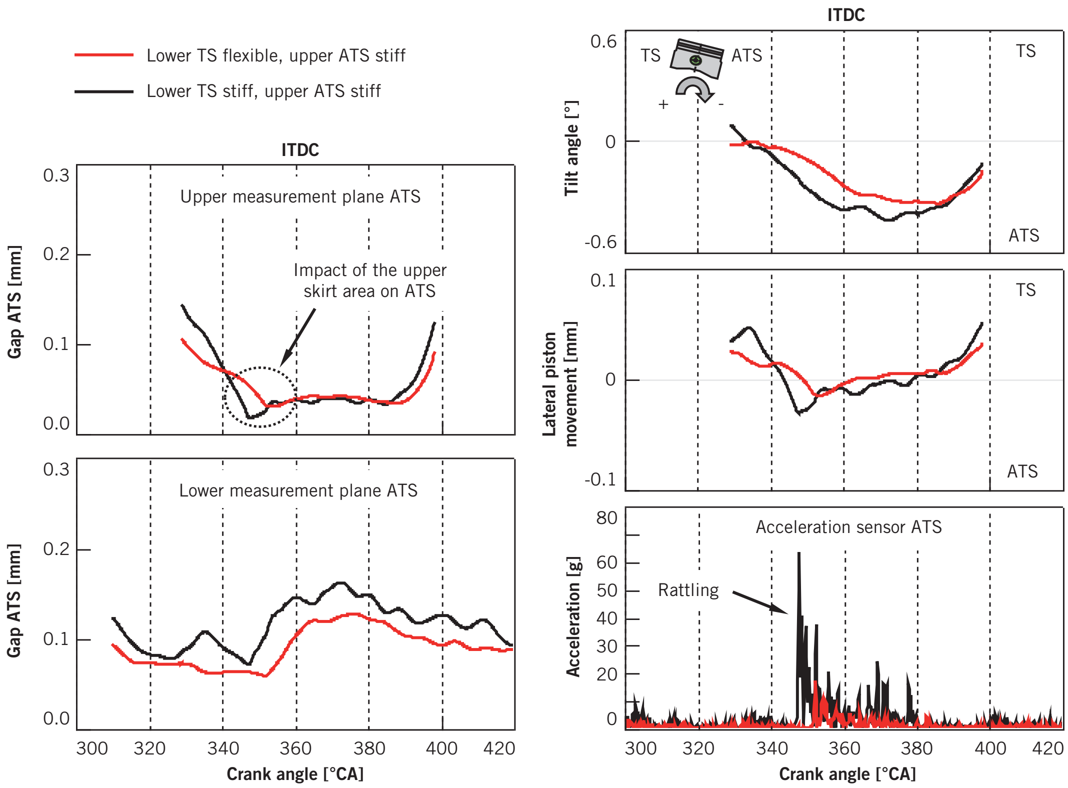
influence of inertial forces at this operating point and the relatively high gas pressure, the piston is in contact with the ATS during the compression stroke. As the gas pressure increases, the piston, with the TS bottom edge leading, starts to rotate into the direction of TS until it makes contact there. The top skirt is thereby supported on the ATS. Due to deformation, this rotation begins earlier and at a lower rotational speed with the piston with reduced skirt rigidity in the



❸ Piston motion behaviour with modified skirt stiffness on the antithrust side in the top skirt area, and thrust-side structure-borne noise excitation, 1600 rpm, 50 Nm, cold engine



❹ Piston motion behaviour with modified skirt stiffness on the thrust side in the bottom skirt area, and antithrust-side structure-borne noise excitation, 2400 rpm, 5 Nm, cold engine

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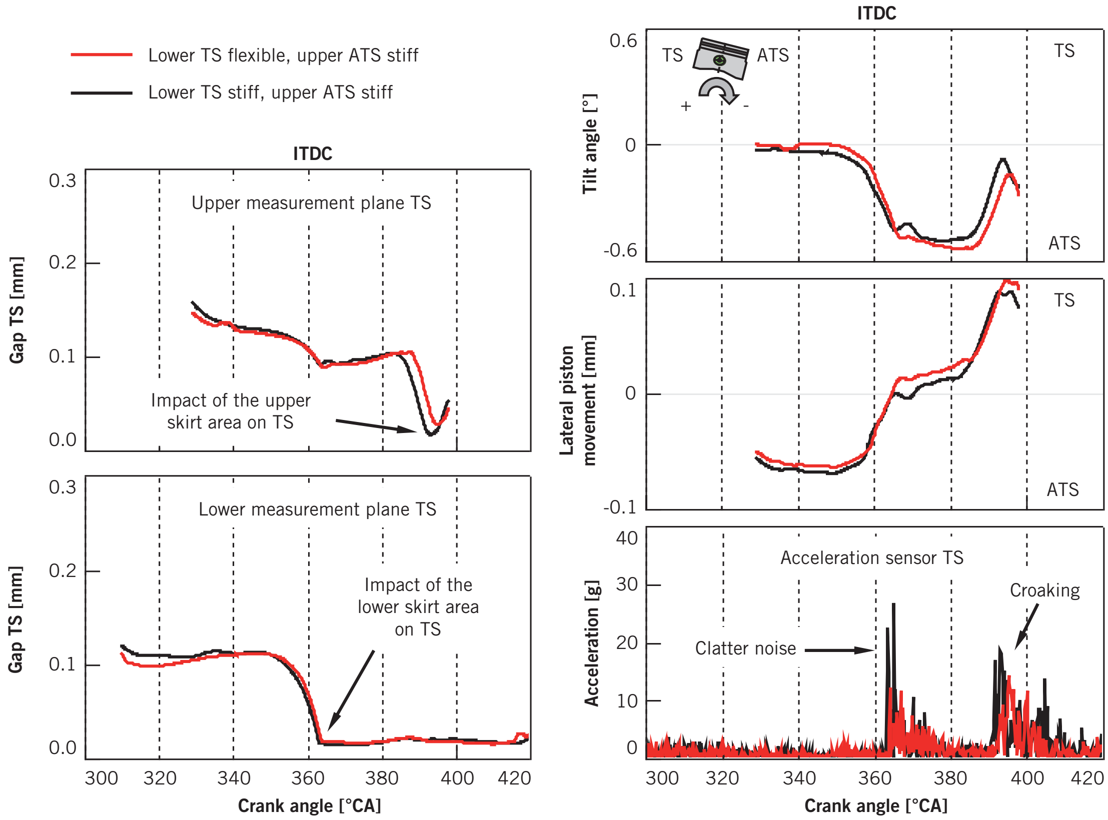


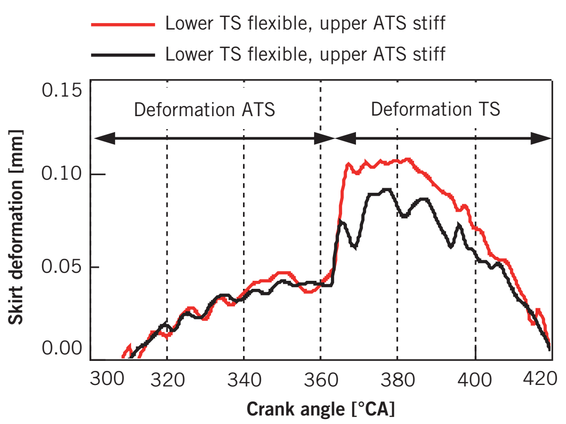
❺ Piston motion behaviour with modified skirt stiffness on the thrust side in the bottom skirt area, and thrust-side structure-borne noise excitation, 1600 rpm, 50 Nm, cold engine

top area of the ATS. As the motion continues, the piston rapidly starts to straighten, and the top skirt area strikes the cylinder wall on the TS at the end of this process, at 393° CA. The backing rotation takes place slightly later for pistons, which are more rigid on ATS, and at a greater tipping angle to ATS, so that the motion stops at an even greater tilt angle and the structure-borne noise excitation can be reduced.

In ④, the piston with the thrust-side reinforcing collar shows significantly greater structure-borne noise excitation on the ATS, starting around 344° CA, compared with the piston, which is more flexible on the TS and does not have this reinforcing measure (rigidity difference on the thrust side is about 35 %, ②). The cause is the significantly different motion behavior of the two pistons. As can be seen from the lubricant gap signal of the top measuring plane on the ATS, the piston with the flexible thrust-side skirt approaches the antithrust-side cylinder wall from a lesser distance, as well as a significantly lower speed. Contact is made with the top skirt area, due to the angle of tilt to the ATS. The top land and ring land have sufficient clearance and show no traces of contact after disassembly. The rigidity of the skirts in both variants is about the same at this location, so the differences in excitation can be attributed almost exclusively to the difference in motion behavior.

The differences in piston motion curves due to differences in skirt rigidity are not the only things affecting the noise excitation. For example, the variants considered in ❺, with varying thrust-side skirt stiffness, have a nearly identical motion curve before the TDC, with significantly differently structure-borne sound amplitudes in the range of 366° CA.

Due to the offset of the crankshaft in the TS direction, the rise in the tilt angle and the contact alteration of the bottom end of skirt toward the TS, just before ITDC, occurs unusually late in a phase of high cylinder pressure at the observed operating point, and is already complete just after ITDC, at about 366° CA. This means that the bottom end of the skirt



❻ Dynamic piston skirt deformation at the height of the lower measuring plane, during operation at 1600 rpm and 50 Nm, cold engine

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strikes the cylinder wall on the thrust side with a great deal of kinetic energy. For the piston without the reinforcing collar, this contact leads to a significant reduction in structural excitation.

Assuming that skirt deformation during gas exchange is negligible at the relatively low observed speeds, the actual running clearances that can be determined from the displacement measurement signals can be used to determine the dynamic skirt deformation during engine operation, as the difference in running clearance between the range around the gas-exchange TDC and the ignition TDC, ❻. Here, the piston without the thrust-side collar shows up to 38 µm more deformation after the bottom of the skirt strikes on the TS, compared with the piston with a rigid skirt.

SUMMARY

The results show that the noise excitation due to the piston can be significantly affected by a targeted design of the thrust- and antithrust-side skirt rigidity conditions. One observed direct effect is that contact by a skirt area having increased flexibility leads to lower vibration excitation in the crankcase. More of the kinetic energy on impact is converted into acoustically advantageous elastic deformation on the piston side, and less into a vibration of the crankcase structure. A change in skirt rigidity also affects the motion well before the actual contact alteration, so that it takes place from a different initial piston position. With a favorable design, this allows a significant reduction in noise. Such acoustic optimisation by skirt stiffness design must take place at a very early phase of the design, because later optimisation means changing the raw part and thus the foundry mold.

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FRICTION REDUCTION IN POWER CYLINDER SYSTEMS OF GASOLINE ENGINES

The power cylinder system, consisting of the piston rings, pistons and cylinder running surfaces, offers great potential for a further reduction in friction. A study by Kolbenschmidt Pierburg quantifies the parameters that cause friction and shows which further developments are particularly effective in minimising friction and wear.

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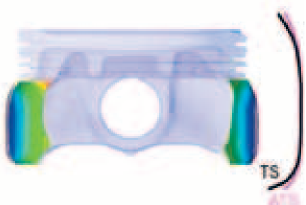
REDUCING FRICTION IN THE SYSTEM

Due to increasing specific power density in connection with the requirement for reduced fuel consumption and emissions, the tribological demands on the internal combustion engine are expanding as well. The power cylinder system (PCS), consisting of piston rings, pistons and cylinder friction surfaces, plays a central role here in terms of wear resistance and friction reduction. This complex tribological system can meet the increasingly stringent requirements imposed on heavily stressed internal combustion engines only through continuous research and development in the fields of materials, simulation, testing and production technology [1]. The desire to minimize friction in the basic engine leads directly to the PCS, which accounts for some 50 % of engine friction. Of course, reducing friction in any system requires high-precision quantification of friction losses. Therefore Kolbenschmidt Pierburg AG uses simulation tools and the 'floating liner' method developed by Furuhama, which enables exact crank angle resolved measurement of friction losses in PCS [2].

PISTON DESIGN FOR MINIMUM FRICTION LOSSES

The lightweight Liteks design already discussed in [3], ❶, supports the requirements of engine developers owing to its typical characteristics, including large ring zone recess with double rib support, heightened section modulus of the piston crown due to concave casing walls that taper towards the base, and – ever since introduction of the 2nd generation – optimum tensile-strength boss surfaces that dovetail with the casing walls.



❶ Comparison of the lightweight concepts Liteks-1 (left) and Liteks-2 (right)

Lightweight and fatigue strength do not depend solely on piston design, of course, but also result from solutions involving innovative materials. Thanks to its special composition, the new high-performance alloy KS309 achieves significantly enhanced resistance to fatigue at 250 °C [1]. At a typical base temperature of 300 °C, the new material results in the fatigue strength some 25 % greater than the alloy used today. Apart from enhanced material characteristics, specific improvements in the casting process have been implemented, including casting of especially thin-walled thicknesses.

Besides a reduction in weight, the Liteks lightweight concept has special friction-reducing features. The already mentioned load adjusted asymmetric skirt widths following the introduction of the Liteks-2 generation have resulted in lower friction losses, ❷ (top) due to the smaller contact surface with the cylinder wall. Moreover, this feature is combined with an optimized asymmetric convex skirt profile. This improves the hydrodynamic formation of the lubricant film as well as reducing the friction surface of the skirt. Here, it proved possible to obtain a lowering in friction mean effective pressure (FMEP) of 20 to 24 % at higher speed rates (2000 to 3000 rpm). This can be seen especially clearly in the friction force diagram, ② (bottom right). In addition, the Liteks-2 concept enables a reduction in size of the pin offset as well as an increase in initial assembly clearance, thus resulting in further advantages with regard to friction behaviour without compromising on noise levels. Combined with the design features the use of polyamideimid-based Nanofriks skirt coating leads to a reduction in mixed friction losses at top (TDC) and bottom dead center (BDC) through the well-placed application of solid lubricants [5]. The high wear resistance achieved here enables greater maximum surface pressure on the piston skirt surface with reduced skirt contact surfaces of the Liteks-2 concept.

FRICTION BENEFIT THROUGH OPTIMIZED RING DESIGN

Optimizing the piston ring package can make an important contribution to minimizing friction in the PCS. In cooperation with Nippon Piston Rings (NPR), a low-friction ring family was developed which

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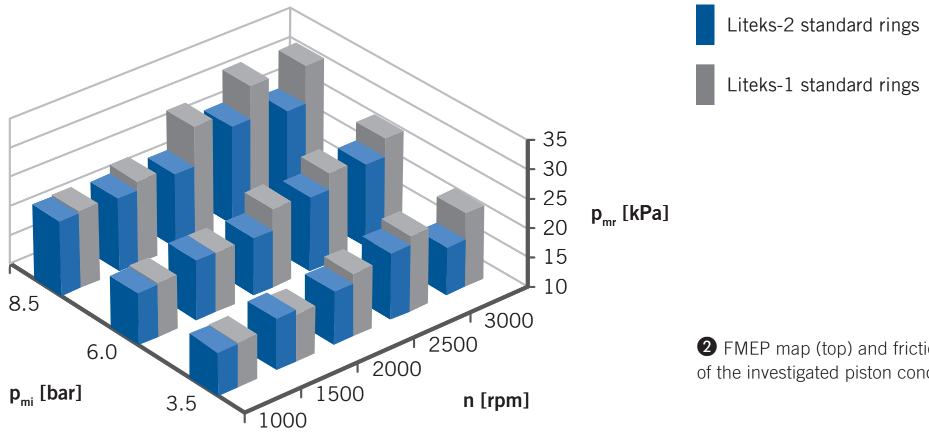
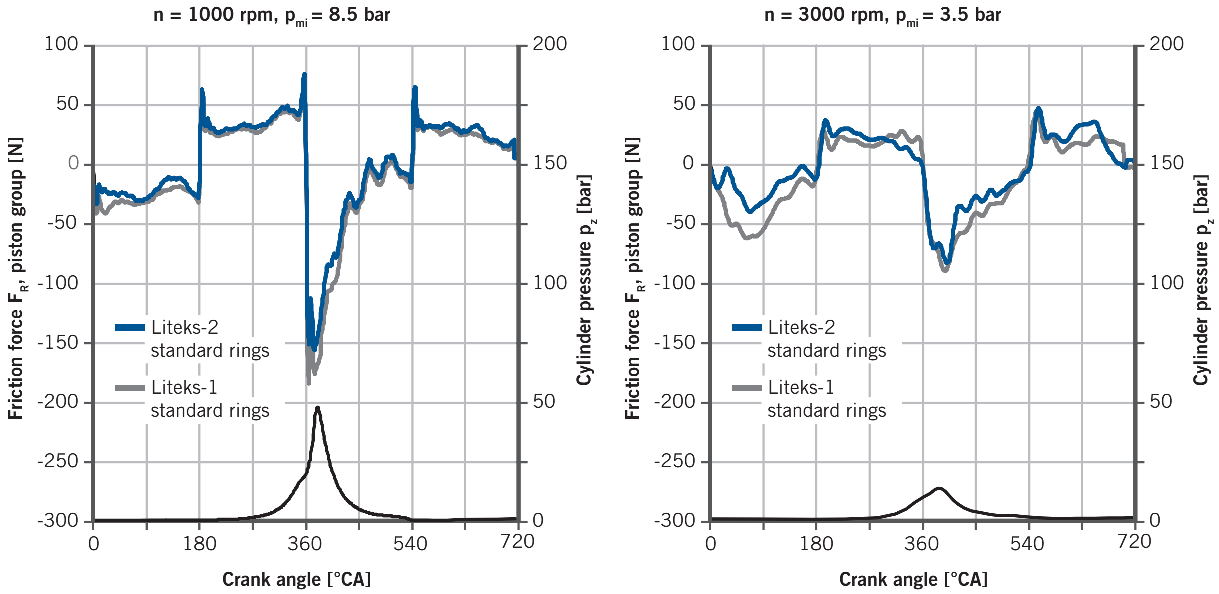


❷ FMEP map (top) and friction force diagram (bottom) of the investigated piston concepts



	Standard rings	Low friction rings	Low friction PVD	Low friction DLC
1. Groove				
Ring type	Barrel faced	Barrel faced with a strong curvature	Barrel faced with a strong curvature	Barrel faced with a strong curvature
h1 x a1 [mm]	1.2 x 2.9	1.2 x 2.7	1.2 x 2.7	1.2 x 2.7
s1 [mm]	0.28	0.28	0.28	0.28
Tangential force [N]	7.6	5.6	5.6	5.6
Surface treatment	Nitrided	Nitrided	PVD	DLC
2. Groove				
Ring type	NM	EM	EM	EM
h1 x a1 [mm]	1.5 x 3.2	1.0 x 2.7	1.0 x 2.7	1.0 x 2.7
s1 [mm]	0.45	0.445	0.445	0.445
Tangential force [N]	8.7	4.9	4.9	4.9
Surface treatment	Phosphated	Nitrided	Nitrided	Nitrided
3. Groove				
Ring type	ES2	SOR	SOR Step land	SOR
Ass. h1 x a1 [mm]	2.0 x 2.45	1.5 x 1.95	1.5 x 1.95	1.5 x 1.95
s1 [mm]	0.35	0.275	0.275	0.275
Tangential force [N]	30.0	12.0	8.0	12.0
Surface treatment	Nitrided	Nitrided	PVD	Nitrided
Total tangential force [N]	46.3	22.5	18.5	22.5

❸ Ring characteristics of the low friction family [4]

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reflects the requirements imposed by various cylinder surfaces. The ring characteristics are compared in ❸. The standard rings represent the current production standard. The low friction rings display characteristics indicating low FMEP on cylinder surfaces consisting of cast iron or AlSi alloys (LF DLC).

An effective reduction in friction requires first and foremost a lessening of the hydrodynamic friction force which sets in at high piston velocities in mid stroke. Here, reducing the contact surfaces to the cylinder and the tangential forces at the ring package makes an essential contribution. In order to lessen the mixed friction in the area of TDC and the accompanying wear, compression rings can be additionally provided with friction-reducing, wear-resistant coatings.

The low friction (LF) Eco ring set with nitrided steel rings display an improved ring conformability. With the top ring, the contact surfaces to the cylinder are reduced through greater curvature in the direction of stroke; with the oil ring, through a lower land height. In order to counter the danger of wear to the top and oil ring and thus to ward off the increase in FMEP over time, the LF PVD ring set is coated with PVD on the running surfaces. Furthermore the oil ring is likewise provided with a reduced, echeloned contact surface accompanied by lower tangential force, meaning that the contact surface pressure and oil consumption remain constant.

In [4], with the AlSi cylinder surfaces, the top rings with DLC coatings display a reduction in mixed friction in the area of the dead center Moreover, they reduce damage to the cylinder surface during running in. The simulation clearly demonstrates the effectiveness of the measures described in reducing the friction force of the ring package, which also enables the breakdown of component friction for the individual operating points via FMEP [4].

❹ shows the FMEP and friction force progression of the PCS with the standard, LF Eco and LF PVD ring set described above on the basis of the Liteks-2 piston. In ④ (bottom), the influence of reduced tangential tension and shorter axial contact lengths of the LF Eco and LF PVD packages is especially clear in the upward stroke, with the system gliding on a relatively thin film of oil here.

④ (top) shows FMEP of the PCS in the characteristic map. The friction advantages of the LF Eco and LF PVD packages

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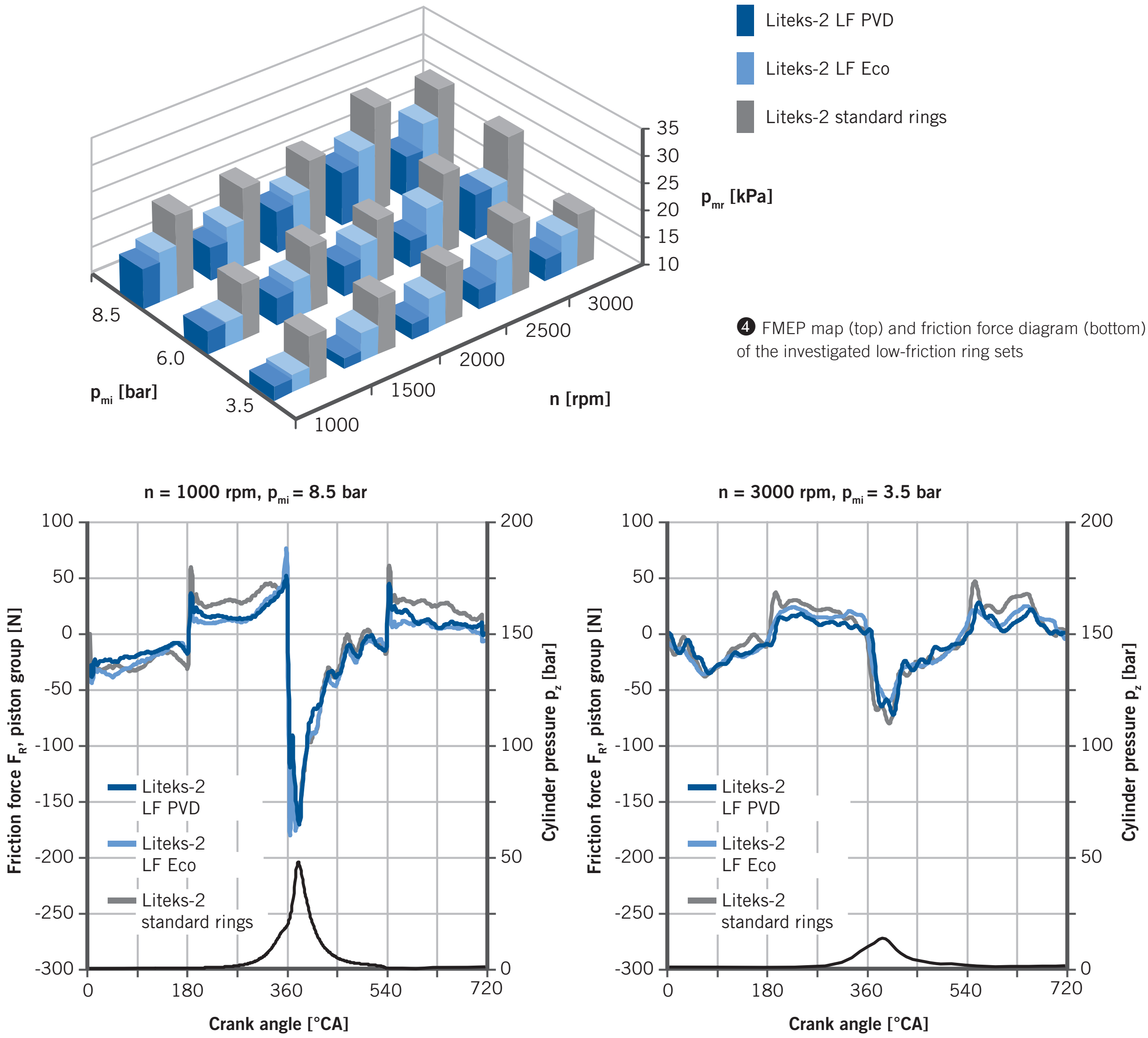


❹ FMEP map (top) and friction force diagram (bottom) of the investigated low-friction ring sets demonstrated in the ring simulation are depicted in the partial load characteristic map. This reveals the friction advantages in FMEP for all operating points starting with a speed of 1500 rpm. Depending on the operating point, with the LF Eco these can amount to 10 % and 22 %, and with the LF PVD, to between 26 % and 37 %.

REDUCTION OF FRICTION AND WEAR ON THE CYLINDER SURFACE

The cylinder surface is a critical element in the design of the piston system, and thus offers potential for optimizing friction and wear in this complex tribological system. Monolithic engine blocks are often based on Alusil technology [6]. Here, the cylinder surface is characterized by the mechanical exposure of primary silicon (standard honing), ❺ (top left). Structure honing represents an advance on standard honing, in which the surface typography displays fine honing grooves, ⑤ (top right).

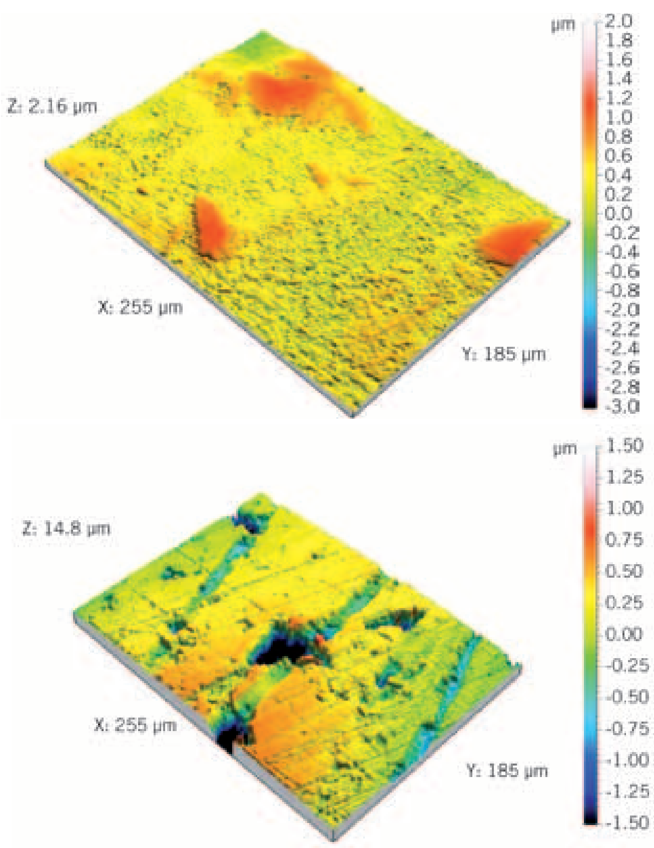
Thanks to the use of quasi-monolithic concepts with thermal injected ferrous-based coatings such as PTWA (Plasma Transferred Wire Arc), ⑤ (bottom), it is possible to combine the weight advantages of the monolithic AlSi engine block with the increased resistance to fatigue of ferrous-based-coated cylinder surfaces. In order to create a strong bond between the Al base material and the ferrous coating as well as optimum coating adherence, honing as described in [6] is necessary, taking the form of an undercut profile with the aid of special thread cutting insert.

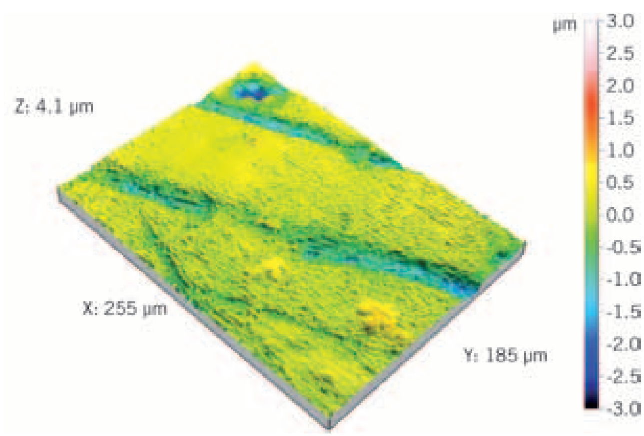
The characteristic FMEP map and the friction force diagram of the cylinder surface technologies discussed are compared in ❻. In FMEP map, structure honing is depicted in comparison with Alusil standard honing, revealing a significant friction advantage in the region of 1000 to 2000 rpm, ⑥ (top). Here, especially at a speed of 1000 rpm, depending on the load, a reduction in FMEP of 14 to 16 % was determined. Another advantage is the significant reduction in piston skirt wear. As seen in the friction force diagram, ⑥ (bottom left), this can be explained by the significantly reduced friction force peaks and lower mixed friction at high cylinder pressures and low rpm rates in the region of 360 to 390° CA. Both at high piston velocities and at the PCS's dead centers, the cross structure and the accompanying increased R_{vk} values and lower R_{pk} values resulted in a reduction of FMEP in the entire FMEP map.

Taking into account knowledge gained in previous investigations of structure honing, optimum honing parameters were determined for ensuring a further reduction in friction with ferrous coating using the PTWA method. Here, porosity depicts the thermally sprayed cylinder surfaces following natural oil retention volumes, and improves lubrication at the contact point [7]. As solid lubricants, iron oxides such as Wustite (FeO) and Magnetite (Fe_3O_4) also contribute to the reduction of solid object friction. In the low rpm range, the combination of technical manufacturing and material influence factors succeeded in reducing FMEP compared with friction-optimized structure roughening by a further 17 to 20 %. This is especially conspicuous at the PCSs dead centers at low piston velocities and maximum cylinder pressure, ⑥ (bottom).

SUMMARY AND OUTLOOK

The Kolbenschmidt Pierburg Group offers the complete development of the piston, piston ring and cylinder surface systems,





❺ Confocal microscope images (50x) – AlSi cylinder surfaces in gusset area, operating period 30 h in both cases: Alusil standard honing (top left), Alusil structure honing (top right), PTWA-coating (bottom)

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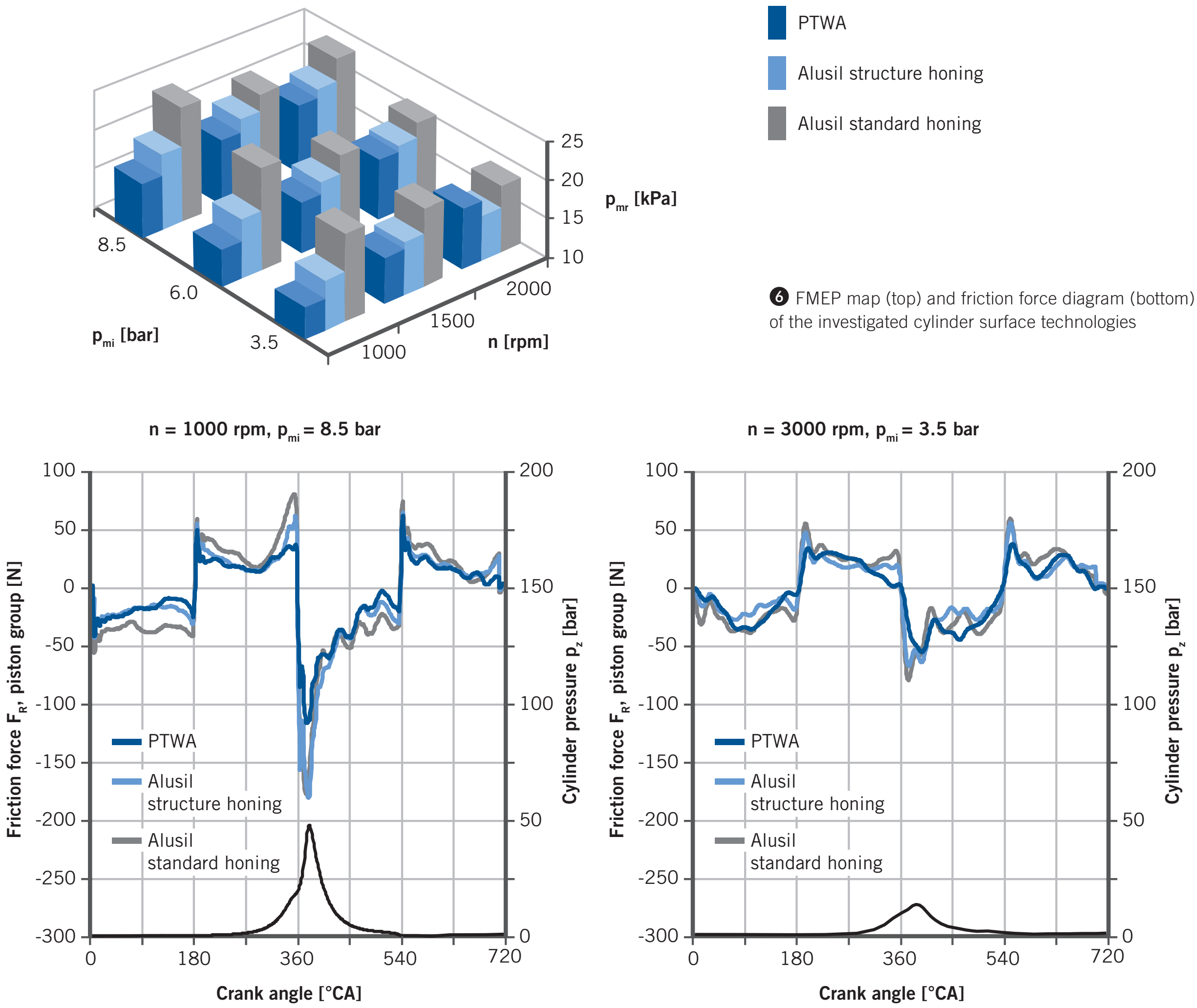

❻ FMEP map (top) and friction force diagram (bottom) of the investigated cylinder surface technologies

with the focus on optimizing friction behaviour and wear resistance of the complete system. By means of the friction force investigations depicted above, the effectiveness of measures for reducing friction losses in the entire PCS were quantified. The catalogue of measures for reducing losses in the pistons includes, among other things, the weight-reduced Liteks piston design with noise- and friction-optimized asymmetric skirt profile and the nano particle-reinforced PAI coating Nanofriks on the piston skirt. Together these measures resulted in a reduction of FMEP of up to 24 %.

In future, other advantages will arise from the lightweight design of the Liteks-3 generation thanks to the new Al-alloy KS309. In the base area, for example, improved material strength will lead to a substantial reduction in wall thickness of up to 25 %. In the window area, ready-cast wall thicknesses can for the first time be reduced to the minimum 2 mm wall-thickness required for strength reasons.

The low-friction ring family from NPR, with reduced tangential forces, lower contact surfaces at the compression and oil ring and modified cylinder surface coating, contributes significantly to minimizing friction, accounting for up to 38 % of the reduction within the power cylinder system.

In nearly all operating points of the FMEP map, structure honing was shown to minimize friction. Furthermore, it was demonstrated that quasi-monolithic concepts with ferrous coatings and corresponding final treatment and intelligently selected surface topography offer additional potential for reducing wear while simultaneously lowering friction in the PCS by up to 36 %.

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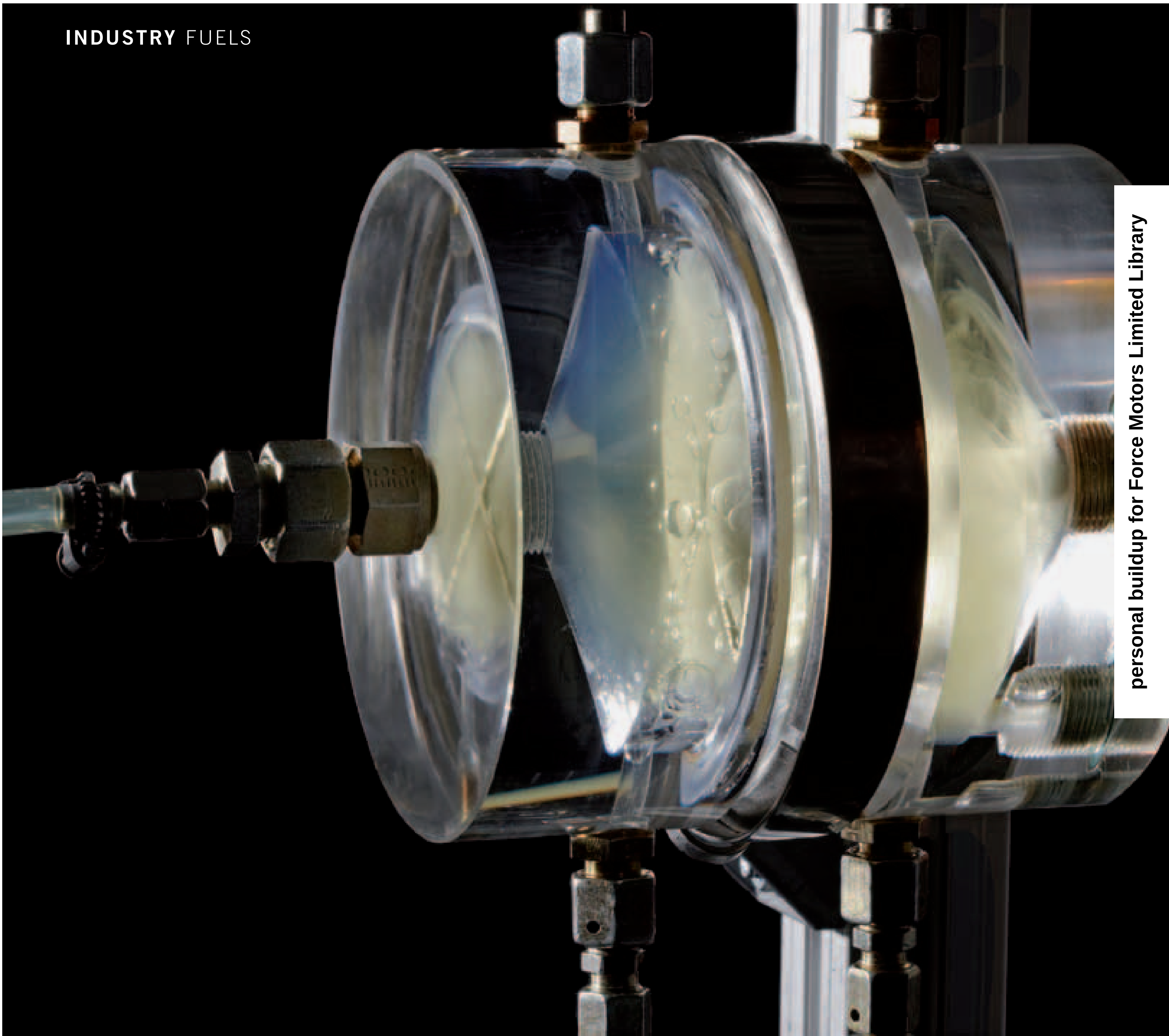

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NEW ROUTES FOR WATER SEPARATION FROM DIESEL FUEL

Highly-developed common rail diesel injection systems place greater requirements on the cleanliness of the fuel than the simple mechanical systems. Free, drop-shaped water impacts the function of the components in a high-pressure injection system and must, therefore, be reliably removed from the system. Mann+Hummel highlights the parameters that result in smaller water droplets and points towards a solution for how these can be enlarged and reliably isolated from the fuel.

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BOUNDARY CONDITIONS

To conform with market requirements and legislation, injection systems for modern diesel engines have evolved considerably with regard to energy efficiency and achievable exhaust gas quality. Common rail injection systems used today work with increased pressures of up to 2500 bar. These systems place markedly higher requirements on the cleanliness of the fuel than the simple mechanical systems of days gone by. Besides the particulate contamination of the fuel, free, drop-shaped water also impacts the function of the components in a high-pressure injection system and must, therefore, be reliably removed from the fuel system.

As well as the mechanical-technical parameters, the composition of diesel fuel has also changed considerably over the last decade [1]. The sulphur content has been reduced from 500 ppm to < 10 ppm while a biofuel content of up to 7 Vol.-% is now prescribed, ❶. This has resulted in the water droplets that build up as the fuel passes through the system having an average volumetric diameter of just 4 to 10 µm. These extremely finely dispersed water-in-diesel-emulsions are very stable and the water can no longer be quantitatively separated from the fuel flow with the classic hydrophobic systems. To achieve the required cleanliness under these altered conditions, the water separating systems must be adapted for future applications. Mann + Hummel is confronting this market challenge by developing a multi-stage water separator.

DIFFERENT FUEL QUALITIES

Injection systems used today require diesel fuels with good lubrication to prevent wear and abrasion. With the consistent reduction of sulphur content in diesel and the introduction of ULSD (ultra-low-sulphur diesel) in the USA and Europe [2], the inherent lubrication properties of diesel have decreased significantly. To counteract this, additives must be used to restore lubrication properties and wear-resistance.

Biodiesel is a cost-effective alternative to synthetically produced fuel additives and enhances the lubricating properties of diesel fuel. The addition of biodiesel leads to the proliferation of biodiesel-ULSD blends. The biodiesel components and additives

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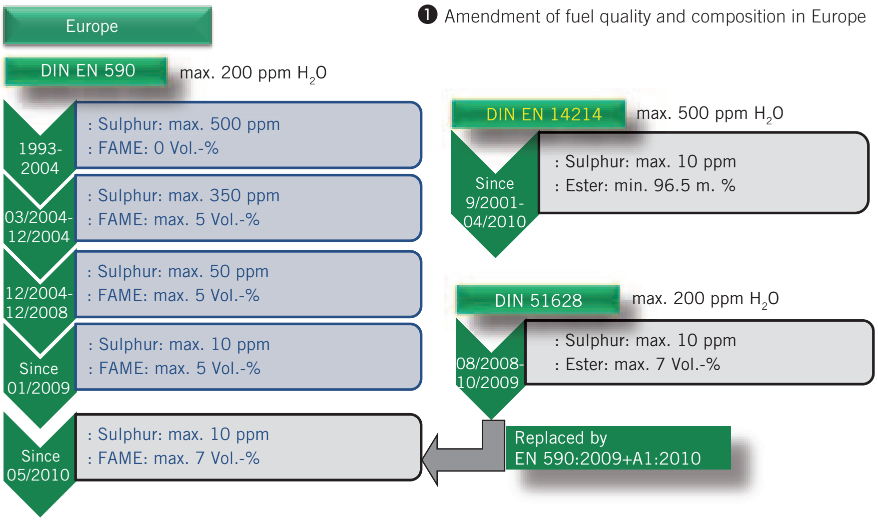

❶ Amendment of fuel quality and composition in Europe

have surface-active properties and impede or prevent the separation of the emulsified water with traditional water-separating concepts based on hydrophobic filter media [3].

INFLUENCE OF BIODIESEL AND BIODIESEL BLENDS

During the synthesis of biodiesel, the raw materials react to fatty acid methyl ester (FAME) and glycerol. These mostly incomplete reactions result in various residues from the raw materials and by-products remaining in the biodiesel. In the fuel system, in the presence of free water, these lead to the formation of extremely stable water-diesel emulsions, from which it is becoming increasingly difficult to re-separate the water droplets as the biodiesel content increases.

A further influence is the varying solubility of water in the various diesel blends. Standard EN 590 prescribes a maximum permissible absolute water content of 200 ppm. For pure biodiesel, standard EN 14214 stipulates a permissible concentration of 500 ppm.

The damage to a fuel injection system through wear and abrasion as a result of free water droplets in diesel is described extensively in relevant literature. It is sensible to consider more closely the varying solubility of water in diesel depending upon temperature. When driving, the fuel temperature is normally ≥ 80 °C. ❷ illustrates the increasing solubility of water in various diesel-biodiesel blends as the temperature increases. ❸ illustrates the solubility of water in pure biodiesel B100.

OPERATING LIMITS OF WATER SEPARATORS ON THE RAW SIDE

Water separators currently available on the market for passenger cars/HGVs principally come in two-stage designs. The filter medium consists of a raw-side hydrophobic polyester meltblown layer and a downstream extra-fine cellulose for particle separation. The altered chemical composition of the diesel results in a fundamentally different separation behaviour of the water droplets. Both the water droplets and the hydrophobic surfaces are covered by the surface-active substances, thus reducing the hydrophobic properties of the media or barrier layers. In addition, surface-active substances considerably alter the emulsifying behaviour of water in diesel. The energy input of the fuel pumps and/or components of the fuel system (e.g. ejector) create a considerably more stable emulsion since very fine droplets are created and these cannot coalesce into larger droplets due to the droplet surface being covered by surface-active substances. In two-stage water separators, the arrangement of the filtration-relevant filter media layers has a detrimental effect. The barrier layer used for water separation is located on the dirty side of the system, i.e. with increasing particle content, the hydrophobic layer alters and partially loses its hydrophoby. The degree of water separation decreases with increasing particle content and/or operation of the fuel filter, meaning that the system components are no longer sufficiently protected.

DROPLET FORMATION AND FURTHER INFLUENCING PARAMETERS

Water droplets form in the diesel fuel circuits of modern engines through partially high energy inputs into the water-diesel system, principally in the low-pressure and high-pressure fuel pump as well as through the abrupt depressurisation of the

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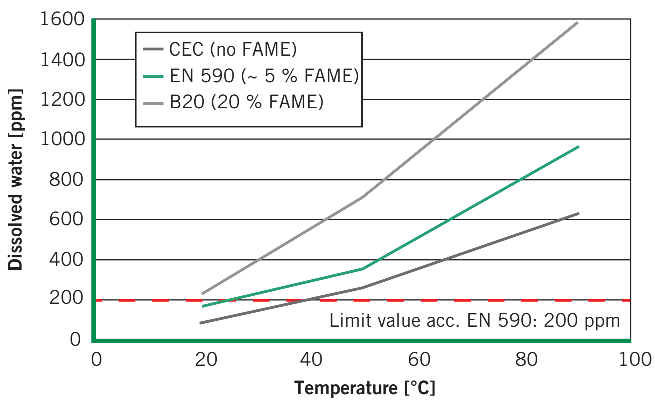
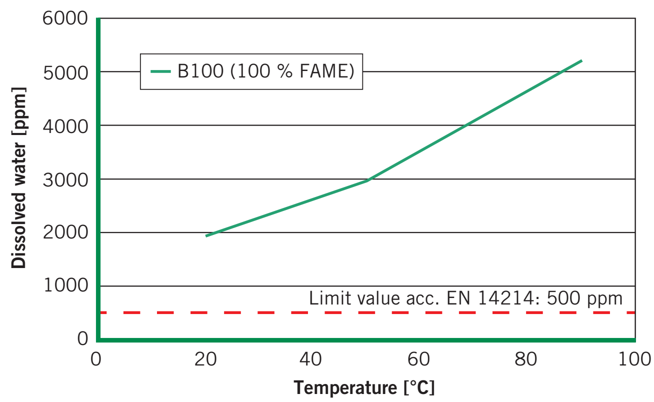


❷ Solubility of water in different diesel-biodiesel blends depending on temperature



❸ Solubility of water in 100% biodiesel depending on temperature mixture in the fuel return during transition from the high-pressure to the low-pressure area. The high circulation rates of the fuel for cooling the components require an energy input into the water-diesel system [4] that repeats periodically and is therefore efficient for the purposes of emulsion formation.

The sizes and size spectrum of the water droplets formed are influenced by the various parameters. The most important material parameter is the interfacial tension, which characterises the energy content of the water/diesel interphase and thus the thermodynamic stability of the emulsion. A low interfacial tension with a similarly low surface energy tends to produce finer water droplets. As the biodiesel content increases, the interfacial tension reduces from approx. 40 mN/m, with pure crude-oil-based diesel, to as little as 5 mN/m.

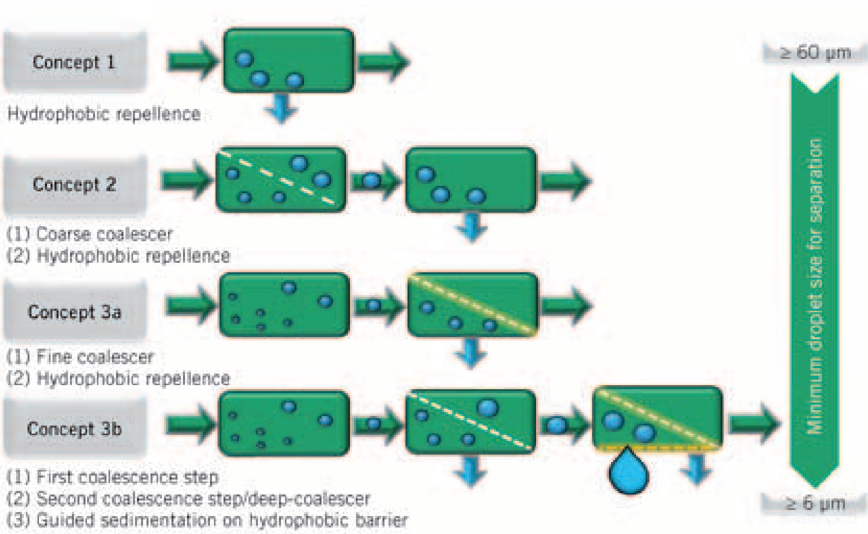
❹ Filter system with water reservoir

DESIGN-RELEVANT PARAMETERS

In the low-pressure circuit, intake systems have been almost universally replaced by pressure systems with electric fuel pumps (EFP). Due to the necessary lower energy consumption, with secondary systems as well, these pressure pumps are increasingly operated on demand. This results, on average, in a lower feed volume flow. This improves the parameters for water separation since lower flow speeds facilitate higher separation efficiency.

In the complete filter, the flow cross-sections must be structured so that no areas of local high fluid velocities appear. Such throttle points cause additional pressure loss and, above all, result in a crushing of the emulsified water drops in the areas with higher shear forces. Since the water droplets enlarged in the coalescer are to be separated from the flow via sedimentation, a lower fluid velocity also supports the physical process here.

Due to their higher density, the water droplets enlarged in the coalescer sediment in the direction of gravity and must be collected in a separate volume, ❹. The water is this area is purified as adhesive diesel droplets separated from the water.



❺ Different concepts for water separation

The water reservoir must be flow-restricted to avoid the risk of water drops being torn out of the phase interface between water and diesel and entering into the injection system with the flow. In modern systems, the water reservoir is normally fitted with a water level sensor that initiates servicing via the on-board electronics when the water level reaches the maximum.

WATER SEPARATION – NEW ROUTES

Based upon the conditions described, it is possible to develop adapted water separation concepts that take into account the current diesel compositions with increased biodiesel content. ❺ shows three basic concepts for water separation with filter media. With new fuel blends, the effectiveness of concepts 1 and 2 is significantly limited. That is why Mann + Hummel has developed concept 3 to meet current requirements for water separation. A very fine filter medium is required, initially to separate very fine emulsified water droplets from the diesel and then to coalesce these to larger droplets. These droplets are enlarged to even larger drops via additional, downstream coalescer layers. The design and resulting conduction of the flow is optimally selected to discharge the large drops from the fuel via gravity. Where necessary, a hydrophobic barrier layer is fitted in the form of a screen, to act as a 'last-chance' filter.

The newly developed water separation efficiency concept has a major influence on future filter sizes and designs and directly influences the filter change interval. Concept 3 in ⑤ is currently being implemented in prototypes and validated via field tests. ❻ shows a cross-section of a corresponding prototype.

TEST STANDARDS FOR WATER-DIESEL SEPARATORS

The separation capacity of water-diesel separators is evaluated based upon national and international standards. The most significant differences are in the droplet generation and size distribution, the proposed fuels, the test design and the water concentration and volume used during testing. ❼ compares the most common standards [4], whose fundamental weakness lies in the fact that either the practice-relevance or reproducibility of the results must be critically evaluated.

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❻ Prototype of a multi-stage water separator for finely dispersed water-in-diesel emulsions

An important parameter for practical evaluation of separators is the drop size distribution of the water droplets. This determines the stability of the emulsion formed and the separating behaviour at the coalescence elements. Measurements taken from vehicles with different fuels show that the average-sized droplets for practice-relevant laboratory tests in area $d_{3,50}$ should be within 4 to 10 µm [4]. Measurements based upon standards that make no specifications in this regard are not capable of creating reproducible results. One method of reliably creating defined drop size distributions is an orifice system in which the water is dispersed in a defined shear and elongation field. The properties of the emulsion are influenced by the nature and concentration of the surface-active substances present. During a test sequence, therefore, the composition of the fuel must not vary significantly, e.g. via extraction of these surface-active substances in the water phase. Slipstream procedures, as proposed in SAE 1488, do not fulfil this requirement.

The test fuel used must be defined with regard to its interfacial properties, characterised by the interfacial tension and emulsion stability. EN 590 diesels with approximately 7 % Fame content have a very low interface tension with water of approx. 8 to 15 mN/m. Biodiesel-free diesels lie within a range of 30 to 40 mN/m and are significantly less critical for separating systems with regard to emulsion stability and the size of droplets created.

A further aspect is the design of the test circuit. In a vehicle, an excess of fuel along with water (where present) is delivered into the circuit and only a small proportion is fed through for combustion (multipass). For practice-relevant tests, it is therefore sensible to replicate this principle in the design.

SUMMARY AND OUTLOOK

Highly-developed, modern fuel injection systems require fuels with a very low free water content. Fuel composition has altered significantly in recent years. Of particular significance for water separation is the increased proportion of biogenic components of up to 20 % and the altered additive composition associated with the reduction of sulphur content. This results in a dramatically altered interaction of water and diesel; in particular, the size of the water droplets distributed in the diesel has reduced to less than 10 µm. This creates very stable emulsions that can no longer be separated with existing separating systems. Mann + Hummel has taken up this challenge and is developing new separating concepts capable of achieving high water separation efficiency reliably and consistently under real operating conditions. The test standards for the qualification of water-diesel separators must also be adapted to the altered parameters. Currently valid standards are no longer in a position to reflect real results. The findings compiled by Mann + Hummel contribute both towards standards and directly towards innovative customer projects for water-diesel separation. These systems are designed so that particles are initially separated, and subsequently the droplets are enlarged in a graded, hydrophilic coalescer so that they can be discharged from the fuel flow. A hydrophobic sieve cloth represents the final barrier for any residual water droplets via three filtration stages. The separating concept presented can be integrated into existing installation spaces despite the complex design.

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Parameter \ Standard	Surface vehicle recommended practice: emulsified water/fuel separation test procedure SAE J1488,OCT-2010	Road vehicles-filter for diesel engines ISO 4020 6.5, 09-1982	Diesel engines – fuel filters – method for evaluating fuel/water separation effiicency ISO/TS 16332, 09-2006
Droplet size	3,500 rpm centrifugal pump ITT gould model 1ST 1E5D4	Not specified in detail, implicit specification via usage of a "Diaphragm pump acc. Annex E.1"	60 µm for pressure side application or 300 µm for suction side application; in alternative customer fuel pump
Water addition	Defined position, continuous	Defined position, continuous	Defined position, continuous
Water flow rate	0.25 % of fuel flow rate	2 % of fuel flow rate	0.15 % of fuel flow rate, optional 2 %
Determination of free water	Karl-Fischer titration	Volumetric/centrifugal method or Karl-Fischer titration	Karl-Fischer titration
Fuel quality and preparation	Diesel, biodiesel; (treatment possible)	Diesel; fuller earth treatment; IFT setting with "additive"	CEC reference diesel saturated; treated as necessary
Temperature test fuel	26.6 ± 2.5 °C	23 ± 5 °C	23 ± 2 °C (inlet filter)
Test duration (sampling time)	150 min (every 20 min DS; every 40 min US)	60 min (every 5 min)	90 min (every 10 min DS; at start and end US)
Sampling procedure	Upstream and downstream sampling with detailed sample preparation	Downstream sampling with detailed sample preparation	Upstream and downstream sampling with detailed sample preparation

❼ Comparison of different test standards for the qualification of water-diesel separators [4]

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CHANCES AND CHALLENGES OF THE ADMIXTURE OF ETHANOL TO DIESEL FUEL

The growing ratio of diesel engines being used not only in the transport sector but also for private transport in Europe leads to a significant increase in the demand of diesel – and a rising surplus of gasoline. By partial use of E85 – initially intended as a substitute for gasoline fuel – in diesel engines, it might be possible to counteract this problem effectively. In light of this, there were first investigations at FEV involving different blends of diesel and E85 in diesel engines.

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POTENTIAL OF BIO COMPONENTS

Since fossil fuels resources are limited and there is an increased sensitivity when it comes to fossil CO_2 emissions, the admixture of biomass-based fuels was stipulated in an EU guideline. For diesel engines, vegetable oil-based methyl ester has become established as a typical bio component; for spark ignition engines it is alcohol based ethanol. However, since admixing of ethanol to gasoline is also increasing, the available surplus of conventionally refined product in this segment of the fuel market is becoming even higher. Previous experimental investigations [1] show that considerable amounts of gasoline-like fuels can be added to diesel fuel, thus rendering the admixture of ethanol to diesel fuel a possible solution [2]. First results show that, from engine's perspective, it is possible to add up to 30 Vol.-% of E85 to diesel fuel in the entire characteristics map without any major adaptations to the engine functions that are relevant for combustion.

TEST OBJECTS AND BOUNDARY CONDITIONS

The HECS combustion system used for investigations on engines with a cylinder displacement of 0.39 l was designed to consistently feature lowest engine-out emission levels and at the same time high fuel efficiency. A compression ratio of 15:1 was selected in order to be able to robustly represent acceptable peak pressures in spite of the increased charge density. The combustion system reached a displacement-specific output of at least 80 kW/l at maximum combustion peak pressures of 190 bar. A common rail system with a maximum injection pressure of 2000 bar was used as injection system. To optimize cylinder filling and flow characteristics with regard to efficiency, one intake port was designed as a filling port, the second one as a classical tangential port. The aligned generation of charge movement was effectively supported by the seat swirl at both intake valves. The combustion chamber geometry had a conventional Ω recessed shape, which was further optimized together with the adjusted nozzle geometry (eight-hole, ks = 1.5) in order to achieve the best possible air utilization in every operation point. Reducing the compression ratio, using a higher maximum cylinder and injection pressure as well as improved exhaust gas recirculation cooling resulted in the lowest possible particulate emissions, so that the engine meets the Euro 6 standard without using any active $DeNO_x$ measures. Additional detailed information on the HECS combustion system can be found in various other publications [3, 4]. Essential characteristic data are compiled in ❶.

In addition to the engine tests, experimental basic investigations were conducted on a high pressure injection chamber with continuous flow under stationary boundary conditions. The distance between the injector nozzle and the tip of the glow pin which is relevant for combustion in the engine was created with the help of a specifically designed injector holder. As part of the preliminary investigations, we determined that we would use a ceramic glow pin without any sleeve and a distance of approximately 2 mm between the jet axis and the glow pin surface as basic configuration. To find successful glow ignitions, the light emitted by the flame with a wide bandwidth was visualized with the help of a high speed camera.

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DIESEL ENGINE	
CYLINDER DISPLACEMENT [cm³]	390
STROKE [mm]	88.3
BORE [mm]	75
COMPRESSION RATIO [-]	15
VALVES/CYLINDER [-]	4
MAX. CYLINDER PRESSURE [bar]	220
INJECTION SYSTEM [-]	Piezo CR System (Bosch)
MAX. INJECTION PRESSURE [bar]	2000
BOOST PRESSURE [-]	max. 3.8 bar abs.

❶ Characteristic data of the HECS combustion system

PROPERTIES	100 % DIESEL	100 % E85	Vol.-% E85 : Vol.-% DIESEL			
			10 : 90	20 : 80	30 : 70	40 : 60
E85 CONTENT [% v/v]	0	100	10	20	30	40
DENSITY (15 °C) [g/l]	839.0	784.6	833.6	828.1	822.7	817.2
LOWER CALORIFIC VALUE [MJ/kg]	42.91	28.99	41.60	40.28	38.93	37.57
CETANE NUMBER [-]	56.0	~10	52.4	39.0	26.9	17.6
CARBON CONTENT [% m/m]	85.9	56.67	83.2	80.4	77.6	74.7
HYDROGEN CONTENT [% m/m]	13.3	13.2	13.3	13.3	13.3	13.3
OXYGEN CONTENT [% m/m]	0.8	30.13	3.5	6.3	9.2	12.0
AROMATIC CONTENT [% m/m]	24.0	5.0	22.1	20.3	18.4	16.5
H/C RATIO [-]	0.15	0.23	0.16	0.17	0.17	0.18

❷ Properties of the investigated fuels

INVESTIGATED FUELS AND ENGINE CALIBRATION

To determine the potential and the challenges of fuels containing ethanol in diesel engines, different kinds of mixtures of E85 and conventional EN590 diesel fuel were analyzed visually and in the engine. In contrast to ethanol, E85 is not hygroscopic and readily available. Another advantage is that the mixture is stable, which makes it possible to eliminate emulsifiers. The properties of the investigated fuel mixtures can be found in ❷. During the engine tests, high blend rates with up to 40 Vol.-% E85 were used, while even higher admixtures were analyzed in the high pressure chamber. With increasing ethanol content, the cetane number decreased and the oxygen content increased. In compliance with future diesel requirements, the H/C ratio rises with higher E85 blend rates, since the aromatics content is reduced.

With the applied combustion system, it is possible in particular to successfully compensate the impact that the longer ignition delay of fuels containing ethanol has on combustion (Closed Loop Combustion Control). A constant center of combustion – with the help of adjusted injection timings – permits the efficient combustion of fuels with low cetane numbers. As a result, deviating emission behavior can be directly attributed to the fuel-specification and is not a consequence of a not optimally adjusted course of combustion. Other operating parameters, such as boost pressure, injection pressure, and charge air temperature, were optimized with regard to engine load during earlier investigations by taking realistic boundary conditions for achieving the Euro 6 standard [5] into consideration. The fuel investigations were carried out in four load points, three of which are in the NEDC range for a flywheel mass class of 1590 kg. ❸ shows the relevant calibrations for the different load points.

RESULTS SINGLE-CYLINDER

The influence of the rising E85 content on the compression-ignition combustion system is shown in ❹ for an Euro 6 nitrogen oxide level for different load points. As a result of the decreasing cetane number, the ignition delay increases continuously with the higher E85 content. Due to unfavorable conditions in the combustion chamber the differences at lower load points are considerably more pronounced here. Stable combustion is not achieved with an E85 admixture of 40 Vol.-% in the lowest load point; a combustion chamber pressure of 25 bar and a temperature of 750 K during injection are no longer sufficient for stable ignition. Because of a homogeneous fuel/air ratio that results from the longer ignition delay and the modified evaporation properties, particulate emissions can be significantly reduced with an increasing E85 ratio. With an increase of the oxygen content in the fuel, we can assume that there will be additional oxidation of the particulate emissions that are forming, thus leading to a nearly soot-free diesel operation in the entire NEDC characteristic map when using 30 % of E85. However, the disadvantages are increased noise, HC, and CO emissions,

	CENTER OF COMBUSTION °CA BTDC	PILOT OFFSET °CA	DURATION PILOT INJECTION µs	RAIL PRESSURE bar	BOOST PRESSURE bar	CHARGE AIR TEMPERATURE °C	EXHAUST GAS BACK PRESSURE bar
n = 1500 rpm, IMEP = 4.3 bar	-6.6 at 0.5 g/kWh $ISNO_x$	10	180	720	1.07	25	1.13
n = 1500 rpm, IMEP = 6.8 bar	-5.8 at 0.5 g/kWh $ISNO_x$	11	140	900	1.50	30	1.60
n = 2280 rpm, IMEP = 9.4 bar	-9.2 at 0.5 g/kWh $ISNO_x$	20	120	1400	2.29	35	2.39
n = 2400 rpm, IMEP = 14.8 bar	-10.8 at 1.0 g/kWh $ISNO_x$	28	120	1800	2.60	45	2.80

❸ Calibration for the different load points

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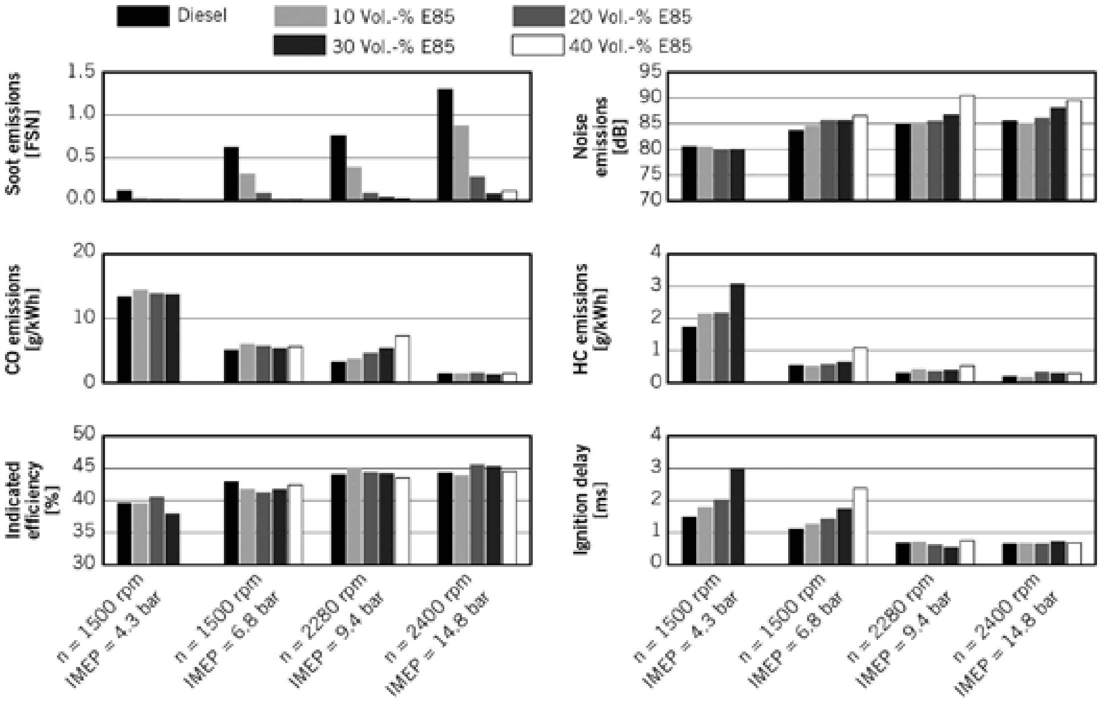

❹ Comparison of emission results in a single-cylinder research engine, with pilot injection and constant center of combustion, Euro 6 NO_x-level

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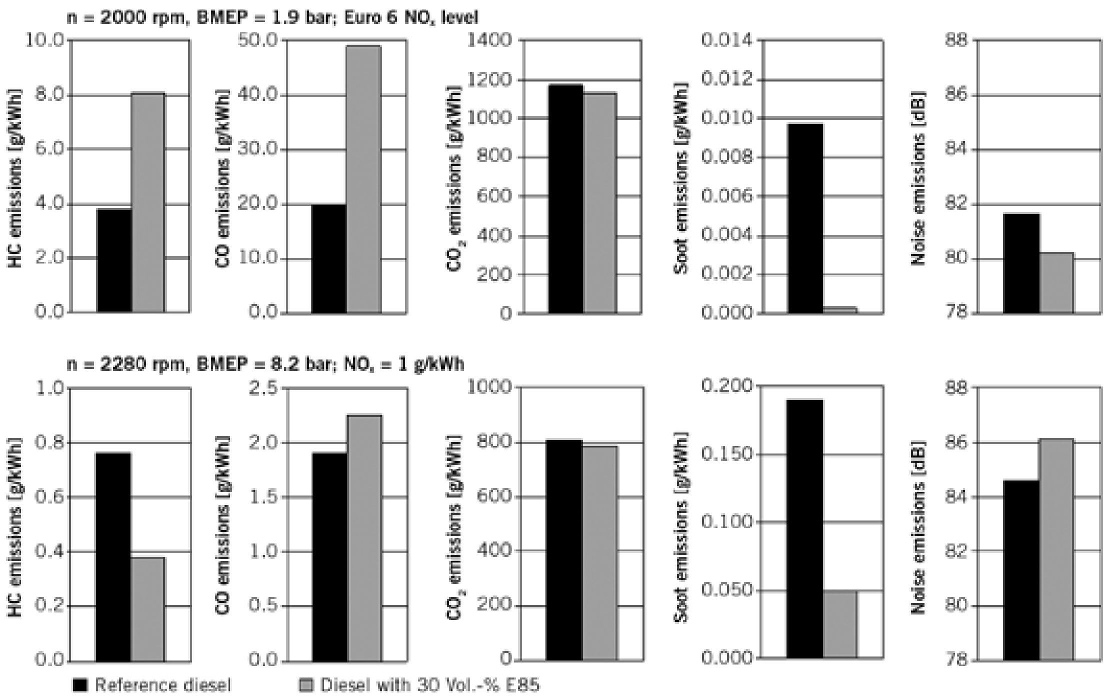
but the advantageous efficiency of the diesel engine's combustion remains even at high ethanol contents.

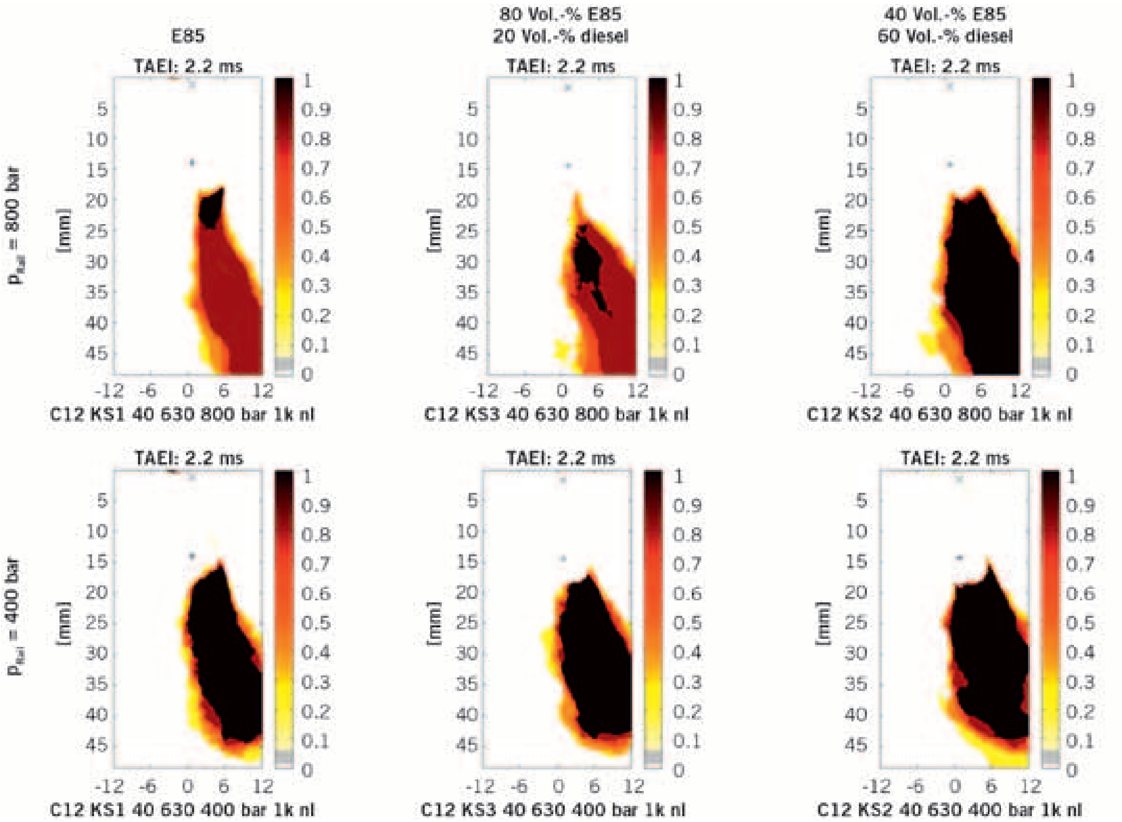
The results show that an E85 admixture of 30 % to diesel fuel is possible without any hardware changes. Additional measures are yet required to further increase the E85 content.

MULTI-CYLINDER ENGINE

The following investigations with ethanol portions on the multi-cylinder engine generally confirm the results obtained from the single-cylinder engines. As you can see in ❺, the particulate emissions for operation at low loads decrease to a minimum level when using 30 Vol.-% of E85 as compared to pure diesel fuel. At the same time, however, HC and CO emissions increase significantly. The combustion noise slightly decreases as a result of the very highly homogenized combustion. By contrast, the degree of fuel/air mixture during operation with higher loads is con-



❺ Comparison of emission results in a multi-cylinder engine, with pilot injection and constant center of combustion



❻ Probability distribution of light emitted by the flame for experiments carried out with E85 and E85 diesel mixtures at injection pressures of 800 bar and 400 bar

siderably lower. Thus, increasing the premixed portion during combustion leads to a distinct increase in combustion noise. The behavior of particulate emissions is the same as during operation with low loads, whereas HC emissions even tend to decrease.

FUNDAMENTAL INVESTIGATIONS IN THE HIGH PRESSURE COMBUSTION CHAMBER

Innovative glow ignition concepts using a continuously operated glow plug offer an option for burning admixtures with a high content of E85 in diesel engines. Former researches, which have been conducted at the institute for combustion engines, prove the high potential of glow ignitions with ethanol [6, 7]. The challenge of glow ignition with ethanol containing fuels is the surface ignition temperature of 1080 K, which is increased by 170 K compared to methanol. To analyze the parameters influencing the glow ignition stability of blended fuels containing E85, experiments were performed on a high pressure injection chamber with stationary flow. ❻ shows the probability distributions of light emitted by the flame 2.2 ms after actuating the injector during experiments conducted with E85 and mixtures of E85-diesel at injection pressures of 800 and 400 bar. According to the color chart shown on the side, black pixels represent a 100 % and white pixels a 0 % probability of light emitted by the flame at the corresponding point. The location of the injector nozzle is marked with an x, the location of the glow pin tip with a + sign. In the upper row of ⑥, the probability distributions of light emitted by the flame are shown for experiments carried out with an injection pressure of 800 bar. We can see that the glow ignition stability and reproducibility of the flame propagation increases with rising diesel fuel content and a stable flame propagation can only be ensured once 60 % by volume of diesel is admixed in all cycles.

In the lower row of ⑥, the probability distributions are shown for experiments carried out with an injection pressure of 400 bar. Compared to the probability distributions with an injection pressure of 800 bar it can be seen that, with an injection pressure of 400 bar, the various fuel compositions have no significant influence on the ignition glow stability and a reproducible flame propagation can be realized with all investigated fuels. This means that low injection pressure is one of the major influencing parameters for improving the ignition glow stability of fuels containing E85.

OUTLOOK

An admixture of E85 of more than 30 Vol.-% in low load points leads to insufficient ignition conditions, without support of additional measures. The long-term goal is therefore the implementation of surface ignition during engine operation. Using the example of the adding 40 Vol.-% of E85, glow ignition was therefore investigated for the lowest operating point of n = 1500 rpm and IMEP = 4.3 bar in the engine. As shown in ❼, the IMEP standard deviation can be lowered continuously with a decrease in injection pressure. Only a rail pressure below 300 bar will lead to a clearly less stable combustion. Furthermore, HC as well as the CO emissions can be reduced with decreasing injection pressure while at the same time increasing efficiency. All these effects con-

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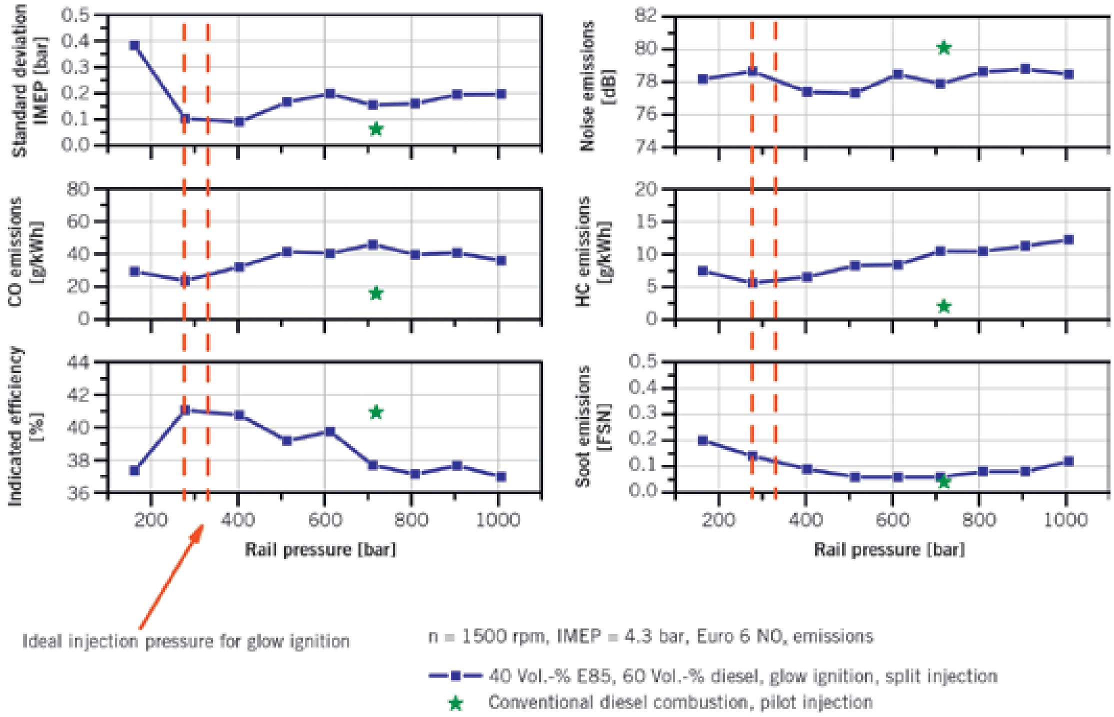

❼ Application of glow ignition with 40 Vol.-% E85 in the single cylinder engine

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firm the higher stability of glow ignition with decreasing rail pressure. The results from the high pressure chamber can therefore be valuably transferred to diesel engine applications.

The influence of the changed fuel properties on the fuel system was not taken into consideration during these investigations. What needs to be mentioned are the poorer lubrication properties as well as the reduced flash point, making further investigations into this matter necessary.

THANKS

The investigations in the high pressure chamber and on the single-cylinder as well as multi-cylinder engine were carried out as part of the Beauty project (Bio-Ethanol Engine for Advanced Urban Transport by Light Commercial & Heavy-Duty Captive Fleets), sponsored by the EU as part of the "Seventh Framework Programme".

CONCLUSION

The investigations show that E85 can also be used as a partial substitute for diesel under the aspect of efficient and low-emission combustion. A 30 Vol.-% admixture of E85 to conventional diesel fuel is possible without any major hardware changes in the entire characteristic map area. The results show that particulate emissions can be almost completely avoided with this blend, maintaining a constantly high engine efficiency. A disadvantage is the higher noise emissions. The E85 portion can be further increased by employing innovative glow ignition concepts. The investigations show that lowering the injection pressure will lead to more stable ignition conditions.

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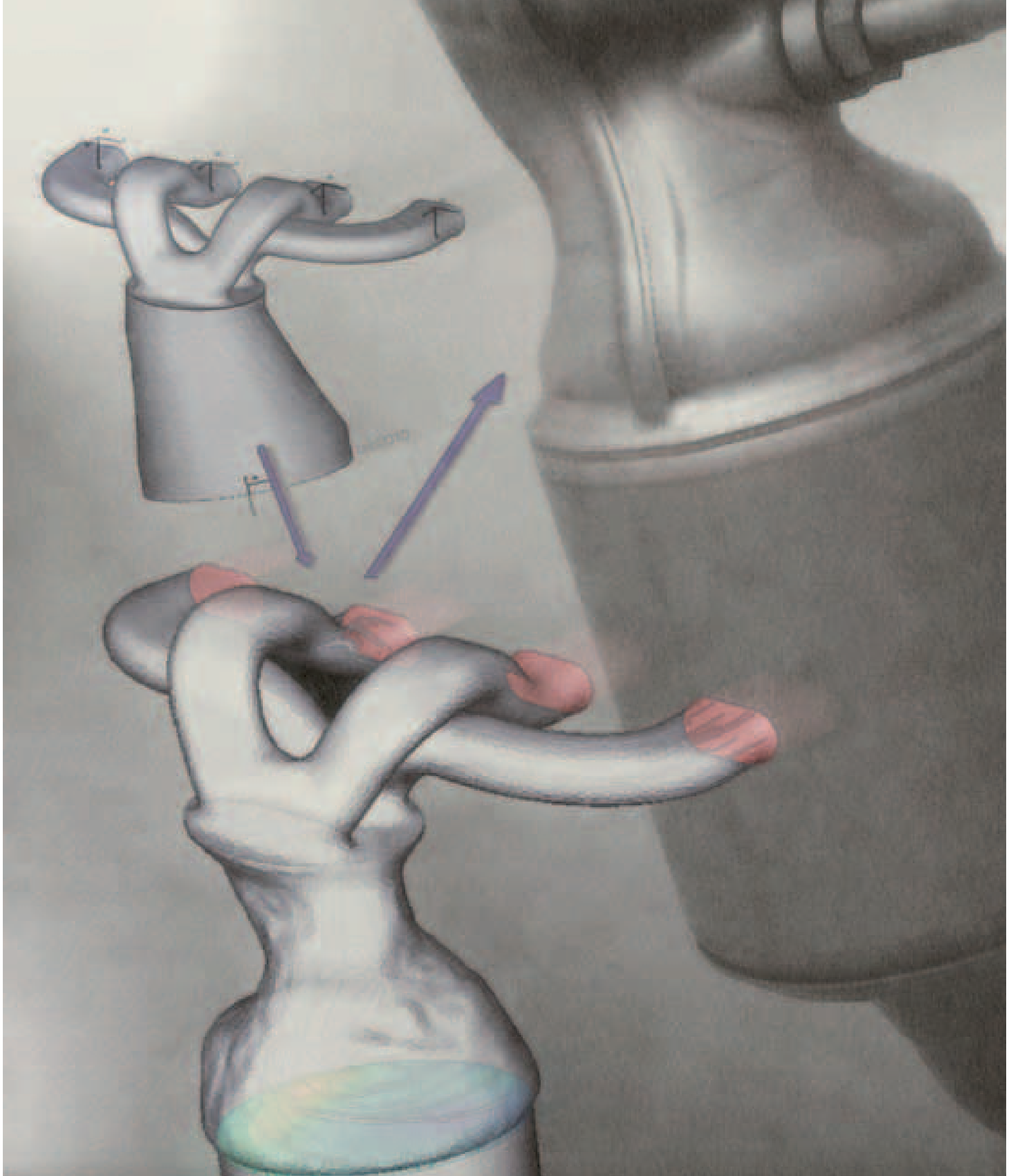

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AUTOMATIC SHAPE OPTIMIZATION OF EXHAUST SYSTEMS

Maximizing the uniformity of the catalyst flow and simultaneously minimizing the backpressure is an essential goal of exhaust system development, which is often complicated by restrictive underhood packaging spaces. A new optimization tool has thus been developed at Faurecia Emissions Control Technologies (FECT) that can automatically determine a flow-optimized draft design for a particular packaging space.

INITIAL SITUATION

Optimizing the flow uniformity of the catalyst flow and the backpressure of exhaust systems normally requires numerous design loops – with iterations between CAD and CFD departments that can be very time consuming. This is particularly the case when optimizing a manifold and close-coupled-catalyst combination. During exhaust blowdown, the gas exiting the engine cylinder can momentarily approach sonic flow, which presents particular difficulties to manage and to avoid undesirable flow separation. A further challenge in the optimization of exhaust systems is posed by the extremely restrictive packaging spaces, coupled with durability, manufacturability and cost considerations.

As an example of conventional optimization, a close-coupled-catalyst and three-in-one manifold from twelve-cylinder engine is shown in ❶. The catalyst position is fixed (based on the packaging space) as is the manifold geometry (based on an existing part). Thus, the only permitted changes were to the inlet pipe/cone region itself. Starting from the original design, ① (a) – which exhibited highly non-uniform flow in the catalyst cross-section – several successive design changes were taken to determine an optimized solution, ① (b). The final design achieved a nearly uniform catalyst flow, which is required for an efficient conversion rate. This example illustrates how relatively minor geometrical changes can result in significant functional improvements.

The continually reduced development times have further increased the demand for an automatic optimization program that is based on the fluid dynamics simulation. Starting from the available packaging space, an optimal solution should be found in a reasonable period of time. Using shape-based optimizing is impractical for several reasons. An obvious disadvantage is the parameterization itself, which may be possible for simple geometries (e.g., pipes) but which is generally non-trivial for the complex free-form geometries used for catalyst inlet cones. Since the flow fields are determined before the shape parameters are adjusted, the optimization is inherently sequentially, which leads to unacceptably long computational times. In contrast, the adjoint method tool presented here has extremely modest calculation times and has the notable advantage that both a painstaking geometry parameterization and a stepwise adjustment of the meshed volume become superfluous. Instead, only the packaging space needs to be supplied as a meshed volume, ❷.

MODELLING

The optimization program Cago (Continuous Adjoint Geometry Optimization) developed at Faurecia is based on the continuous adjoint CFD method from Othmer et al. [1] implemented in Openfoam [2]. The calculation method used in Cago is only briefly described here, with details to be found in [3], while the theoretical basis of the adjoint process is described by Othmer in [4].

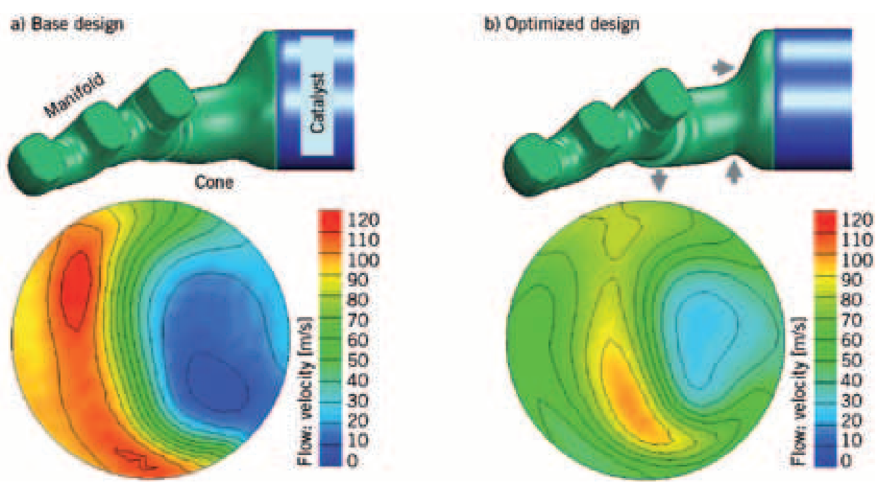
For the purposes of the automatic geometry optimization, the turbulent flow in the exhaust system is treated as being incompressible and a standard high-Re k-ε turbulence model [5] is used. The geometry itself is described using an immersed boundary method. As shown schematically in ② (b), the meshed region is characterized by flow (fluid) and non-flow (solid) sub-regions. Since the volume mesh itself only provides a stepwise representation, the geometry is based internally on a level-set-approximation, which yields an exacter and smoother geometry description. Using this approach greatly improves the overall computational efficiency since it precludes any need for mesh movement or remeshing.

COST FUNCTIONS

The optimization of the inlet cone geometry is expressed in terms of cost functions to be minimized. The primary goal is a high flow uniformity combined with a low backpressure. Correspondingly, the cost function includes the deviation of the current velocity profile from the target velocity profile as well as the energy loss between cone inlet and outlet. Adding extra weighting to the velocity deviation in the outer circumference of the catalyst allows the centricity of the catalyst flow to be included into the cost function.

ADJOINT CFD METHOD

The optimization method is based on idea that the sensitivity (i.e., gradient of the cost function) with respect to the geometry change can be calculated for every point on the surface geometry. It also accounts for the influence of the geometry changes on the flow field. Viewed mathematically, the fluid dynamics equations are treated as a constraint (via a Lagrange function) for the optimization problem. For this optimization method, the Lagrange multiplier is the so-called adjoint flow field. This adjoint flow field has no physical meaning, but is a mathematical construct that allows the sensitivity of the cost function with respect to local changes in the geometry to be calculated. The decisive advantage of this method is that no additional and costly flow solutions are required since the sensitivities can be calculated directly as a scalar product of the primary



❶ Closed coupled catalyst and a three-in-one manifold of a twelve-cylinder engine

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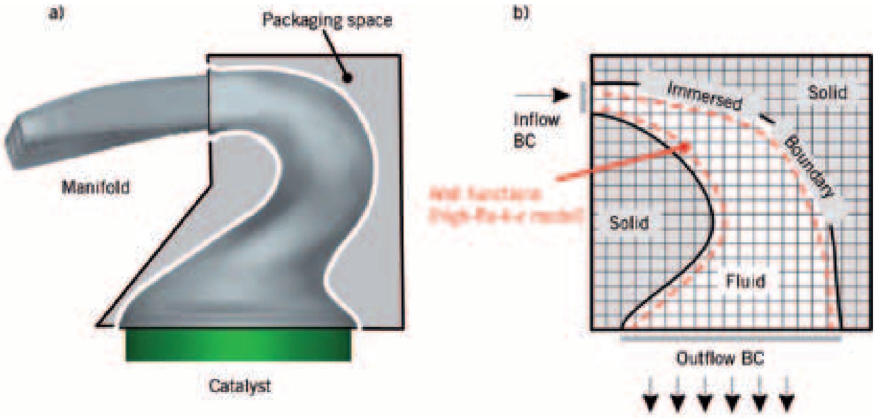
and adjoint flows. The simultaneous calculation of primary and adjoint flow fields, as well as the corresponding geometry adjustments, is associated with simulation times for the entire optimization process that is on the same order of magnitude as a single (normal) flow calculation.

APPLICATION

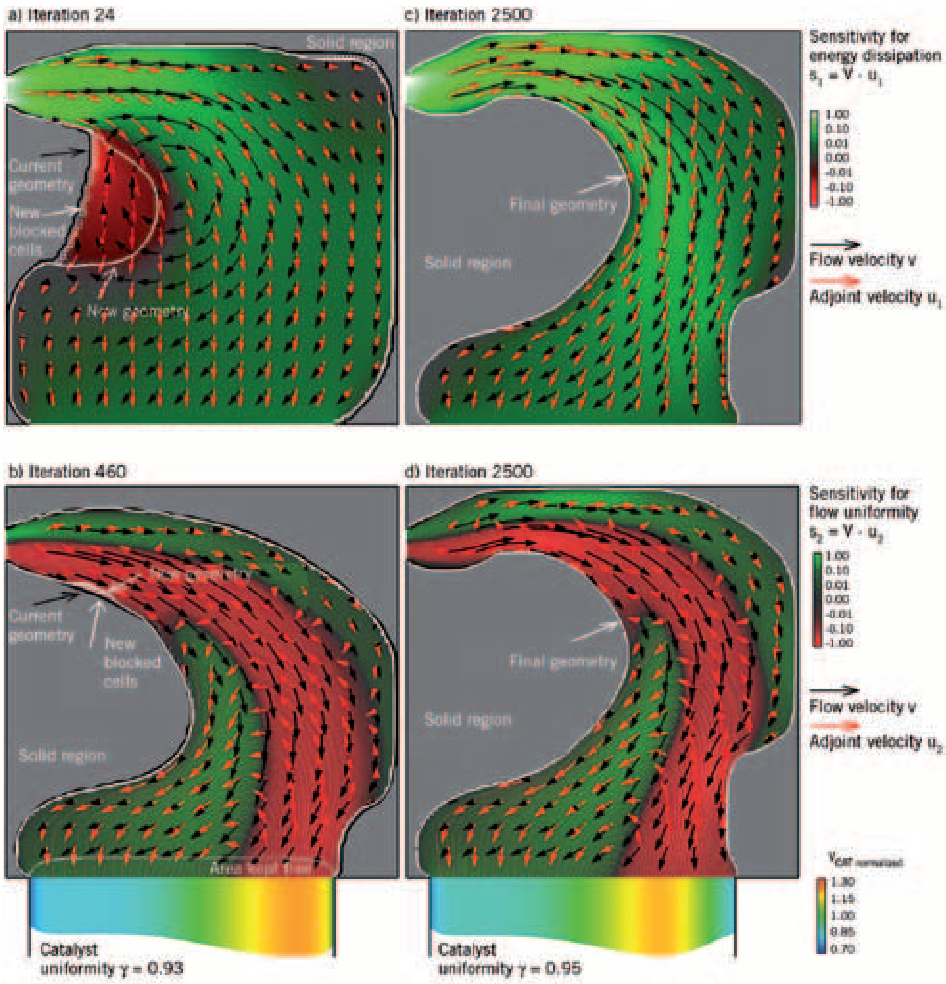
As an example of the optimization process, a simple packaging space with a transverse inflow, given schematically in ② (b), is taken. The development of an optimized geometry and the sensitivities calculated by the adjoint method are shown in ❸, with sensitivities for backpressure shown in ③ (a and c) and sensitivities for the flow uniformity shown in ③ (b and d). The red regions are negative sensitivities, which degrade the respective target function. It is thus possible to identify a recirculation zone after 24 iterations, ③ (a), that will cause backpressure and can therefore be successively blocked. The geometry after 460 iterations is shown in ③ (b) along with the instantaneous velocity profile at the front of the catalyst. The excessive velocities on the right side of the catalyst are reflected in the sensitivities for the uniformity. The negative sensitivities (red regions) define regions in which the introduction of additional obstructions would improve the catalyst flow uniformity. This leads to an indentation in the geometry (immediately after the inlet) that improves the uniformity. After further iterations, the geometry reaches a final form as shown in ③ (c and d). A comparison of the uniformity and backpressure sensitivities reveals the conflicting goals: a further indentation of the geometry would improve the uniformity but at the cost of higher backpressure.

The geometry optimization for a more complex packaging space is shown in ❹. The packaging space was meshed with 300,000 polyhedral cells. The optimization was complete after ten hours on a standard workstation (3.0 GHz Intel Core Duo CPU). With a slightly higher administrative effort, the optimization can also be run in parallel on a cluster, which reduces the calculation time to approximately one hour.

The optimization with Cago of a catalyst inlet cone for a naturally aspirated engine is illustrated in ❺. The inlet cone is the central geometric element for redistributing the flow from each of the cylinders. Since it is important that the flow from each cylinder satisfies the uniformity target, each cylinder is calculated individually as steady-state within Cago. For the automatic geometry optimization, the geometric sensitivities for backpressure and uniformity for all cylinders are combined, with the sensitivities for the cylinder with the poorest uniformity being weighted most heavily. This optimizes the



❷ Sketch of package space (a) and of CFD model (b)



❸ Sensitivities for backpressure in inlet cone (a, c) and for flow uniformity across the catalyst (b, d) after 24 iterations (a), after 460 iterations (b) and after 2500 iterations (c, d); final cone geometry after 2500 iterations (c, d)

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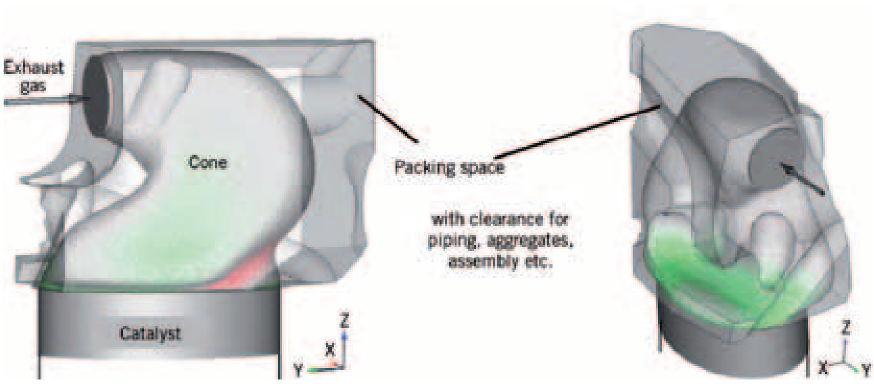
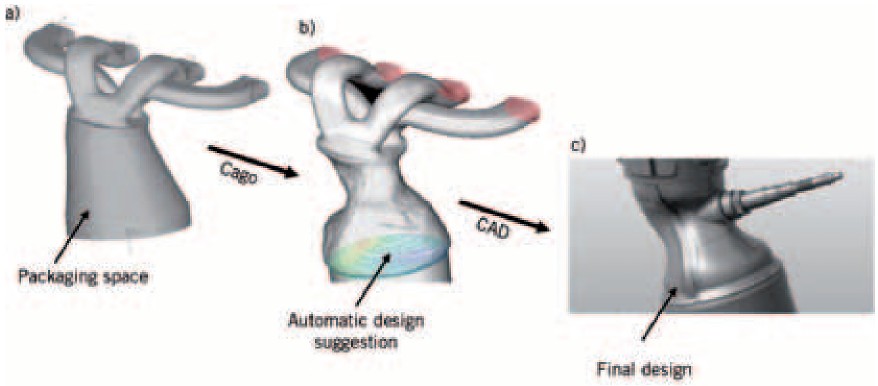
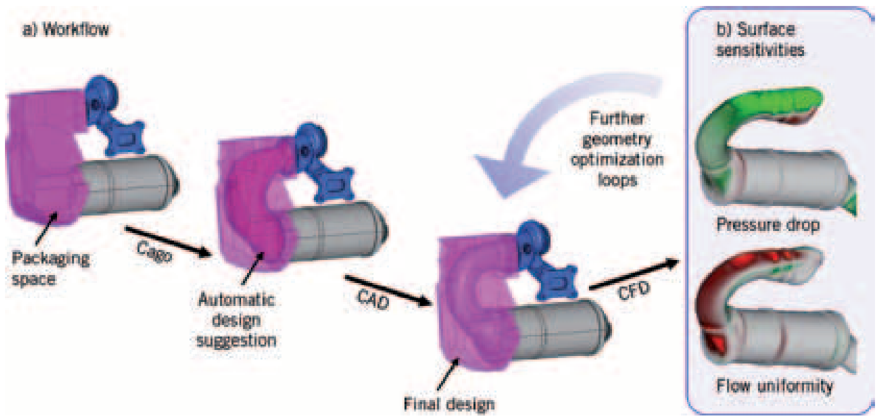


❹ Automatically generated inlet cone for the provided packaging space



❺ Catalyst inlet cone for four-cylinder naturally aspirated engine



❻ a) Workflow for catalyst cone development, b) surface sensitivities for subsequent geometry optimization

uniformity for individual cylinders as well as for the average flow.

WORKFLOW

The workflow for the development of a pipe/cone inlet geometry is shown schematically in ❻. The available packaging space is initially defined, meshed and an optimal draft design is calculated via Cago. The draft geometry is exported in IGES format, which can be read in a CAD system and serves as a guide during the design process, which also includes manufacturability and other design aspects. The final CAD model is subsequently verified with conventional (non-optimizing) CFD, whereby the adjoint flow equations for backpressure and uniformity are also calculated. The resulting surface sensitivities, ⑥ (b), highlight the geometry regions in which the cost function is particularly influenced. This information flows into the final design and is also used for the optimization of geometry details.

CLOSURE

An automatic geometry optimization method based on adjoint methods has been presented. The practical feasibility has been demonstrated for complex packaging spaces. Although the current treatment addressed catalyst systems exclusively, the optimization method can be used for a variety of applications. The calculated sensitivities of the cost function steer the automatic adjustment of the surface geometry and also provide a guideline for the CAD designer during the design process. The process is extremely efficient, thus the time for the entire optimization process remains on the same order of magnitude as a conventional CFD calculation. This allows the automatic geometry optimization to be an integral part of the product development process.

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FRICTION REDUCTION POTENTIALS IN CHAIN DRIVES

In recent years, designing chain drives in combustion engines has increasingly focused on reducing friction. By selecting better materials, using different production techniques and modifying the design of all chain-drive components, it has been possible at Iwis Motorsysteme to verify CO_2 reductions of up to 2 g/km many times over.

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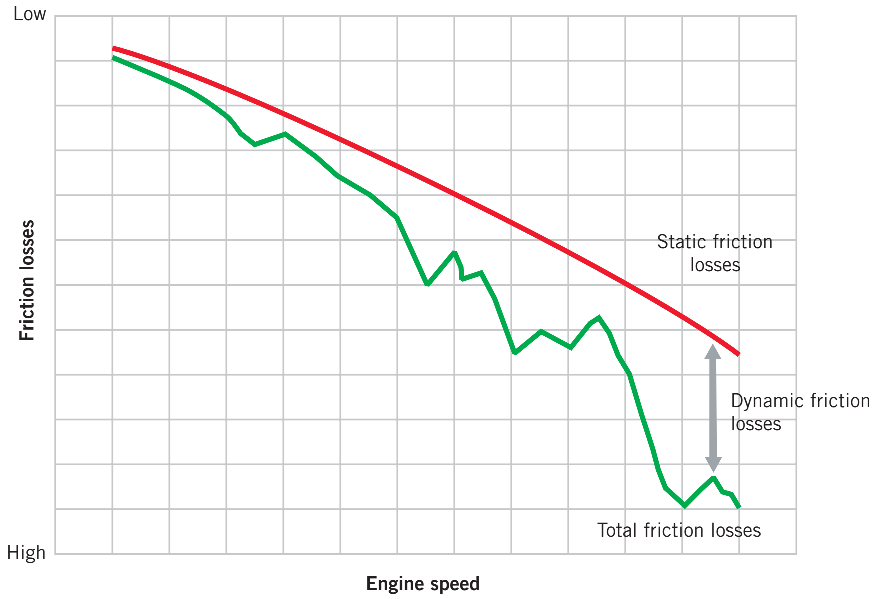


❶ Calculated friction losses

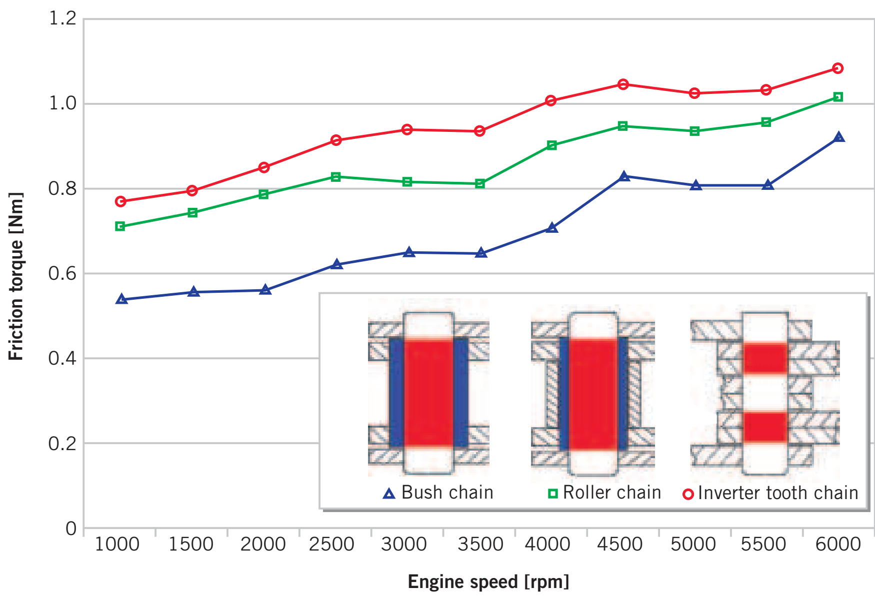
BASICS

It is first necessary to ascertain where exactly friction is produced in the chain drive and how these components contribute to overall friction. Computations and tests have revealed that most friction occurs at the point of contact between chain and rails as well as in the chain joints themselves, with only a minor percentage resulting from the chain meshing with sprocket toothing. The way in which friction can be influenced in the chain drive largely depends on whether friction losses are static or dynamic. A typical spread of these friction contributors is shown in ❶. The basic chain-drive layout, the chain type employed, the materials used for the friction linings on the rails as well as lubrication of the chain drive are the principal variables influencing static friction losses. Excitations from crank-train and valve train, chain-tensioner damping, the mass and stiffness ratios of chain-drive components as well as the phase angle of injection pumps and macro-geometry of the sprockets mainly influence the dynamic component of frictional losses.

FRICTION IN THE TIMING CHAIN

Friction inside the chain joints is extensively influenced by the tribological partners (pin link-plate in the case of the tooth chain and pin bush in the case of the roller or bush-type chain, and lubricant used) along with the type and magnitude of loading. ❷ shows principle-related differences in chain-joint geometry between the three chain designs – bush-type chain, roller-type chain and tooth chain. The clear system-based drawback can be seen for the tooth chain which, for the same design width, exhibits a joint surface relevant to wear that is 35 to 40 % smaller than for the other chain designs. Investigations on engines motored using the strip method reveal a clear frictional advantage for the bush-type chain whereas friction torque is as much as approximately. 30 % higher for the tooth chain, ②. This means the roller-type chain provides a well-balanced compromise between good tooth-chain acoustics on the one hand and the friction and wear-related advantages of the bush-type chain on the other. In these investigations, however, it is imperative to ensure that the chainline (i.e. chain run in the timing drive, including the radii used for the tensioning and guide elements) remains unchanged for the different chain types. For any variation in the rail geometries would have a direct effect on the normal forces between chain and guide element and consequently, of course, on the frictional forces – with an unchanged friction coefficient being assumed.

After defining the chain type with the most favorable frictional properties, the question arises as to whether friction can be reduced further still by changing the pin coating. Following intensive development activities at Iwis Motorsysteme, it was recently possible to introduce the innovative IC+ technology in mass production, reducing friction torque by 10 to 20 % over the standard process (IC technology). Both technologies are based on a 10 to 20 µm chrome-carbide coating that is applied to the martensitic pin in a complex process. Representing an advancement of the inchromizing process, IC+ technology optimizes the parameters relevant to friction and wear, ❸.



❷ Friction torque of different chain types (same pitch = 8 mm)

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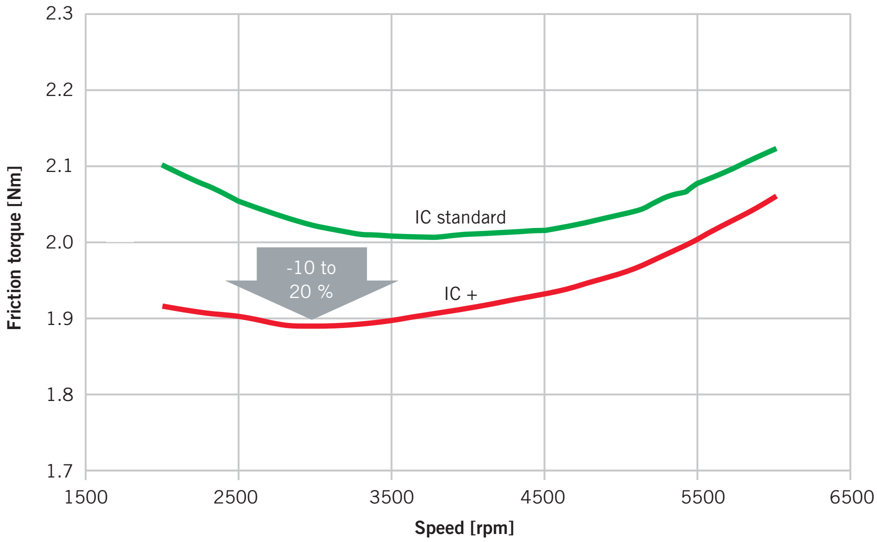
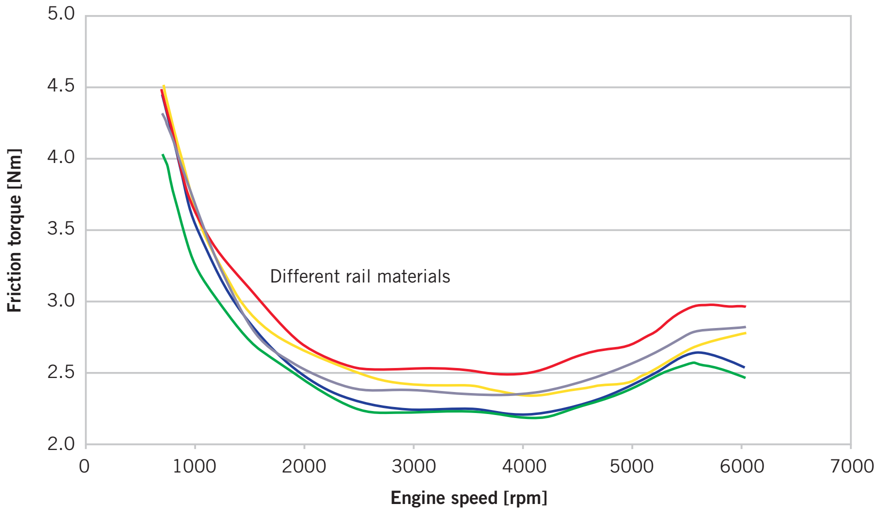
By combining this IC + technology with the fine blanking process for making chain-link plates – replacing the cutting-and-shaving process that is established on the market – it is possible to reduce frictional loss by approximately 55 W at 2000 rpm or by a level of approximately 145 W at 5000 rpm for a V-engine. Needless to say, Iwis Motorsysteme will be examining further pin coatings that increase this improvement even further while at the same time significantly enhancing the resistance of chains to wear (specifically in the engine's demanding lubrication environments).

FRICTION BETWEEN TIMING CHAIN AND RAIL

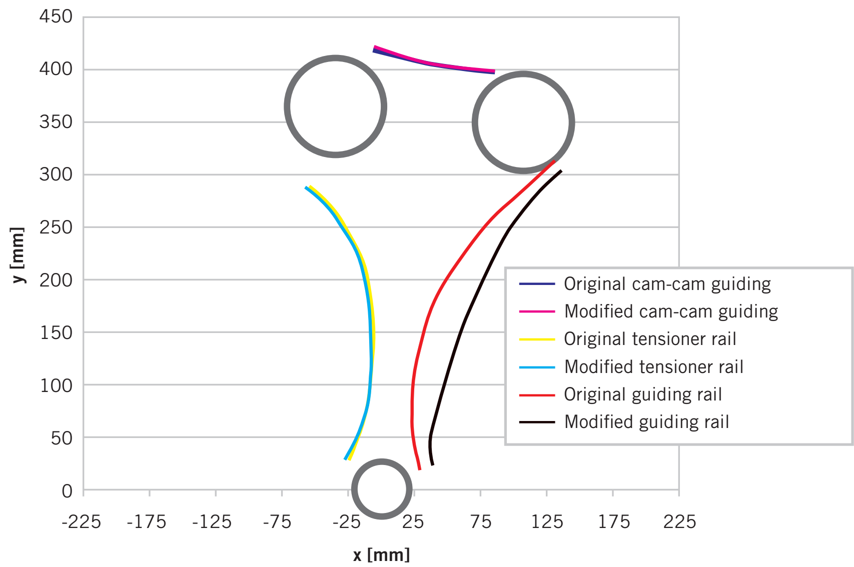
The second main component of chain-drive friction occurs at the point of contact between the timing chain and the guide elements (tensioning rail and guide rail). Low-cost polyamides (PA66 or PA46) injection-molded with a high surface quality have become established as materials for providing the contact surface on guide elements. Many different suppliers also use a variety of compounds that promise further improvements in reducing friction. This is why iwis motorsysteme set out to test the functional performance of the various material combinations found on the market and measure the differences in friction. For this purpose the set-up designed specifically for testing the components that were produced under real conditions using the injection-molding technique was used. Particular attention was paid to lubrication conditions, oil temperature and the introduction of force into the chain. Selecting the appropriate material is shown to be capable of reducing friction by as much as 10 % at the point of contact between chain and rail, ❹. Further-reaching studies into optimizing the structure of the rail surface are part of current research projects aimed at tapping further potentials by creating the ideal surface (recesses, grooves, segments etc.). In cooperation with a research institute, a plastic compound is also being developed that is formulated specifically for use in chain drives and takes into account the conditions prevailing in the combustion engine. The results of these two activities will be published in a separate report at the given time. However, initial findings suggest it will be possible to achieve further improvements.



❸ Measurement results



❹ Chain drive friction losses



❺ Optimization of guiding geometry

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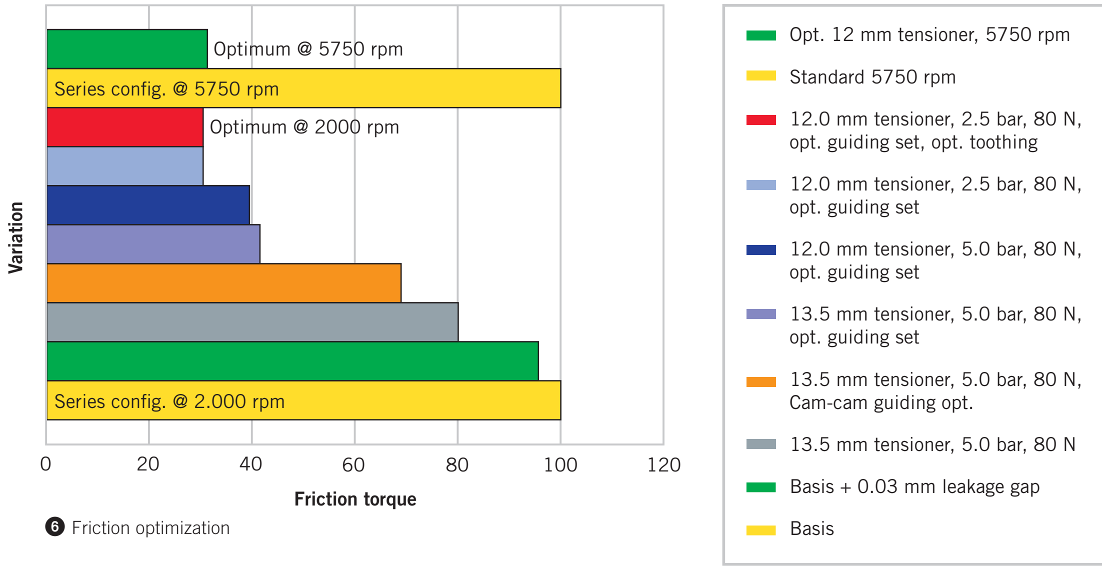


❻ Friction optimization

IMPLEMENTING DESIGN MEASURES TO REDUCE FRICTION

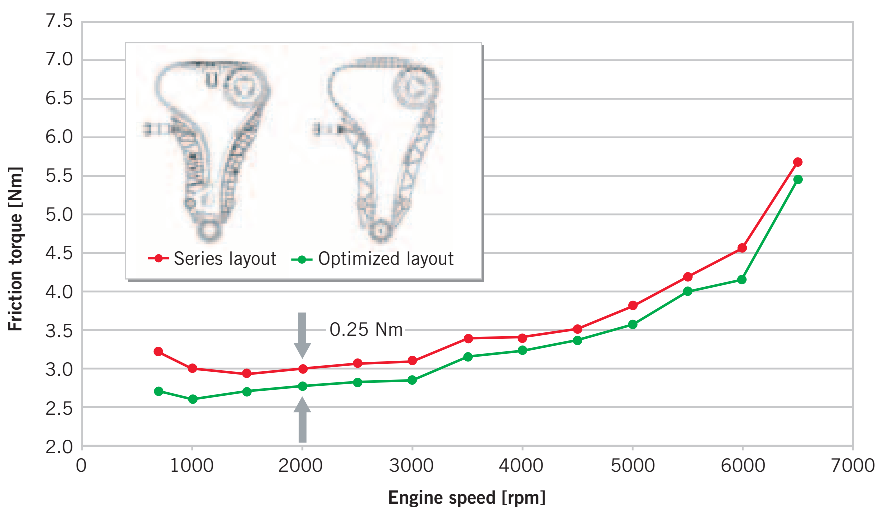
The designer can exercise by far the greatest influence on reducing friction in the chain drive by optimizing the chainline. Avoiding tightly curved tensioning and guide rails reduces normal force on them from the chain and, for design reasons, lowers the frictional forces that occur. Proceeding from a chain drive with tightly curved guide rail, ❺, – that has found its way into mass production on account of package restrictions – an optimized design was created and then assessed by simulation. This measure is shown to be capable of reducing friction loss by up to 70 % if it is also accompanied by adjustments to chain tensioner and tensioning-system oil supply. ❻ clearly shows that the main improvement is attributable to optimizing rail geometry (light blue bar compared to the orange bar), whereby optimizing the chain tensioner then opens up further potential. A system comparison between toothed-belt drive and chain drive published in October 2008 and showing the toothed belt to provide benefits in terms of reducing CO_2 consequently delivers no fundamental statement but merely contrasts an optimum toothed-belt drive layout with a chain-drive layout that involves compromises. Yet comparing the optimized chain drive, ❼, with the optimum toothed-belt drive fails to confirm the statements made and instead verifies functional advantages for the chain drive. The theoretical potentials were examined in tests by IAV GmbH in Chemnitz, demonstrating the influence optimizing guide-rail geometry has on friction. ⑦ reveals that straightening the guide rail permits a reduction in friction torque by 0.25 Nm that can be maintained across the engine-speed bandwidth at an oil temperature of 90 °C.

For this reason, it is extremely important to examine the effects on chain-drive friction while laying out the engine and defining the principal dimensions, and to incorporate the findings into the design as early as possible.

SUMMARY

Over recent years, Iwis Motorsysteme has constantly made improvements to all components of the chain drive as well as to the basic layout concept. By selecting appropriate materials and production methods for chain and guide elements as well as modifying the chain drives, friction loss can be cut by 500 to 1000 W which can reduce CO_2 emission by approximately 2 g/km. Close consultation between the engine designers on the one hand and Iwis Motorsysteme as a development partner for chain drives on the other is crucial in this context.

Further potentials were tapped by analyzing plastic materials as well as by optimizing the topography of the guide elements. Beyond this, innovative chain-pin coatings permit further improvements with regard to friction and wear.



❼ Measurement results (oil temperature main gallery: 90 °C)

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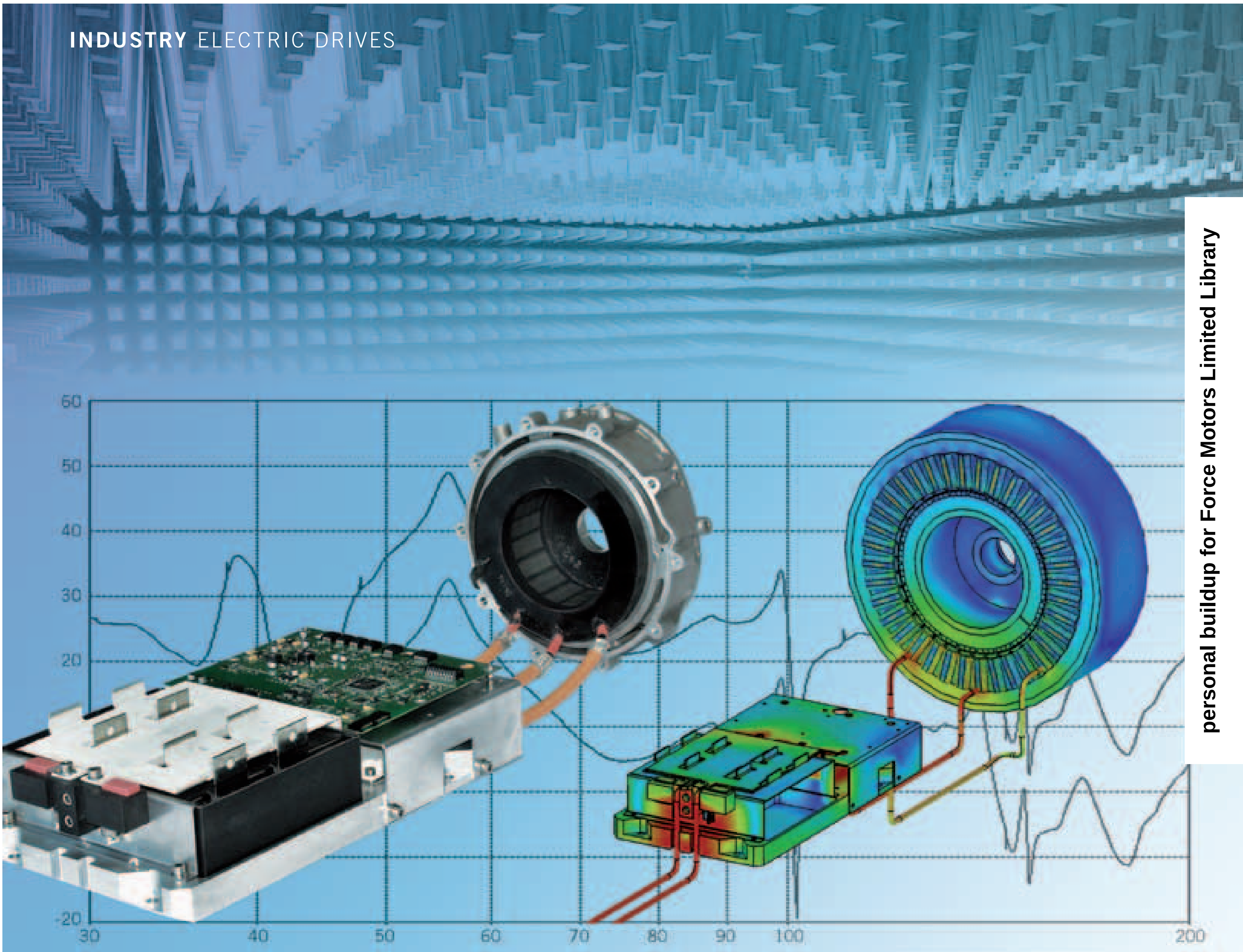

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DEVELOPMENT OF EV / HEV SYSTEMS SUITABLE FOR EMC SUPPORTED BY SIMULATION

EMC requirements regarding emission and immunity valid for systems are also valid for all involved system components. To ensure electro-magnetic compatibility AVL-Trimerics follows a systematically approach.

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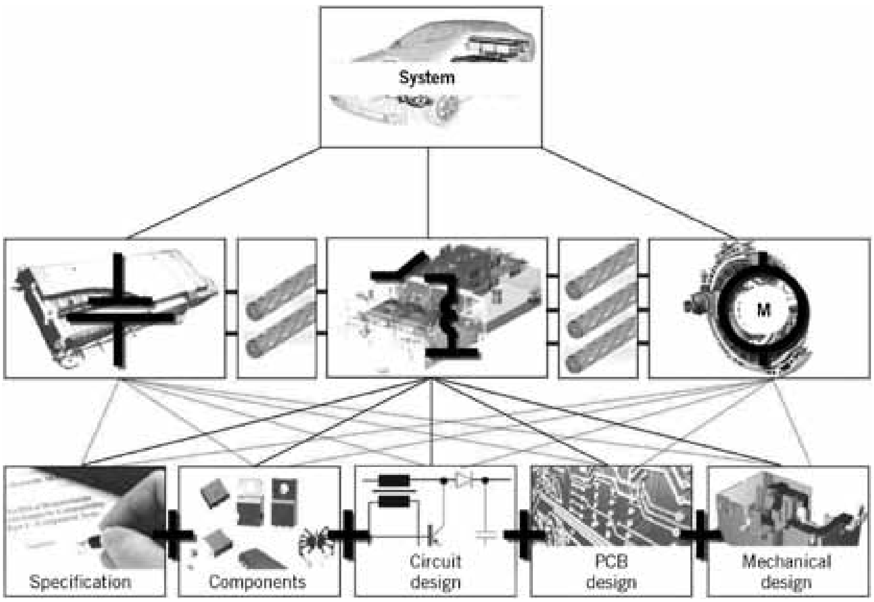
CONSIDERATIONS OF SYSTEM COMPONENTS

These considerations are structured as followed: analysis and assessment of the functional requirements and the EMC requirements – the used active and passive components – the circuitry design – the PCB design – the mechanical components – the enclosure – the wiring harness – the loads – as well as the vehicle specific environment. Especially the descript below simulation approaches, created for the development of electronic devices in the automotive industry must be applied. This becomes necessary, because the problem of energy transfer in the entire vehicle and its useful distribution must be added to the known problem of signal transfer. A system analysis is required to be able to assess the EMC behavior of such systems. This will provide information about the contribution of a system component to the system emission behavior or the level of the attenuation between the power supply network and the high voltage supply network. Detection and counteraction of the root cause is the aim, not to work against the effects, which usually means very high effort. A systematically approach concerning the system components is illustrated in ❶. The shown system components, indicated as subsystems must systematically be analysed and evaluated based on one another. For the sake of completeness the system requirements which have to be fulfilled are listed. A feasibility study provides a basis for a system development.

REQUIREMENT BASED SIMULATION ANALYSIS

An upcoming question might be: “Will the functional requirements based on the specification affect the requested EMC requirements, or is it impossible to fulfill them?” Here is a first possibility to run simulation. The specified values for voltage and current profiles in the specification will be represented in the frequency domain caused by the shapes and amplitudes in time domain. These voltage or current profiles generated by a system component will be transferred via wires to corresponding sensors, actors, loads, etc. The example in ❷ shows, that the functional requirements of a considered system component will cause a conflict with the EMC requirements. The high emission is caused by the function and could not be compensated with filtering at circuitry design level or at PCB design level. Filtering will affect the requested voltage and current profiles; PCB design will not show a reduction of emission in the lower frequency range. The only countermeasure would be to use shielded wires. It would be conceivable to start renegotiations about the requested limits for emission or revi-



❶ Schematic presentation of involved system components, based on one another

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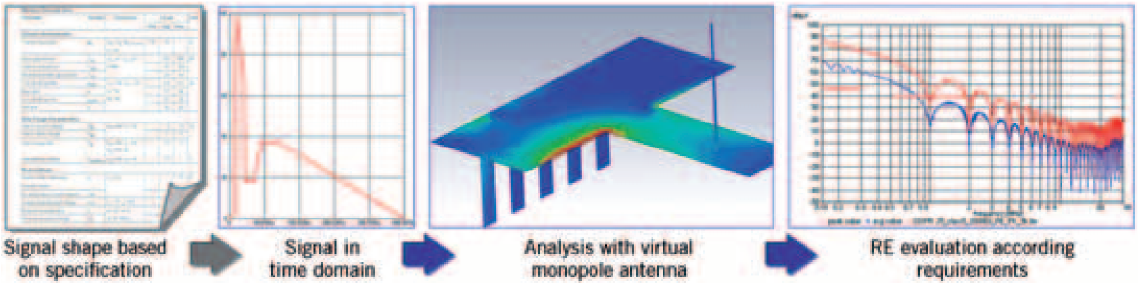

❷ Example of a requirement based simulation analysis: radiated emission with virtual monopole antenna

❸ Example of a schematic based simulation analysis: conducted emission with virtual LISN

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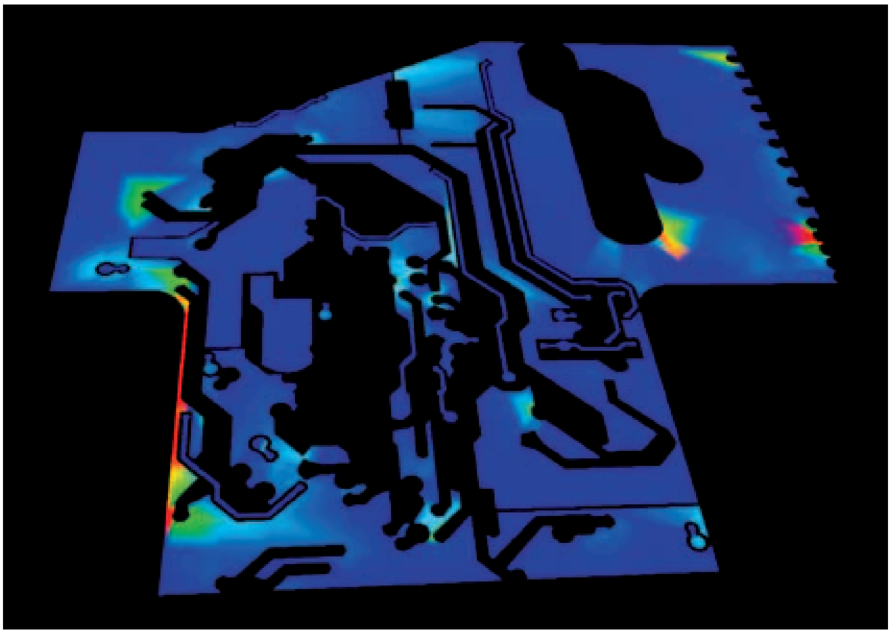
sion of the function. This approach continues for each involved system component (subsystem). It is always essential to identify the source of disturbance or a conflict. Appropriate countermeasures at this subsystem must be applied at an early stage. The elementary model of: source path and sink should be applied at each subsystem not only at system level. An additional example clarifies this statement: The poor emission behavior of a DC/DC converter caused by insufficient circuitry design (circuitry design level) could not be improved by bettered PCB design (PCB design level) in the lower frequency range. After a followed EMC measurement for validation, it is necessary to add filtering or shielding to realize improvements (ECU level, wire harness level). This approach represents the usual standard.

SCHEMATIC BASED SIMULATION ANALYSIS

A new method could be achieved by using more suitable simulation approaches, which allows determining the emission of the circuitry design, already. The schematic based simulation analysis, ❸, enables the user to recognize problems in very early stages of the development. With this approach it is possible to have a run through of various countermeasures to evaluate their impacts on EMC effectively in short time. This could be done without producing diverse samples. Intentionally the impact of the PCB design has not been considered yet. But consistently it will be considered later, when the circuitry design promises good results based on the simulation. These analyses are designed for investigations below 100 MHz, because useful parameters at PCB level are not relevant. Another motivation to concentrate on the lower frequency range is the fact that filter components are expensive and large. An unnecessary implementation should be avoided and the filtering limited to a minimum. In the focus is not the entire schematic diagram, but more previous selected functional groups or modules where high emission levels are expected.

ANALYSES OF COMPONENTS

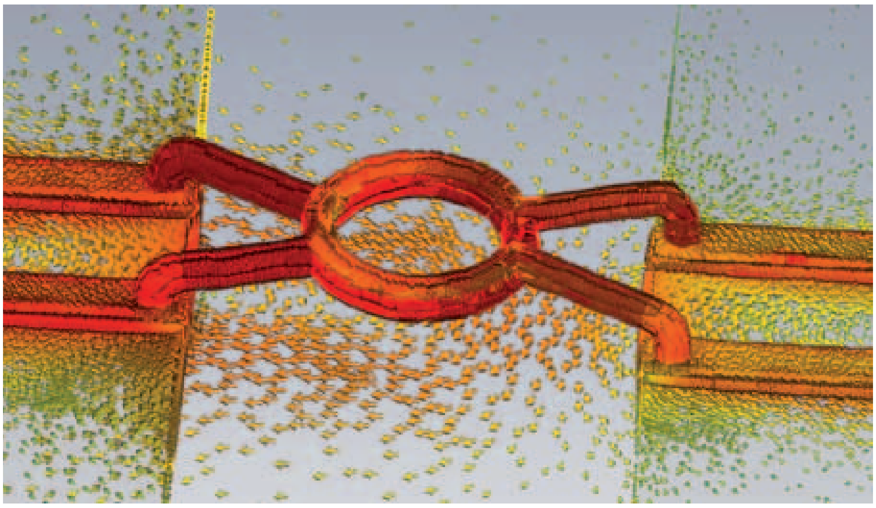
Beside the interpretation of the function of each component of a circuitry under



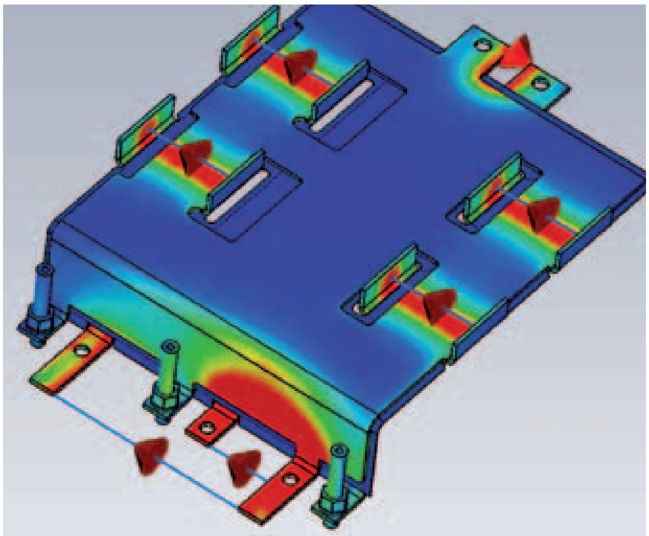
❹ Current distribution within ground system investigation the electrical characteristic has to be added. The analysis of the active and passive components represents in many cases a challenge, because not every relevant parameter for EMC could be estimated from the outside. But, the component analysis is important to perform a schematic based simulation. High frequency parameter must be supplemented to the electrical parameters of each component to enhance the depth of simulation and to specify the simulation results. Beside the generally applicable factors, like material and dimension, the real electrical characteristic has to be determined. This could be done by measurements with vector analyzers or by analysis of the internal structures of complex components, e.g. carrier substrate of an IGBT (Insulated Gate Bipolar Transistor). The real electrical characteristic will be transferred into appropriate models to be able to process them with a simulator. Active components, like microcontrollers or ASIC's are often noticed as the source of disturbance, but power stages with MOSFET transistors or IGBT's as well, because they will show significant effects in the frequency spectrum. Very fast switching operations and high voltages and current in time domain are responsible for these effects. Today, information (Generic IC EMC specification) is available about the EMC behavior of active components in the formats, like IBIS models (Input-Output Buffer Information Specification); or occasional ICEM models (IC Emission Model). These could be used for a deeper analysis of the circuitry design. An optimized circuitry concept, including the involved components done in an early phase represents the key of an adequate EMC behavior. Additionally unnecessary costs and design iteration could be avoided.

PCB DESIGN ANALYSIS

The PCB design analysis is crucial, because the PCB board could be designated as the largest electrical component. The interaction of geometrical properties of all copper structures and the dielectric properties of the substrate determinate the frequency behavior of the PCB board. The placement should be as compact as possible, to minimize the trace length. This could be accomplished by using previously defined functional groups or modules. Requirements regarding immunity requiring considerations about the dimensions of trace lengths or current loops in the area of $\lambda/30$. Therefore the acceptable dimension of a trace loop should not be larger than 10 mm for the frequency of 1 GHz (using realistic dielectric material, $\varepsilon r = 3\text{-}6$). This represents a rudimentary PCB design analysis. Additional aspects like, impedance ratios, the traces themselves, among themselves and within the available layers, the impact of the dielectric material to guarantee signal integrity, resonance effects of ground and supply systems, different modes propagated in ground structures, cross talk and attenuation attributes. Often the PCB design represents a trade-off of space requirements, costs, manufacturability and the high frequency characteristics, thus the EMC behavior. Deep analysis is required to be able to evaluate such compromises. This analysis is particularly for all system components containing a PCB board. The creation and the observance of PCB design guidelines should be indicated here. Simulation tools aid this type of analysis. They provide a visualization of effects, ❹, or parameterize effects and therefore they provide support to find the best solution. In certain difficult cases, simulation could be the only solution to solve the problem; e.g. visualization of the pure capacitive or inductive coupling within a PCB board.



❺ Example: visualization of eddy currents in a metal structure cause by a transformer



❻ Example: visualization of current distribution at a high voltage bus bar

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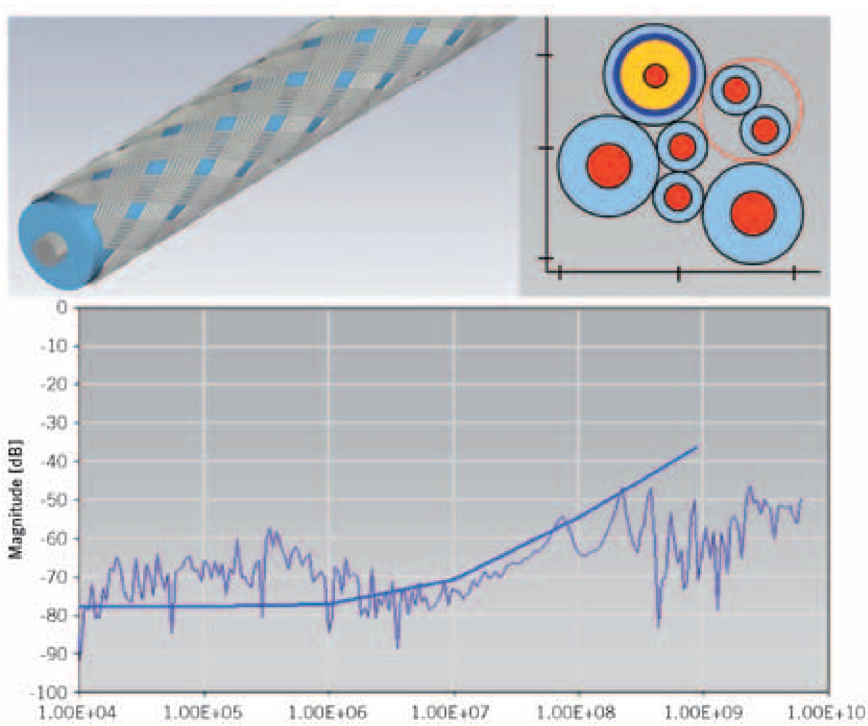

❼ Example: estimation of high voltage cable parameter; analysis of cable bundle

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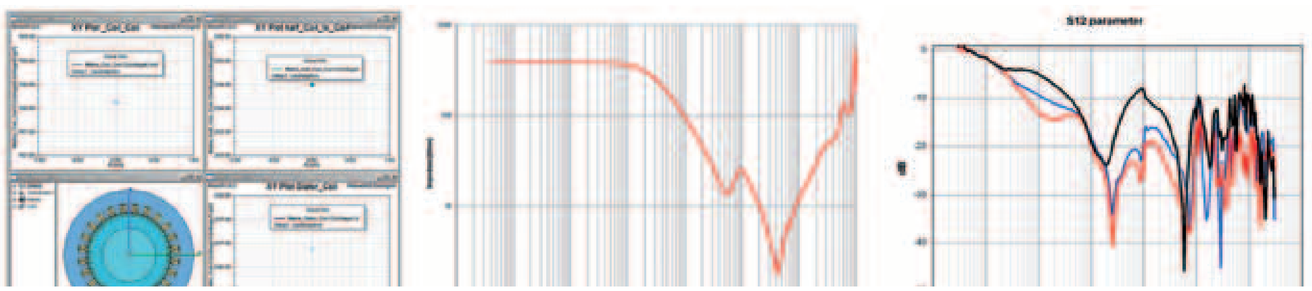
MECHANICAL COMPONENTS ANALYSIS

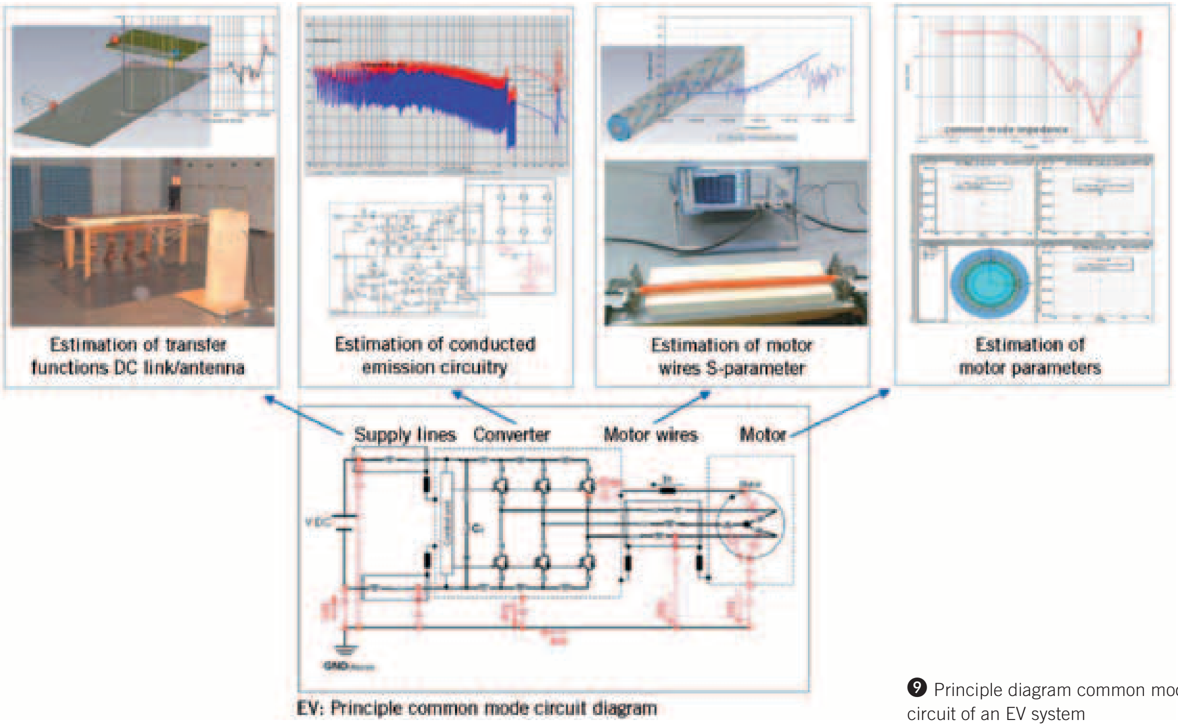
Especially in the area of power electronic, mainly conductive parts or for thermal purpose used components signify an important influence to the emission and immunity behavior of an application. Mechanical components, like: heat sinks, lead frames, enclosures of bus bars must be analyzed to determine their capacitive and inductive characteristics. To give a few examples: Disadvantageous ratios of length and width will increase the inductance of bus bars inside a converter. The consequence could be that the link-circuit capacitor could not deliver the needed current at a specific frequency. It will be taken from the battery, which has a large distance to the converter. This will lead to higher emission. Filter components connected via lead frame of bond wires symbolize a high impedance connection. The desired attenuation will be reduced to a minimum, because at frequencies more than 100 MHz the inductance becomes an invincible barrier. Only a few nH inductance prevent a wanted low impedance filter connection. Power devices mounted to a metal enclosure of a system component for cooling. The capacitive coupling of such a power device to the enclosure could increase the emission. Heat conducting materials (e.g. folio) with a high dielectric constant (εr) supporting this coupling. With the aid of simulation all conductive structures could be analyzed, ❺ and ❻, and solutions for enclosure connections, resonances of enclosures, current distributions for system components compiled.

WIRING ANALYSIS

A wiring system attached to the application identifies a coupling path, the load usually a sink. How much energy will be distributed via the cabling and coupled into other structures or antennas? The available shielded cables for high voltage applications offering a high shielding effectiveness, but the electrical parameters should be known, ❼. Particularly the connection of the shield to ground potential becomes interesting. How high will be the impedance of this connection? Should there be a single sided or double



❽ Assessment of high frequency influences E-motor with simulation and measurements



❾ Principle diagram common mode circuit of an EV system

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sided shield connection? To answer these questions the transfer impedance and the S-parameter of the used cable have to be estimated. To perform simulations at system level, this estimation must be made. Furthermore there are virtual measurement methods available for investigations and estimations of cable parameters via computer simulation.

E-MOTOR ANALYSIS

The attached E-motor to an EV and HEV systems represents the load. In general it could be understood as an enclosed metal housing connected to three shielded cables. The control inside the inverter provides the three phases for the E-motor which occur in the best case as sinusoidal waveform. This pictures the functional character, but not the EMC character of such a construction. Also for an E-motor the high frequency characteristic must be estimated because high potential of disturbance leaving the inverter is feed to the E-motor. Common mode currents will be distributed within the system caused by capacitive couplings inside the windings, the winding core, to the stator, to the rotor and to the metal housing, ❽. The estimation could be done with computer simulation or by suitable measurements.

SYSTEM ANALYSIS

It is indispensable to have the knowledge about all system components, as previously described, to do an analysis at the system level, ❾. Additional analyses in terms of system integration are only possible, when the contribution to emission or immunity of each system component is known. Thus, an EMC assessment at component level takes place, evaluated with international normative measurement methods or customer specifications. It is not always possible to apply this evaluation at real existing system components, but with simulation this is not an obstacle. For system integration within a vehicle additional influencing factors have to be considered: situation of installation space, the rest of the cable harness and the impact of the chassis (e.g. bead).

SUMMARY

A System analysis represents the prerequisite to make a meaningful statement about EMC behavior at the system level. After all system components have been analyzed and the emission behavior of the sources, of the propagation paths and the sinks is known, further analyses at higher level could be performed. With an exchange procedure the system components could be analyzed and their contribution to system behavior predicted. Essential model descriptions, which have to be strongly simplified to keep the calculation time in reasonable range, are imaginable. By using qualified simulation tools coupled with expert know-how upcoming and challenging questions could be answered: “What could be the maximum distance between E-motor and inverter? What type of net topology for high voltage nets must be chosen? What is the level of the attenuation between the power supply network and the high voltage supply network?

THE NEW 1.8 L TFSI ENGINE FROM AUDI
PART 2: MIXTURE FORMATION, COMBUSTION METHOD AND TURBOCHARGING

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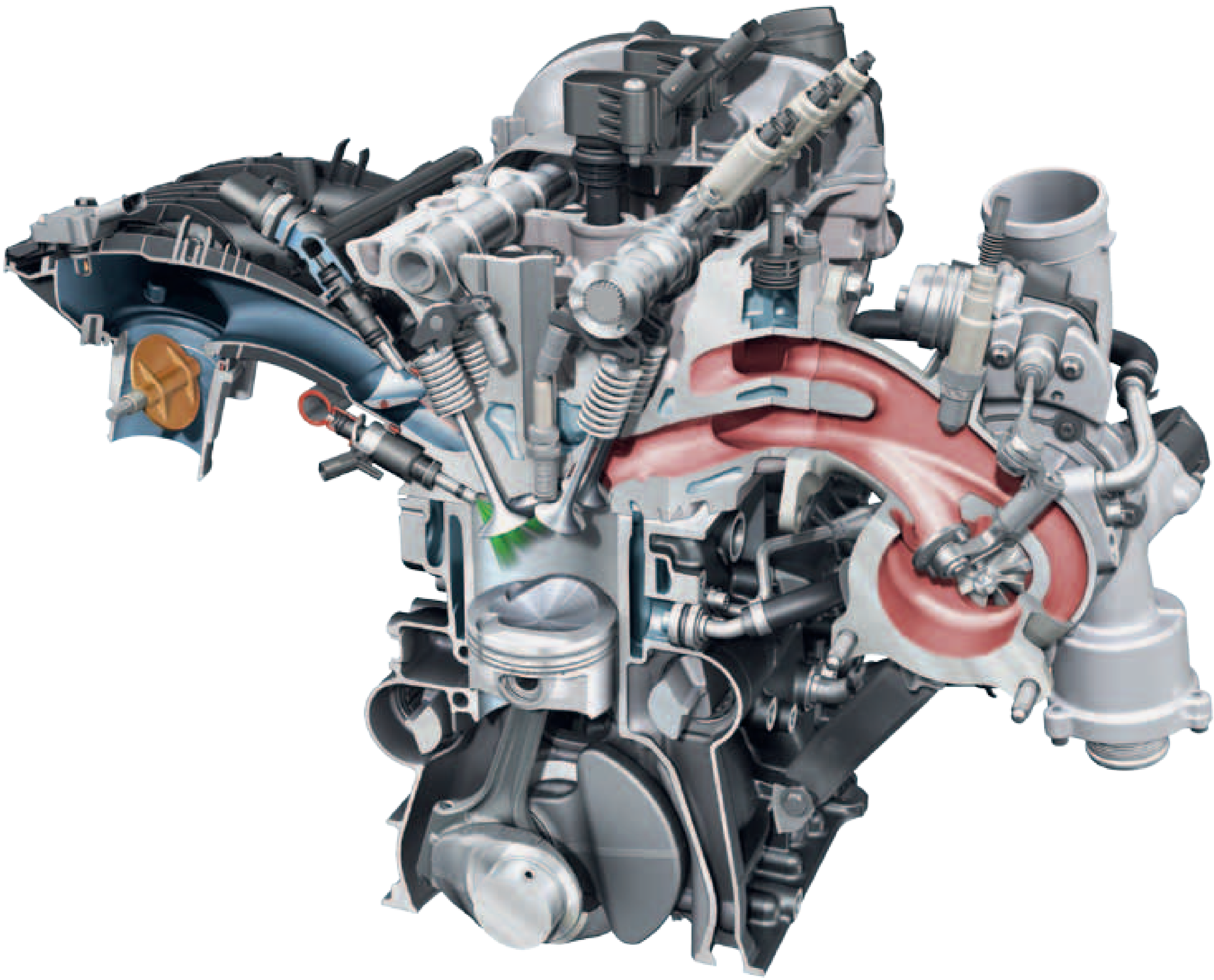

The launch of the new 1.8 l TFSI engine marks the third generation of the successful four-cylinder gasoline engine family from Audi. With consistently reduced frictional losses, the advanced combustion process and new mono-scroll turbocharger and electric wastegate technology, the engine represents a new benchmark in terms of performance and fuel-efficiency. The power plant has already been configured to meet even the strictest future emissions standards worldwide. The mixture formation, the combustion process and the turbocharger of the new engine are described below. The base engine and the thermomanagement system were described in the first part of this article in MTZ 6.

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MIXTURE FORMATION

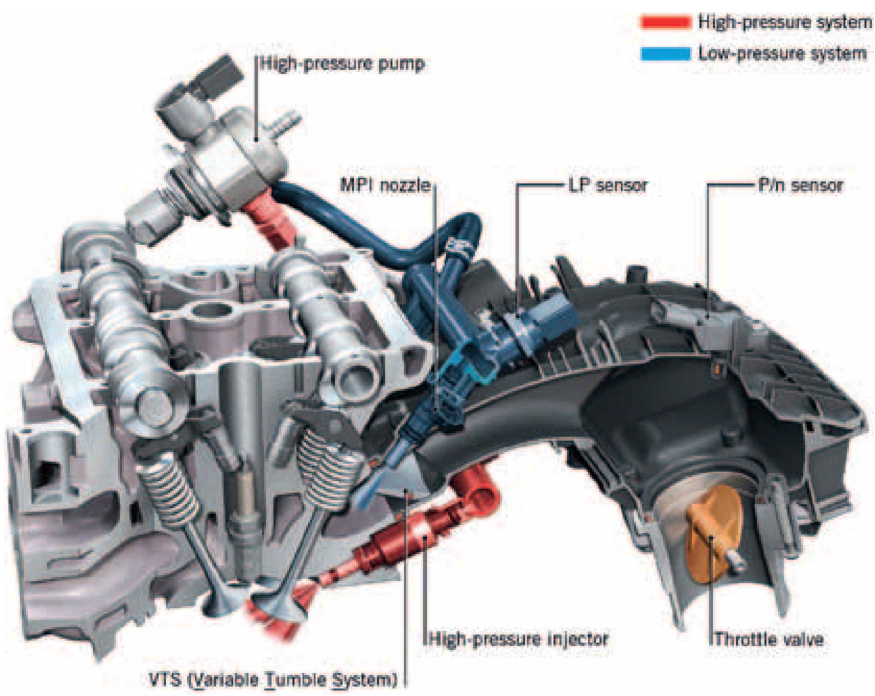
The newly developed high-pressure injection system has been completely revised with regard to maximum system pressure (raised from 150 to 200 bar), acoustics and individual costs. In order to satisfy the limits in terms of particulate mass and number for the Euro 6 requirements and to develop further CO_2 potential, an additional MPI injection system has also been adapted, ❶.

The MPI valves are supplied via a flush connection through the high-pressure pump (HPP) in order to guarantee internal cooling of the HPP during MPI operation. There is a choke built into this flush connection in order to minimise the pulsations transferred to the MPI rail by the HPP. Fuel is supplied to the MPI valves by means of a rail made of plastic. The rail has an integrated low-pressure sensor to regulate the injection volume. The MPI valves are integrated in the VTS flange (Variable Tumble System) and adapted with regard to spray targeting to the location of the VTS partition plates in the cylinder head. The HPP and the high-pressure injectors (HPIs) had to be adapted for the system pressure of 200 bar. The injectors are acoustically decoupled from the cylinder head by means of steel spring washers. Fuel is supplied to the engine under partial load by means of the MPI valves. Under greater loads and when starting off the high-pressure injectors are employed. The fuel supply to the HPIs is achieved by means of a high-pressure rail, which is decoupled from the inlet manifold and bolted directly onto the cylinder head. The system has a pressure sensor with a modified pressure range for the purpose of high-pressure regulation.

The newly developed injection system opens up new levels of freedom for engine applications. In addition to single, double and triple direct injection, mixture formation is carried out by means of the MPI valves in the partial load range of the engine mapping. This results on the one hand in further improved fuel consumption and on the other hand in outstanding particulate emission figures. This engine already meets the coming Euro 6 requirements. Various assessment criteria such as the following are employed in order to coordinate the different injection modes:

- efficiency, knocking
- emissions, especially particulate mass and number
- manifold wall condensation, fuel in the engine oil
- smooth running.

Triple direct injection is used when cold-starting at extremely low temperatures



❶ Dual injection system to meet future Euro 6 emissions standards

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and under greater loads during the warm-up period. This guarantees minimal fuel in the engine oil and minimal emissions, particularly with regard to particulates. On engine start-up and in the catalytic converter heating phase, the mixture formation is carried out by means of double direct injection. Smooth running, robustness in the case of fuel quality variations and minimal emissions are the most important optimisation parameters in this context. Double direct injection is also employed under heavier loads. This injection strategy ensures optimum anti-knock characteristics, particulate emissions and oil dilution in the corresponding mapped range.

The VTS system integrated into the inlet manifold has been revised because of the higher boost pressures. The cranked, one-piece stainless steel shaft guarantees maximum torsional rigidity for the dished flaps in the inlet port. The positioning of the flaps is detected by means of a non-contact rotation angle sensor. When open, the dished flaps are held in the main casting so that excitation by the air flow is minimised. The shaft is electro-pneumatically switched by the engine control unit by means of a vacuum-controlled capsule (two-point control).

COMBUSTION PROCESS

The proven TFSI combustion process has been optimised in many respects for the new, third-generation EA888 engine, both in order to further enhance the engine's robustness with regard to knocking and pre-ignition at mean pressures increased to 22 bar, as well as to optimise the stability of combustion under the modified general conditions with regard to the residual-gas behaviour and λ resulting from the cylinder head with integrated exhaust gas cooling.

As a result of the introduction of the cylinder head with integrated exhaust gas cooling, the period for energy conversion U05-U50 in the standard combustion process is increased by 1 to 2° CA. In conjunction with that the σ_{pmi} becomes worse at 3000 rpm in particular. Thanks to the redesign of the inlet port (increased tumble), it is possible to compensate entirely for this behaviour and to achieve greater potential fuel savings at higher engine speeds. In addition to this, the charge motion induced by the inlet port without the tumble flap activated has been increased once again. The operation of the port when the tumble flap is close, which is of particular importance for catalytic converter heating mode, has remained virtually unchanged thanks to selective improvements in geometry. As a result of the optimised, slightly retracted position of the high-pressure injector, mixture homogenisation has been further improved, and at the same time a positive side-effect has been achieved in the reduction of the temperature load on the injector.

In order to achieve the required increase in power output together with improved spontaneity and optimised full-load fuel consumption, the Audi valvelift system familiar from the 2.0 l TFSI predecessor engine (two-stage valve-lift on the exhaust camshaft) has been adopted and for the first time combined with a camshaft adjuster on the exhaust side, in order to offer maximum freedom in the control of the gas flow. The exhaust camshaft timing has been adapted to the various gas flow requirements of the 1.8 l TFSI in the full load and partial load range, at 180°/195° with an "exhaust closing" from -24° to 6° after TDC. In this way it is possible on the one hand to combine the objectives of outstanding responsiveness and impressive fuel consumption under full load (be < 250 g/kWh) and on the other hand to exploit the good residual gas compatibility of the combustion process under partial load for the purpose of reduced fuel consumption, ❷.

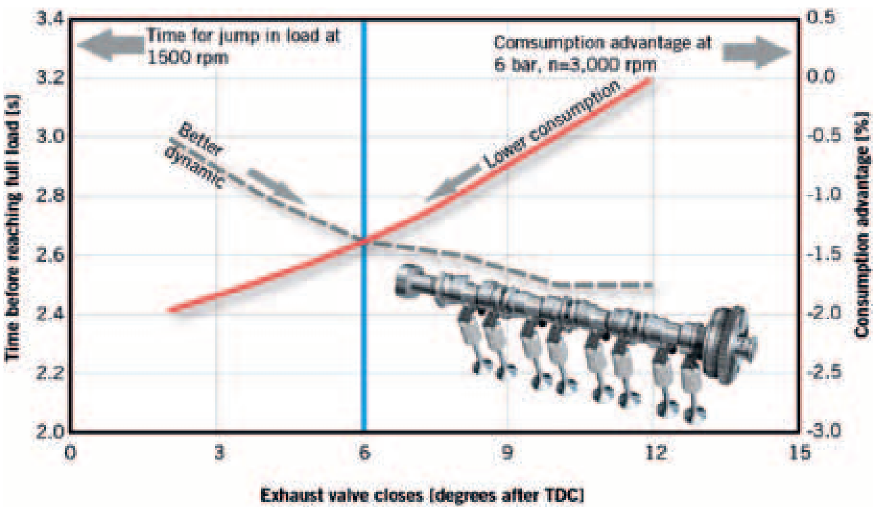
Thanks to the integrated exhaust gas cooling as described above it is possible to achieve operating levels of λ = 1 over wide ranges. Optimum consumption in the engine map is below 230 g/kWh, but what is far more important is the extremely wide engine map range with very good consumption figures of below 250 g/kWh, which guarantees good fuel consumption even if the customer has a rather dynamic driving style, ❸.

TURBOCHARGING

An entirely newly developed mono-scroll turbocharger is used as a charging system, ❹. The aim of the design was to combine good low-end torque with maximum power output and very high performance. The basis for this design was the RHF4 turbocharger from IHI, with various improvements made to the rotor assembly, the spirals, the housings and to all the flow-carrying parts and components. The turbocharger is distinguished by the following features:

: electric wastegate actuator
: lambda sensor ahead of the turbine
: compact cast steel turbine housing with double-flow inlet on the cylinder head flange
: compressor housing with integrated pulsation silencer and electric dump valve
: Inconel turbine rotor, designed for T3 = 980 °C
: bearing housings with standard connectors for oil and water
: milled compressor rotor.

The turbine housing is made of 1.4837 cast steel. This guarantees dependable fulfilment



❷ Conflicting aims: Dynamics and partial load fuel consumption

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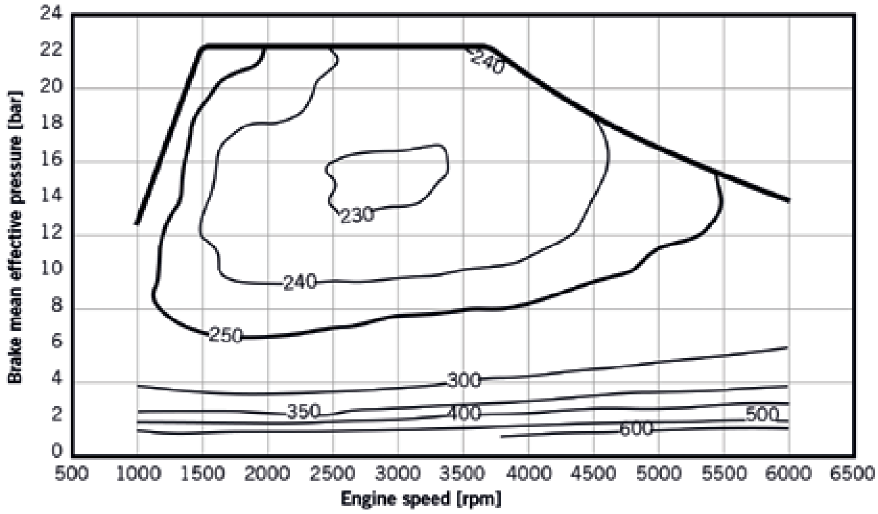
of the functional requirements over the lifetime and makes possible the positioning of the lambda sensor in the turbine housing ahead of the turbine, coupled with an exhaust gas temperature of 980 °C. The best possible ignition sequence separation has been represented by means of dual ducting as far as just ahead of the turbine. Thanks to the integrated exhaust gas cooling concept the total mass of the turbine housing and hence the use of the high-alloy, nickel-based material has been reduced by about 40 %. As a result of the compact design of the turbine housing, it is bolted to the cylinder head by means of standard studs and nuts. For the first time at this high exhaust gas temperature it has been possible to produce the turbine wheel from Inconel 713 C (nickel-based alloy) instead of MAR. Extensive preliminary trials were carried out into the creep characteristics of the rotor.

The very intricately designed compressor housing is made of die-cast aluminium. Integrated into the compressor housing are the pulsation silencer, the electric dump valve and the inlet point for the gases from the crankcase and fuel tank ventilation. The compressor housing has been structurally strengthened for the considerably increased actuation forces of the electric wastegate actuator. The compressor rotor is milled from a solid block, which delivers advantages such as greater high-speed strength and better acoustics.

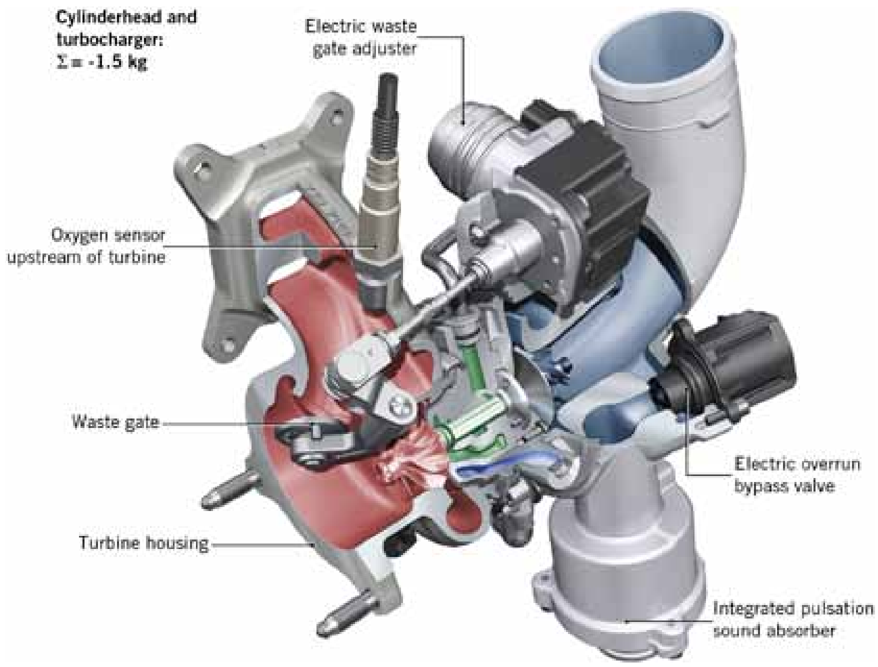
The newly designed wastegate actuator is a faster and more precise actuator than the previous pressurised actuator. It can be activated independently of the current boost pressure. The key advantages of the now active control system are set out below:

- As a result of the greater closing force, the maximum torque of 320 Nm is achieved at a lower engine speed of 1400 rpm.
- With the active opening of the wastegate under partial load, the basic boost pressure can be reduced. This enables savings of approximately 1.2 g CO_2/km in the NEDC cycle.
- The active opening of the wastegate during the heating of the catalytic converter results in a 10 °C higher exhaust gas temperature ahead of the catalytic converter, which leads to lower cold-start emissions.
- As a result of the higher operating speed of the electric wastegate actuator, it is possible to achieve an immediate drop in boost pressure in the event of negative load changes, which has a particularly positive effect on the acoustic behaviour of the turbocharger (spitting, chattering). The responsiveness of the engine to positive load changes is also improved, as a result of the faster dynamic build-up of boost pressure.

For the first time at Audi the lambda sensor has been positioned ahead of the turbocharger turbine. This enables a considerably earlier end to the dewpoint and hence to an early release of lambda regulation after engine start-up as well as good individual cylinder recognition. Attention was given to good gas flow with the lowest possible temperature loads when defining the position of the lambda sensor.



❸ Fuel consumption map (effective specific fuel comsumption [g/kWh])



❹ Compact turbocharger module with lambda sensor ahead of the turbine and electric wastegate actuator

CAE OPTIMISATION OF THE TURBOCHARGER

Comprehensive CAE optimisation measures were carried out both on the turbine side and on the compressor side. ❺ shows a diagram of both CFD simulation models. On the turbine side the CFD simulations were

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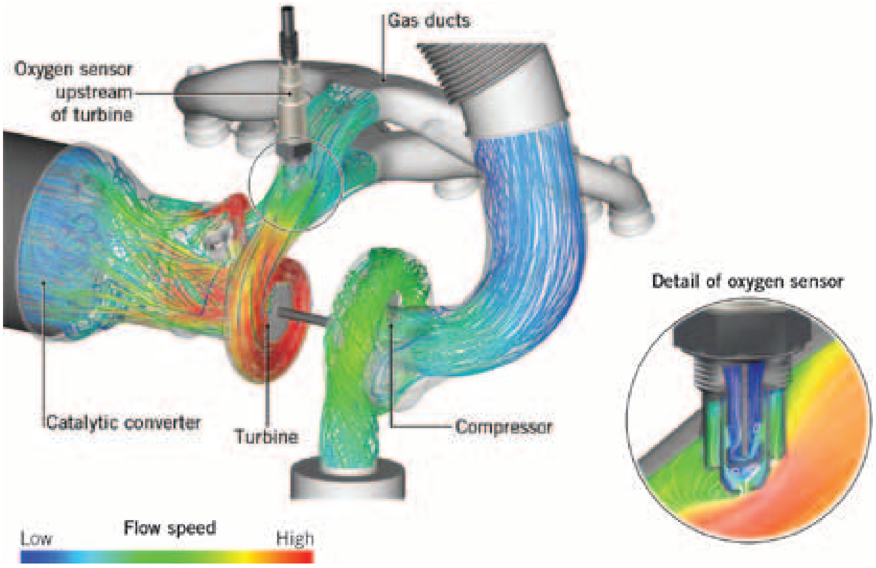
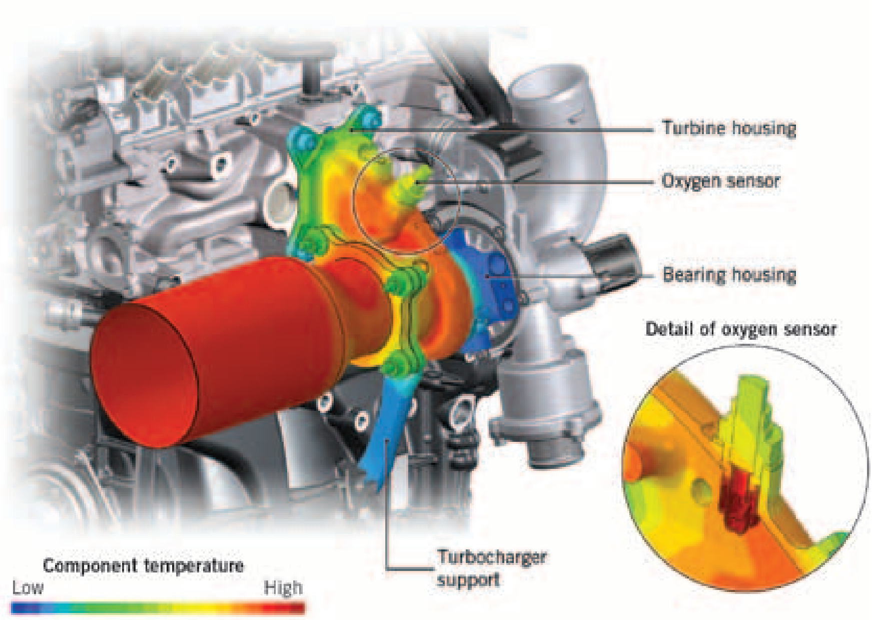


❺ CFD simulation in the region of the turbocharger (left); detailed calculation of the lambda sensor (right)



❻ FEM simulation of the turbocharger and detailed calculation of the lambda sensor (temperature distribution)

observed in the entire system including the integrated exhaust gas cooling gas duct in the cylinder head, the turbine housing including rotor, wastegate, lambda sensor ahead of the turbine and the exhaust system as far as just after the close-coupled primary catalytic converter. The aims were to optimise the integrated exhaust gas cooling gas duct in conjunction with the turbine inlet, the gas flow to the lambda sensor, the wastegate design and to achieve very good gas flow to the catalytic converter. On the compressor side the CFD model includes the air induction, the compressor including all inlet points (e. g. from the crankcase ventilation) as well as the dump valve and the charge air duct. The aim was firstly to develop gas flow to and from the compressor that was as loss-free as possible and neutral with regard to the performance of the compressor, and secondly to find the best possible position for the inlet points and the dump system. In this context the simulations identified considerable potential both with regard to pressure losses and in terms of compressor efficiency.

In conjunction with FEM simulations, the thermo-mechanics and the durability of the turbine housing were significantly improved, ❻. The result of the CFD simulation was used in this context and applied as an ancillary condition for the calculation of the temperature map. On the basis of the calculated temperature map and additionally imposed forces such as bolt-tightening forces, the loading of the component was determined and optimised in several design phases.

The gas flow at the lambda sensor, the temperature distribution and load on the sensor were also simulated in detail. On the one hand extensive transient CFD simulations were carried out to check functions under partial load, ⑤ (right), and on the other hand detailed thermo-mechanical calculations of the sensor embedded in the turbine housing were used in order to ensure the durability of the component by means of optimum material pairings and an optimum installation location, ⑥ (right).

PERFORMANCE AND FUEL CONSUMPTION

The various optimisation measures with regard to turbocharger design, the combustion process and gas flow are ultimately reflected in stationary and dynamic full-load behaviour. The new 1.8 l TFSI already achieves its maximum torque of 320 Nm at 1400 rpm and has a wide power output band of 125 kW between 3800 and 6200 rpm, ❼ (left). Further increases in power output are promised. Furthermore the time taken to reach maximum torque has actually been reduced once again compared with the predecessor engine, despite considerably reduced mean pressure and higher maximum power output, ⑦ (right). This guarantees spontaneous response and superb performance.

The Performance Feel Index (PFI) is used at Audi to assess the performance and driving dynamics of a vehicle. It is a measure of the acceleration capability of a car and is often depicted in conjunction with fuel consumption. The full-load behaviour in the case of the third-generation 1.8 l EA888 engine (125 kW/320 Nm) results in considerably improved performance compared with the preceding second-generation 1.8 l engine (118 kW/250 Nm), despite wider gear ratios for improved fuel consumption. ❽ clearly shows the performance gain (+12 %) along with significantly reduced fuel consumption (-22 %). Even compared

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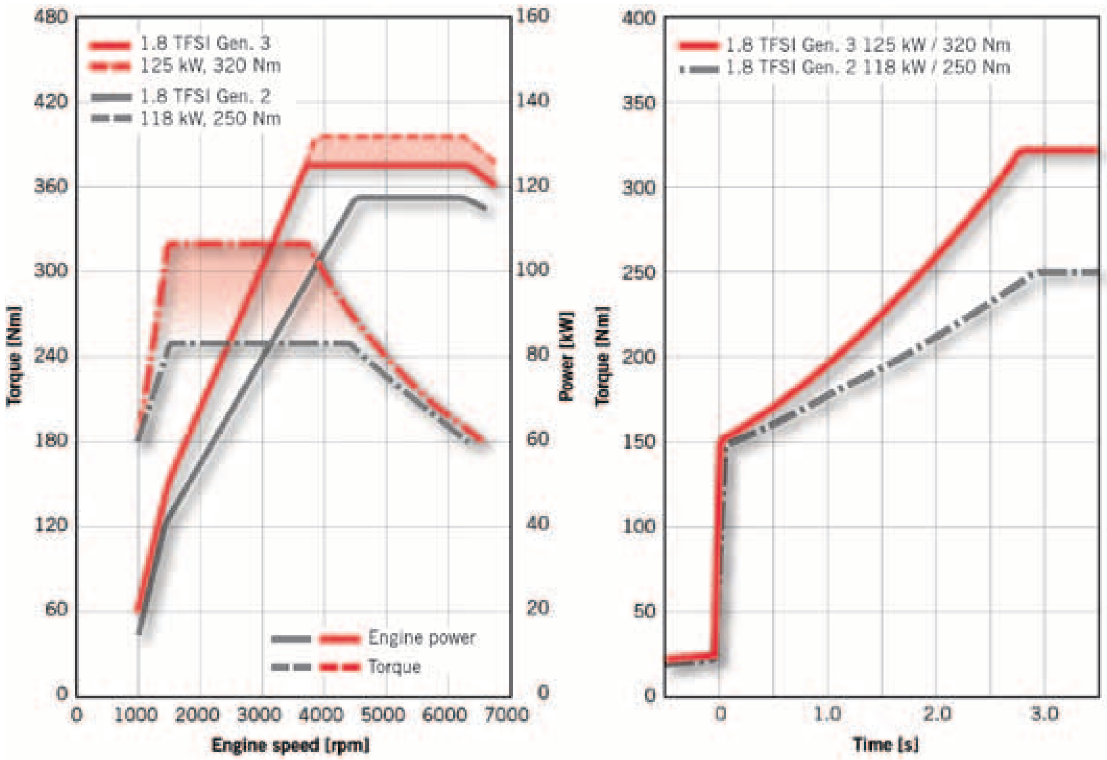

❼ Power output and torque curves as well as dynamic torque increase at 1500 rpm speed in comparison with the predecessor engine [1]

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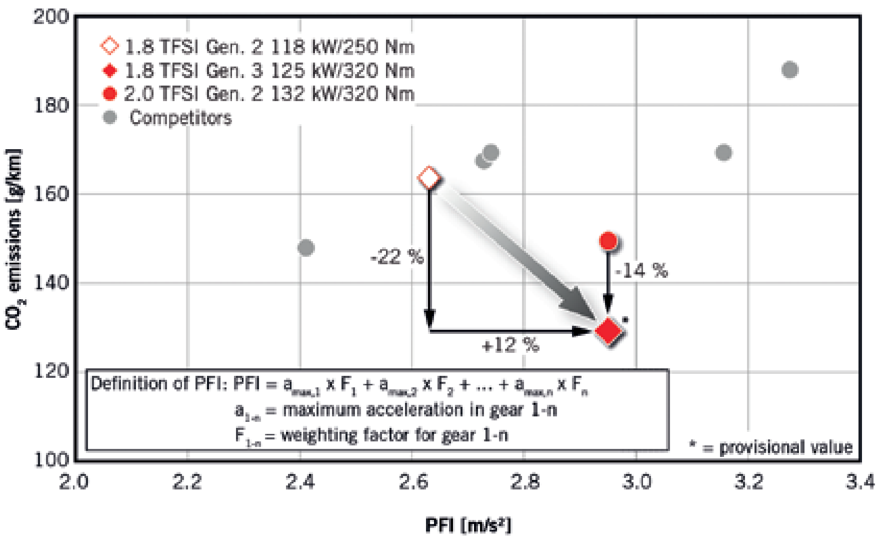
with the second-generation 2.0 l EA888 engine with the Audi valvelift system (132 kW/320 Nm), there is still a clear, 14 % improvement in fuel consumption, with the same performance.

Thanks to down-speeding, frictional and thermomanagement measures and the many and varied thermodynamic improvements, in conjunction with further accompanying vehicle-related measures, the NEDC fuel consumption of the new Audi A4 with the third-generation 1.8 l engine has been significantly reduced. In particular the exhaust camshaft adjustment in conjunction with the exhaust-side Audi valvelift system, the electric wastegate actuator and the dual injection system improve efficiency under partial load and also contribute to the achievement of the ambitious fuel consumption targets.



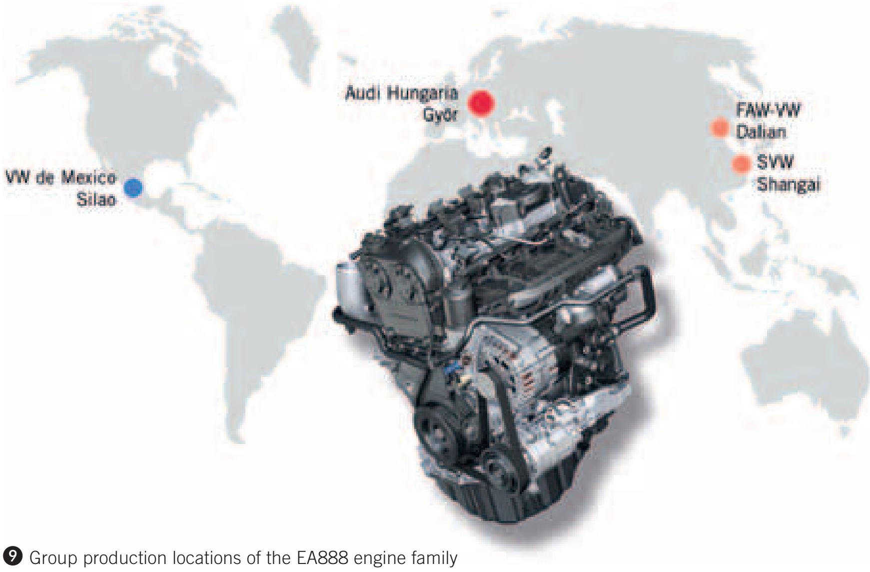
❽ CO_2 emissions versus performance (PFI – performance feel index, weighted average acceleration ability in all gears)

EMISSIONS APPLICATION

The Euro 6 target is achieved by the optimisation of raw emissions by means of an increase in the injection pressure of the FSI system from 150 to 200 bar, the optimised positioning of the high-pressure injectors for improved mixture formation and the exhaust tract designed to achieve minimum surface area. Rapid lambda sensor readiness and dynamic detection of the λ-value are resulting from the fitting of the lambda sensor ahead of the turbine.

In addition the exhaust treatment is further developed by means of a new thin-wall ceramic monolith rated at 400 cpsi and with a wall thickness of 3.5 mil as well as the newly developed JM835 precious metal coating. Thanks to the use of the thin-wall substrate, the exhaust back-pressure is significantly reduced with simultaneously reduced light-off time. The bypassing of the turbine as a result of the opened wastegate under partial load also ensures that the maximum amount of exhaust energy is made available for heating the catalytic converter when starting from cold.

In start mode triple FSI injection is employed in the compression phase, and warm-up is achieved by means of an FSI double injection in the induction and com-



❾ Group production locations of the EA888 engine family

pression phases with moderately retarded ignition. As a result of the DoE-based application of the injection, ignition and camshaft parameters, a balanced compromise has been found with low gaseous emissions and minimal particulates along with very smooth running of the engine. Following the warm-up, the MPI injection system comes into play in the non-knock limited range for optimum partial load efficiency. For the further reduction of CO_2 emissions, in addition to the aforementioned thermomanagement system, the familiar stop-start and recuperation measures are also employed. These measures ensure consistent compliance with the Euro 6 emissions figures.

SUMMARY AND OUTLOOK

The third generation of the 1.8 l FSI engine marks the start of the third generation of the successful EA888 "global engine" family from Audi. The third-generation 1.8 l engine presented here thus represents the springboard for other versions and for other increases in power and torque. As a result of the thorough reworking of the engine, significant progress has once again been made compared with the already very well positioned second generation of the EA888 engine series, thus achieving outstanding positioning of the engine in the competitive environment. The new 1.8 l TFSI engine thus represents a further milestone in Audi's down-sizing and down-speeding strategy. It has been possible to significantly improve all the engine-related properties of the EA888 engine family thanks to the application of:

: consistent optimisation of frictional losses and lightweight construction in all subassemblies
: a cylinder head with integrated exhaust gas cooling
: the Audi valvelift system on the exhaust side with inlet and exhaust camshaft adjustment
: the innovative thermomanagement system with fully electronic coolant control
: the dual FSI/MPI injection system with 200-bar high-pressure injection
: a new, compact turbocharger module with cast steel turbine housing, electric wastegate actuator and lambda sensor ahead of the turbine.

The third generation of the EA888 "global engine" is thus ideally equipped to meet the high requirements for worldwide use in all markets. In order to be able to cover the high production numbers in the VW Group, there are already several factories worldwide for the production of the EA888 engine family, ❾. Production of the third-generation engines will start this year at the Audi engine plant in Györ (Hungary) with subsequent start-ups in Mexico and China. The engine plant in Györ is one of the world's biggest engine plants and is the "lead plant" for all new engine start-ups of the EA888 family.

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POTENTIALS OF A MECHANICAL FULLY VARIABLE VALVE ACTUATION AT BOOSTED SI ENGINES WITH DIRECT INJECTION

Future spark-ignition engines, particularly those with a reduced displacement due to downsizing concepts, will increasingly make use of exhaust gas turbocharging, fully variable valve timing and homogeneous direct injection. The University of Kaiserslautern and the Karlsruhe Institute of Technology (KIT) are jointly carrying out research to determine the influence that the individual measures of this combination have on the engine's target parameters. The following report presents some of the results of this research.

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1 INTRODUCTION

The reduction of fuel consumption and thus of CO_2 emissions is in the focus of ongoing research and development activities. In view of different fuel qualities two concepts found their way into modern vehicles for worldwide application. On the one hand this is the concept of downsizing, where high specific power is reached by turbocharging. To reach a good response behavior, these engines are more and more equipped with direct fuel injection and homogeneous mixture distribution [1, 2]. On the other hand high efficiency is reached by reducing friction and pumping losses. An effective way to reduce pumping losses is the unthrottled engine operation at part load with mechanical fully variable valve actuation. This technique was introduced by BMW in 2001 and later introduced by several car manufacturers like Toyota, Mitsubishi, Nissan and Honda in the last years with their own valve train systems [3]. The direct fuel injection with stratified mixture distribution and turbocharging has high potential to reduce fuel consumption but has high demands on mixture preparation and in-cylinder flow. This technology is in the focus of research and development activities [4, 5, 6]. Future concepts to increase the efficiency will combine different technologies like direct injection, boosting and variable valve actuation. BMW as precursor of mechanical fully variable valvetrains realized already the combination of DI, turbocharging and CVVL (TVDI) and offers such engines on the market [1, 7].

At the Technical University of Kaiserslautern and the Karlsruhe Institute of Technology (KIT) researches are going on to answer which impact such different technologies have on the engine efficiency, performance and emissions. In this paper the potential of boosting and CVVL on gasoline engines with DI and PFI are presented.

2 VARIABLE VALVE TRAIN

For the research activities the mechanically fully variable valve train system Uni Valve from Kolbenschmidt Pierburg AG is used. The system was developed at the Technical University of Kaiserslautern with the support of the company Entec Consulting GmbH. Similar to the Valvetronic, the Uni Valve mechanism is a curvature gear to adjust the valve lift from zero-to maximal lift [3]. The base valve train system is a roller finger arm to reduce the friction. In between camshaft and roller finger arm, intermediate levers with a working curve are positioned by an eccentric shaft and by a guide. The position of the intermediate lever and over that the valve lift height is adjusted by the eccentric shaft. The position of the intermediate lever controls which part of the working curve comes into contact with the role of the roller finger arm. To guarantee low friction levels, especially at part load of the engine and that means at part lift, the contact of the intermediate lever to the eccentric shaft and to the cam lob is done over roller contacts. As these rollers are beared over needle bearings, the friction level of the Univalve system is reduced at full lift on the level of standard roller finger valve train system.

In an engine with four valves it is additionally possible to run the two inlet valves with different valve lifts and timings. This so-called valve lift phasing generates an intensive in cylinder swirl flow through which the engine's residual gas compatibility is increased [8]. The actual implementation of the valve lift phasing can be achieved by two different eccentric shafts' contours, so that it only occurs at part load. Therefore, the full load behavior is not influenced by the valve lift phasing. A cylinder disabling (in the US called displacement by demand) is also possible in this system.

In ❶ the possible variation parameters with a fully variable valve train and cam phase adjustment on the inlet-and outlet side are shown. On the inlet side, this allows to separately modify either the

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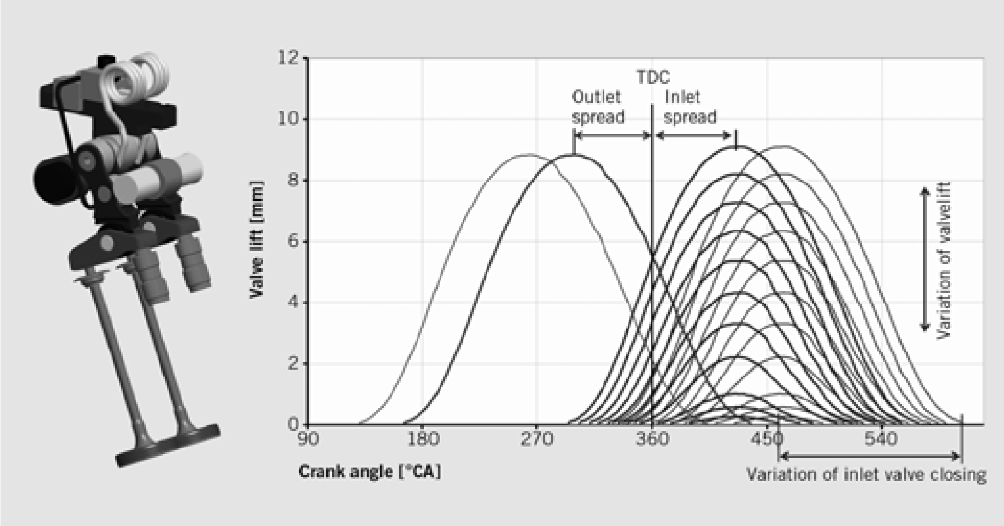


❶ Variable valve actuation Uni Valve

inlet spread (IS) or the inlet valve lift and in the case of the outlet side the outlet spread (OS). The fixed relation between valve lift and timing can be uniquely configured at the valve drive's arrangement.

3 MEASURES TO REDUCE FUEL CONSUMPTION

3.1 DOWNSIZING WITH GASOLINE DIRECT INJECTION

A reduction of the displacement results in a loss of torque and power. To reach a comparable power and a comparable torque with a lower displacement, these engines are generally boosted. The high specific power of such engines makes it possible to adjust the overall gear ratio, so that the engine is operated with reduced engine speed. This is called downspeeding and is usually combined with downsizing concepts.

Basically the fuel consumption declines with increasing the downsizing level. The only change to homogeneous direct injection leads to a decrease of the fuel consumption of about 4 to 12 %. The combination with reduced engine displacement and boosting leads, depending on the downsizing level, to a decrease of fuel consumption of about 15 to 30 % [9, 10]. The direct injection assists the filling at higher loads and the knocking behavior by reduced mixture temperatures. Additional benefits in fuel consumption can be reached with variable valve actuation, lean mixture composition, high boosting and different EGR strategies. How far a variable valve actuation can be used to reduce fuel consumption and to increase the response behavior, is analyzed during these investigations.

3.2 FUTURE TECHNOLOGIES

The combination of boosting and stratified fuel injection has the highest potential to reduce fuel consumption and to further increase the engine efficiency. Turbocharging enables a further increase of the air-fuel ratio with a corresponding increase of the thermal efficiency and an extension of the operating rang in stratified mode. Results of experimental investigations with boosting and spray guided fuel injection are shown in ❷. During these investigations the operating point with an engine speed of n = 2000 rpm and an indicated mean effective pressure of IMEP = 7 bar was held constant by adjusting the injected fuel mass. These investigations were performed on a single-cylinder research engine with an injection pressure of 1000 bar. The high injection pressure leads to an extreme shortening of the injection duration with very high vaporization rate. At first investigations to the leaning behavior were done without boosting and with homogeneous mixture preparation. The mixture can be leaned up to an air/fuel ratio of $\lambda = 1.55$ at 2000 rpm and IMEP = 7 bar, until misfire occurs.

Based on stoichiometric engine operation, the indicated specific fuel consumption is reduced of 7 % with higher air-fuel ratio from ISFC = 242 g/kWh to 225 g/kWh. Based on this operating point with maximum leaning ($\lambda = 1.55$) the boost rate was increased at 2000 rpm and IMEP = 7 bar. The boost rate is defined as the pressure after compressor related to the pressure before compressor. With increasing the boost rate to 1.5 the specific fuel consumption is reduced significantly to ISFC = 208 g/kWh. Thus the fuel consumption is further decreased with boosting and spray guided stratification of about 8.5 % compared to homogeneous lean engine operation and of about 15 % compared to homogeneous stoichiometric engine operation. In consideration of the high engine load, this enhancement is remarkable. During these measurements the cyclic variations of the IMEP were on the same level like engine operation with homogeneous stoichiometric mixture. Also the HC, NO_x and CO emissions were on the same level. The Bosch smoke degree is due to the high injection pressure at the detection limit.

4 DOWNSIZING WITH FULLY VARIABLE VALVE TIMING

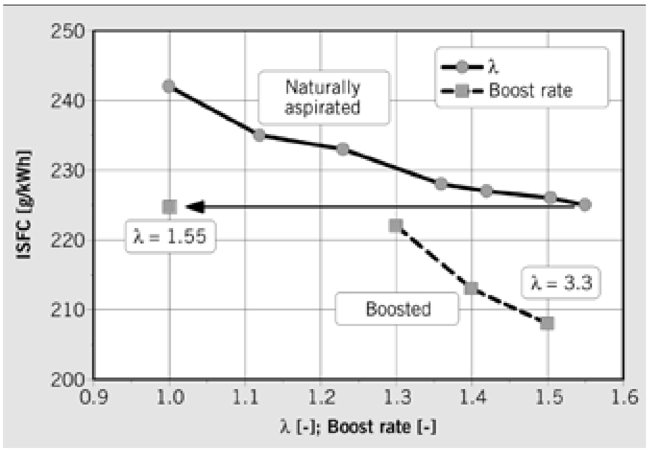
At the Technical University of Kaiserslautern a 2.0 l four-cylinder gasoline engine was equipped with turbocharging and a mechanically fully variable valve train (CVVL). Two similar engines were designed and run respectively with interior and exterior mixture preparation. This project was made possible through generous financial support by the companies Audi and Entec. In both engines, the fully variable valve train Uni Valve was used.

4.1 TEST RESULTS AT FULL LOAD

❸ (left) shows a comparison between the maximal torques for the three different concepts at engine speeds between 800 rpm und 2000 rpm. The maximal torques of the turbocharged engine with gasoline direct injection and CVVL (Concept 1: CVVL-T-DI) are compared here with those of the turbocharged engine with port fuel injection and CVVL (Concept 2: CVVL-T-MPI) and those of the comparable engine with turbo charging and gasoline direct injection, but without CVVL (Concept 3: T-DI).

In the engines with fully variable valve trains, the inlet valve lift and the inlet spread was modified according to the different engine speeds. The diagram illustrates the respective best performances. The comparison between the base engine (T-DI) and the engine run with MPI and Univalve (CVVL-T-MPI) shows an increased full load torque at very low engine speeds with the variable valve train in action. The maximal torque was already reached with lifts ranging between 6 mm. Starting from an engine speed of 1250 rpm, both engine designs show a comparable torque behavior. In reference to the base engine, the engine which was equipped with both gasoline direct injection and a mechanically fully variable valve train (CVVL-T-DI) has with an increase of approximately 10 % the highest torque over the whole area displayed.

In ③ (right) the speeds of the turbocharger for the different engine concepts are compared. What becomes apparent here is that



❷ Specific fuel consumption with varying the air-fuel ratio and the boost rate [14] at n = 2000 rpm und IMEP = 7 bar

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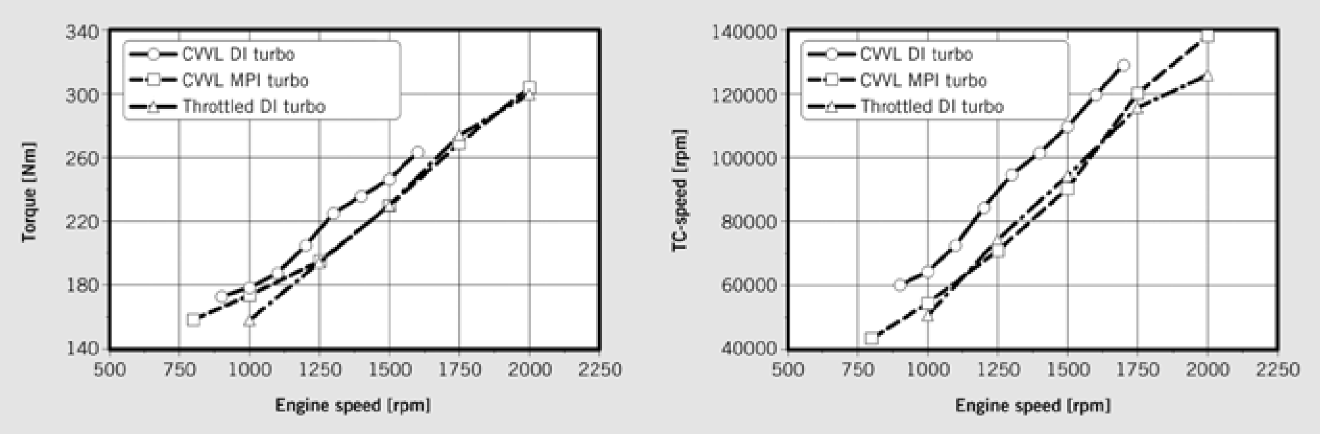

❸ Comparison between the torque and the ATL-engine speed of the different engine concepts at full load

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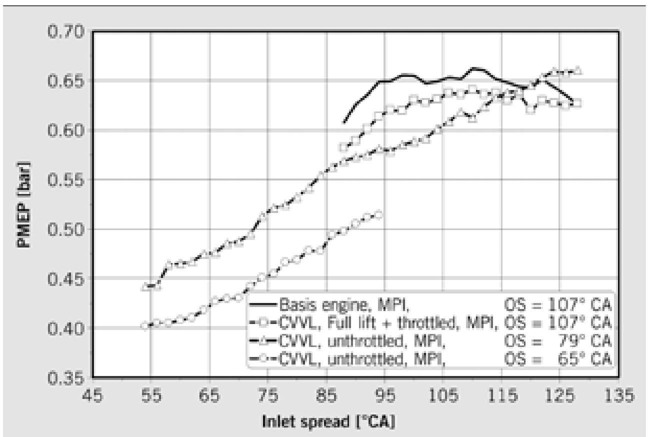
the turbocharger speed from concept 1 equipped with turbocharger, CVVL and gasoline direct injection is approximately 15,000 rpm higher than seen in the other two designs. The highest turbocharger speed and therefore also the highest torques are accomplished with the CVVL-T-DI engine. In the range of low engine speeds, specifically between 1200 rpm and 1500 rpm, the torque is approximately 10 % or respectively 20 Nm higher than in the CVVL-T-MPI engine. These higher turbocharger speeds in the CVVL-T-DI engine are generated by an increased cylinder air mass, which is connected to higher exhaust gas mass flow and a higher cylinder pressure when opening the outlet valves.

The influence of the outlet spread on the torque at full load was not examined in this project. As such, the question remains unanswered whether a phase adjustment or a CVVL on the outlet side has the potential to further increase the maximal torque. In the test engines scavenging did not occur at lower engine speeds. In fact, the longer outlet duration and the increased outlet spread were disadvantageous for scavenging. A maximal trapping ratio was achieved through short engine timings which could be run with low engine speeds with Uni Valve at full load. The extended variability at full load further reduced the amount of residual gas. By reducing the temperature of the cylinder load, the risk of knocking decreased which accomplished earlier crank angle for the peak cylinder pressure. This has positive effects on the fuel consumption at full load and the torque. The evaporation of fuel in the combustion chamber that occurs with gasoline direct injection leads to a further decrease of the in cylinder temperature. These resulting advantages are further benefits of the fully variable valve train.

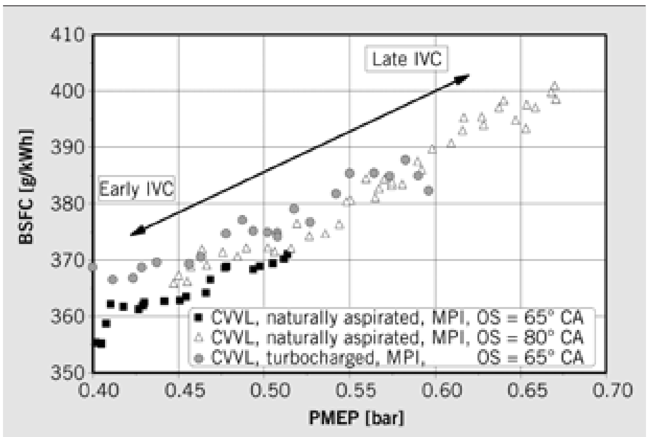
4.2 TEST RESULTS AT PART LOAD

The fuel consumption at part load with throttle-free load control can be either reduced through a decrease of the pumping mean effective pressure PMEP (charge cycle work) and an increase of the amount of residual gas or a reduction of friction. The pumping mean effective pressure at throttle-free load control can be for example influenced by the valve lift duration and the spread of the in- and outlet side. ❹ shows the influence of the inlet spread on the PMEP for throttled as well as throttle free modes with MPI and two different outlet spreads for an engine operating point of n = 2000 rpm and a break mean effective pressure of BMEP = 2 bar. It becomes clear that the throttle-free mode considerably reduces the pumping mean effective pressure.

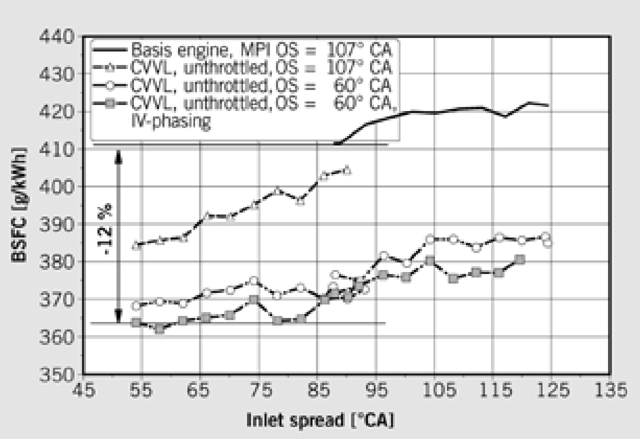
In ❺ the relation between brake specific fuel consumption and PMEP is shown for natural aspirated engines as well as for turbo



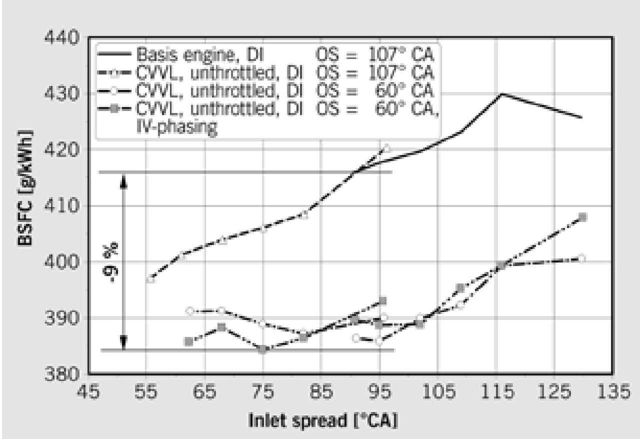
❹ PMEP over inlet spread at n = 2000 rpm und BMEP = 2 bar



❺ Specific fuel consumption over PMEP at n = 2000 rpm und BMEP = 2 bar



❻ Brake specific fuel consumption of the CVVL-engine with port fuel injection at n = 2000 rpm und BMEP = 2 bar



❼ The CVVL-engine's brake specific fuel consumption with direct fuel injection at n = 2000 rpm and BMEP = 2 bar

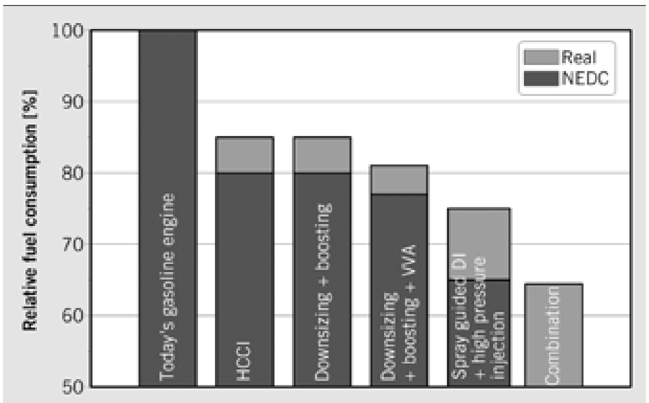
charged engines with different inlet spreads for an engine operating point at n = 2000 rpm and BMEP = 2 bar. The outlet spread was additionally modified for the natural aspirated engine. What can be seen is that a decrease in the PMEP directly reduces the specific fuel consumption. This occurs independently of the outlet spread or the chosen engine concept. An additional increase of the outlet spread leads to a reduced brake specific fuel consumption at PMEP due to an increased residual gas fraction and an increased effective compression ratio. The internal residual gas fraction is mainly controlled by the outlet spread, in detail by the timing of exhaust valve closing in throttle free engine mode [11]. In the tests the amount of internal residual gas fraction was approximately 19 % at an outlet spread of 65° CA and about 14 % at an outlet spread of von 80° CA.

In ❻ is shown the brake specific fuel consumption over the inlet spread of the CVVL-T-MPI concept with two different outlet spreads for an engine operating point at n = 2000 rpm and BMEP = 2 bar. Also inlet valve lift phasing was evaluated. As a result of the throttle-free load control the brake specific fuel consumption decreases by 12 % in this concept in contrast to a turbocharged gasoline engine in throttled mode. On the one hand, the decrease of fuel consumption is due to reduced losses in the charge cycle work. On the other hand, better mixture preparations as result of smaller inlet valve lifts and higher flow velocity in the valve gap are responsible for the decrease in fuel consumption. It becomes clear that when the inlet spread is earlier modified, the specific fuel consumption continuously decreases.

❼ displays the CVVL-T-DI concept's specific fuel consumption over the inlet spread with two different outlet spreads for the same engine operating point at n = 2000 rpm and BMEP = 2 bar as previously used. It is here again that inlet valve lift phasing was utilized with a small outlet spread. Due to throttle-free load control, the specific fuel consumption decreases in this concept by 9 % in contrast to a charged gasoline engine with throttled engine mode. The lowest fuel consumption can be observed at an outlet spread of 60° CA and an inlet spread of 75° CA with inlet valve lift phasing. Whilst a further reduction of the inlet spread led to a decrease of the charge cycle work, the interior mixture preparation made additional cuts in the fuel consumption impossible. That the reduction in fuel consumption was 3 % lower than in CVVL-T-MPI concept is on the one side attributable to the higher friction losses as a result of the high pressure pump. On the other side, this lower percentage goes back to the insufficient mixture preparation due to direct fuel injection. An increase in the injection pressure as well as a further increase of in cylinder flow may very well bring about further advantages concerning the mixture preparation and the combustion. The question whether an increase in the compression ratio that can be accomplished through direct fuel injection due to interior fuel condensation might lead to further reductions of the fuel consumption will be investigated in future research. The the potential of higher in cylinder charge motion to improve the mixture preparation with the help of fully variable valve trains is also of interest for future research.

5 CONCLUSIONS

❽ shows an estimation of the potential to reduce fuel consumption of different engine concepts in real world driving and in the NEDC compared to a modern vehicle with gasoline engine and PFI. The different technologies were evaluated according to the load spectrum which results from the NEDC. The real fuel consumption is on a higher level depending on the driving characteristic and the higher power demand although the efficiency of the ICE is on a higher



❽ Estimation of the fuel consumption benefit of different engine concepts

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level [12]. With further development of the SI engines a decrease of the fuel consumption of 30 % to 50 % is possible. Corresponding values can be found in [13].

Downsizing as a method to increase the engine efficiency found the way in series application of many automotive manufacturers. Additional technologies like GDI and VVA will help to further increase the efficiency and the response behavior. With homogeneous mixture preparation the maximum torque can be increased especially at low engine speeds with boosting, VVA and GDI. With a VVA a filling optimized IVC can be realized at every engine operating point. At part load the valve timing and the valve lift can be optimized to reduce the pumping losses. So the fuel consumption can be reduced with boosting and GDI compared to the initial throttled engine of about 9%, with PFI of about 12% in which the reduced inlet valve lift enhances the mixture preparation. With GDI the friction losses are higher. The mixture preparation with DI can be enhanced by optimizing the injector location, injection parameters and increasing the injection pressure. These results shows, that with combining innovative combustion systems with a mechanical VVA the efficiency of ICE can be increased significantly. The effects of additional variability's like a variable compression ratio is also analyzed at both research facilities [8]. Analyzing the additional potentials is an exciting development and research task for future internal combustion engines.

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COMBUSTION PROCESSES FOR A LOW PRESSURE GASOLINE DIRECT INJECTION

The gasoline direct injection becomes more and more important for small engines used in passenger cars. Lowering injection pressure and thus costs could enable the introduction of this technology in the small engine segment offering an alternative to today's MPFI engines. The results of these investigations made at TU Graz show the potential of low pressure direct injection in terms of improvement over standard MPFI engines (fuel economy, emissions, and engine operation stability) and reduced costs compared to high pressure GDI systems.

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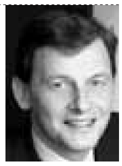

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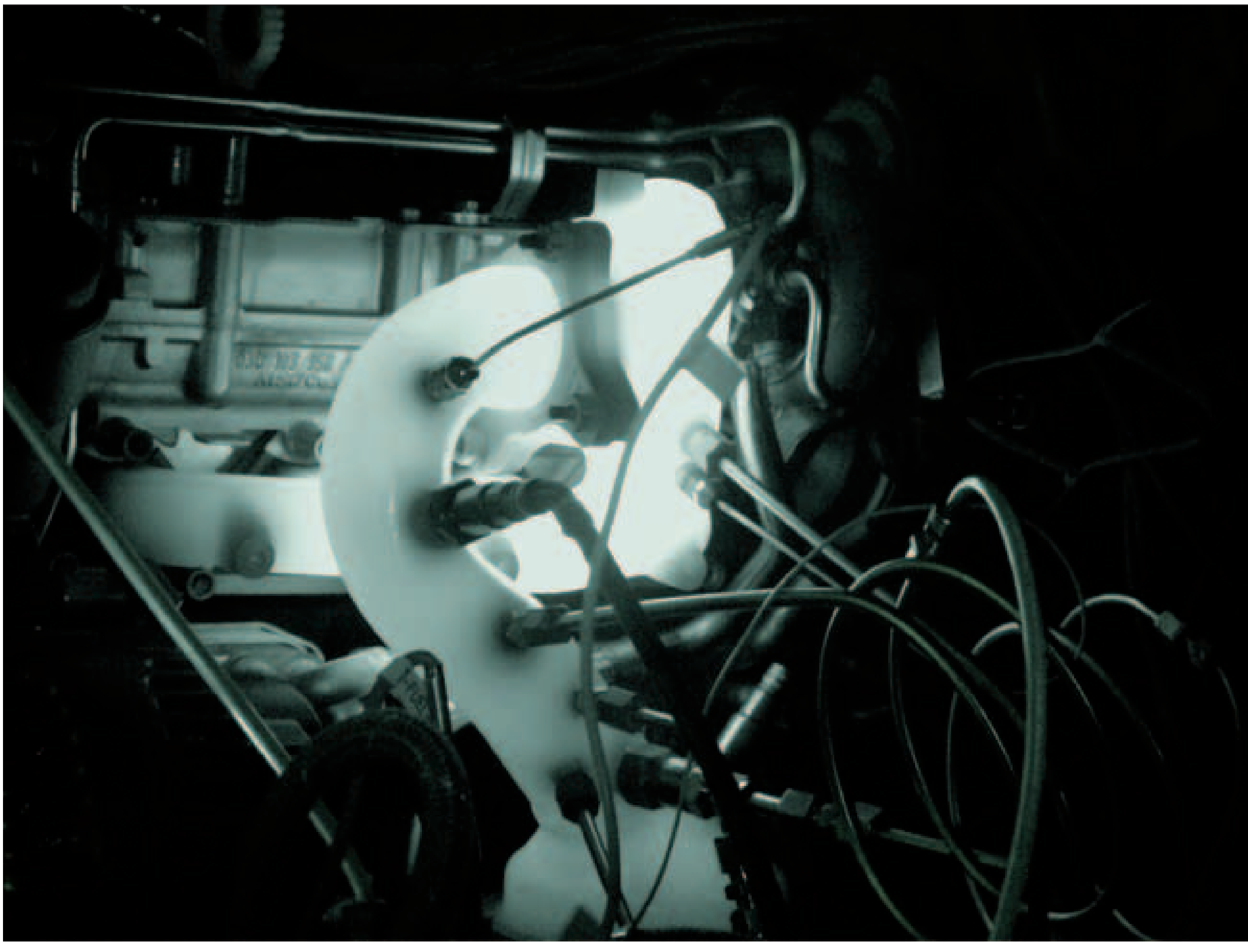

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1 REDUCED FUEL CONSUMPTION FOR SMALL VEHICLES

In Europe the share of automobile vehicles powered by GDI (Gasoline Direct Injection) engines is relatively high and is expected to grow further. Especially the combination of direct injection with turbo charging is well established because this technology allows for downsizing the engine capacity and therefore reducing the CO_2 emission. Smaller total engine displacements appear on the market and with this trend GDI technologies are penetrating also into the smaller vehicle segments.

Besides the downsizing strategy, several other technologies like variable valve timing, spray guided internal mixture formation or HCCI (Homogeneous Charge Compression Ignition) have been investigated and developed with the goal of CO_2 reduction. All these technologies including GDI are of high potential for fuel consumption reduction but they are expensive. Although the GDI technology would reduce the CO_2 emissions of today's MPFI engines (Multi-Point Fuel Injection) in the small vehicle class, due to the intense pressure on costs within this segment, the use of these high-priced technologies is delayed or in fact may not happen. Therefore an alternative strategy for the low cost small engines is of interest. The lowering of injection pressure of GDI engines is such a possibility. The high pressure pump, injectors, rail and the high pressure connectors are expensive components within the DI system and their costs are at least partially dependant on the maximum system pressure.

A GDI system for smaller vehicle classes using significant lower injection pressure could be a competitive alternative to today's MPFI engines offering reduced fuel consumption and emissions compared to port fuel injection. The purpose of this study is to investigate such a novel GDI system with low injection pressure derived from a standard automotive high pressure GDI system. The main focus is put on the question of the minimum injection pressure with respect to fuel consumption, emission and engine operation stability.

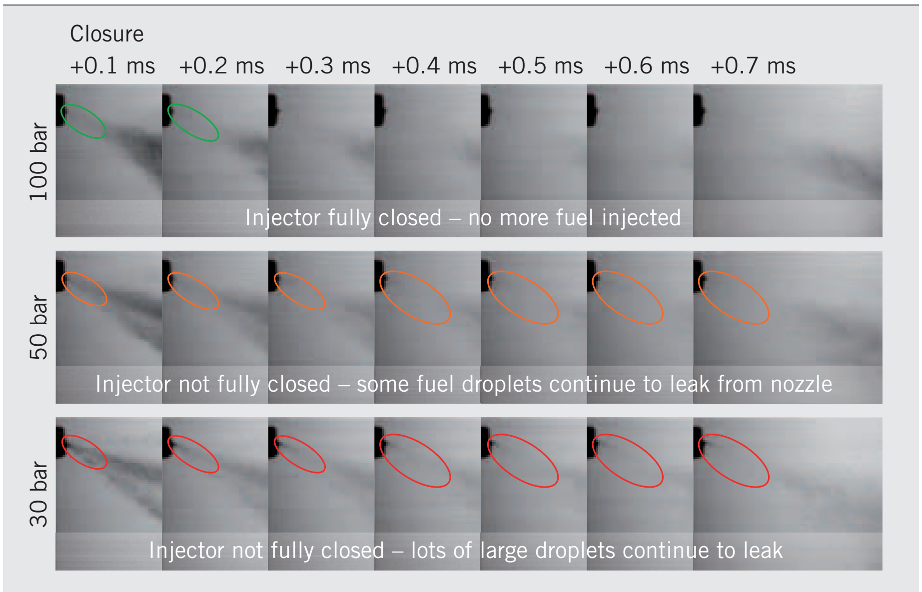
2 APPROACH

The following investigations have been conducted on a 1.4 l series production engine fulfilling Euro 4 emission limits. The engine is turbo charged, equipped with an inlet cam phaser and uses high pressure direct injection (max. 100 bar) with a homogenous combustion concept (stoichiometric).

In order to assess the potential of the LPDI (Low Pressure Direct Injection) combustion system, two benchmark combustion systems were defined. One is the underlying standard high pressure GDI system. The target engine segment is that of MPFI engines, a typical representative of this engine class was used as the second benchmark. Based on these benchmark engines, acceptable limits for emission and fuel consumption have been defined for the investigations with low pressure injection. Due to the easier handling of experimental results this publication mainly shows comparisons between HPDI (High Pressure Direct Injection) benchmark and LPDI combustion system.

3 CHARACTERISTICS OF LOW PRESSURE INJECTION

Prior to the layout of the low injection pressure DI combustion system, a profound knowledge of the behaviour of the injection spray with lowered injection pressure has to be gained. As the mixture formation of DI engines is considerably influenced by the injection pressure driven spray breakup and the transport of fuel via the charge motion [1], the investigations of these effects were conducted first. Reasonably, the models of the 3D CFD simulations have to be calibrated



❶ Injector closing at different injection pressure

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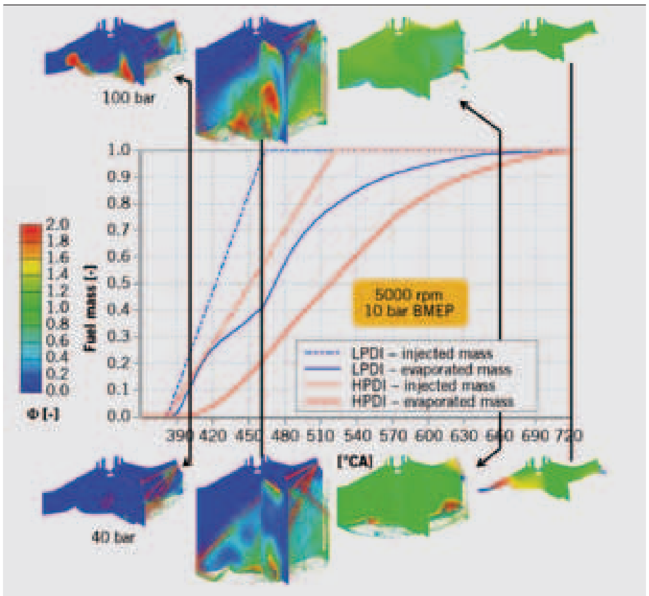


❷ Comparison of mixture formation for high and low pressure injection

to injection pressure [2]. Spray measurements of non-modified automotive injectors with a high speed camera have been performed for the calibration of CFD models and for a detailed understanding of the quality of the spray breakup at different injection pressures.

The higher relative velocity between spray and air results into a faster atomization and better evaporation of the high pressure injection compared to low pressure injection. Characteristic for the low pressure injection is the distinct liquid core at the nozzle area, large droplets around the jet tips and a reduced penetration depth [3]. Characteristic for the high pressure injection is the fine atomisation beside the jets due to the high relative velocity. At DI engines the injection has to be adjusted to the charge motion in order to reach a sufficient mixture formation and avoid cocking. Due to the different spray breakup behaviour and penetration depth of the low pressure injection the engine layout (like spray targeting, port design or piston bowl) has to be adjusted for an LPDI system. The interaction of the fuel jets with the charge motion in the cylinder is the main influencing factor for the mixture preparation of LPDI.

The injector closing behaviour also depends on the injection pressure. The tested series production injector is designed to work typically with pressures around 100 bar. At low injection pressure the closing behaviour is changed and it is speculated that the needle closes more slowly. Certainly large, slow droplets are observed, ❶. Due to changed spray breakup and the closing behaviour of the injector, the low injection pressure results in larger and slower droplets. This affects the mixture formation negatively.

Possible improvements could be: more and smaller holes, an adapted closing and higher charge motion and turbulence in the cylinder. These changes have not explicitly been investigated during this study, but a re-optimized standard GDI injector showed considerably better results for injector closing.

4 ASSESSMENT METHODOLOGY OF THE MODIFIED COMBUSTION SYSTEM

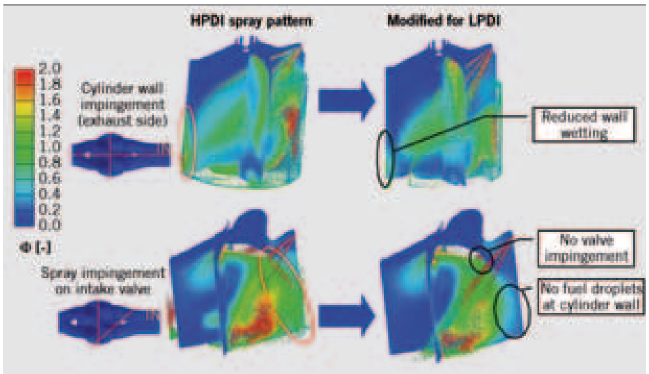
The layout of a direct injection system is very complex due to the numerous parameters which influence the mixture preparation. During this research project comprehensive 3D-CFD calculations were performed in order to understand the local processes and the behaviour of a DI concept. Various engine hardware parameters (like piston bowl geometry, spray pattern or injection pressure) were investigated by the CFD simulation focusing on the three operational points: 2000 rpm and 2 bar BMEP, 2500 rpm and 10 bar BMEP as well as 5000 rpm and 10 bar BMEP. WOT operational points have been additionally calculated. In a first step the influence of injection pressure using standard automotive hardware was analysed [6]. Afterwards different spray targets were developed for low pressure injection by the use of 3D-CFD investigations and in the next step different piston bowl geometries were investigated. For the assessment of the CFD results several criteria were defined:

: subjective 3D-homogenity at ignition timing: indicates the overall quality of mixture formation
: equivalence ratio around spark plug at ignition timing (0D result): indicates the stability of combustion, robustness and emission
: non evaporated fuel at ignition timing (0D result): indicates emission, smoke and fuel consumption as well as under-/overmixing
: equivalent wall film factor for the cylinder (integral of 0D result): indicates oil dilution, smoke and unburnt emission components
: equivalent wall film factor for the piston (integral of 0D result): indicates smoke emission and unburnt emission components
: further subjective criteria: fuel droplets at injector area (coking).

Based on these assessment criteria the engine hardware with highest potential was chosen for prototyping and testing on engine test bench and verification of CFD simulation. Particular attention was put on the cylinder wall wetting and the wetting of the piston surface with fuel. The cylinder wall and piston surface wetting represents a source of smoke formation and oil dilution [5].

5 PRE-LAYOUT BY SIMULATION

The injector targeting was based on the series production high pressure injector, using the same nozzle hole diameter. The jets of the high pressure injector are distributed over the whole combustion chamber in order to get an equal distribution of fuel in the cylinder.



❸ Improved wall wetting and improved spray-valve contact

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Boundaries						Results				
Engine speed	BMEP	Injection pressure	Injector	Piston	Calibration	Homogenisation 3D results (subjective)	Equi. ratio spark plug at ignition	Non evaporated fuel at ignition	Wall film factor/ liner	Wall film factor/ piston
rpm	bar	bar	–	–	–	–	–	% of injected	–	–
2000	2	HPDI	Series production	Series production	Series production	Hom., isolated droplets	0.69	0.11	2.63	1.64
2500	10					Very inhom., many droplets at int. side	1.29	0.82	7.65	5.50
5000	10					Hom., isolated droplets	0.91	0.02	7.13	2.14
2000	2	LPDI	Modified	Series production	Modified	Hom., isolated droplets	0.79	0.14	5.01	2.29
2500	10					Hom., droplets at intake side	1.02	0.51	6.43	4.09
5000	10					Hom., droplets at int. and exh. side	1.08	0.74	4.16	2.03

Modified

Good
Medium
Worse

❹ Matrix of CFD results

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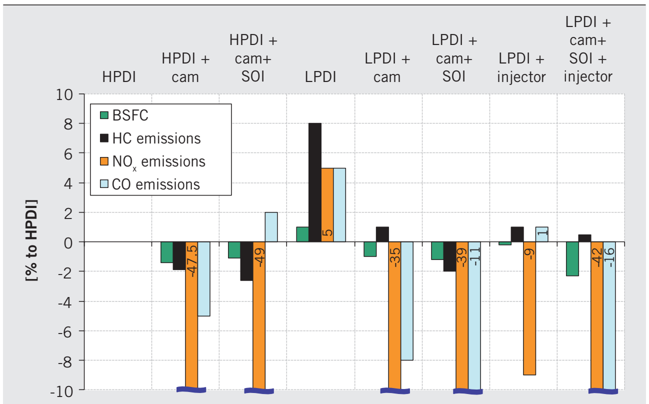
As the injection starts early, the piston is then close to the cylinder head and therefore the original spray pattern has a small inclination to reduce the contact with the piston bowl. This spray pattern leads to a very good mixture formation with high injection pressure. With low injection pressure (longer injection time), the jets cannot locate the fuel in the preferred area of evaporation in the middle of the combustion chamber [5]. Consequently the fuel is not equally distributed in the cylinder, ❷.

Three jets of the series production injector with lower injection pressure hit the liner during injection. The increased contact of the fuel with the cylinder wall leads to higher emissions, high fuel consumption, and oil dilution. Additionally, two jets of the original spray pattern collide with the intake valves during injection, ❸ (left). At high injection pressure this has little influence on the mixture formation, but with low pressure the collision has an intensified impact on mixture formation.

As an improvement a spray pattern was developed where the jets are directed farther towards the piston top, so that the wall is only hit at the end of very long injection times, ③ (top). Another positive effect of the higher inclination is that the jets are better captured by the tumble at high turbulence operational points (high rpm and high load). Nevertheless, the transport of the larger droplets caused by lower pressure to the cylinder wall cannot be completely avoided without modifications of the intake port geometry. Another modification is that the jet of the new spray pattern avoids the contact with the valve and the upwards oriented air flow beneath the valve again supports evaporation, ③ (bottom).

The previous spray investigations showed a major influence of charge motion on the mixture formation at low pressure injection. Mixture formation is less influenced by wall-spray guiding at low pressure direct injection but is primarily depending on in-cylinder flow velocity and direction. Several piston bowl variants have been investigated via CFD in order to improve the fuel transport and charge motion. Generally the design of the spray pattern showed a more intense impact on the combustion system as the design of the piston bowl geometry. The piston bowl design significantly influences operational points such as catalyst heating (homogenous split injection) but has little influence on higher rpm and load points.

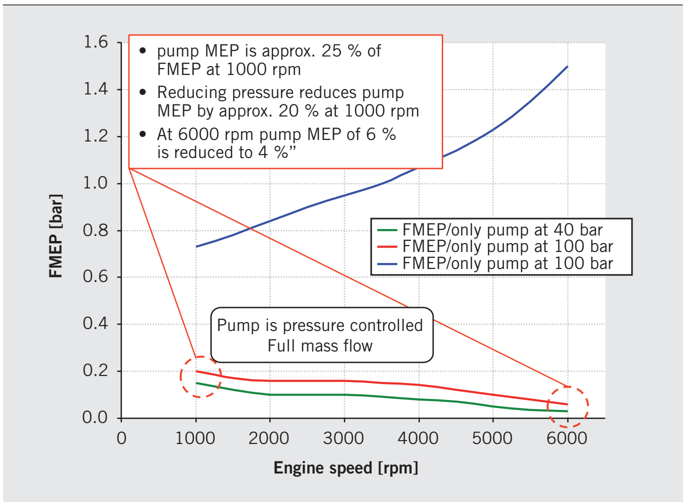
A modification of both the intake port and the piston bowl can improve the mixture formation of low pressure direct injection. Using standard intake port geometry, the series production piston bowl geometry showed best results of all investigated variants for low pressure mixture formation.



❺ Experimental results 2000 rpm and 2bar BMEP

6 EXPERIMENTAL INVESTIGATIONS

For verification experimental investigations with modified hardware have been conducted. The first experimental investigations [4] were carried out with the standard engine hardware and unchanged engine components. This includes: solenoid injector with six jets, side-positioned injector, standard high pressure pump, and cylinder head. The concluding experimental investigations, which are presented in this publication, have been performed with an improved combustion system and series pro-



❻ Friction mean effective pressure of the fuel pump for high and low pressure

duction calibration as well as improved low pressure calibration (SOI, intake cam phasing and ignition timing).

The results of the CFD investigations of several hardware modifications and engine calibrations were evaluated using the assessment criteria and are displayed in a matrix. ❹ shows this matrix for the final hardware. The comparison of high and low pressure direct injection shows disadvantages and benefits for both pressure levels. At the operation point, 2500 rpm and 10 bar, both LPDI and HPDI show rather high piston impingement resulting in high wall film factors.

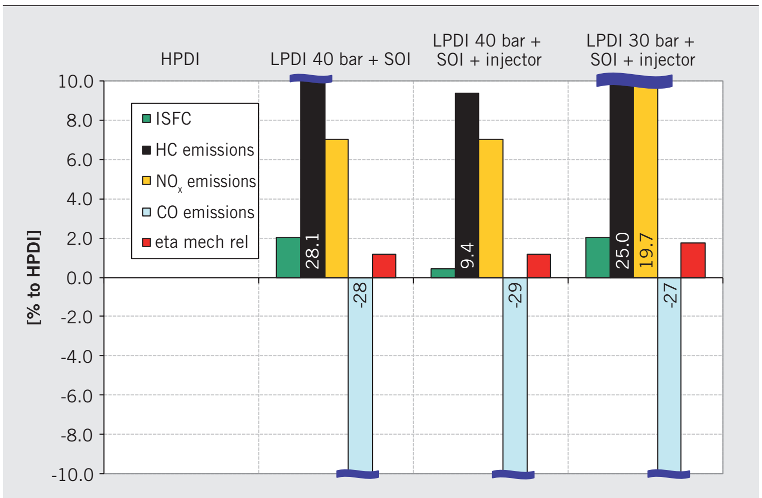
7 EMISSION AND FUEL CONSUMPTION BEHAVIOUR

Exemplary for the operation map two demonstrative operational points are presented. The operation point 2000 rpm and 2 bar demonstrate fuel consumption and emission behaviour at part load area and 5000 rpm and 10 bar BMEP shows the behaviour at higher load and higher rpm. In the following, the emission and fuel consumption results of two hardware variants of LPDI and HPDI with optimum calibration settings are compared to the series production engine at 2000 rpm and 2 bar BMEP. CFD results, ④, indicate increased HC and smoke emission due to increased wall film of the liner and piston. This trend has been verified by experiment using unmodified engine setting and hardware.

Equal or slightly worse (but within the defined limits) HC emissions are reached by using the prototype injector in combination with the series production piston at 40 bar injection pressure and modified injection timing. The modified intake cam phasing, realizing more EGR, is the main parameter for the improvement of NO_x emissions, this is true for high and low pressure injection, ❺. Adjusting the injection timing resulted in decreased CO emissions.

Smoke and COV_{IMEP} showed equal or only slightly degraded results for the warm engine and low load at all injection pressures and therefore are not displayed in a separate figure. For the cold engine and higher loads, the smoke emission rises with lower injection pressure, beginning with 20 to 30 bar, till 30 bar FSN is below 0.1 [6].

All variants showed improved fuel consumption due to modified cam phasing (more EGR) and earlier injection timing. The ISFC of the prototype injectors in combination with adjusted calibration



❼ Experimental results 5000 rpm and 10 bar BMEP

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reach the ISFC level of the series production engine. Due to the improved mechanical efficiency resulting from the lower required power to drive the GDI pump at lower injection pressures, the LPDI results in better BSFC than the production HPDI engine.

The mechanical efficiency is both positively and negatively influenced due to a reduction of injection pressure. The lower the fuel pressure, the less pump work is required but concurrently the injection time increases at low pressure and thus the injector requires more energy. The sum of these two effects was positive, ❻, whereas a none mass flow but pressure controlled pump was used. Depending on the pump and injector design further factors influence the mechanical efficiency. Due to the layout of the HPDI pump, which is designed for a considerably higher pressure level than used for the LPDI, it can be assumed that a modified pump layout would gain additional benefits regarding BSFC. Specialized low pressure GDI injectors using low voltage/current could further improve the mechanical efficiency. For several operating points a mass flow controlled pump was used and showed similar results.

At 5000 rpm and 10 bar BMEP the production engine has the lowest ISFC compared to the variants at low pressure injection, ❼. Using modified calibration parameters and considering the better mechanical efficiency the BSFC is similar to the level of the series engine. The reduced injection pressure results in considerably higher HC and NO_x emissions for the high load/speed point for all hardware variants, even with modified calibration settings, while the CO emission is reduced.

8 OIL DILUTION

Oil dilution (fuel enters the crankcase and contaminates the motor oil) exceeding a certain level is harmful to the engine and its components. The influence of injection pressure on oil dilution was investigated using the stationary operational point 2500 rpm and 5 bar BMEP in cold (40 °C coolant temperature) and warm (95 °C) condition. The test duration was 3 h and after each 1 h operation time an oil sample was taken. The oil samples have been chemically analysed regarding their gasoline fraction and viscosity. Three different hardware variants were investigated using the series production calibration.

Tests have been performed at high and low pressure injection and in warm and cold condition. In warm condition no influence of injection pressure on oil dilution was recognizable. The standard HPDI engine operated at cold engine conditions showed a considerable oil dilution of 4 % vol. fraction, the lowered injection pressure of 55 % increased the fraction to 6.5 %. Using the prototype injector and the series piston with low pressure at cold engine condition lowered the level of oil dilution to similar values of the HPDI system. Using the modified piston variant had negative effects on the oil dilution. ❽ shows the comparison of experiment in terms of oil dilution gradient [%/h] and the wall film factor calculated by the CFD simulation.

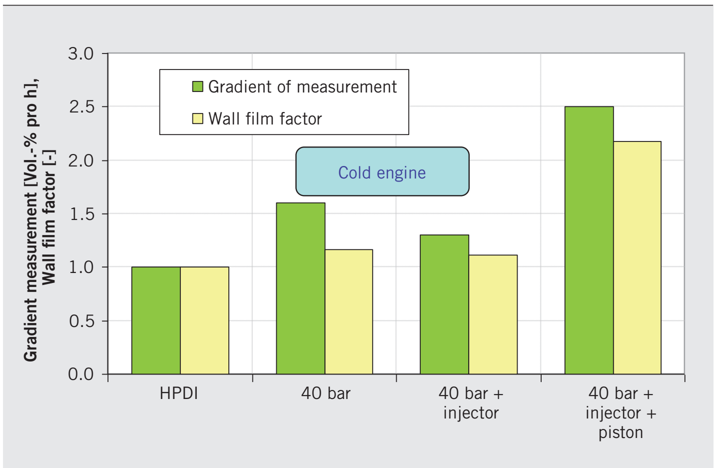
9 CONCLUSION

As the trend to smaller vehicle classes requires low cost but high efficient powertrain systems, the transfer of modern engine technologies from top vehicle classes becomes more important. In order to realize the potential of SI direct injection combustion systems but avoid the cost of a high pressure system, this study aimed the development of a low pressure direct injection system. A reduction of injection pressure induces both positive and negative effects on the internal combustion engine. On the one hand mechanical losses are positively influenced by the use of a lower injection pressure as well as by the decreasing demand of fuel pressure thus leading to a possible reduction of costs for the fuel injection and delivery system. On the other hand atomization and mixture preparation is negatively influenced.

For this study, the system was based on the hardware of an automotive standard high pressure direct injection combustion system. The development was performed using 3-D CFD investigations and experimental tests from the baseline and the new combustion system. The results showed that using a reduced injection pressure direct injection can improve engine out emission, BSFC as well as operation stability of standard MPFI engines. In order to utilize the potential, the following modifications had been considered:

: injector layout
: spray pattern and nozzle flow
: high charge motion is required.

The "acceptable" lowering of the injection pressure is highly influenced by the engine load and temperature; especially fuel consumption and HC emissions are deteriorated at higher loads compared to the high pressure injection. The injection pressure



❽ Comparison of simulation and measurement gradient of oil dilution

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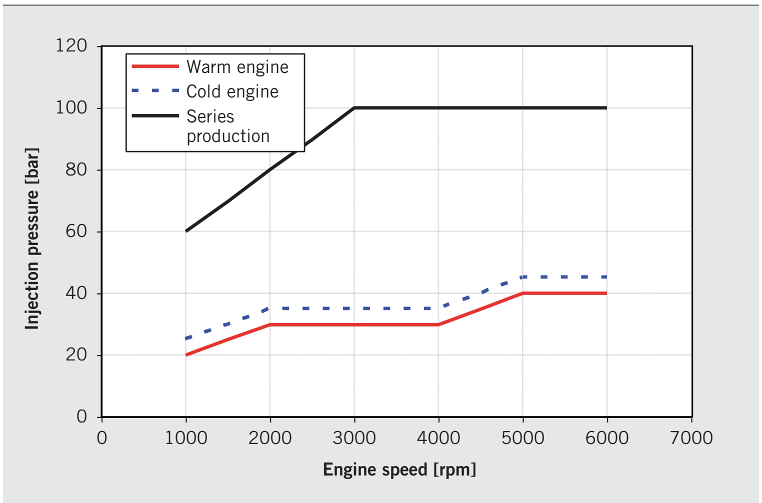


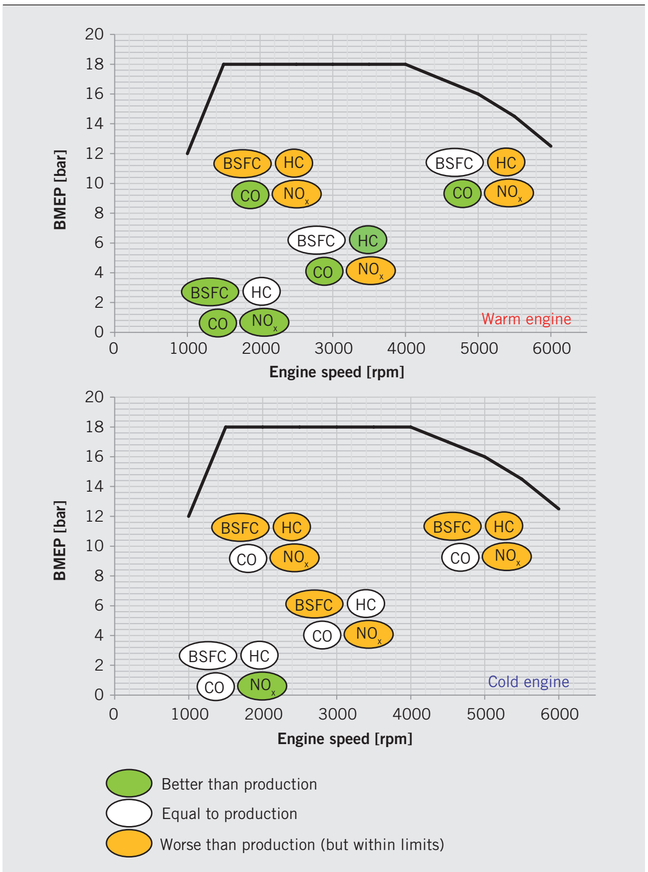
❾ Final injection pressure for the LPDI system

does not influence the smoke emission when limiting the pressure to 20 to 30 bar. Beneath that level the smoke emission increases steeply. The combustion stability is not influenced by the injection pressure due to the high turbulence level of the engine. The modified spray pattern (prototype injectors) showed best results at low injection pressure. Via modifying the piston bowl geometry no further improvements were possible. The CFD results indicate that the layout of piston bowl should be adjusted to the intake port geometry. Modifications of both, derived from an existing HPDI system, are of high potential to improve a low injection pressure combustion system.

❾ displays the final injection pressure target and ❿ shows the general trends of emissions and fuel consumption based on this injection pressure level and the necessary calibration parameter adjustments. The level of injection pressure depends on the engine speed and load. In cold condition the required injection pressure level is approximately 5 bar higher. The results are based on the prototype injectors in combination with the series production pistons and with final calibration settings.

Due to the effect of reduced atomization with lower pressure, intense charge motion and a high turbulence level are required for sufficient mixture formation. The turbulence level of the investigated LPDI system is high. With a lower turbulence level, as used in MPFI engines, a deterioration of mixture formation is presumable.

The results indicate that a low pressure GDI injection system offers benefits regarding emission reduction (compared to standard MPFI engines) as well as good performance/cost trade off. These advantages combined with better transient and catalyst warm up behaviour as well as excellent supercharge ability could be an interesting concept for tomorrow's small and downsized engines.



❿ Results of experimental investigation – final hardware and calibration settings

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