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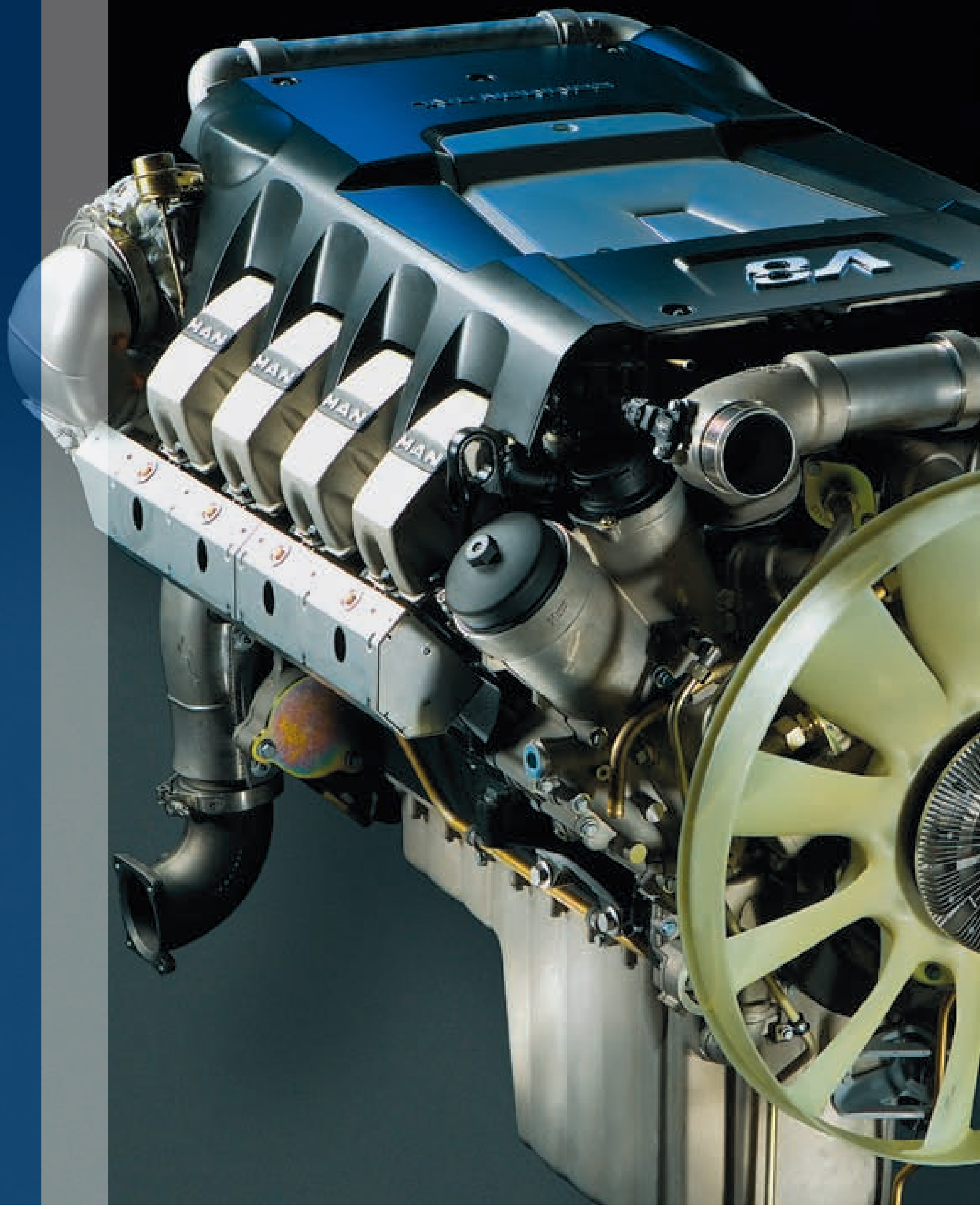
Innovations in Commercial Vehicle Engines



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COVER STORY

Innovations in Commercial Vehicle Engines

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Commercial Vehicle Engines have to meet constantly increasing requirements regarding fuel consumption and emissions. Vehicle manufacturers and their suppliers are working on innovations that will make the commercial vehicle of the future more fuel-efficient and cleaner.

AGR versus SCR

Dear Reader,

2012 will see the introduction of the Euro 6 standard for commercial vehicles. It will be very interesting to see, which technical solutions commercial vehicle manufacturers and their suppliers will use to comply with the new emissions limits. Until recently, the SCR system was seen as the ultimate solution. It not only removed NO_x from the exhaust but also made the vehicle more economical, as the engine could be designed for lower fuel consumption.

But now, Behr and AVL have carried out joint research into low-temperature EGR. Their findings show that two-stage cooled EGR has the potential to meet the Euro 6 standard. Further optimisations and developments are, of course, still required on the way to possible series production readiness. The supporters of SCR point to the high cooling performance for the EGR system, which can amount to up to 70 % of engine output, as well as the higher fuel consumption. According to their calculations, SCR pays off for vehicle operators after only a few years due to the savings in fuel consumption. As is so often the case, the truth is probably somewhere in the middle, and the best solution will be a combination of both technologies. In other words: the lowest possible engine-out emissions due to the sensible use of EGR and then aftertreatment with a simple and low-cost SCR system.

This example once again shows that it is better for politicians to prescribe ambitious targets rather than to declare and stipulate specific technologies as a standard. After all, no-one would continue researching into EGR systems if the SCR system were legally prescribed. Competition among different systems is always the best corrective. I am pleased that MTZ can continue to focus on presenting technical innovations rather than technical regulations.

Yours sincerely

Richard Backhaus

Richard Backhaus
Wiesbaden, 23 July 2008

Richard Backhaus
Chief Correspondent MTZ

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Prof. Dr.-Ing. E. h. Heinrich Buschmann and
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Organ of the Fachverband Motoren und Systeme im VDMA, Verband Deutscher Maschinen- und Anlagenbau e. V., Frankfurt/Main, for the areas combustion engines and gas turbines

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Cooperation with the STG, Schiffbautechnische Gesellschaft e. V., Hamburg, in the area of ship drives by combustion engines

The New V8 Diesel Engine from MAN

MAN has developed a new V8 engine for the 16-l class with an output of 500 kW and 3,000 Nm of torque for its TGX and TGS ranges of heavy trucks. To reduce NO_x, MAN has applied an SCR system with AdBlue injection. This article describes the engine concept, the design of the main components, the development of vehicle-specific add-on parts and the work carried out to optimise the combustion system and exhaust aftertreatment.

1 Introduction

The highlight of the new TGX and TGS series of heavy trucks is the new V8 engine for the very highest powerband requirements. This eight-cylinder V engine puts out 500 kW with a maximum torque of 3000 Nm or 2700 Nm, depending on engine characteristic, for long-haul or heavy-duty transport.

The newly evolved V8 engine (internal designation: D2868) is one of the best achievers in the class of 16-l commercial-vehicle engines. At this time, the TGX V8 from MAN is the most powerful series-production truck in Europe.

The new V8 engine ensures peak average speeds, even when route topography is at its most challenging. This is also the key to economy in goods-vehicle logistics. The chief strength of the TGX V8 lies in time-critical freight forwarding in international long-haul transportation. The V8 engine utilises the MAN AdBlue exhaust system, so pollutant emissions are compliant with Euro 5, the strictest limits now in place.

2 Design Concept

The V8 is the first in a new series of V engines based on a design concept developed by MAN over a period dating back to 2001. These new engines will gradually replace the current MAN V engines in all applications.

The D2868 LF is a radically new development and was designed specifically for the new TGX-series vehicles. The design brief stipulated that all components had to be engine-mounted without necessitating major alterations to the engine-to-vehicle interfaces vis-à-vis the requirements for installation of an inline engine. There were no compromises that could disadvantage the customer: usable space in-cab and on-frame remains unchanged with the V8 installed.

Designed for peak pressures as high as 240 bar, the D2868 has potential for high specific power and compliance with future emissions-control standards. The brief also called for compact size and minimal weight. The **Table** lists the principal dimensions.

As regards many of the modules it integrates, the V8 engine is an evolution of developments previously incorporated into the straight-six engines (D20/D26). This translated into shorter development times and reduced development costs. Parts shared with the tried-and-tested D20 and D26 engines also mean benefits in terms of stocking spares.

Table: Main dimensions

main dimensions	bore	128	mm
	stroke	157	mm
	displacement	16.16	l
	cylinder capacity	2.02	l
	compression ratio	19	–
crankcase	cylinder offset	170	mm
	block height	440	mm
connecting rod	con-rod length	282.5	mm
	con-rod ratio	0.277	–
main bearing	diameter	112	mm
	width	28.8	mm
connecting rod bearing	diameter	96	mm
	width	31.7	mm
piston	compression height	79	mm
	fire land height	12	mm
	pin diameter	55	mm
	pin length	80	mm

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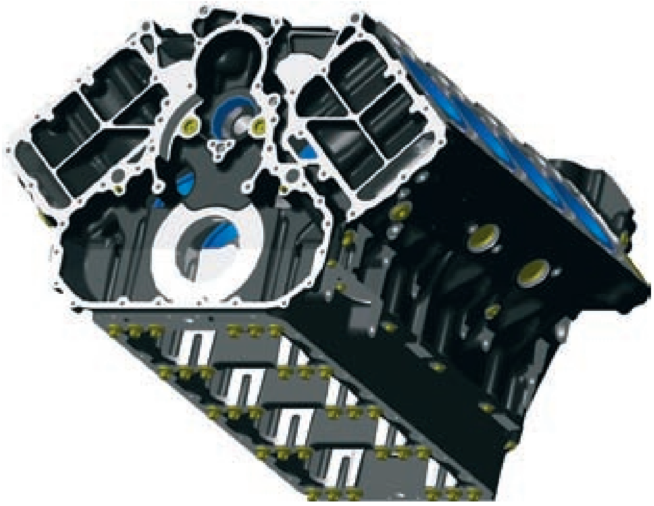


Figure 1: Crankcase

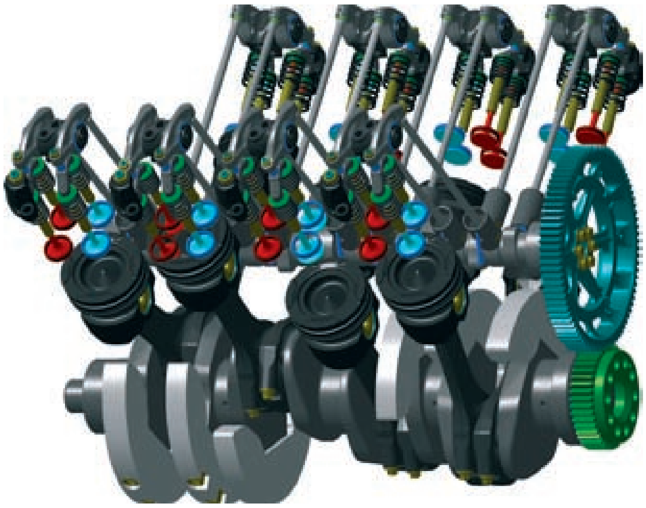


Figure 2: Crankgear and valve gear

However, the specifics of a V configuration combined with the high requirements applicable to specific power and durability necessitated new departures in the design of certain key components. Major subassemblies of the new V engine are outlined below.

3 Basic Engine

In 2002, a joint-venture contract for the development of the basic-engine components was signed with Liebherr. MAN also co-operates with Liebherr on purchasing and production of the shared components. Each company bears responsibility for assembly in its own engine plant.

3.1 Crankcase

To meet the high requirements inherent to a design capable of dealing with ignition pressures up to 240 bar, this MAN engine is the company's first implementation of a bedplate design, see **Figure 1**. This design dispenses with the cross-bolting common to most V-engine configurations. At the same time, the bedplate maximises engine rigidity for absorption of the ignition forces.

High-strength materials are used to keep the engine block compact. The crankcase is a GJV casting, the bedplate is made of GJS. Finite-element analysis was employed to study and optimise the structure of the block. The chosen materials are adopted specifically to achieve maximum strength at the locations where operating loads are at their highest. In addition to building in strength and rigidity, another important aim in design development was to optimise the routing of the oil and coolant galleries in the engine block.

3.2 Running Gear

The rigid crankshaft, **Figure 2**, of the V8 has six bolt-on counterweights for full balance, and in combination with the uniform ignition spacing this ensures smooth engine operation. The firing sequence is the same as that proven in other MAN V8 engines.

The precision-forged conrods are made of high-strength steel and are delivered to MAN for cracking and finishing. Modern-design steel pistons are used in the new V engines on account of the stringent requirements for power and durability. The one-piece forged pistons have an internal cooling duct. The special-cast-iron cylinder liners are of the top-stop design and are fully enveloped by the coolant jackets.

3.3 Cylinder Head and Valve Gear

A new single-cylinder head, **Figure 3**, was developed, so that the same component

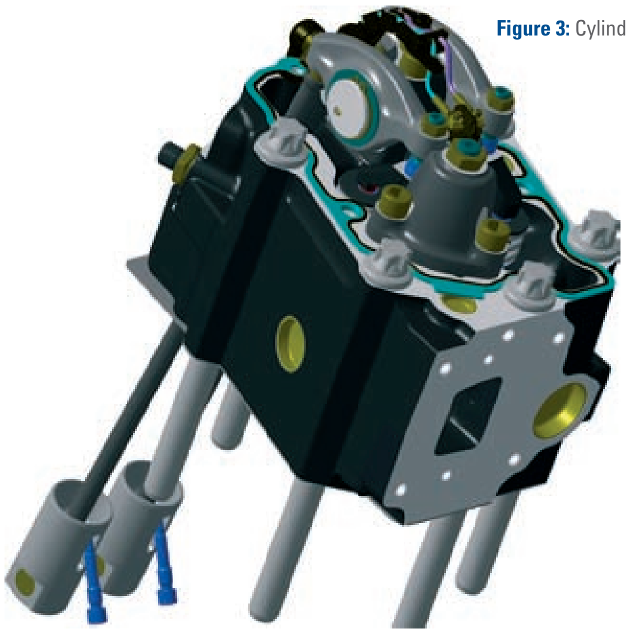


Figure 3: Cylinder head

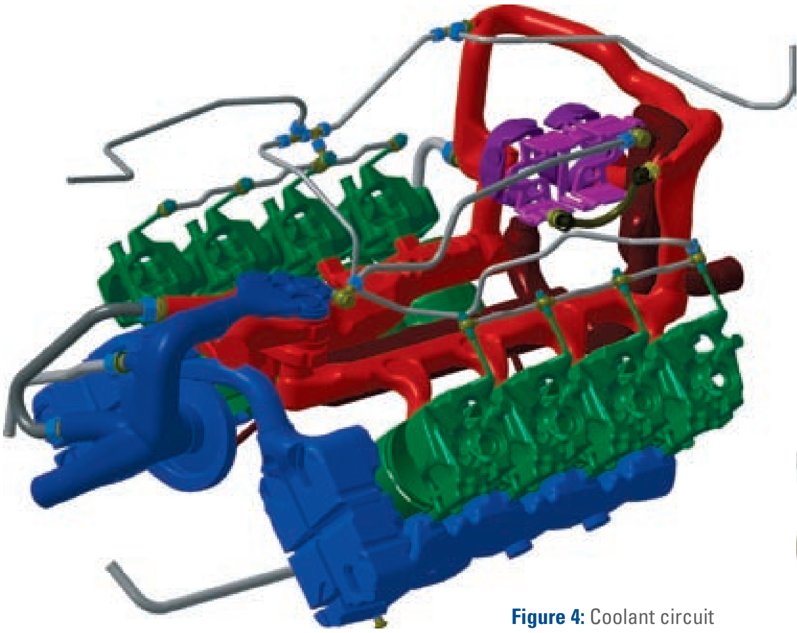


Figure 4: Coolant circuit

Figure 5: Oil module (left-hand cylinder bank)

could be used for all applications across the entire range of new V engines. The fastener concept spaces the six cylinder-head bolts uniformly around the combustion chamber, while allowing the cylinder heads to be positioned closely beside each other. The cylinder-head walls are high and stiff, so the hold-down force applied by the threaded fasteners is utilised to the best possible effect to contain the high ignition pressures.

The rigidity of the cylinder-head/cylinder-block construct permits a single-thickness cylinder-head gasket to be used. Elastomeric lips dependably seal the coolant and engine-oil penetrations.

The valve star is offset vis-à-vis the pattern of cylinder-head threaded fasteners, enabling the intake and exhaust ducts to be routed for optimised flow. The exhaust valve brake (EVB) required for certain vehicles is an option that can be mounted on the cylinder head.

There are four valves per cylinder, actuated by an overhead camshaft. Roller tappets, pushrods and rocker arms are the components of the dependable, low-loss valve actuating train.

3.4 Coolant Circuit

The coolant circuit of the new V engines was designed to permit the requirements of the many and varied applications to be met optimally by modifications to no more than a minimal number of system components. **Figure 4** shows the coolant circuit for the vehicular engine.

The coolant pump is centred at the front of the engine and has two delivery ports, so the rate of supply is the same to each cylinder bank.

Side distribution channels and the sizing of the cross-sections along each cylinder bank ensure uniform distribution of the coolant from the first to the last cylinder. Flow through the cylinder heads and the crankcase block is parallel.

In each cylinder bank the coolant collects in a longitudinal gallery and, controlled by restrictors in the thermostat housing, is directed in full or part flow to the rear of the engine where it is used to cool mounted parts or ancillaries. The coolant ducted off at the rear of the engine is returned to the thermostat housing along a central coolant duct in the crankcase.

Delivery rates from 800 to 1600 l/min are needed for the various applications from the V8 vehicular engine to the V12 yacht engine. Compliance with this requirement is achieved by having different impeller sizes and two possible idler-gear transmission ratios.

The two-part idler gear in the geartrain to the coolant pump has built-in elastomeric elements that effectively decouple the pump‘s drive shaft from the rotary vibrations transmitted by the engine‘s running gear.

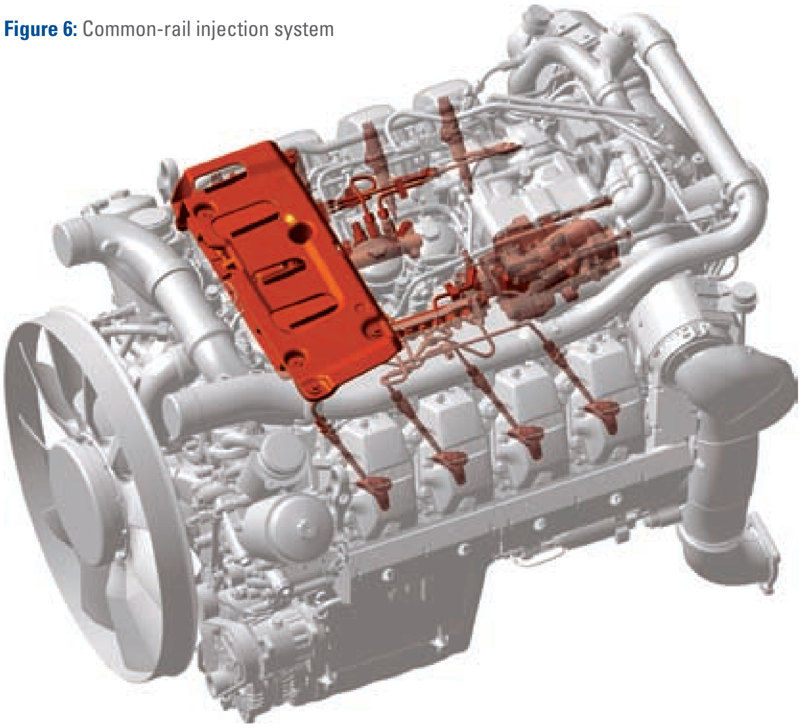
3.5 Oil Circuit

The new V engines feature two oil pumps, driven directly by the crankshaft and working in parallel. Geometry and design of the inner rotor and the eccentrically located hollow gear are taken from the D20 engine. Unlike the D20 design, however, each pump has a separate aluminium housing and both mount on the inside of the gearcase cover at the front of the engine.

Delivery rate is varied by changing the width of the impeller in the oil pump or by altering the transmission ratio in the drive from the crankshaft. In this way, each pump can be set to meet delivery-rate requirements in the range from 150 to 220 l/min.

The pumps carry the oil to the two oil modules, **Figure 5**, one at the end of each cylinder bank. The components for oil

Figure 6: Common-rail injection system



filtration, cooling, separation and pressure control are all integrated into these oil modules. In the coolant circuit, moreover, the oil modules are the interfaces between coolant pump and crankcase.

The plate-type heat exchangers for cooling the oil are at the bottom of the oil modules. Each oil module can accommodate one or two plate stacks of up to twelve plates; the configuration depends on the cooling power required and on the quantity of oil in circulation.

The cooled oil flows to the filter cartridges which – as is the case with the D20/D26 engine – are screw-fitted into the top of the oil module. A dump valve in the filter housing allows the engine oil to drain back into the oil pan to facilitate filter removal. Ports in the bottom of the oil module carry the streams of unfiltered, filtered and return oil from and to the gearcase cover.

Two drilled main oil galleries in the crankcase take the filtered engine oil to the oil spray nozzles for piston cooling and to the running-gear and valve-drive bearing seats. Pressure and volume equalisation of both main oil galleries is ensured by ducts to the crankshaft bearings. Consequently, a sensor in only one gallery is sufficient for oil-pressure monitoring.

Fine-oil separation from the engine's blow-by gases is by the maintenance-free cyclone separators downstream from the initial-separation chamber.

4 Vehicular Engine

On account of the new main dimensions of the basic engine on the one hand and the tight restrictions on space underneath the floor of the cab on the other, the injection system and the exhaust and charge-air systems were among the most important for which new designs had to be developed, and the same applied to the positioning of the ancillaries. In this context full compliance was necessary with the requirements for both long-distance haulage and heavy-duty transport up to 250 metric tons gross weight.

4.1 Injection System

The new V engine boasts a second-generation common-rail injection system, **Figure 6**. The system operates with injection pressures up to 1600 bar. These pressures are produced by an oil-lubricated CP3.4 high-pressure pump flange-mounted to the crankcase in the V between the cylinder banks. This pump is directly camshaft-driven via a transmission gear.

The compressed fuel initially enters the series-connected rails, each of which is assigned to a cylinder bank. The fuel is carried to the injectors by short injection pipes to the unions, one in the side of each cylinder. Two control units set underneath a step plate at the top front of the engine meter the supply of fuel to the individual cylinders. The control units operate in a master-slave configuration and without extra cooling.

The fuel service centre (Kraftstoffservicecenter, KSC) is located in the engine V in front of the high-pressure pump. In addition to the two fully incinerable filter cartridges of the D20, the KSC also integrates the port for the flame-start system and the heating element for fuel preheating.

4.2 Exhaust System and Charge-air System

In order to superimpose good drivability even at the low end of the engine rpm range and high peak power, a configuration with one exhaust turbocharger per cylinder bank was selected in the concept-design phase. This solution with two relatively small turbochargers also has advantages with regard to the restricted space available for the power plant and accessibility to the two auxiliary PTOs at the rear of the engine.

Charge air from the air filter arranged transversely behind the engine is channelled left and right to the compressors. The compressed-air flows come together on the left side of the engine, as viewed in the forward direction of travel. After flowing through the vehicle-mounted intercooler, the stream of charge air is split to the cylinder banks and flows through the two headers to the cylinders.

The standard flame-start system with a heater plug in the elbow in front of each header can preheat the charge air for cold starting and when ambient temperatures are low.

Double-pass, two-part headers carry the combustion gases from each cylinder bank to the turbocharger's turbine. Rocker posts provide additional support for the exhaust turbochargers. The exhaust gases exit the turbine to an elbow and flow down through an adapter to compensate for thermally induced expansion to the exhaust-gas aftertreatment unit and the silencer system. On account of close proxim-

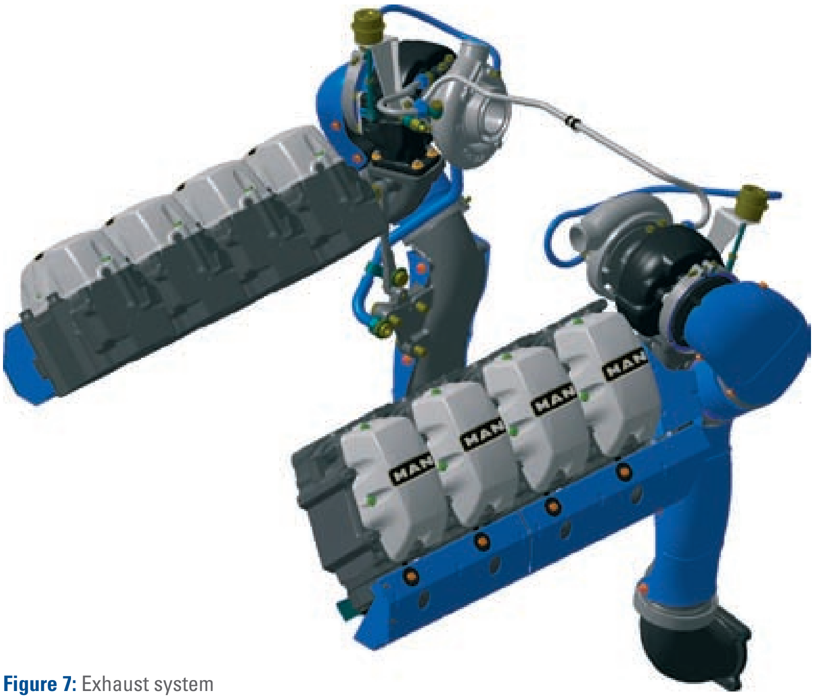


Figure 7: Exhaust system

ity to sensitive components, the entire exhaust system, **Figure 7**, including the turbochargers is well insulated with heat shields and integral insulating material.

4.3 Ancillary Units

At the front of the engine, the air-conditioning compressor and the alternator are driven directly off the crankshaft by an eight-groove Poly-V belt. Belt tension is kept constant by an automatic tensioner of the same design as that featured by the D20.

Because of the on-frame height of the engine and the requirement specifying the same fan connections as for MAN inline engines, the centre-to-centre distance between fan and crankshaft was a mere 147 mm. This excluded the possibility of driving off the geartrain behind the vibration damper with a diameter of 340 mm. Given that fan power could be specified at ratings as high as 52 kW, the width of the belt drive was increased by 40 mm. The fan runs in a maintenance-free taper roller bearing, mounted on an extremely rigid GJS carrier bolted to the crankcase.

The V8 engine has a two-cylinder compressor with a capacity of 720 cm^3 to supply compressed air for the vehicle‘s brake system. The drive end of the compressor is carried by an intermediate housing mounted on the flywheel bell housing, and the non-driven end is supported off the crankcase, as shown in **Figure 8**. A compressor crankcase with integral liquid cooling was developed specially for this horizontal-installation situation. All the internals and the compressor cylinder head are parts taken from the D20/D26 engine. Drive is directly off the camshaft sprocket via a split idler gear with elastomeric tensioners.

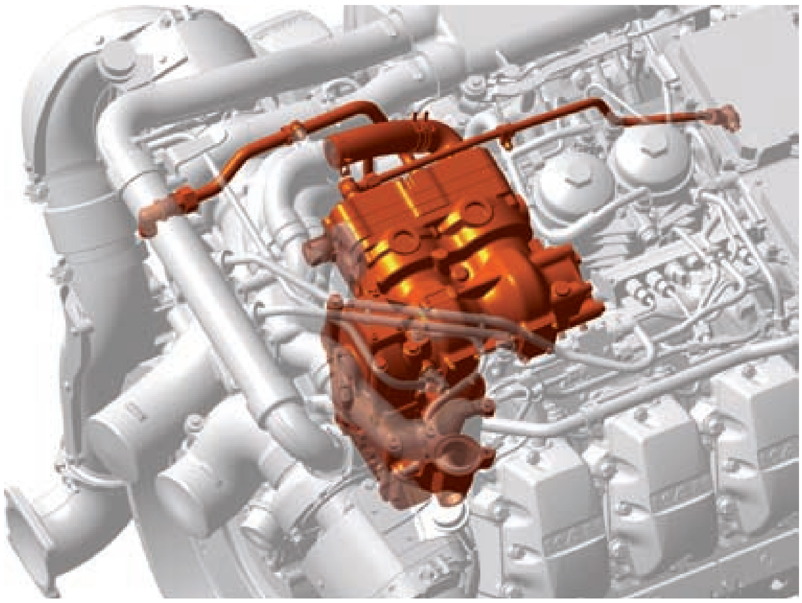


Figure 8: Compressor attachment

The standard power-steering pump is driven off the right PTO. The second power-steering pump for a heavy-duty transporter configuration attaches to the end of the compressor and is driven by the compressor crankshaft. A PTO can be installed at the left rear of the engine if necessary to drive extra hydraulic pumps for a heavy-duty transporter, for example.

5 Combustion and Exhaust-gas Aftertreatment

5.1 Combustion Development

As stated above, the V8 vehicular engine is used in two different applications. Tractive power is critically important for moving ultra-heavy loads, whereas high average speed and economical fuel consumption are two of the most important aspects factoring into all designs for long-distance haulage. In heavy-duty transportation, the engine often operates at rated output for lengthy periods of time. Long-distance haulage up to 40 metric tons gross weight, on the other hand, generally requires an engine of this size to operate most of the time in its part-load range at low engine speeds.

The Euro 5 emission limits set the bar for both applications. In order to achieve low untreated emissions concomitantly with minimum fuel consumption, in-depth studies were conducted with vari-

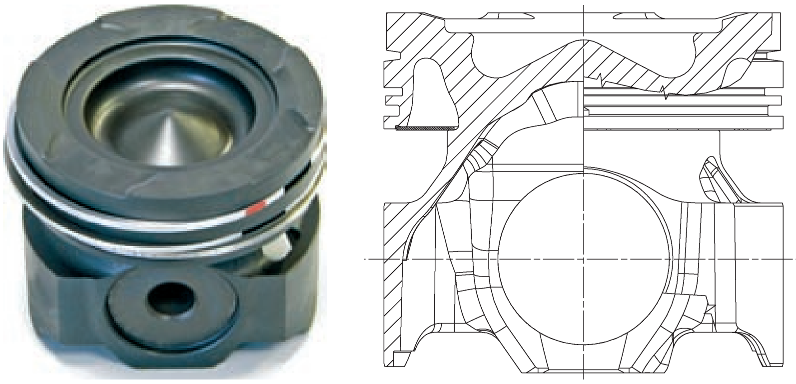


Figure 9: Steel piston with stepped combustion chamber

ations on piston-recess geometry and with different injection nozzles. DOE methods were employed in order to minimise the complexity deriving from the multiplicity of variable parameters and thus to shorten development time and reduce costs [1].

A piston recess with stepped combustion chamber, **Figure 9**, combined with an eight-hole nozzle and 146° spray angle was identified as the ideal configuration for volume production. The load collectives obtained from field tests in the vehicle were then taken into account in compilation of the characteristic map for electronic engine management.

5.2 Exhaust-gas Aftertreatment

Since a target engine power rating of 500 kW was a given and the engine had to operate with the same in-vehicle cooling system as the straight-six predecessor, a concept involving exhaust-gas aftertreatment was pursued right from the beginning. The engine-combustion parameters of the design are such that particulate raw emissions are below the limit permitted by Euro 5, so exhaust-gas aftertreatment is limited to NO_x reduction by selective catalytic reduction using AdBlue.

A hydrolytic catalytic converter is employed to optimise SCR. The catalytic converter is situated in a part-flow exhaust-gas conduit: this arrangement increases dwell time, allowing virtually complete conversion of the urea into carbon dioxide and ammonia – the reduction agent as such [2]. The main proportion of the exhaust gas flows through the parallel pre-converter that derives the nitrogen dioxide necessary for acceleration of the SCR reaction and for particle oxidation from the nitrogen monoxide discharged by the engine.

The deflectors in the stream of exhaust gas are designed to achieve this division of the exhaust mass flow so as to optimise distribution to the catalytic converters at the engine's operating points. The hydrolytic converter's tendency to cool the partial stream of exhaust gas, which would have a detrimental effect on urea decomposition, is countered by jacketing the partial-flow conduct within the main exhaust conduit.

The silencer to which the converters discharge accommodates ten catalytic inserts, each 230 mm long, where nitric-oxide reduction takes place. In order to prevent ammonia discharges caused by transitory saturation in non-stationary engine operation, the residual NH_3 is oxidised in a downstream blocking catalytic converter.

A V8-specific silencer system had to be developed to accommodate the interface specifics – the twin-turbocharger concept also means that the two streams of exhaust gas have to be combined upstream of the reducer – and also because the space available in the vehicle frame is limited. **Figure 10** shows the configuration for long-distance haulage vehicles, with the exhaust discharge downward and to the left as viewed in the forward direction of travel, in other words directed toward the middle of the road.

By comparison with the specific diesel consumption of the V10 engine that was used with Euro-3 parameters in the heavy-duty tractor unit, specific consumption of the new V8 engine with Euro-5 parameters is lower by approximately 4 % in the ESC cycle. This result is weighted for the requisite quantity of AdBlue.

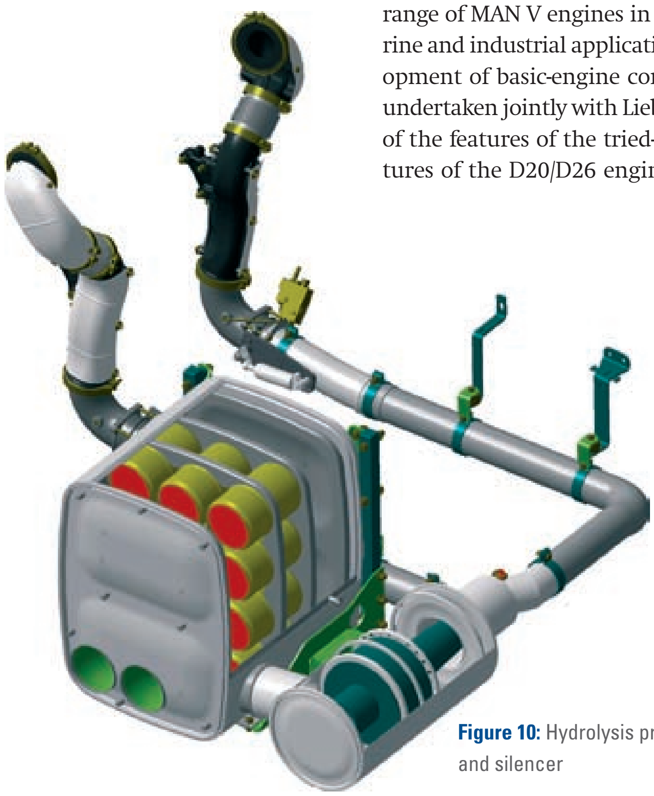


Figure 10: Hydrolysis pre-catalyst and silencer

6 Summary

A new series of V-configuration engines is being developed to supersede the existing range of MAN V engines in vehicular, marine and industrial applications. The development of basic-engine components was undertaken jointly with Liebherr. As many of the features of the tried-and-tested features of the D20/D26 engines as possible

Figure 11: Configuration for heavy-duty tractor

were incorporated into conceptualisation and design of the new development.

The first engine in the new V series is the D2868 LF with 16.16 l displacement, 500 kW (680 bhp) and a maximum torque of 3000 Nm between 1100 and 1500 rpm. It is for use in the heavy-duty tractor unit for gross weights up to 250 metric tons, **Figure 11**, and in the TGX-V8 premium tractor-semitrailer unit, **Figure 12**. The most powerful standard truck engine in Europe is characterised by outstanding tractive power in combination with excellent smoothness and extraordinary drive dynamics. The Euro-5-compliant design reduces consumption by another 4 % approximately over the Euro-3 V10 engine of the predecessor series.

The article describes conceptualisation, the design of the principal basic-engine components, the development of the vehicle-specific mounted parts and the design effort invested in the combustion process and exhaust-gas aftertreatment.

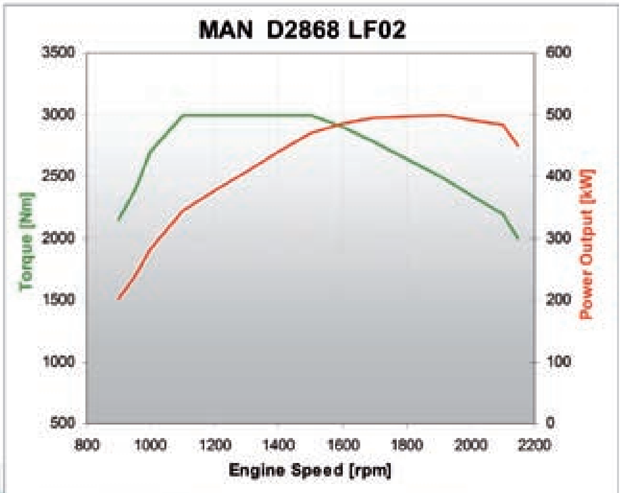


Figure 12: Configuration for premium tractor

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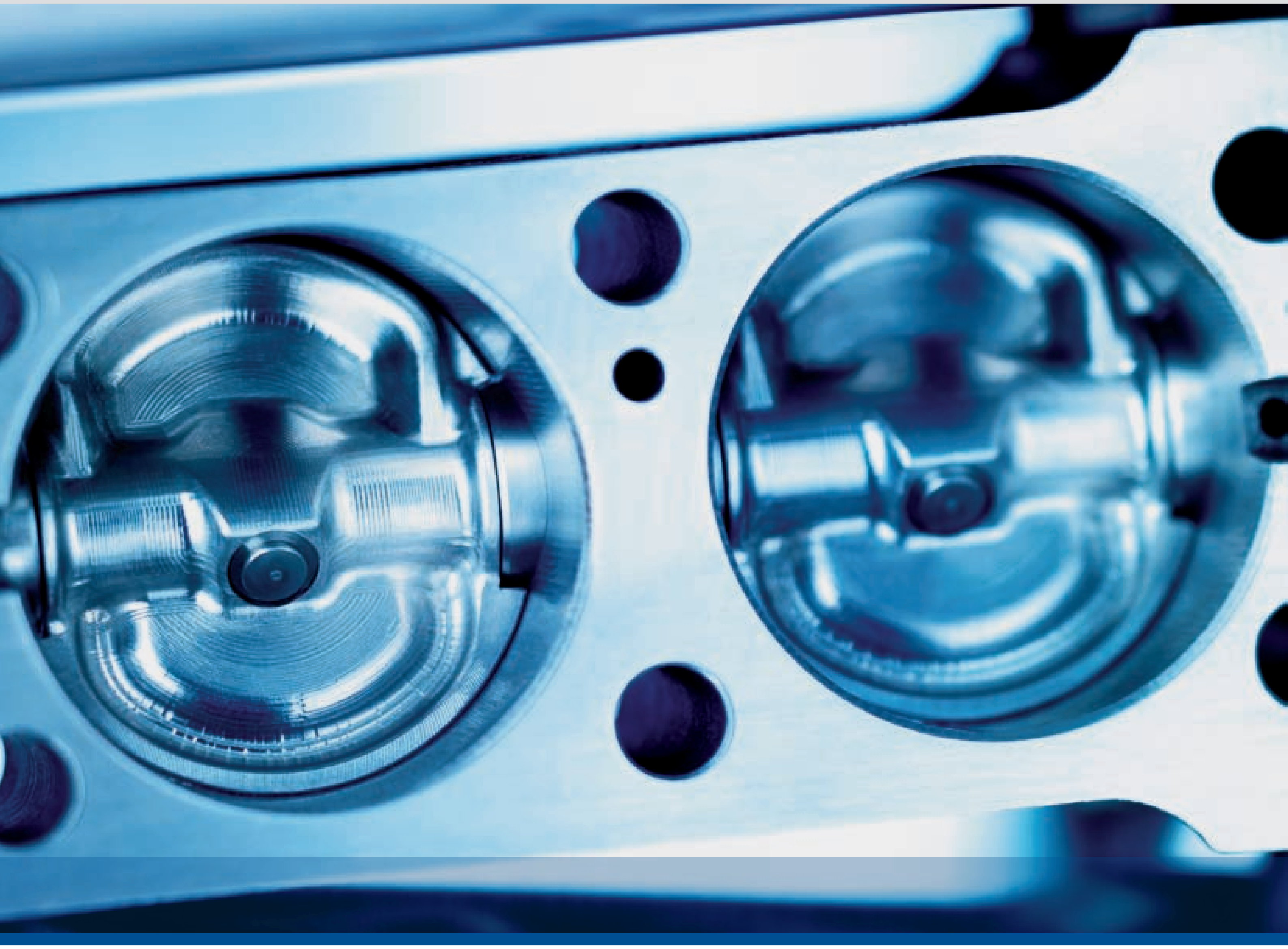
[2] Döring, A.; Jacob, E.: „GD-KAT: Abgasnachbehandlungssystem zur Verringerung von Partikel- und NO_x-Emissionen bei Nutzfahrzeug-Dieselmotoren" (Exhaust-gas treatment systems to reduce particulate and NO_x emissions of commercial-vehicle diesel engines); HDT-Essen 2001, MAN Nutzfahrzeuge AG



Mechatronic Exhaust Gas Recirculation Valve
for Commercial Vehicle Applications

The emission optimization of commercial diesel engines, both on-highway and off-highway, will need further measures to meet the requirements of worldwide strengthening emission laws. In addition to exhaust gas after-treatment systems also the external exhaust gas recirculation established in passenger cars with diesel engines offers substantial potential for lowering the engine-out emissions of vehicle engines. To this effect, Pierburg GmbH in Neuss, Germany, developed a completely electrically actuated exhaust gas recirculation system for heavy-duty applications.

1 Introduction

The worldwide strengthening emission laws for vehicles driven by combustion engines, in particular for commercial diesel vehicles, require further measures to reduce the untreated emissions in order to achieve an efficient and stable attainment of the critical values through an after-treatment of the exhaust gas [1,2]. **Figure 1** illustrates the development of emission standards for particulate matter and nitrogen oxides in Europe that, in future, will largely converge with the standards applicable in the US.

A well-known and very efficient instrument for reducing the NO_x emissions of car engines is the external recirculation of exhaust gas (EGR). For commercial diesel applications, similar to the demands of passenger cars with diesel engines, a continuous and precise admixture of recirculated exhaust gas with high system dynamics will have to be guaranteed. It is almost certain that only a completely electrically operated exhaust gas recirculation valve will be suited to meet these demands. It allows for a simplified application due to the fact that no auxiliary energy such as compressed air is needed and thus also permits to use this technology in different vehicle segments and fields of application, be it on-highway or off-highway.

2 Concept and Design

The conceptual design and development of the exhaust gas recirculation valve are in particular determined by the demand specifications of the application range of commercial diesel vehicles. The **Table** shows some parameters for the valve type illustrated in **Figure 2**.

A key requirement is owed to the system architecture of typical EGR loops with a "hot" EGR valve, i.e. a valve installed upstream of the EGR cooler. This place of installation is advantageous with a view to reduced heat dissipation into the cooler when the EGR is switched off and also with a view to the transient engine behavior due to less dead spaces upstream of the turbine of the exhaust gas turbocharger.

2.1 Valve Group

The high thermal, mechanical and chemical loads prevailing in hot installation require the use of highly temperature-resistant and corrosion-proof materials. In addition, functional aspects such as low flow losses and good controllability, in particular of small mass flow rates, are basic requirements for system acceptance.

After examining a multitude of valve types and considering the goal of minimized actuation energy, the decision was made for a pressure-compensated

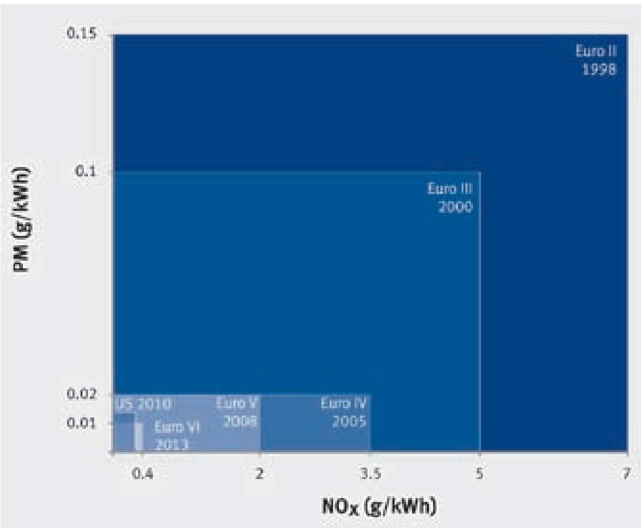


Figure 1: Limit values for NO_x and particle emissions pursuant to European emission standards (EU 6 as per proposal 12/2007; US2010 for comparison)

The Authors

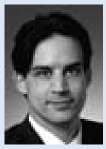


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Table: Data of the electrically actuated heavy-duty EGR valve

Parameter	Unit	Value
Diameter flow duct	mm	42
Height	mm	175
Length	mm	180
Width	mm	75
Mass	kg	3.8
Mass flow open valve per flow, air, Δp = 30 mbar, standard conditions	kg/h	295
Opening time t_{90}	ms	130
Closing time t_{90}	ms	85
Supply voltage	V	12/24
Typical power intake – hold	mA	500
Maximum power intake	mA	1800
Communication of set value (input) and status (output)	PWM %	5 … 95
	CAN	

Figure 2: Double-flow electrically actuated EGR valve in heavy-duty design, duct diameter 42 mm

butterfly valve. With flow simulations and test series the control behavior of the butterfly valve at small opening angles was improved to enhance small volume control.

This optimization was achieved without compromising the maximum throughput. Since the valve is intended for use in heavy-duty diesel engines in the most diverse configurations, single-flow as well as double-flow concepts were realized.

2.2 Electrical Actuator

The distinctly higher requirements in terms of service life and operational safety, in comparison to passenger cars, make the use of a contactless electrical drive indispensable. As a result of an intensive benchmarking of various electrical drives with this characteristic, an electrically commuted direct current motor (EC motor) in combination with a compact gearbox was found to be goal oriented.

To transform the torque of the EC motor into a suitable range for the respective application, a 2-step planetary gearbox is arranged on the motor shaft, **Figure 3**. This type of gearbox has been proven million-fold in the broad-range passenger car series production and allows for large gear reductions with little external forces, high efficiency and low space demand. In this gearbox, a space-saving, contactless sensor for measuring the angle position of the drive shaft has been integrated.

A restoring spring was arranged concentrically around the planetary gearbox to support the restoring spring arranged on the valve side and to allow for a separate fail-safe position of the actuator. Thus, the electrical actuator as a separate unit including fail-safe function has also been configured for other applications.

The control electronics form an integral part of the actuator and undertake the function of the electrical and logic interface to the customer control device.

As the set point value for the internal position control it receives the CAN data or a pulse width modulated voltage signal and returns status notifications in the same manner. The actuator and electronics are designed for a voltage supply of both 12 V and 24 V, so that also specific regional demands can be met.

Optionally, the actuator can be connected to a cooling circuit in order to prevent thermal overload of the electronic components even under highly unfavorable operating conditions.

2.3 Overall System

In principle, the electrical actuator and the valve body are separate modules so that the most diverse demands for a modular system can be fulfilled. This arrangement also benefits the thermal isolation that is indispensable with regard to the high exhaust gas temperatures. The parallel arrangement of the valve axis represents a favorable solution with regard to the package requirements in the vehicle.

A coupling rod is used as a transmission element for the torque between actuator and valve group. The transmission between drive and output was optimized through an adequate definition of the effective levers. The focus was on good controllability, specifically in the case of small flow rates, and on the efficient utilization of the motor torque in respect of the holding and breakaway torque.

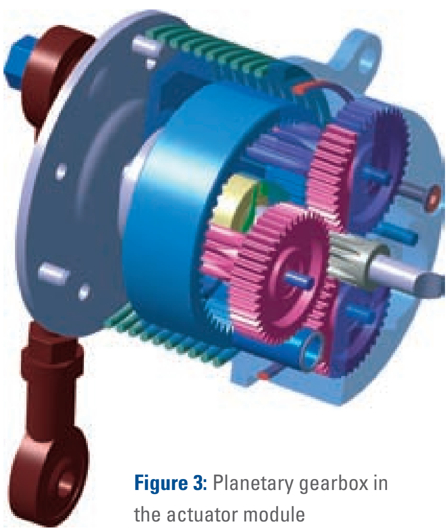

Figure 3: Planetary gearbox in the actuator module

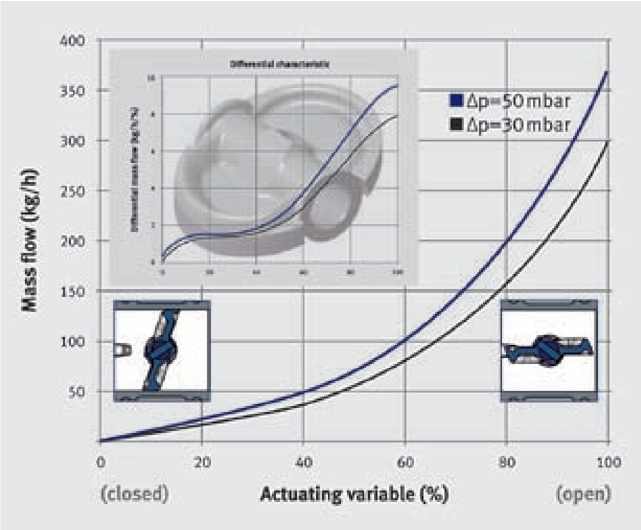

Figure 4: Gas throughput characteristic on a flow of the EGR valve and its derivative as a function of the control value (inset with valve)

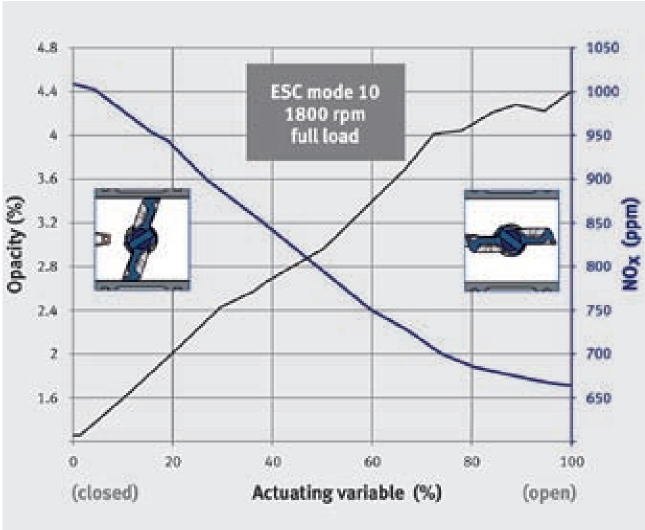

Figure 5: Emission characteristic in ESC mode 10 with EGR rate variation via the valve position

3 Test Results

In addition to comprehensive component tests also various tests on an engine test bench were conducted with the EGR valve under conditions relevant for practical use. To this effect, a 10.5 l, 6-cylinder common rail Euro III diesel engine with cooled external exhaust gas recirculation was used. Instead of the pneumatically actuated series valve the electrically-actuated EGR valve was arranged in the flow direction of the exhaust gas upstream of the EGR cooler with an optional connection to a cooling water circuit with a throughput of 5 l/min and a flow temperature of 85 °C.

3.1 Controllability of Engine Emissions

It was interesting to see how the different designs intended to improve controllability of the valve throughput affected the actual controllability of engine emissions. **Figure 4** shows a measurement of the gas throughput characteristic of an individual valve flow with a diameter of 42 mm and air feed at ambient temperature against standard pressure as a function of the actuating variable. The selected valve contour and the kinematics of the coupling ensured an initially flat, steady increase of the throughput in the first third of the actuating variable range. The typical flattening of the characteristic for butterfly valves in the higher throughput range did not occur. In combination with the high control accuracy of the electrical actuator of ± 0.35 % of the set value interval, a precise setting of the mass flow through the EGR valve is guaranteed over the complete control range.

On the engine test bench, the exhaust gas opacity and further emissions were measured as a function of the actuating variable of the valve in the ESC operating modes. **Figure 5** illustrates a typical example of the development of the opacity and the NO_x concentration in mode 10 (full load, 1800 rpm) as a function of the valve position.

As expected, the nitrogen oxides and opacity develop in opposite directions during the EGR rate variation according to the well-known trade-off. The described throughput characteristic with its excellent resolution, particularly in the range of smaller exhaust gas mass flows, results in the largely linear dependency of emissions from the valve control value as shown in Figure 5, which means optimum controllability across the complete emission range, thereby allowing for even the smallest shifts in the trade-off line.

3.2 Dynamics

Just as important as the decrease of stationary emissions is the reduction of the exhaust gas emissions in transient conditions where high dynamics of the EGR control unit are important. On the engine test bench, a sudden load variation from 40 % to full load at 1200 rpm was tested. Here, the time lag between the step-wise load increase and the closing signal to the valve for an abrupt reduction of the EGR rate was varied. Following the ELR test, the mean value of the maximum opacity value of several sudden load variations was calculated as an evaluation criterion for the soot emissions, **Figure 6**. During the test the anticipated rise of the opacity peaks with the artificially prolonged time lag was observed. A delay of the switching point within the first 100 ms obviously did not show any effect relevant for emissions, which demonstrates that the actual closing speed of the valve, less than 100 ms, **Figure 7**, is sufficient.

Of course, the electrically actuated EGR valve is not limited to sudden profile switches. Instead, the interacting parameters rating, torque, consumption and exhaust gas emissions will be optimized in practice using freely selectable valve position curves [3].

3.3 Endurance and Component Tests

At the place of installation, the exhaust gas temperatures prevailing during the endurance test in the selected operating mode (ESC mode 6, 1200 rpm at 75 % load) were in the range of 600 °C. During the complete test period involving more than half a million switching operations, the valve position and the flow intake of the overall module were monitored

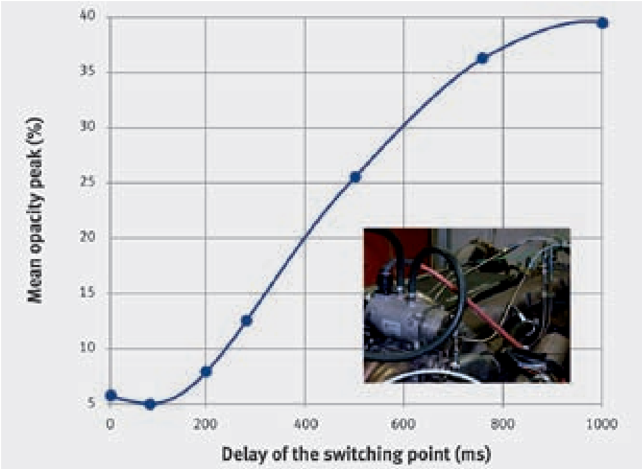

Figure 6: Mean opacity peaks during sudden load change (40 % of the full load) with artificial delay of the switching point of the EGR valve

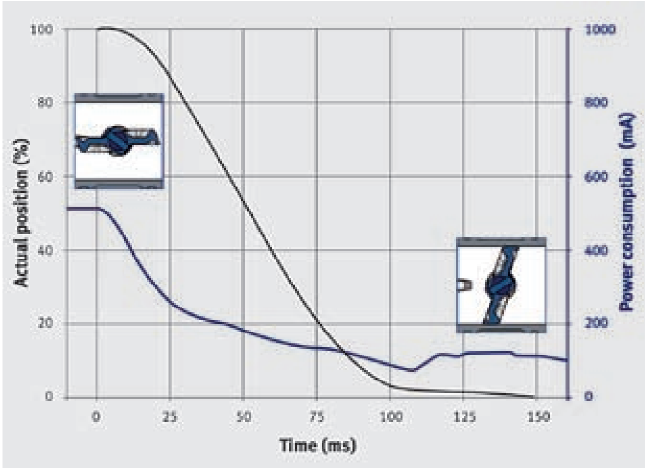

Figure 7: Development of the valve position with a nominal value change of 100 % (fully open) to 0 % (closed)

whereby both parameters did not show any measurable drift. Separate leakage measurements on the valve prior to and after engine bench tests did not produce any significant changes either. The emission values of the closed valve corresponded to those of a full mechanical closure of the EGR tube, which attests to the good tightness of the selected valve system.

On the engine test bench, particular attention was paid to the temperature distribution across the module during operation, which was monitored by additional thermal sensors. Both the hot gas test and the engine bench test showed that the temperature of the electronic modules, which is key to ensuring the function, is stringently related to the (externally measured) temperature of the cooling medium due to the cooling water routing.

In the tests on the engine test bench, closing of the cooling water flow during operation even resulted in a reduction of the electronics temperature; in these cases, the cooling water must still have dissipated some heat into the module. Consequently, even post-heating effects after switching off the engine and media flow do not result in critical conditions. However, only the respective thermal conditions at the place of installation will determine whether connection to the cooling water can be omitted in a specific application.

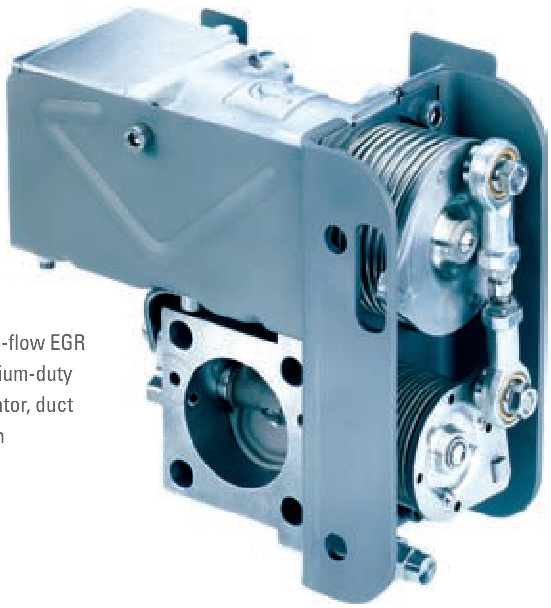

Figure 8: Single-flow EGR valve with medium-duty electrical actuator, duct diameter 42 mm

4 Summary and Outlook

The development of the exhaust gas recirculation valve shown here fully accommodates the future requirements regarding an improved emission control with a simplified application and enhanced functionality. The modular structure consisting of a separate valve group and an electrical actuator allows for an easy configuration of the overall system for different applications. **Figure 8** shows an example of a medium-duty design of the EGR valve with a smaller valve group and an adapted actuator. Thanks to the "Actuator Toolbox" developed by Pierburg, positioning tasks in other fields of application can also be fulfilled.

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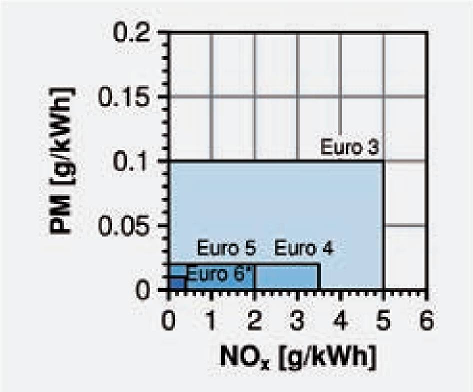

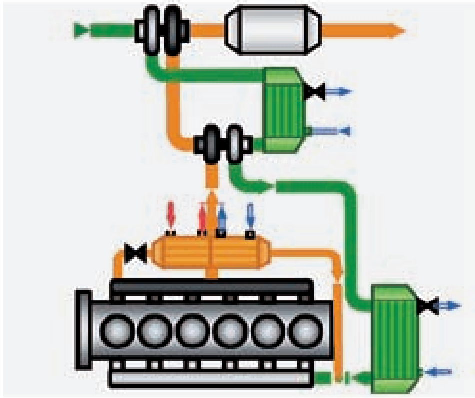

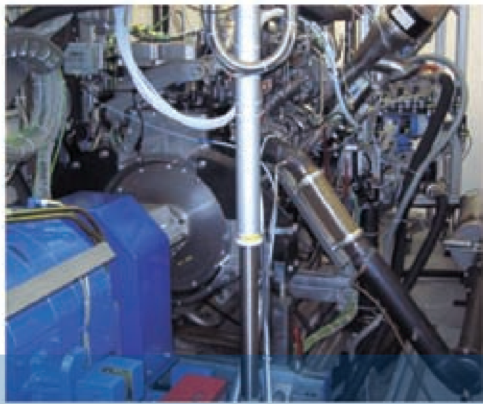

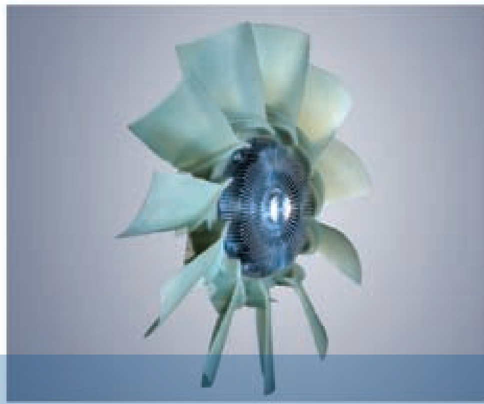

Emissions Concepts and Cooling Systems for Heavy Duty Trucks at Euro 6

In order to achieve the likely Euro 6 NO_x emission limits two technologies can be considered: selective catalytic reduction and cooled exhaust gas recirculation. Although the final emission limits have not been set, it is likely that a combination of these technologies could reach the targets. However, the question remains whether one of the technologies alone could reach the targets. In collaboration with AVL List, Behr has investigated the NO_x and fuel consumption reduction potentials of both these systems for Euro 5 and 6.

1 Introduction

In **Figure 1** the current and future legislation limits for NO_x and PM in Europe, Japan and the USA are shown. It should be noted that the relevant emission test cycles are different in each region [1]. In Europe there is an on-going discussion in relation to the limits and test cycle for Euro 6. The latest suggestion from the EU Commission, in December 2007, included limits of 0.4 g/kWh for NO_x and 0.01 g/kWh for PM based on the existing ETC. Should the World Harmonized Transient Cycle be used as the new test cycle, as is currently being discussed, then the suggested NO_x limit would be changed. However, independent of the current discussions, it is clear that new emissions concepts will be necessary, and this will mean further development of the cooling system.

2 Euro 5 Emissions Concepts

For the investigation on a 10.5 litre engine the following emissions concepts were chosen, **Figure 2**: selective catalytic reduction (SCR), cooled exhaust gas recirculation (EGR) with particulate aftertreatment and low temperature EGR (LT-EGR) without aftertreatment. The engine had a rated power of 290 kW and was fitted with common rail fuel injection. For the tests over the steady state (ESC) and transient (ETC), an engineering margin of 15 % was chosen at Euro 5, which mean emissions targets of 1.7 g/kWh NO_x and 0.017 g/kWh PM. Each emissions concept was calibrated to allow measurements over both the ESC and ETC. As an example, **Figure 3** shows the EGR rate engine maps for the EGR and LT-EGR concepts. On the test bed, the exhaust backpressure was set to represent the relevant exhaust aftertreatment. For the heat

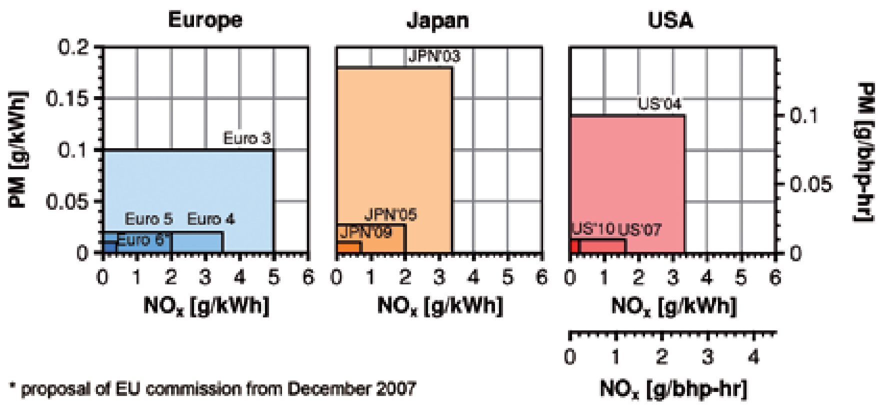

Figure 1: Emission limits for heavy duty trucks in Europe, Japan and the USA

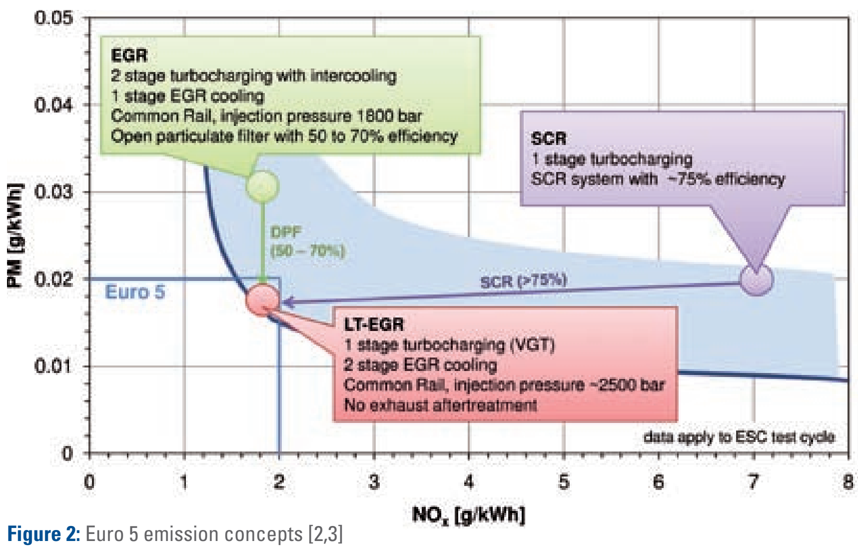

Figure 2: Euro 5 emission concepts [2,3]

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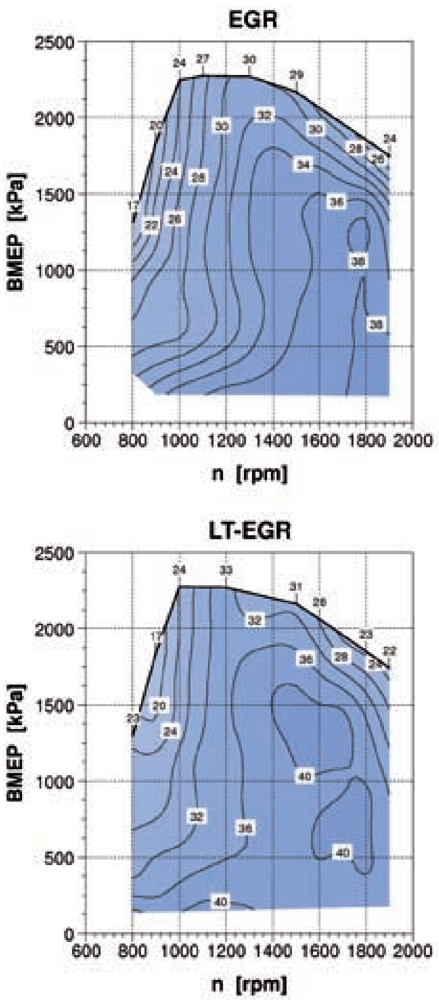

Figure 3: Engine maps showing the EGR rate of the EGR and LT-EGR concepts

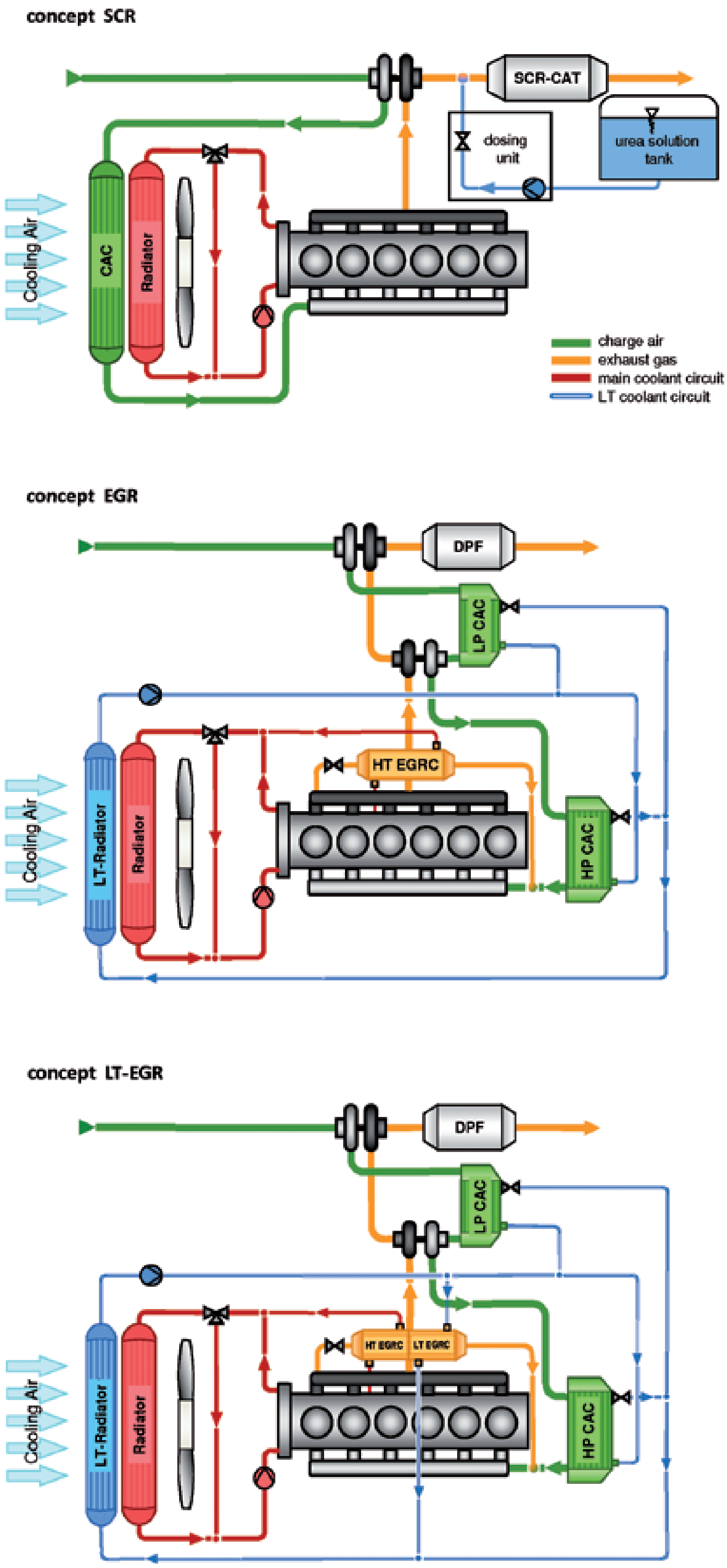

Figure 4: Schematic layouts of the SCR, EGR and LT-EGR concepts with the corresponding cooling systems. HP CAC: high pressure charge air cooler; LP CAC: low pressure charge air cooler, HT EGRC: high temperature EGR cooler; LT EGRC: low temperature EGR cooler

balance measurements each heat exchanger had a separate cooling circuit with its own conditioning. As such, the temperatures in each heat exchanger could be set, as would be expected at an ambient temperature of 25 °C.

2.1 Selective Catalytic Reduction

An efficiency of 70 to 80 % was assumed for the SCR concept. Consequently, engine out ESC or ETC NO_x emissions of 7 to 9 g/kWh could be accepted. This meant that a fuel economy optimised calibration could be used with PM emissions under 0.02 g/kWh such that no PM filter was needed. A single stage fixed geometry wastegate turbocharger was used. In **Figure 4** (top) the layout of the engine and the cooling system is shown.

2.2 EGR

The NO_x emissions limits could be reached engine out with single stage EGR cooling. For the PM emissions an open particle filter (DPF) with an efficiency of 50 to 70 % could be chosen. Therefore, engine out PM emissions of 0.04 to 0.06 g/kWh were acceptable. The turbocharging and the charger air cooling were two stage; both charge air coolers were engine mounted and cooled with a low temperature cooling circuit. In Figure 4 (middle) the layout of the concept including the engine cooling circuit, which included the EGR cooler, and the low temperature cooling circuit is shown.

2.3 LT-EGR

In this concept the EGR is cooled in two stages, Figure 4 (bottom): in the first stage with the engine coolant, in the second stage with the low temperature coolant. Significantly lower EGR and intake charge temperatures were reached with this two stage cooling. These lower temperatures were used to reduce the engine out NO_x emissions.

2.4 Results of the Engine Tests

2.4.1 Fuel Consumption

The fuel consumption values for each of the three concepts at 1.7 g/kWh NO_x were determined over the ESC and ETC. The LT-EGR concept showed a 3 to 4 g/kWh fuel consumption improvement compared to the EGR concept over the ESC and ETC respectively, **Figure 5**. That is about 2 %. As such, the LT-EGR concept had the same consumption as the SCR concept, once the economic equivalent urea solution consumption was included. The LT-EGR boosting system was the same as that for the EGR concept: with an optimization of the boosting system for the LT-EGR concept a further 1 to 2 g/kWh fuel consumption benefit is to be expected. In **Figure 6** the fuel consumption maps for the EGR and LT-EGR concepts are shown.

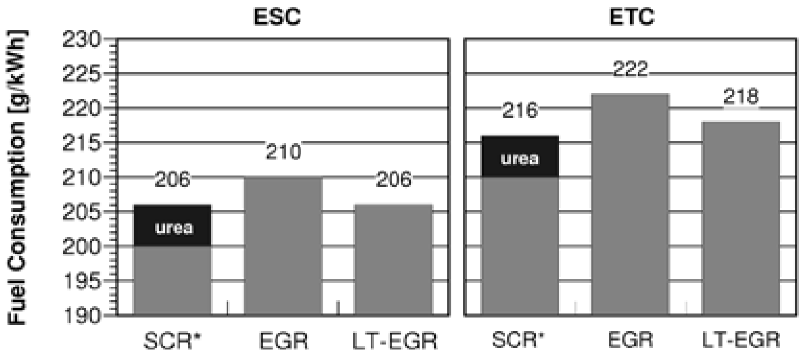

Figure 5: ESC and ETC measured fuel consumption for investigated concepts

* fuel consumption of SCR concept contains fuel cost equivalent urea solution consumption of 1.40 €/l for diesel fuel and 0.60 €/l for urea solution

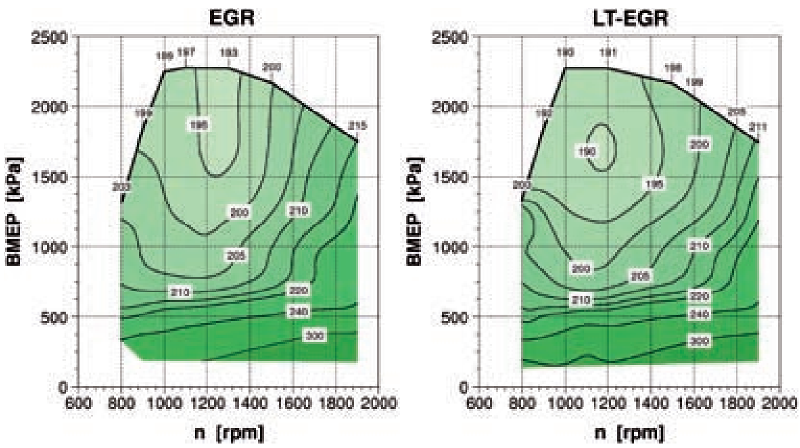

Figure 6: BSFC maps for EGR and LT-EGR concepts

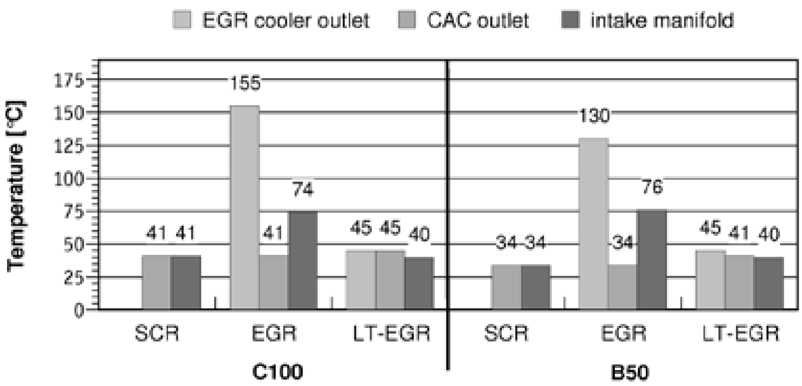

Figure 7: Measured temperatures in the air system for the C100 and B50 operating points

2.4.2 Temperatures in the Intake System

In **Figure 7** it is shown that the charge air temperatures for the LT-EGR concept are about 5 K warmer than for the EGR concept. This is a consequence of the greater heat load in the low temperature cooling circuit because of the two stage EGR cooling. However, the temperature of the mixed air and EGR charge is about 35 K lower in the LT-EGR concept. This is primarily a result of the improved EGR cooling, secondly a result of the evaporation of the exhaust gas condensation. This condensation occurs in the LT-EGR cooler: when the EGR is mixed with the dry charge air there is a steep concentration gradient between the fluid phase and the unsaturated air, resulting in the evaporation of part of the condensation. This results in a cooling of the charge in the intake manifold of about 5 K. A further interesting aspect of the LT-EGR concept is the increased stability of the charge temperature during the ETC, **Figure 8**. This brings advantages during the engine calibration.

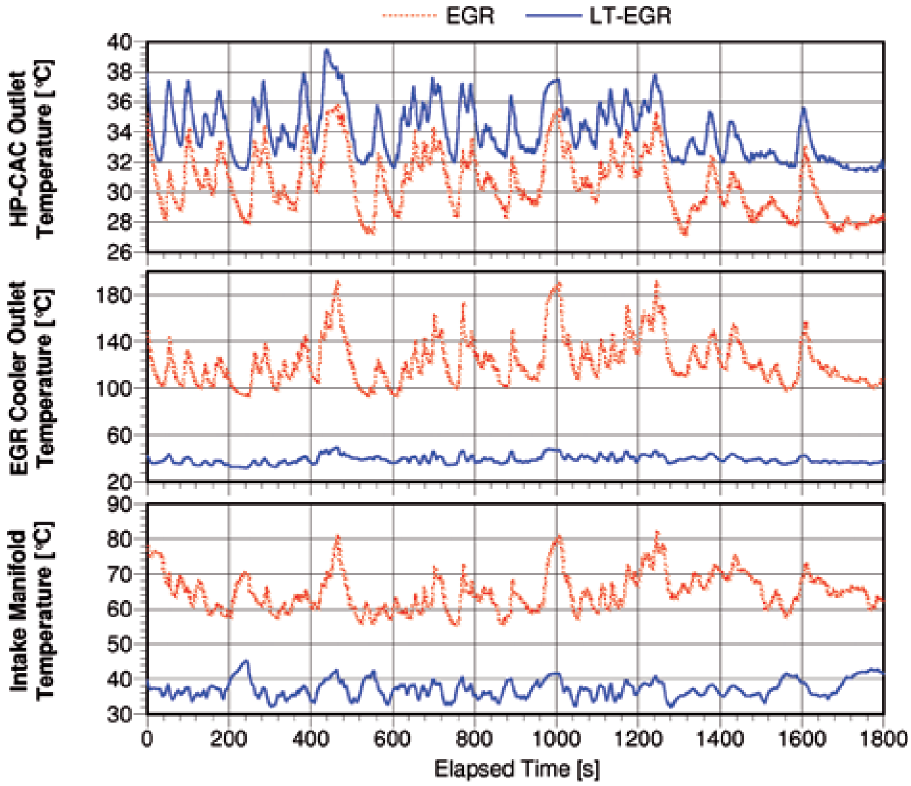

Figure 8: Plots for charge air cooler outlet, EGR cooler outlet and intake manifold temperatures during the ETC test

2.4.3 Heat Quantities

The necessary heat rejection with the LT-EGR concept at the C100 operating point is about 37 % higher than with the SCR concept but 6 % lower than with the EGR concept, **Figure 9**. Despite the fact that more heat is extracted from the EGR, the LT-EGR concept needs a lower boost pressure due to the higher density of the cooler charge air. Hence, the charge air cooling heat rejection is 10 kW lower compared to the EGR concept. Furthermore, the heat rejection from the engine is about 20 kW lower with the LT-EGR compared to the EGR concept. 63 % of this reduction is because of the lower component temperatures (as a result of the lower charge temperatures), 23 % because of evaporation of the condensation in the cylinder and 14 % because of the reduced engine fuel consumption.

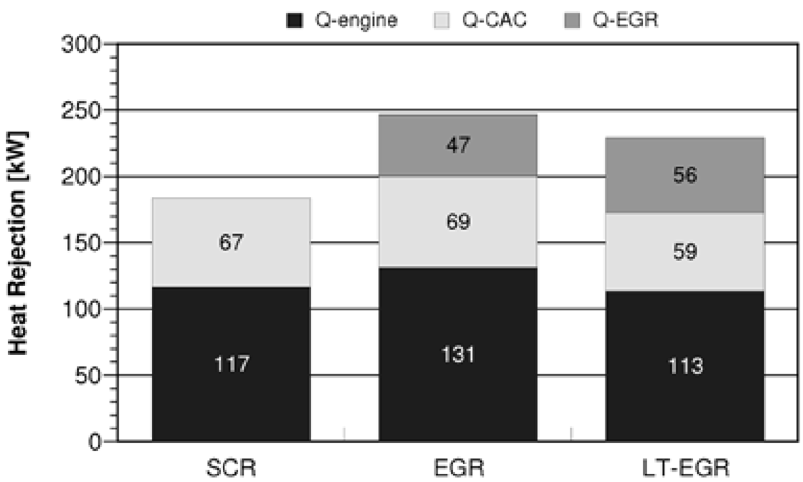

Figure 9: heat rejection for C100 operation mode of the investigated concepts

2.4.4 EGR Cooler Fouling

Normally, with continuous operation of an EGR cooler there is a build up of exhaust residue on the tubes, which reduces the EGR cooler performance. This effect was not seen with the low temperature EGR cooler: the cooler performance remained high. It is suspected that the condensation in the low temperature cooler leads to a cleaning effect: the fouling is washed away. The performance reduction experienced in the first stage of the cooler was, to a large extent, compensated for in the second stage, leading to a constant cooler performance over the duration of the testing.

3 Layout Strategies for Euro 5 and Euro 6

Test bed results from Euro 5 engines with lambda control for EGR rate show that an increase in the EGR cooler outlet temperature of 10 K at the C100 point results in an increase in the NO_x emissions of about 0.1 g/kWh over the ESC. The test bed results also show that a variation of the EGR cooler outlet temperature of ±15 K around the mean value must be expected for a Euro 5 engine as a consequence of the fouling and regeneration of the EGR cooler under different operating conditions. This temperature variation leads to a variation in the NO_x emissions over the ESC of about 0.15 g/kWh, about half the engineering margin.

With an assumed Euro 6 limit of 0.4 g/kWh NO_x and a 15 % engineering margin, the development target would be 0.34 g/kWh. If it is further assumed that 50 % of the engineering margin must be used to compensate for EGR cooler fouling and that the correlation between NO_x and EGR outlet temperature described above remains valid, then the acceptable variation in the EGR cooler outlet temperature becomes ±3 K at the C100 point. It is clear to see that the fouling behaviour of EGR coolers, as known from Euro 5 engines, will not meet the requirements for Euro 6. The results shown here for the LT-EGR concept show a variation less than ±3 K at the C100 point. The LT-EGR cooler ran for about 200 hours on the test bed: a longterm assessment of the stability of the cooler in relation to fouling still needs to be done.

4 Technology for Euro 6

4.1 SCR System Developments

The developers of SCR systems for Euro 6 face a challenge: whilst it depends upon the engine out NO_x emissions, significantly greater NO_x coversion rates will be required compared to Euro 5. If it is assumed that the engine out NO_x will be 5 g/kWh, then a conversion efficiency of more than 90 % over the cycle will be necessary. Due to the lower engine speeds and loads in the likely WHTC, there are lower exhaust gas temperatures. This makes the hydrolosis of the urea more difficult and reduces the efficiency of the SCR process, which is especially true during a cold WHTC test. Therefore, an SCR only concept for Euro 6 would have a more complex SCR than currently for Euro 5. For example, additional hydrolysis and clean-up catalysts could be needed.

4.2 LT-EGR with Two Stage Turbocharging

In the ESC tests NO_x values of 0.85 g/kWh were recorded without changes to the hardware. In order to reach the Euro 6 NO_x limits with an EGR only concept, significantly higher full load EGR rates are necessary when compared to Euro 5: about 35 to 40 %. In order to keep the charge temperature as low as possible, the EGR cooling should be two stage, **Figure 10**. In addition, in order to supply enough air to the engine, two stage boosting will be necessary, preferably with intercooling. In addition to the EGR and boosting systems other engine components must be optimised. Most importantly the fuel injection system must have increased pressures, up to 2500 bar and perhaps beyond, in order to keep the PM emissions under control at the high EGR rates.

ESC calculations suggest that, in order to reach 0.34 g/kWh NO_x over the cycle, a value of 0.5 g/kWh at the C100 point is necessary, **Figure 11**. If the NO_x emission target at this point is reached through LT-EGR and a recalibration, then results suggest that the fuel consumption will increase by about 7 g/kWh (3.5 %) compared to the Euro 5 value. This is mostly a result of the later fuel injection timing. The EGR valve is only 50 % open at the C100 point: the limiting factor for further NO_x reduction is the soot value of more than 0.3 g/kWh. With the introduction of fuel injection pressures of more than 2500 bar and an optimization of the boosting and EGR systems, it is likely that the fuel consumption penalty would be held at 2 % with an acceptable soot value.

At lower engine speeds the EGR rate is limited by the available pressure difference across the engine. However, the challenges to reach the Euro 6 limits without NO_x aftertreatment, are greater in transient than steady state engine operation. For Euro 3 without EGR the ratio of NO_x emissions in the ETC to the ESC

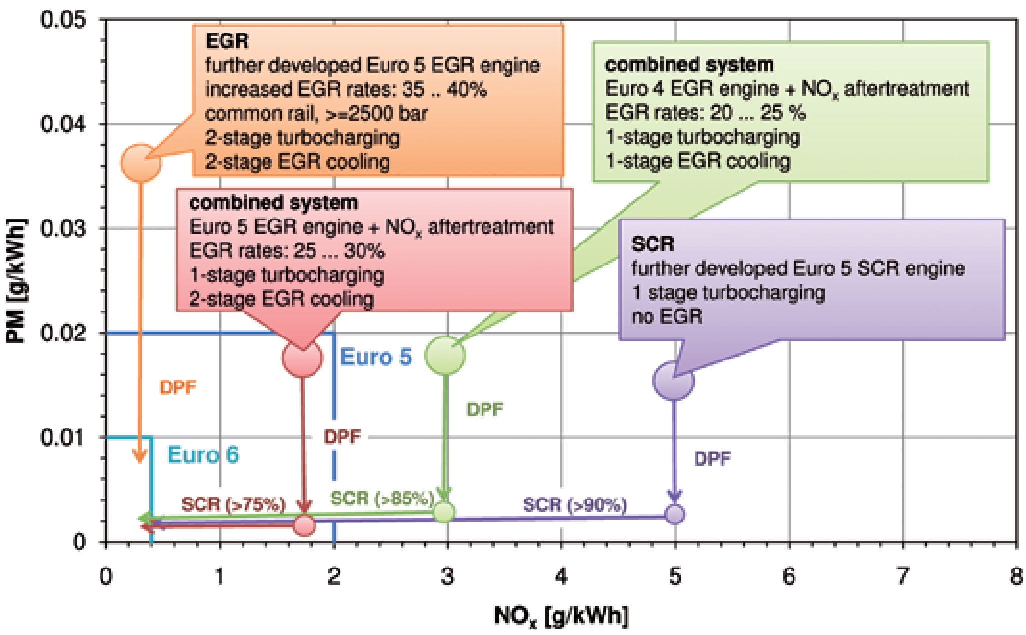

Figure 10: Emission concepts for Euro 6

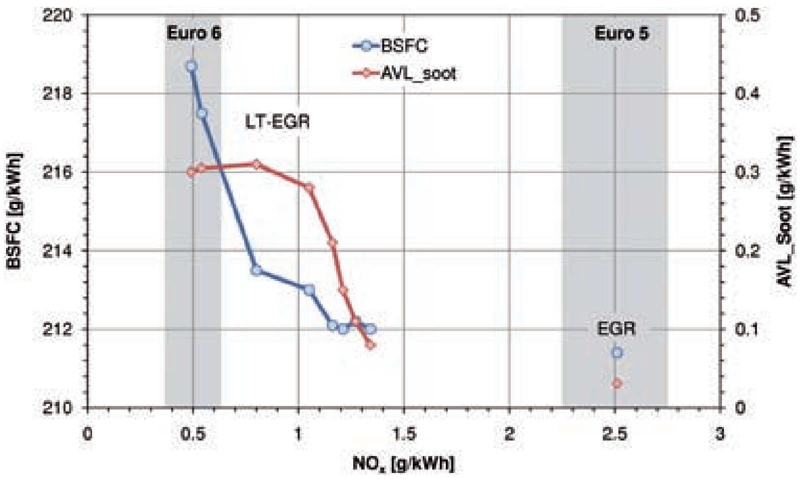

Figure 11: Test results for fuel consumption and smoke at the C100 operating point for a variation of engine out NO_x emissions

was approximately 1, injection timings and pressures were comparable, **Table**. For Euro 4 and beyond this ratio has increased, since the EGR rate during the transient operation must be reduced in order to achieve an acceptable transient torque response and smoke emission. Since there is only a difference in the PM limits between the ESC and ETC, the NO_x limits in the ETC must be achieved through a reduction in the injection pressures and later injection timing, if the EGR rate is reduced. At Euro 5 over the ETC transient correction factors in the EGR valve setting (rate reductions) occur 63 % of the time. For 28 % of the time is the EGR valve closed, thereof 50 % during motoring operation.

For Euro 6 the ratio of ETC to ESC NO_x emissions will probably be about 2. If this is to be compensated for by injection pressure reductions or later injection timings, then the fuel consumption penalty would be unacceptably high. In order to enable a higher EGR rate during the transient operation, an optimised boosting system, two stage or with compressed air injection, is necessary. Alternatively a hybrid driveline could be envisaged, in order to reduce the transient operation of the engine.

4.3 Combined Concepts

Combined concepts consist of the obligatory DPF plus EGR and SCR systems. Depending upon the relationship of the NO_x reduction via EGR or SCR, different base engines have to be used. In Figure 10 two examples are shown. In the first example (bottom left) a Euro 5 engine is used, with an engine out NO_x of 1.7 g/kWh. In this case an SCR system with a conversion efficiency of about 75 % would be necessary to reach the likely Euro 6 NO_x limit. This efficiency is similar to what is currently used at Euro 5. The EGR cooling would be two stage: in the second stage low temperature cooling would be used. The concept could have single stage turbocharging, but two stage turbocharging with intercooling is also possible. The second example (top right in Figure 10) would use a Euro 4 engine as its basis. With an engine out NO_x emission of about 3 g/kWh an aftertreatment efficiency of about 85 % would be necessary. In this case both the turbocharging and EGR cooling would be single stage.

All combined systems are more complex than EGR or SCR only concepts and need more packaging space. In general, there are also likely to be disadvantages in terms of weight and cost. However, such systems do have the best potential to reach Euro 6 at the moment.

Table: Ratio of NO_x emissions in ETC and ESC for different emission levels

Assumptions: constant injection timing and pressure and reduction of EGR rate during transient operation in ETC to get a comparable transient behavior

Emission Level	EGR Rate @ C100 [%]	NO_x ratio ETC / ESC
Euro 3	~ 0	1
Euro 4	~ 15	1,2
Euro 5	~ 25	1,5
Euro 6	~ 35	2

5 Cooling Systems for Euro 6

The engine cooling system will face different challenges depending upon the concept for Euro 6. Whereas for the combination concepts and for the SCR only concept the specification of the cooling system will only change slightly, for the EGR only concept it is expected that there will be an increase in the required heat rejection. Measurements at the C100 operating point with a Euro 6 suitable NO_x level of 0.5 g/kWh show an increase in the required heat rejection of about 20 % compared to Euro 5. This means that the cooling system must also be developed further. In order to achieve this, improvements in the cooling system as described in [5] are needed: optimization of the fan, the undercab air flow and the radiator core. Perhaps a waste heat recovery system should also be considered. In addition, more space for a cooling system with a larger face area and fan might be necessary for vehicles with the most powerful engines.

Depending upon the chosen emissions concept for Euro 6, the cooling system can provide further functionality than just removing heat from the coolant, charge air and EGR. For example the aftertreatment efficiency can be increased by raising the intake charge air and hence the exhaust temperature. During the cold start test cycle, the exhaust gas temperatures are too low for sufficient SCR NO_x conversion efficiency or continuous regeneration of the particle filter. In this situation the cooling system can help via the following means:

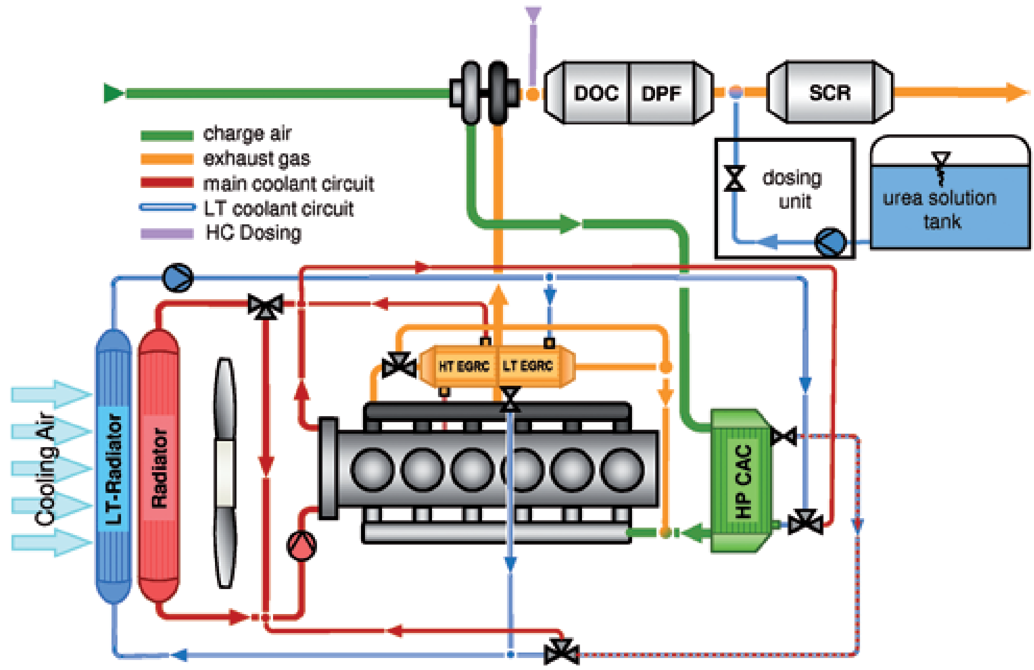

Figure 12: Cooling system for a combined system

- Reducing the charge air cooling: with indirect charge air cooling the coolant flow through the cooler can be stopped, as long as boiling does not occur. With direct charge air cooling a bypass can be installed. Either of these measures is especially applicable to the cold start phase.
- Heating of the charge air: by connecting the low temperature coolant circuit to the main coolant circuit, as shown in **Figure 12**, it is possible to bring the charge air up to the level of the coolant temperature when the engine is warm.
- Reducing the EGR cooling: just as with passenger cars, it is possible to support the aftertreatment during a cold start or in low engine load conditions by bypassing the EGR cooler. With two stage, LT-EGR cooling, it is possible to choose whether both or just the second stage should be bypassed, Figure 12.

6 Summary and Perspective

The significant potential of two stage recirculated exhaust gas cooling to reduce NO_x emissions has been shown in this paper. With a Euro 5 combustion system it was possible to reach 0.85 g/kWh NO_x over the ESC without a fuel consumption penalty in comparison to a Euro 5 engine with single stage EGR cooling. Further development of the combustion system, with fuel injection pressures to 2500 bar and two stage boosting, which allows higher EGR rates, will determine whether NO_x reduction with exhaust gas aftertreatment is necessary. Indirect charge air cooling will supplement the two stage boosting, since this improves the boost delivery during transient operation. The reduction in EGR temperature variation with two stage cooling and indirect charge air cooling will further support the development of the future combustion systems. Finally, the lower rate of fouling in the low temperature EGR cooler will help to keep the needed engineering margin to the emissions limits at an acceptable level.

The fuel consumption relevant advantages of low temperature EGR cooling are the significantly reduced inlet manifold temperatures, the reduced boost pressure requirements (due to the increased EGR density) and the reduction in the cooling requirement compared to conventional EGR engines. These advantages would also be present if a combination, EGR and SCR, strategy was used: compared to a conventional engine reduced fuel and urea solution consumption could be expected.

For the SCR-only solution it will be necessary to achieve high NO_x conversion rates with the lower exhaust gas temperatures of the new test cycle. In this case controllable charger air cooling, as is possible with indirect charge air cooling, would be helpful.

Through the development of cooling systems for Euro 5, Behr can now offer technology for Euro 6 which offers a significant contribution for all emissions reduction strategies: they are unavoidable for a strategy that relys only on EGR. The improved cooling system performance that is necessary, is part of the ongoing development activities and will be available for application at Euro 6.

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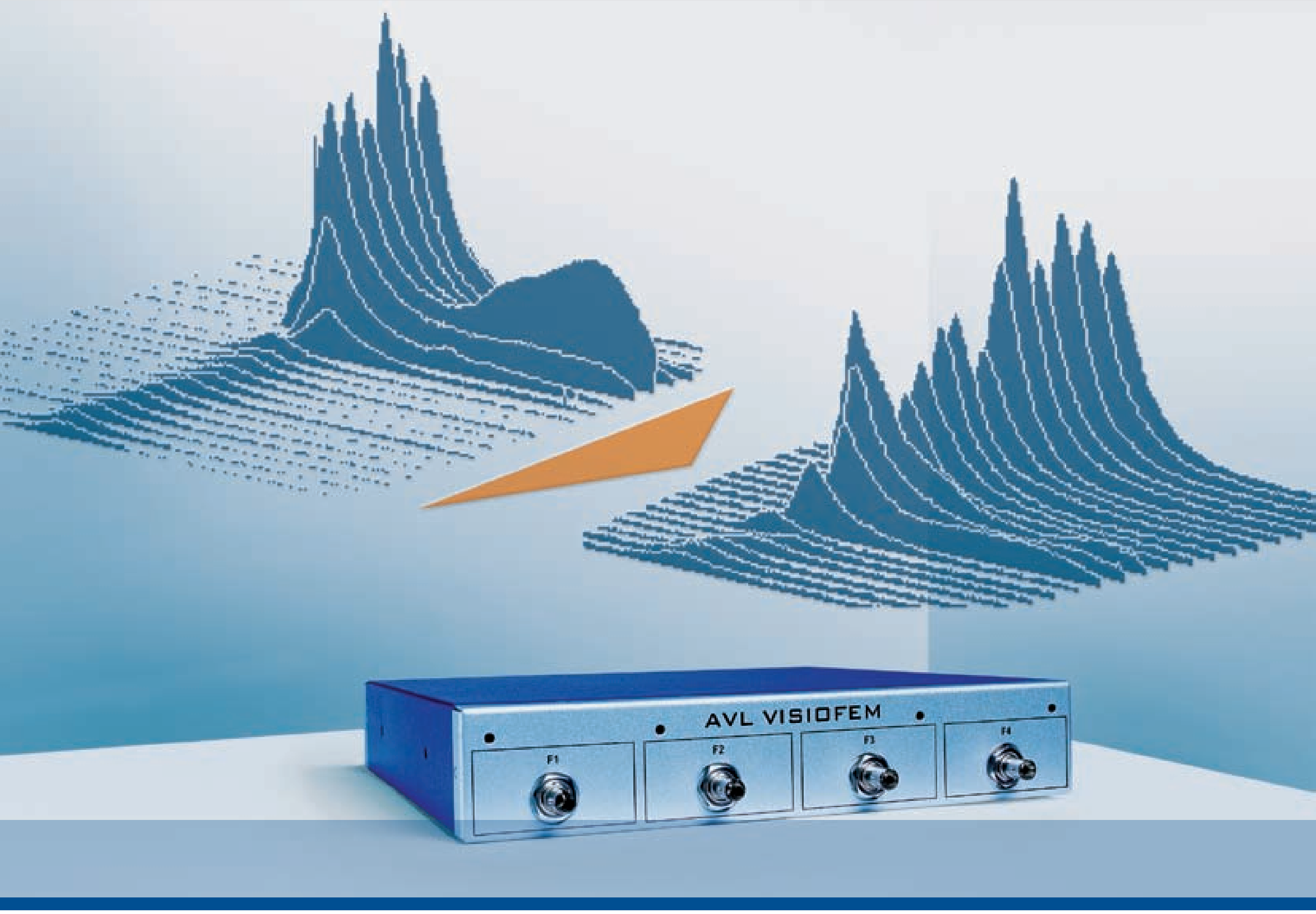

Transient Mode Calibration – Mixture Quality Evaluation

AVL List GmbH presents a measurement technique which enables evaluation of mixture formation quality in order to support calibration of emissions relevant SI engine operation. The method is especially suited to identify individual cycles' particulate emissions contributions at engine start and in transient operation. Sensors, signal recorders and signal evaluation are part of standard test bed and on-board analysis systems.

1 Introduction

"Gasoline engines have perfect emissions quality in stationary part load operation – how to ensure such ideal operation also at engine start, at tip-in tip-out, and at full load conditions?"

Key to the solution of these issues is the optimum use of components for mixture formation, combustion and exhaust gas aftertreatment. For this all, gasoline direct injection (DI) offers an ideal basis, as fuel metering for each cycle and cylinder is under the active control of rail pressure and injection duration. Such precise control, however, must also support mixture formation quality to enable best combustion.

2 The Mixture Quality Measurement Technique

The technique exploits radiation features of premixed versus diffusion flames. In the ideal case, combustion of a stoichiometric, homogeneous mixture in a SI engine yields a premixed flame. Deviations from premixed flame features are identified by the measurement system. The in-cylinder measurement technique yields cycle and cylinder specific data records, which is a prerequisite for the evaluation of transient engine emissions tests. Its application is focused on supporting actuator calibration in test cell and on-board emissions optimisation.

3 Mixture Quality – Flame Radiation in SI Engines

Premixed combustion has the primary feature of almost synchronous rise and fall of rate of heat release and flame radiation intensity [1].

At diffusion combustion, liquid fuel evaporates under influence of hot in-cylinder gas and burns in a fuel rich environment. Lack of oxygen results in sooting diffusion flames with high radiation intensity. Such diffusion flame radiation can easily surpass premixed flame radiation and, furthermore, is largely unrelated to rate of heat release. Identification of such diffusion flames is the basis for evaluation of mixture formation quality [1].

Flame radiation is transmitted with a wide aperture optical sensor and is converted with a broad band photo diode. The crank angle resolved flame signal together with the cylinder pressure trace allows identification of premixed and diffusion flame features. Signal examples for part load combustion in a DI gasoline engine are given in **Figure 1**. Both cycles have nearly identical rates of heat release. Flame radiation, however, shows significantly different features. In cycle (A), heat release and flame radiation signals are essentially synchronous, with the falling rate of heat release, flame radiation as well is decreasing. In cycle (B), however, the flame radiation signal continues to rise long after heat release shows that the mixture has been consumed.

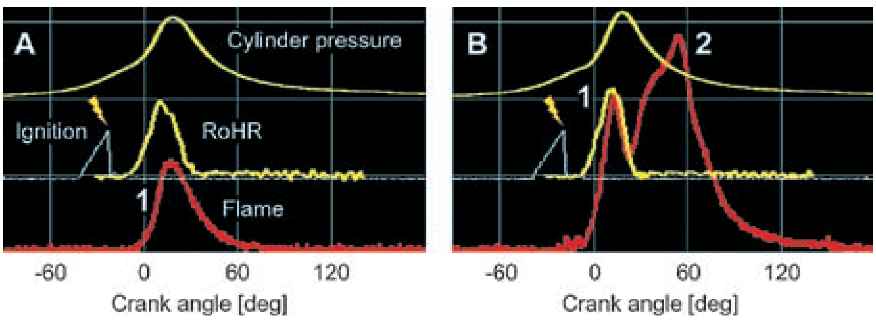

Figure 1: Cylinder presse and flame signals identify flame quality

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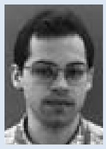

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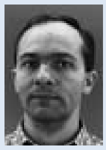

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Such simple combustion data can be summarized with two essential points:
- the luminosity of a flame contributing to noticeable rates of heat release is essentially synchronous with the rate of heat release
- any significant difference between flame luminosity and rate of heat release signals is a marker for imperfect mixture formation.

Figure 2: Sensors for pressure and flame measurement

4 Measurement System Requirements

Any useful engine measurement technique has to provide informative data and yield correct and consistent advice. Moreover, in an engine calibration environment, such tasks must be handled under quite restrictive circumstances. These requirements are fulfilled with
- a combustion chamber sensor to simultaneously measure cylinder pressure and flame radiation signals
- a signal evaluation procedure identifying premixed and diffusion combustion on basis of pressure and flame signals
- a results classification method to identify mechanisms responsible for mixture formation features in the cycles of interest.

Figure 2 shows pressure-flame sensor types and dimensions, sensor applications have been described in ref. [1]:
- VPSP (VisioPressure Spark plug): Spark plug sensors to be applied in standard engines, no adaptations on engine required.
- VPS (VisioPressure Sensor): In engines with suitably large cylinder pressure bores.

Components of a suitable measurement chain are given in **Figure 3**.

5 DI Gasoline Engine Evaluation Examples

5.1 Full Load Tip in Sequences

Combustion signal sequences under full load tip in conditions in a vehicle acceleration test are shown in **Figure 4** (top). Signals were recorded with a VisioPressure spark plug. In the following, the indicating system's cycle number counts are used to identify the sequential events of the tip in tests.

The engine load step starts with cycle no. 172. Cylinder pressure evaluation shows an IMEP increment from 4 bar (c 171) to above 21 bar within 3 combustion cycles (c 174). Thereafter, IMEP is slightly reduced. After 15 more high load cycles, engine load is cut off (c 185). Before tip in, the flame radiation signals recorded together with cylinder pressure, show the typical premixed signal pattern. After tip in, premixed combustion is found from cycle 179 onwards. What is happening in cycles 172 throughout 178 ?

The signal diagrams in Figure 4 (top) and the comments in **Table 1** make it evident that in these few cycles part of the injected fuel stays unburned long into the expansion stroke. This late combustion does not contribute to heat release, but provides noticeable, luminous flame radiation. The timing of this luminous flame is well separated from the premixed flame signature. The incremental growth of the late flame signal for some 4 cycles and the subsequent decrease within 3 more cycles suggests the gradual agglomeration of a fuel reservoir and its consumption with rising cylinder temperature.

A repetition of the tip in event in Figure 4 (bottom) shows that fuel agglomeration is significantly reduced as injection is suitably adjusted.

5.2 Engine Start

At start, a cold engine imposes extreme conditions for the formation of ignitable and combustible mixture. As the heat to evaporate fuel only becomes available from compressing the in-cylinder air, fuel droplets, after injection, must stay floating in air to benefit from this compression heating. This is supported with fuel injection systems and injection strategies which allow considerable reduction of fuel enrichment in starting engines [2].
- Criteria: A start sequence aims at reliable initial firing cycles, fast speed up, catalyst heating and engine stabilization. Cat lightoff is to be achieved in due time at lowest HC and particulate emissions, misfire must be avoided.
- Development procedures: Calibration of starting parameters is iterated at consecutively reduced engine temperature [1, 2].

5.2.1 Engine Speed at Start

After start, engine speed should follow the schematic given in **Figure 5**. Individual cylinder contributions in the speed up phase are evaluated with crank angle

Figure 3: Pressure and flame measurement system modules

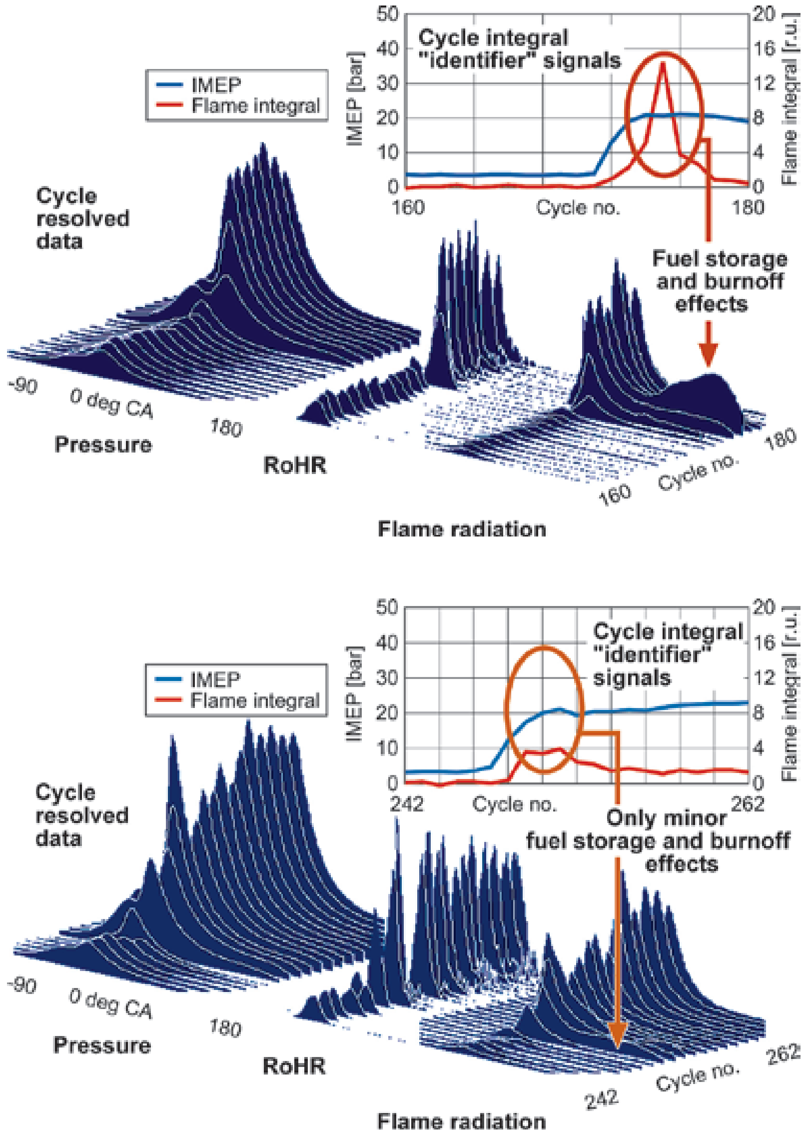

Figure 4: Cylinder pressure, rate of heat release and flame radiation provide a cycle by cycle analysis of mixture quality. Recognizing fuel storage events in (top) enables improvement of injection calibration to avoid transient soot (bottom)

Table 1: Test cycle analysis on basis of cylinder pressure, rate of heat release and flame radiation signals

Cycle no.	IMEP [bar]	Premixed combustion only	Late flame radiation event
171	4	yes	no
172	13		small
173	19		rising
174	21		rising
175	21		large, dominant peak
176	21		falling
177	21		falling
178	20		small
179	20	yes	no

resolved engine speed measurements. An example for the incremental speed up following the firing sequence of an in line 4-cylinder engine is shown in Figure 5. The graphs show the deceleration as each piston approaches firing TDC and the subsequent acceleration in the firing stroke. The example in part C of Figure 5 shows an inadequate start calibration. Cylinder 3 yields no speed increment, cylinder 4 shows a significant drop of engine speed until cylinder 2 again is firing up.

5.2.2 Faults and Root Causes

The analysis of faulty start cycles is accomplished with a cycle by cycle evaluation of speed, cylinder pressure and flame signals. **Figure 6** shows the pressure – flame data records for cylinder 4 with a cycle sequence including misfire and irregular ignition in the early compression stroke. The evaluation of these data is summarized in **Table 2**.

The flame and pressure signal evaluation shows excess fuel to be the root cause for misfire and irregular ignition events. Thus a clear advice is given to reduce overfuelling in these particular cycles. The above example shows that regular combustion only occurs after the agglomerated fuel is burned off in the irregular combustion cycle no. 7. Furthermore, with the measurement of pressure as well as flame signals, a refined interpretation of misfire cycles such as appear in cycle 5, 6 and 7 becomes available.

6 Calibration Environment

Engine and vehicle calibration tools access ECU parameters and record ECU signals as well as signals made available from exterior sensors. The data recording capabilities usually include analog signal inputs. This provides a simple platform to add the pressure and flame radiation signals to the calibration data records – see **Figure 7**. Mixture quality evaluation is then achieved with an IndiCom/Concerto access to the calibration system data.

In vehicles or test beds instrumented with on board indicating systems, the usual crank angle based signal recording procedures are employed. Results are immediately available from on line evaluation procedures.

Table 2: Evaluation of pressure, rate of heat relase and flame signals of Figure 7 data records

Cycle no.	IMEP [bar]	Flame signal	Signal evaluation
5	0	no	Misfire: no heat release, no flame
6	0	After ignition	Misfire: no heat release, combustion of rich mixture (wall film)
7	-1	In early compression stroke	Irregular ignition by residual combustion kernels (deposits) from cycle 6
8	8	Bright diffusion flame	Regular starting cycle, with rich diffusion flame
9	5	Bright diffusion flame	Regular starting cycle, with rich diffusion flame

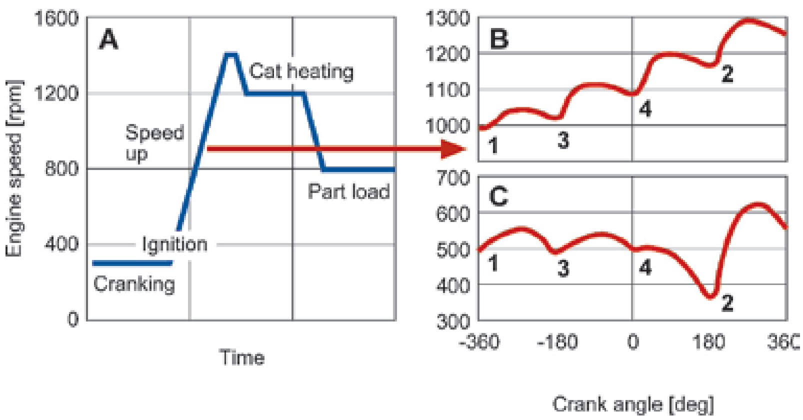

Figure 5: Engine speedup at start

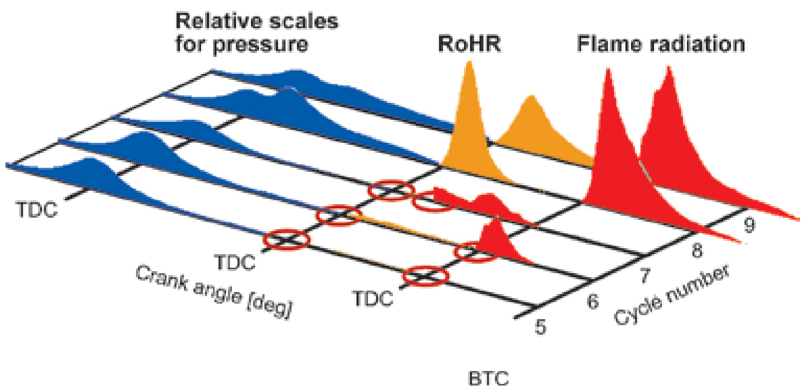

Figure 6: Cylinder pressure and flame analysis of misfire start events

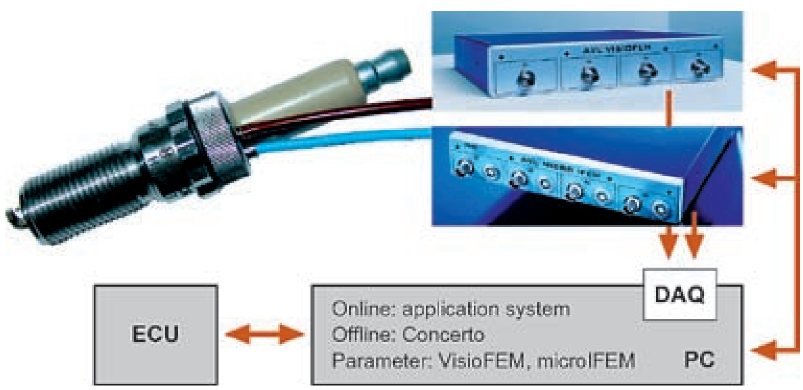

Figure 7: Configuration of on-board application system

7 Summary

Cylinder pressure, rate of heat release and flame luminosity signals provide the basis to identify premixed and diffusion flame combustion in gasoline engines. Diffusion flames in gasoline engines are either the signature for an articulate stratified combustion, or they are the result of insufficient mixture formation. Finding diffusion flames at nominally premixed combustion conditions, thus is evidence for the necessity to improve mixture formation.

In today's gasoline engines, emissions quality is first of all an issue for transient operation. This makes identification of emissions relevant individual cycles and cycle sequences a key factor in further engine out emissions improvements. This task is supported with the simultaneous measurement of cylinder pressure and flame luminosity signals and with on line signal analysis procedures.

Easy handling of such sensors allows straightforward application and usage by non-expert personnel. Data reduction provides a cycle specific "combustion quality index" with on line identification of critical combustion cycles. This especially enables engine calibration tasks to be handled with high precision and effectiveness.

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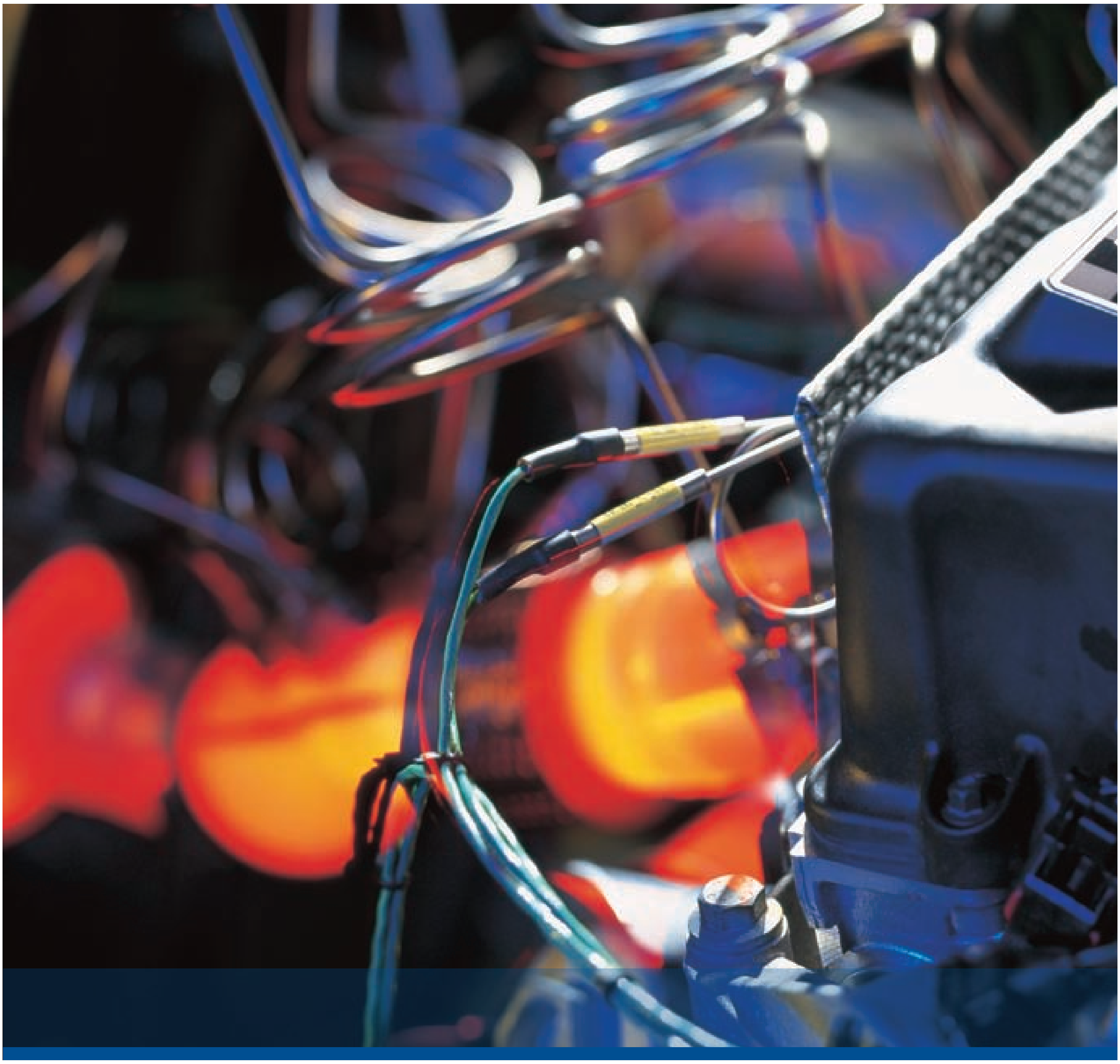

Additional Energy Converters in Future Vehicles

The current discussion of CO_2 emissions and global warming has brought energetic efficiency of vehicles into the focus of the efforts of developers and manufactures. Today's challenge is to harness the branched energy flow through vehicles more than hitherto considered necessary or even possible. The parameters governing the energetic and economic meaningfulness of such proposals are many and linked in a complex manner, as a research by Ricardo shows.

1 Introduction

The 2nd law of thermodynamics is a statement of the fact that conversion of heat to mechanical work never happens without emission of heat out of the converting system. In the case of combustion engines as applied to road vehicles, there is usually one input channel of energy, i.e. the supply of fuel which releases high temperature heat for conversion to mechanical work, which in turn propels the vehicle. There are a number of channels by which the emitted heat leaves the vehicle, the major two being the exhaust gas and the coolant circuit. Some of the work generated by the cycle is converted back to heat and emitted together with the exhaust and coolant heat. The proportion between useful work consumed in moving the vehicle and the energy rejected to the atmosphere by the process of conversion varies widely depending primarily on the operating conditions of the vehicle. The technical potential for improving the fraction of the energy input of which good use is made can be identified according to the notion of exergy and by the usefulness to the general operation of the vehicle. While large scale energy conversion is usually configured according to considerations of efficiency, vehicle systems often are dimensioned for bottleneck conditions and over time operate at compromised efficiency. Further consideration is due to certain aspects of the drive train e.g. exhaust gas after treatment systems which in some cases require energy input (heat to attain operating temperature) and in others are able to deliver energy in the form of heat. Compliance with regulations, safety and attractiveness in the market are things that cannot be ignored and many simplistic approaches that promise energy savings are thus invalidated. A number of technological options shall be presented here that might be building blocks in systems which can improve the utilisation of fuel. Drive train hybridisation is not a subject of this paper while some overlap with it nevertheless exists.

2 Background

The energy expended from well to wheel is utilised to fulfil a number of functions. While transport of persons and/or goods is the primary purpose, there are a number of secondary functions that need energy supply in order to maintain or enhance the usefulness of the vehicle. Lights to enable night driving, power assistance for steering and braking, heating or cooling for the cabin (in some cases for cargo as well) communication and entertainment. In a classical application case all these functions are supplied by the main engine, which is driving a number of auxiliaries to cover these needs and a few that are necessary for its own functioning, such as pumps for lubricant and coolant. Others like electric generators, hydraulic and vacuum pumps support vehicle systems. The requirements for all these auxiliaries, determined by e.g. their operation at a constant speed ratio to the engine while delivering their output dependent on parameters decoupled from this, are often in conflict with energetic efficiency. Assessing the dimensioning of pass car AC systems or calculating the cost of a kWh of electricity in a car and comparing this to the price in a household is quite revealing in this respect. Optimal utilisation of the fuel therefore calls for a systematic assessment of the character, the magnitude and the timing of each and every vehicle function that requires energy in order to find a rational system design.

The largest proportion of the requirements of the vehicle will be the work

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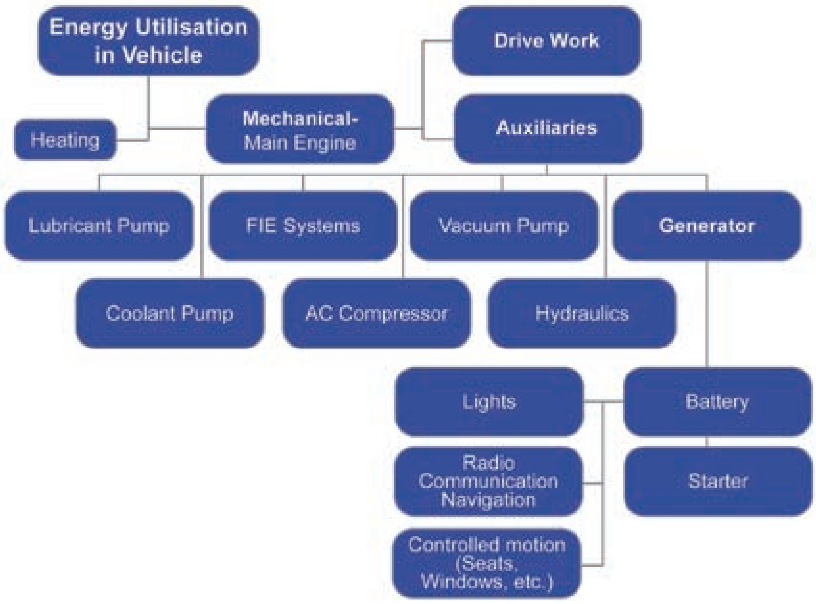

Figure 1: Energy utilisation in vehicle; the paths along which energy flow through the vehicle happens are helpful to identify possible forms of utilisation

done driving the vehicle. This is widely variable (Start-stop systems are an approach here). The ratio of the input of fuel energy and utilisation of mechanical work to drive the vehicle forward varies between the best efficiency of the engine and effectively zero (vehicle stationary, engine idling). Recent proposals to drive coolant and lubrication pumps electrically to match the requirements exactly go in a direction considering this. The advantage of lower load on these auxiliaries must be balanced against the longer chain of loss mechanisms though.

In order to have a basis for comparison between different vehicles and drive trains drive cycles have been defined (NEDC, FTP 75 etc.). Fuel consumption and emissions are compared based on measurements over these cycles. The actual figures in client hands differ considerably from those measured in this way. The need to identify the consequences of drive train hybridisation has brought about new drive cycles (ADAC, Artemis). It can be expected, that assessing the relative advantages of differing energy management systems will make further activities in this direction necessary. Currently developments in energy management and recovery concentrate on cases where the advantage is quite obvious like long haul trucks where exhaust gas heat converted to work can be fed to the drive train and the result is an increase in overall efficiency. **Figure 1** shows the energy flow through conventional vehicle architecture. The quest for better utilisation of energy in vehicles is guided by the exergy that the various energy flows out of the main engine carry. The highest exergy is obviously found in the exhaust. The stream of high temperature exhaust gas is thus the prime candidate for conversion into mechanical work. Work thus generated can then either be fed to the drive train or further converted into electricity. Storage of mechanical energy is generally possible in kinetic (flywheel) or potential (spring tension, gas pressure) form. The main challenge of mechanical energy storage lies in loading and unloading the storage and in the necessary mechanical arrangements to do this. Storage of electricity is a main branch of technical development work (batteries, capacitors, flywheel devices). Utilisation of stored electricity is usually more flexible than storing and recovering mechanical energy. Decisions must be determined by usability for the required function along criteria of power density, energy density and the relevant boundary conditions.

3 Combined and Bottoming-cycles

Combined cycles are dividing the temperature difference to be utilised into two parts. A bottoming cycle is one that converts the heat emitted by another cycle into work. The consideration for the individual cycles is determined by the economic, energetic and technological constraints governing the operation and development of these machines. In many cases such concepts are driven by peculiarities of the heat transfer apparatus and opportunities to shift temperature levels thus increasing the overall span. Depending on the use made of the output of the bottoming cycle sometimes it can be justified to compromise the efficiency of the main engine to improve the bottoming cycle, thus enhancing overall efficiency.

One instance for this is the Thermo Efficiency System (TES) promoted by MAN B&W Diesel [1] for large Diesel engines in ships, where timing on the main engine is changed and an exhaust bypass redistributes the exhaust gas heat from high amount/low temperature to low amount/high temperature. The efficiency of the main engine is somewhat reduced thereby. A boiler and steam turbine is fed from the exhaust gas heat and generates electricity used on board. The lower heat sink temperature for the steam cycle improves energetic efficiency.

In the prevailing situation it may become necessary to assess alternatives which are not well known. Thermodynamics supplies methods for the assessment of such combined cycles. It is obvious that the greater part of the available temperature span should be utilised by the cycle promising the higher figure of merit and that the optimum of the combined cycle is not a combination of the respective optima of the individual cycles, unless they coincide. In any case the efficiency of real engines is lower than implied by ideal cycles and often determined by peripheral apparatus like heat exchangers. A school of thinking [2, 3] that can be traced back to the work of Adrian Bejan [4] is proposing methodical approaches that allow to take such limiting effects into account without relying on concrete results won on actual machinery. **Figure 2** shows the technological options to utilise exhaust gas heat.

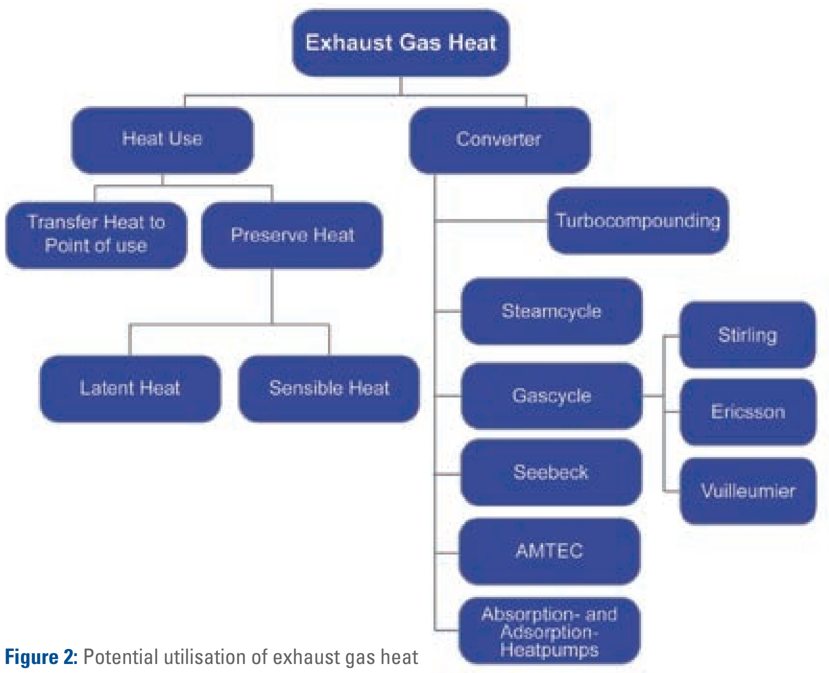

Figure 2: Potential utilisation of exhaust gas heat

4 Technological Options

A number of options exist as to what kind of device can be developed and how its output can be utilised for to harness energy flow that would otherwise leave the vehicle without being utilised.

4.1 Heat Use

The cabin heating by the engine coolant is the oldest and most obvious use of heat to fulfil vehicle functions. Also the issue of timing has surfaced with this. Since it takes some time after a cold start, for the coolant to reach useful temperatures separate heating systems have been conceived burning fuel to heat the cabin and the engine. Recently a system has been proposed [5] that utilises exhaust gas heat via a heat exchanger to heat up the coolant of the engine quicker (which also benefits the cabin heating, should this be required).

Various technologies have been proposed at times for the storage of heat. Depending on the Source and the intended use, either as sensible heat, i.e. by increasing the temperature of the storage medium, which does require effective thermal isolation, or by utilising substances (PCM) which show phase changes at appropriate temperatures, enabling them to absorb heat at constant temperature (e.g. by melting) and releasing this heat at a different time (by freezing). A carrier circuit is necessary to control time and place of heat supply and utilisation. The energy density of the whole device and its space requirement are determining factors for the applicability of such systems. Depending on the concrete demands of the heat utilisation the power density of the whole system is also a prime consideration.

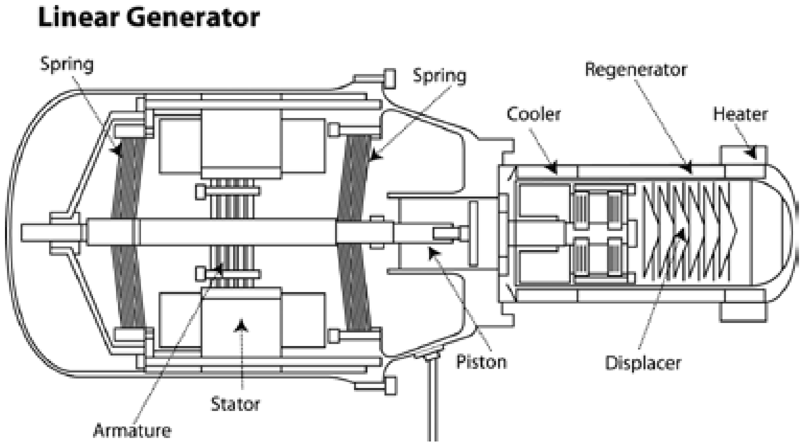

Figure 3: Free Piston Stirling Engine combined with linear electric generator as commissioned for deep space missions (Source Sunpower)

4.2 Conversion

In turbocompound engines the exhaust gas energy is utilised by means of a turbine which adds its work to the same shaft as the piston engine. Often a gearbox is used to synchronise the different speed of the machines. During the transition phase in aeroplane propulsion from piston engines to gas-turbines such turbocompuond engines were quite common, famous examples being Wright R 3350, Napier Nomad and Allision V1710E22. Currently turbocompounded engines are entering series production in heavy duty trucks [7]. The advantage in efficiency due to turbo compounding can not be evenly realised over the whole mapping to the same extent, depending on the calibration etc. A variant is the so called electric turbocompounding [6]. An electrical machine, which can work as a generator as well as a motor is integrated to the shaft of the turbocharger. This enables improved transient operation. The electricity generated at times when the exhaust energy is greater then the need of the Turbocompressor can be stored and/or fed to any sensible application. A comparison between turbocompounding and electrical turbocompounding is therefore strongly dependent on the operational profile of the vehicle.

Steam cycles have been proposed by various workers [8] and recently reduced to a practical demonstration by BMW [9]. In this case heat contained in the coolant of the main engine was also utilised. The advantage of steam cycles is good adaption to the heat source and sink temperatures by intelligent choice of the working fluid. Compact heat exchangers are achievable by utilising nucleate boiling and film condensation respectively. The process itself can be influenced by applying mixtures of working fluids. Reciprocating pistons and turbines can be used equally well as expanders. Piston steam engines show their maximum torque at standstill. If the power of the steam cycle is to be fed into the drive train, proper matching of speed is necessary. Limitations of the applicability of steam cycles exist in the complexity of the actual hardware and some peripheral considerations like the stability of lubricants in piston engines, depending on the working fluid.

Gas cycles comprise a variety of possible realisations. Open and closed systems using turbines or reciprocating pistons can be envisaged. Open cycle gas turbines are considered to be covered by turbocompounding.

A Stirling engine is a closed cycle regenerative machine. Hence the choice of working fluid and pressure level is free within certain boundaries. Stirling engines can be conceived with kinematical mechanisms or with free pistons, in single acting or double acting form. The efficiency of the cycle depends on the grade of regeneration and the formation of the working spaces by reciprocating pistons, in single acting engines piston plus displacer. The cycle efficiency, i.e. the figure of merit multiplying the Carnot factor can be high (> 0.7). Most important here is the dimensioning of the regenerator for the cycle. The overall efficiency is dependent on the interfaces to heat source and sink. Careful matching, especially of the heater to the exhaust gas flow is crucial when deciding on swept volume, working fluid and pressure level. The consideration of capacity flows is called for. It is expected that a well matched and designed Stirling engine can yield a higher work output at a less complex installation than a system based on a steam cycle. Also the matching of operating speed to the main engine seems possible. The heat rejecting circuit must be separate from that of the main engine. **Figure 3** shows a schematic diagram of a free piston engine similar to those commissioned for the NASA deep space missions.

Ericsson engines differ from Stirling engines being open cycle regenerative machines. A fresh charge of air is admitted through the regenerator for every cycle of operation. Their potential advantage over Stirling engines is that the heat load on the rejection circuit is lower. The main disadvantage is that the free choice concerning pressure level and working fluid is not possible.

The Vuilleumier machine is a closed cycle regenerative machine like the Stirling engine. In three working spaces with two regenerators a cycle absorbing heat at a high temperature (driving heat flow) and a low temperature (useful cooling) while rejecting heat at a intermediate temperature. It is therefore a heat driven heat pump. The aggregate volume of all three working spaces is constant over time. By choosing bores and strokes it can be made variable, so that the torque required to move the pistons is covered by a superimposed 'quasi Stirling cycle'. This could potentially extend to give an engine that produces cooling and work simultaneously. The dimensioning and design follows Stirling engine practice closely. The main issue with this is the increased space requirement of a Vuilleumier engine over a Stirling. **Figure 4** shows a Vuilleumier heat pump conceived for domestic heating. While in all cases mechanical work produced by engines can be converted into electricity incurring another loss due to the efficiency of the generator, it is also possible to generate electricity directly from heat.

Seebeck cells are semiconductors which generate electricity dependent on the temperature difference. Recent advances in materials science are raising expectations, that significant efficiency can be reached. The figure of merit combined from the Seebeck-coefficient, the electric and thermal conductivity and the Temperature level has been improved by a factor of more that three in the last years. The main advantage is the absence of any moving parts. It should nevertheless not be forgotten that just like any other heat exchanger the heat transfer rate is coupled to the pressure differential across the receiving element.

The Alkali metal thermal to Electric Conversion, Amtec is an energy converter that originated in GM research in 1968. Its working principle is that sodium vapour becomes ionised when heated, the ions are then able to permeate the solid electrolyte (Beta Alumina Solid Electrolyte – BASE). The electrons flow through the external electric load. Some development effort [10, 11] was made in the mid to late 90s and while the temperature required tends to be high (600 °C) an efficiency of 20 % is reported as being demonstrated, while projections went as far as 30 to 40 %. **Figure 5** schematically shows the principle of the Amtec.

Absorption and Adsorption cycles have been proposed to utilise exhaust gas heat to produce cooling power. The working principle is well known. A pair of substances either refrigerant and solvent (absorption) or refrigerant and sorbens (adsorption) are utilised to change between two states and thereby absorb and emit heat at different locations of the apparatus. There are quite some industrial and household applications currently in the market. Cold production in a vehicle has two possible uses. First it can be utilised for the air conditioning and so reduce the load on the main engine by the amount normally consumed by the refrigerant compressor. Secondly there is an opportunity to improve the operation and efficiency of the main engine (especially a gasoline engine that is knock limited) by cooling the intake air below a temperature that could be achieved by a charge air cooler.

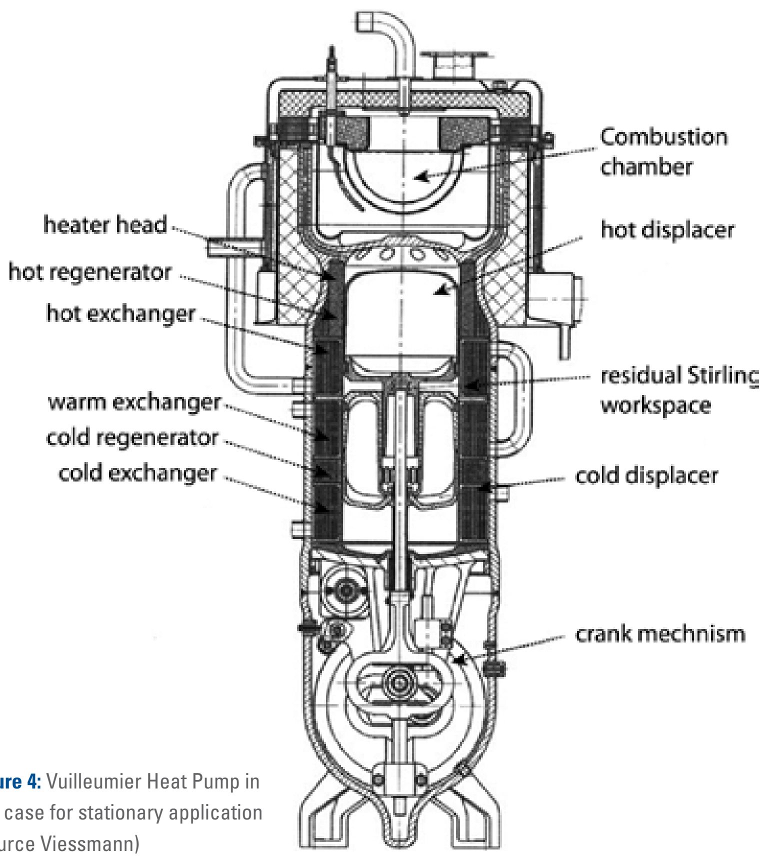

Figure 4: Vuilleumier Heat Pump in this case for stationary application (Source Viessmann)

5 Conclusion

It has been shown that a great variety of options exist to improve the energetic efficiency of vehicles by adding an energy management system comprising energy converters. While many development efforts have already been started, some of the technologies are little known or have hitherto not been applied to automotive

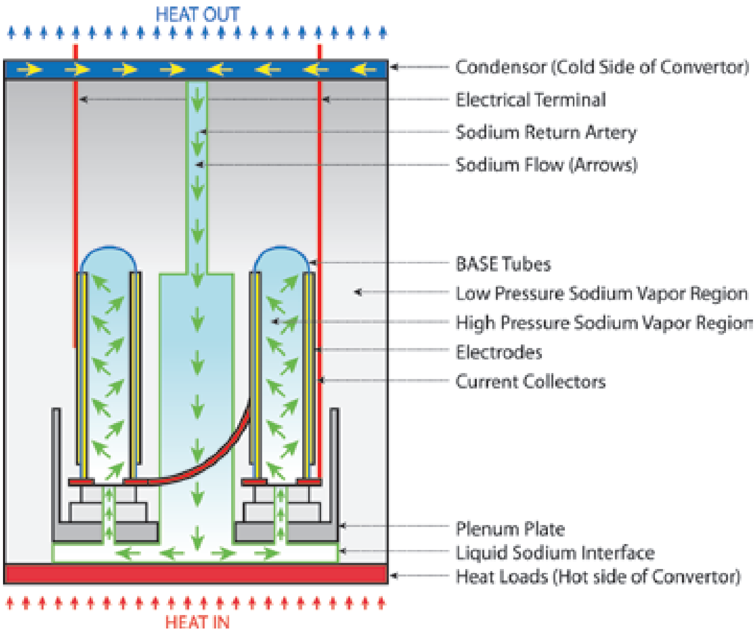

Figure 5: Amtec generator (source [10])

applications. Simple and easy to achieve yet effective measures like heating engine and transmission oil with exhaust gas heat will likely be the first implementations in passenger cars. Other early adopters will be applications where the operational regime allows simple and robust assessment of expected advantages. The prime example being long haul trucks where exhaust gas heat recovery to be converted into additional drive work. Considering conversion into mechanical work will have to take into account how the demand for energy based functions and services is related with vehicle use. Energy storage will play an important part in this. Stored electricity can serve all purposes that mechanical power can. The advantage is full flexibility in timing and matching the respective demand. The downside of this is the added complication and cost of electric motors and storage capacity. A thorough analysis of energy flows that can be tapped and functions and services, the demands of which are decoupled from main engine running conditions, is called for. The vehicles for which considerations are due vary widely, from the new cheap little passenger cars with modest comfort requirements to long haul trucks with additional functions like cooling the load bay. Hence there is no single solution for a system, or even a converter to cover all, or even a majority of applications. It is expected that the differentiation of vehicle architectures will continue.

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Subaru Boxer Diesel

First Four-Cylinder Horizontally-Opposed Diesel Engine

The integration of the horizontally-opposed engine and the functional principle self-ignition is a technical challenge. With the series production of the first boxer diesel engine for passenger cars Subaru has achieved a technological breakthrough. While producing maximum power of 110 kW and maximum torque of 350 Nm, the carbon dioxide (CO_2) figure of the 2.0 l four-cylinder diesel engine is 148 g/km. Low noise, low vibration and compactness are among its features.

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1 Introduction

Subaru, the automotive brand owned by Fuji Heavy Industries, has delivered horizontally-opposed petrol engines to the market since 1966. In 1989, the second-generation horizontally-opposed petrol engines were developed and installed sequentially on Legacy, Impreza and Forester. These engines have convinced with driving pleasure and comfortable touring space, as well as driving performance represented in the World Rally Championship. The Subaru boxer diesel engine has been newly added to the line-up, **Figure 1**. It integrates the technologies of fuel injection, supercharging and exhaust gas aftertreatment, together with the technology of horizontally-opposed engines. Clear target was the development of a diesel engine without sacrificing the benefits of the Symmetrical All Wheel Drive (AWD). The core-technology is not possible without a boxer engine: Its low height ensures a low centre of gravity and a nearly ideal weight balance. The horizontally-opposed diesel engine is provided to European markets since March 2008 on the models Legacy and Outback.

2 Aim of the Development

The aim of the development of boxer diesel engine was to create an environmentally-friendly engine with specific Subaru-performance, driving dynamics and low CO_2 emissions. With the latest common rail system, a variable geometry turbocharger and a large Exhaust Gas Recirculation (EGR) cooler, it responds to the requests in Europe for environmental protection and engine performance. In order to make use of the features of horizontal-

Figure 1: Subaru boxer diesel engine

Table 1: Specifications of the engine

		EE20 Boxer Diesel	EJ20 (Petrol)	EZ30 (Petrol)
Displacement	cc	1998	1994	2999
Maximum Power	kW (PS)/rpm	110 (150)/3600	110 (150)/6000	180 (245)/6600
Maximum Torque	Nm (kgfm)/rpm	350 (35.7)/1800	196 (20.0)/3200	297 (30.3)/4200
CO_2 Emission	(g/km)	148 (Saloon MT)	202 (Saloon MT)	286 (Saloon MT)
Fuel Economy	l/100 km	5.6 (Saloon MT)	8.6 (Saloon MT)	12.1 (Saloon MT)
Compression Ratio		16.3	10.2	10.7
Bore x Stroke	mm	86.0 x 86.0	92.0 x 75.0	89.2 x 80.0
Bore Pitch	mm	98.4	113	98.4
Bank offset	mm	46.8	54.5	46.8
Deck height	mm	220	201	202
Diameter of Journal	mm	ø 67	ø 60	ø 64
Diameter of Pin	mm	ø 55	ø 52	ø 50
Con-rod Pin Distance	mm	134	130.5	131.7
Diameter of Piston Pin	mm	ø 31	ø 23	ø 22
Compression Height	mm	43	33.5	30
Fuel System		Common Rail	MPI	MPI
Turbocharger		Variable Turbo	–	–
EGR		Liquid Cooling	–	–
DPF		Open Type	–	–
Engine Length	mm	353.5	414.8	438.4

ly-opposed engines, the development placed emphasis on the followings:
- low noise, low vibration and high liveliness
- direct and sporty response
- reduced weight
- compactness
- high rigidity
- low centre of gravity
- fuel economy.

3 Specifications of the Engine

Table 1 shows the boxer diesel engine's major specifications in comparison with petrol engines. **Figure 2** shows a cross-sec-

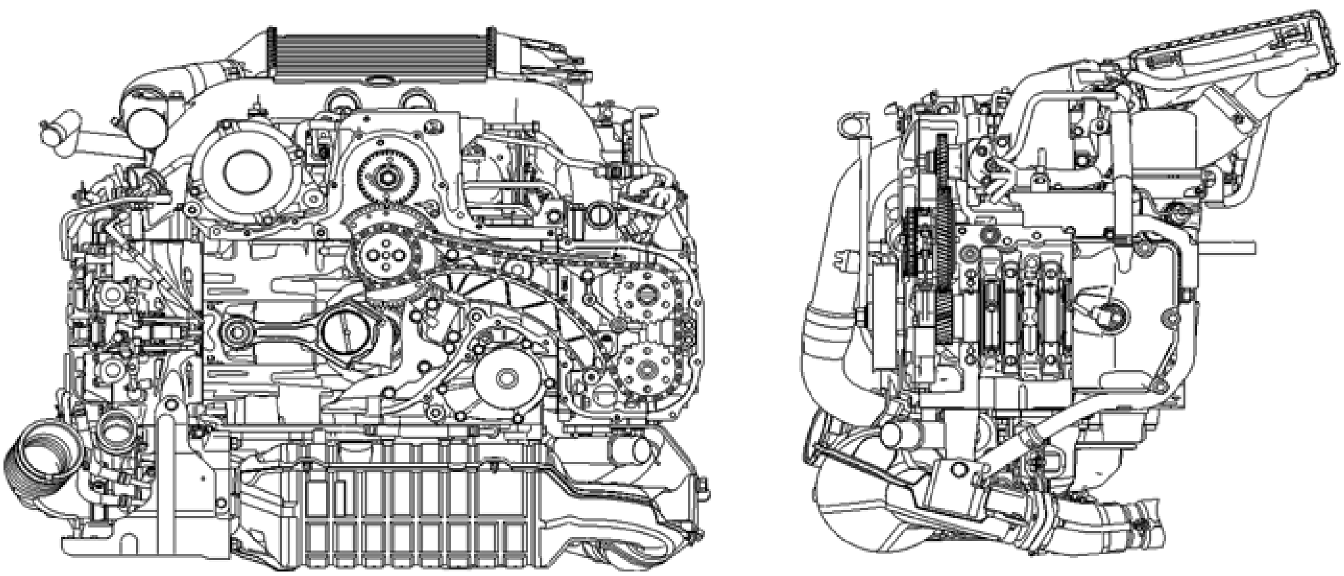


Figure 2: Cross-section of the Subaru boxer diesel engine

Table 2: Comparison of inertia force and moment

		H4 engine
Unbalanced inertia force	1st-order	0
	2nd-order	0
Unbalanced moment	1st-order	0
	2nd-order	$-2m_{rec}\,\omega^2 rs1/\lambda\cos 2\theta$
		L4 engine
Unbalanced inertia force	1st-order	0
	2nd-order	$4m_{rec}\,\omega^2 r1/\lambda\cos 2\theta$
Unbalanced moment	1st-order	0
	2nd-order	0

m_{rec}: reciprocating weight; ω: crank angle speed; θ: crank angle;
r: crank radiator; λ: connecting-rod lenght / crank radiator; s: bank offset

tion of the engine. In order to fulfil the aim of the development the following technologies have been adopted:
- compact design
- low friction and low inertia moment
- lightweight and highly-rigid cylinder block and crankshaft
- common rail system with 1800 bar fuel injection pressure
- lightweight, compact and dedicated solenoid injectors
- variable geometry turbocharger mounted underneath the engine
- exhaust gas aftertreatment system installed downstream of the turbocharger.

Owing to its mechanical system properties, a horizontally-opposed engine does not produce any unbalanced inertia forces from the reciprocating piston system weight. **Table 2** shows the unbalanced inertia forces and unbalanced moments – first- and second-order – compared between a four-cylinder horizontally-opposed (H4) engine and a four-cylinder inline (L4) engine [1]. Note that engine's rolling moments are not included in Table 2, both for H4 and L4.

The second-order unbalanced inertia force of an L4 diesel engine is large, more than 10 kN at 4000 rpm. On the other hand, the second-order unbalanced moment of the boxer diesel engine (H4) is around 250 Nm at 4000 rpm, having little impact on engine vibration. Thanks to such good engine vibration characteristics, the boxer diesel engine does not need a balance shaft, and realises low inertia moment, low friction and reduced weight, **Figure 3**.

For achieving good diesel combustion, the piston stroke of the boxer diesel engine has been extended compared to that of an H4 petrol engine. The engine overall width is kept equivalent to the H4 petrol counterpart, ensuring the mounting capability to existing vehicles. With a shortened bore pitch, the engine overall length is 61.3 mm shorter than that of an H4 petrol engine. This has made the boxer diesel engine more compact than its H4 petrol counterpart.

For better performance of exhaust gas purification and a lower centre of gravity, the turbocharger layout has been revised. In an H4 petrol engine, the turbocharger is placed above the back right side of the engine. In the Subaru boxer diesel engine, the turbocharger is placed underneath the front right side of the engine. In addition, the variable geometry turbocharger enables efficient supercharging in all operating ranges, controlling the exhaust turbine's vane opening angles according to the operating conditions.

Placed closely downstream of the turbocharger is the exhaust gas aftertreatment converter with an oxidation catalyst and an open Diesel Particulate Filter (DPF) inside. This layout has shortened the warm-up time of the catalyst to activate in a cold condition, enhancing the emissions conversion performance in wide ranges. The oxidation catalytic converter has been positioned below the engine together with the turbocharger. It separates unburnt fuel into water and CO_2. The unit has been made compact enough to be activated soon after the engine has been started. If the temperature rises to 250° C under certain driving conditions, the oxidation catalytic converter generates nitrogen dioxide (NO_2) which oxidise the collected diesel particulate inside the DPF. In addition, an EGR cooler is adopted for Nitrogen oxide (NO_x) reduction to comply with Euro 4 emissions standard.

The fuel system adopts a common rail system with maximum fuel pressure of 1800 bar [2]. It consists of a high-pressure fuel pump and eight-hole solenoid injectors placed at each cylinder, capable of multiple injection events. Compact-size solenoid injectors were developed exclusively for the boxer diesel engine, in order to make the engine width equivalent to that of petrol engines.

The combustion chamber is of a re-entrant type. Together with atomisation characteristics and swirls in the combustion chamber, it has allowed the fuel economy, engine performance and emissions performance to be satisfied at the

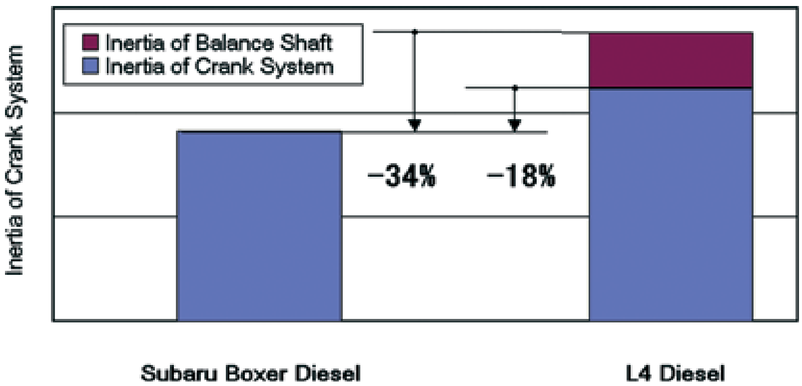


Figure 3: Comparison of the inertia moment on the crankshaft system

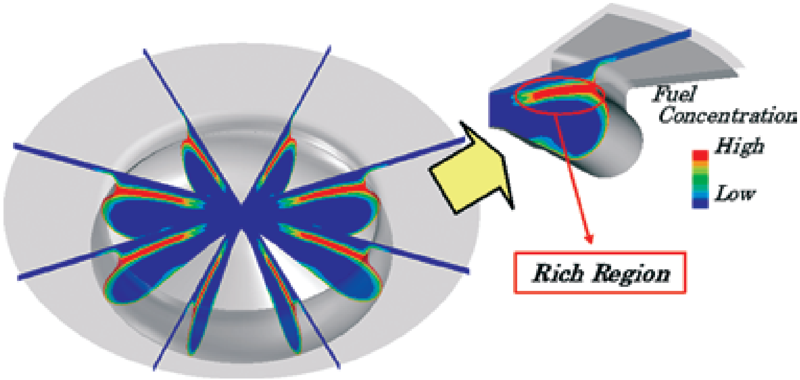


Figure 4: Study of combustion chamber shape by CFD

same time. The compression ratio is 16.3:1, and the maximum combustion pressure is 18 MPa. For the study of combustion chamber shape, Computational Fluid Dynamics (CFD), **Figure 4**, has been utilised.

4 Characteristics of the Engine

4.1 Low Noise, Low Vibration and High Liveliness

The combination of the Symmetrical AWD and the boxer principle has maximised the potential of the engine that works favourable to noise and vibration performance, and has successfully decreased the noise specific to diesel engines. This realises small noise and vibration in all ranges from idling to high speed.

4.1.1 Highly Rigid Drivetrain by Symmetrical AWD

To deal with the large torque fluctuations of diesel engines, a dedicated dual mass flywheel clutch system is employed to reduce noise and vibration in the drivetrain. In many of the front-engine rear-drive vehicles, the rubber coupling mechanism of propeller shafts absorbs noise and vibration in the drivetrain. This mechanism, however, reduces the rigidity of the drivetrain. Even if a dual mass flywheel clutch system is employed, many resonance points are generated, which would cause the booming noise of 100 Hz or less. On the contrary, the Symmetrical AWD system can absorb the diesel engine's large torque fluctuations with its four wheels. Therefore, the vibration at each part that suspends the drivetrain is small. In addition, since it is a highly rigid drivetrain system that does not use rubber coupling, there are only a few resonance points that cause the booming noise of 100 Hz or less. Thus, a pleasant interior noise is realised from the very start of the driving, with low vibration and no booming noise.

4.1.2 High Balance Performance by Boxer Layout

A boxer engine balances out the inertia force of reciprocating parts. Therefore, a secondary balancer system is unnecessary. Compared to an L4 engine with secondary balancer system, the second order engine vibration is low, from low- to high-speed ranges, **Figure 5**.

4.1.3 Low Vibration by Highly Rigid Crankshaft

The crankshaft of a boxer engine is shorter in overall length, and thus more rigid, compared to that of an L4 engine. When the rigidity is low, the crankshaft causes resonant torsional vibration in high-speed range of actual driving, which results in engine vibration. On the contrary, with a boxer engine, the crankshaft's resonant torsional vibration does not occur in high-speed driving, **Figure 6**. Therefore, the interior noise is clear even at high speed.

4.1.4 Reduced Combustion Noise

For reducing combustion noises that are specific to diesel engines, dedicated acoustic shields are employed for the cylinder block, intake manifold and transmission housing case, all of which are main sources of radiation. This has decreased the combustion noise of the power unit as a whole, **Figure 7**, and realised a quiet diesel engine at idling and in all speed ranges.

4.2. Direct and Sporty Response

4.2.1 Engine Output Performance

The engine develops a maximum output of 110 kW at 3600 rpm and a maximum

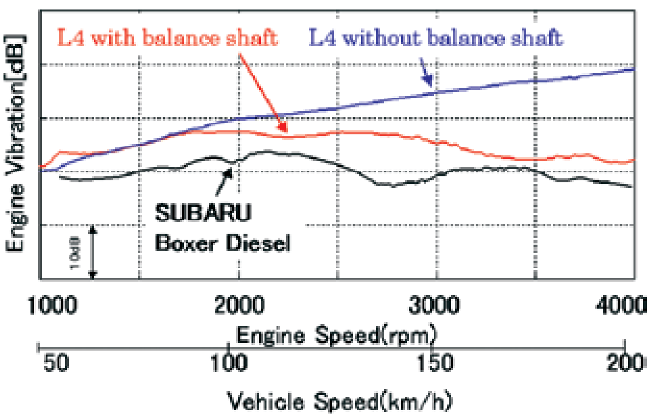


Figure 5: Engine vibration (second order)

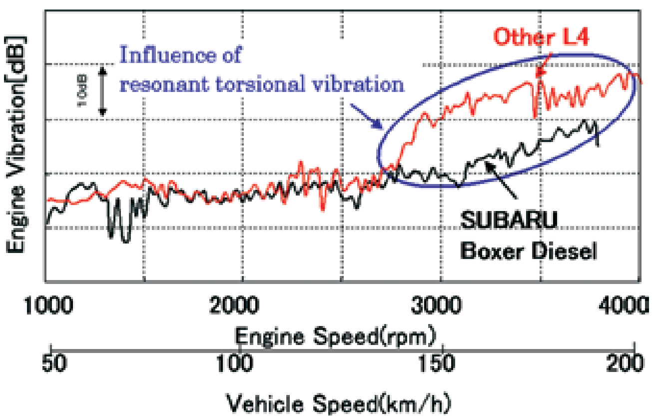


Figure 6: Engine vibration (sixth order)

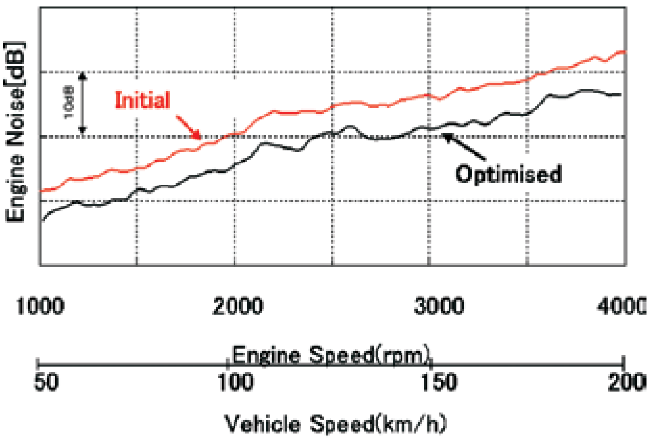


Figure 7: Engine noise (octave sound pressure level 2 kHz)

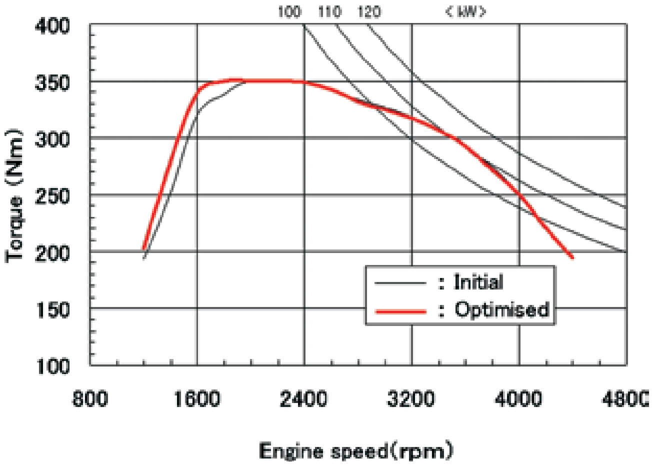


Figure 8: Engine output performance

torque of 350 Nm at 1800 rpm. While the torque curve gradually starts to descend in the latter half of 2000 rpm, the output keeps increasing, even after the peak of 3600 rpm, the engine will revolve briskly up to the rev limit of 4750 rpm, **Figure 8**. This is characteristic for the Subaru boxer diesel engine.

Compared to an L4 engine, the crankshaft is 50 % lighter in weight and the rotational inertia is reduced by 18 %. Considering the elimination of the balance shaft, the inertia moment is reduced by 34 %. With such low inertia, the boxer diesel engine has achieved good response to the accelerator pedal position.

4.2.2 Turbocharger

A variable geometry turbocharger is employed, which is capable of an optimum control of the exhaust gas flow at the turbine inlet according to the operating range, **Figure 9**.

While the engine extensibility at high speed was already sufficient with the engine specification in the mid-development stage, emphasis was put further on the turbine size and the specification of variable vanes in order to realise the engine flexibility, responsiveness and driving feel for controllability at low speed. Thus, the torque at 1800 rpm and below by a maximum of 20 Nm could be improved, as well as the transient characteristics.

An intended low-speed performance has been realised, while maintaining the maximum output and other high-speed performance.

4.2.3 Acceleration Performance

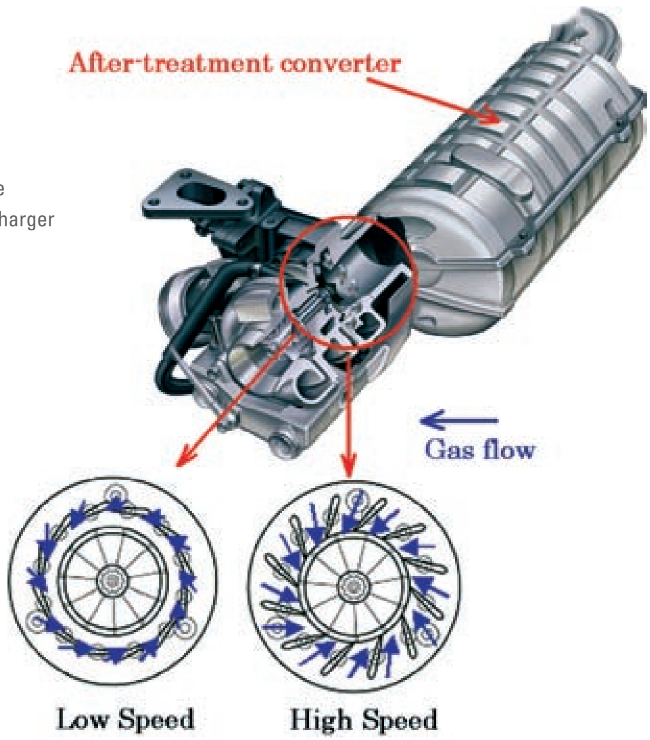
The developers of the boxer diesel engine focused on intensive tuning for controllability, aiming at an acceleration feel linear to the acceleration pedal stroke. The acceleration time in second and top gear is the evaluation index for the engine muscularity in low- and mid-speed ranges frequently used in European roundabouts and Autobahns. The acceleration performance of the boxer diesel engine outweighs that of a naturally-aspirated petrol engine of Subaru, and is superior to those of other passenger car diesel engines of the same displacement.

4.3 Compactness and High Rigidity

4.3.1 Basic Configuration of the Engine

The basic configuration of an H4 engine enables a compact, lightweight and highly rigid design compared to an L4 engine. It offers a good noise and vibration behaviour, engine performance and fuel economy. In order to make the

Figure 9: Variable geometry turbocharger



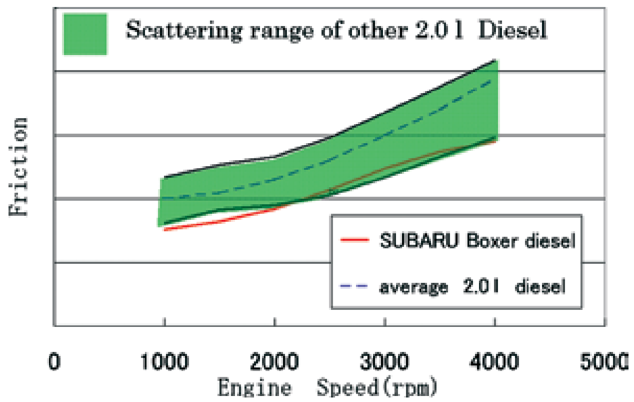


Figure 10: Motoring friction

stroke longer than that of an H4 petrol engine while keeping the engine overall width equivalent to a petrol engine, the service hole in the cylinder block, which served as a guide in assembling the piston pin and the connecting rod, was eliminated and instead a diagonally fracture-split connecting rod employed, whose cap is assembled from the oil pan side. This minimised the increase of the cylinder deck height.

The large ends of the connecting rods feature an asymmetrical profile, which increases precision during assembly and in roundness of the surface connecting the crankpin for reduced friction. It has also contributed to minimise the rotational path thus enabled to employ an extended piston stroke inside the compact cylinder block. Moreover, the compression height has been reduced by adopting the pistons made of high-strength aluminium alloy and by lowering the piston temperature with forced oil cooling using cooling channel. In addition, the cylinder head height has also been reduced. All together the new boxer diesel engine is compact with an overall width equivalent to that of an H4 petrol engine.

4.3.2 Cylinder Block

The cylinder block in a horizontally-opposed engine is lightweight and highly rigid. For higher strength and rigidity all five main journals in the cylinder block incorporate metal matrix composite journals which are inserted during the casting process, resulting in higher levels of quietness due to its high rigidity and similarity in the thermal expansion ratio to that of crankshaft. This is a technique that has already been validated in H4 petrol engines. The block design uses the semi-closed deck type that has proven its durability in the turbocharged petrol models. This increases rigidity around the head gasket mating areas.

4.3.3 Crankshaft

The deflection of the crankshaft under a high combustion pressure is smaller in a horizontally-opposed engine than in an in-line engine, since the span of the main journal receiving the load is shorter. Therefore, the load at the bearing end is reduced, where the oil film thickness is at smallest. This has led to higher reliability with a narrower width of the bearing compared to an in-line engine.

Krupp ahead of Subaru?

The world's first boxer diesel engine for a passenger car, which was described as a "revolution" by Subaru, does in fact have a predecessor, although not in a car. Exactly 75 years ago in 1933, Krupp launched a boxer diesel for light trucks. Like the Japanese engine, Krupp's version also had a cast aluminium crankcase and, to save weight, the designer Adolf Roth (1892-1966) used air to cool the engine. The result was the industry's lightest engine (325 kg, compared to Subaru's 160 kg, wet) and a power-to-weight ratio of 6.5 kg/PS (Subaru 1.07 kg/PS).

A further highlight was the fact that the diesel engine could be easily converted back into its original form as a petrol engine (1932) by replacing a few parts. Turbocharging, which gives the Subaru engine an extremely agile response, was not introduced by Roth until 1934 – for the two-stroke version, which had a specific output of 20 PS/l (Subaru 75 PS/l). Although the figures prove that a good deal of evolution has taken place, Subaru can count itself lucky that Krupp never built cars. *Erik Eckermann*

4.3.4. Cylinder Head and Valve System

High strength cylinder heads have been used to withstand the high combustion pressures. Compact and low friction end pivot type roller rocker arms have been used in combination with a Double Overhead Cam system. The diameter of the intake valves has been optimised for enhanced breathing performance and swirl ratios, resulting in improved combustion efficiency. The combination of an intake swirl port system and optimised intake valve diameter results in ample swirl performance. A highly durable chain system has been used to drive the camshaft to handle the variations in torque produced by the diesel engine.

4.4 Fuel Economy and Engine Friction

For less CO_2 emissions, the friction loss is kept at a low level. Since a horizontally-opposed engine counteracts the second-order acceleration of reciprocating inertia weights such as pistons, and thus allows the absence of the balance shaft and the system that drives it, the friction loss is reduced. The roller rocker arm is adopted in the valve system and also low-viscosity lubricating oil is employed. All together, a friction of the boxer diesel engine could be achieved, which is among the lowest level among the 2.0 l class cars, **Figure 10**.

The specific fuel consumption has been lowered in all speed ranges, by reducing friction in the engine hardware, and by calibrating the engine to satisfy both fuel economy and performances in a most balanced way. As a result, the CO_2 figure of the Legacy is 148 g/km, which is among the lowest in 2.0 l class cars, despite the fact that it is an AWD saloon.

4.5 Overview of the Control System

4.5.1 System and Hardware

The Engine Control Unit (ECU) employs the supplier's standard ECU hardware, allowing a significant cost reduction. Systems other than the fuel injection system, such as the communication system, the air conditioner, the radiator fan and the cooperative control with the Antilock Brake System and the Vehicle Dynamics Control systems, are shared

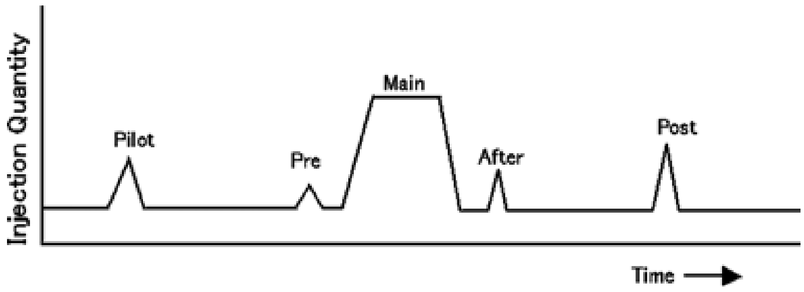


Figure 11: Multiple injection events

with the current petrol engine vehicles so that revisions are minimised and the original ECU potential suited to the Legacy is maintained.

4.5.2 Controls

The injector is capable of multiple injection events, up to five injections in one stroke, **Figure 11**. It is important for the injector to inject the fuel in high accuracy so that low noise and vibration of the boxer diesel engine is realised. There are two measures taken for this purpose:

– Injector Qick Response (QR) Code written into ECU: The dispersion data of each injector's properties are printed in the form of a QR Code (a two-dimensional bar code) at the top of the injector. In the engine plant, the code reader reads the QR Codes for all cylinders of the assembled engines. Later in the vehicle plant, these data are written into the ECU during vehicle assembly to conduct the correction of injection quantity.
– Learning control of micro-injection quantity: It is important to enhance accuracy of pre-injections and after-injections. To fulfil this, injectors undergo learning controls. During stable idling, the dispersion of injection is measured for each injector of all four cylinders, and then used for correcting the injection amount. Since this learning control is an essential factor for the product performance of the vehicle, it is conducted at the time of shipment from the plant, and also on regular basis when the customer is driving the vehicle. Based on the precise setting and calibration of fuel injection pattern, the tailing process, the gradually changing of the injection quantity at each injection event at the time of fuel injection pattern change, the correction control of cylinder-to-cylinder dispersions and the learning control for micro injection quantity as described above was adopted.

5 Summary

Subaru has launched the series production of the first boxer diesel engine for passenger vehicles and has begun to provide it to European customers in 2008 on the models Legacy and Outback. In addition to the high engine performance and responsiveness, the low noise and vibration and the fuel economy have been balanced at a high level. Although Subaru was a latecomer in diesel engines, the developers focused on the proprietary technology, which led to the creation of an unprecedented horizontally-opposed diesel engine that possesses sufficient competitiveness and product performance for success in the international markets.

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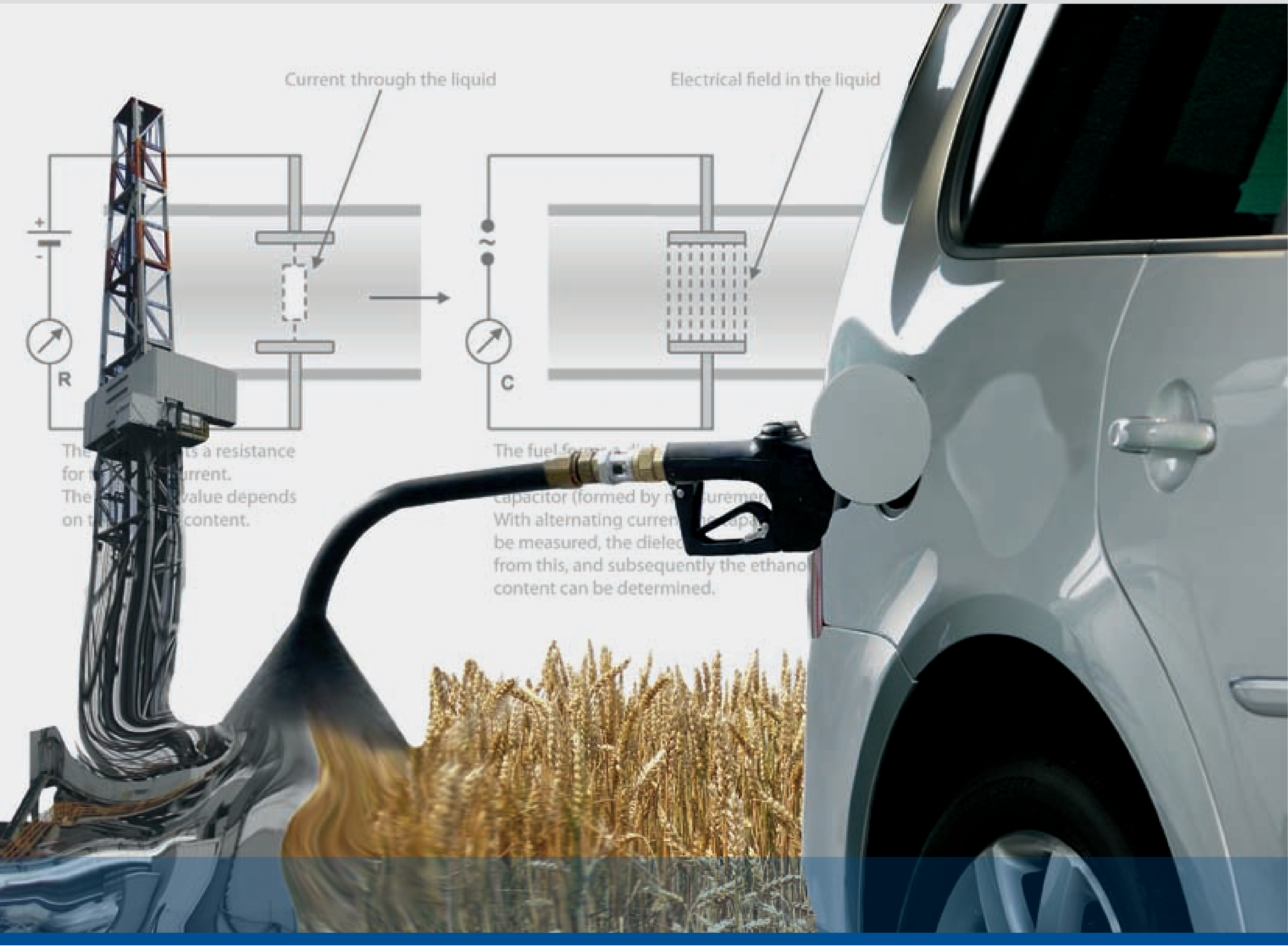

Ethanol Sensors for Flex Fuel Operation

Motor fuels are increasingly being mixed with ethanol, i. e. plant production alcohol, to reduce CO_2 emissions. "Flex Fuel" vehicles are particularly good at running on high ethanol levels (up to 85 % vol. – also known as E85). The engine's control system must, however, be adjusted to the mixture poured into the tank; the precise determination of the ethanol amount is a critical issue here. At Altran the combination of a virtual and an electrical sensor is currently considered to be the best way forward in this respect. The different measurement methods are examined in more detail in the following.

1 Introduction

The production of ethanol is currently the subject of strong (and justified) criticism because of the competition situation with regard to foodstuff production. One possible solution would be to produce ethanol from cellulose, whereby inedible plant parts are used. Numerous technical hurdles are, however, holding back any foreseeable industrial implementation in this respect, which is why it is expected to be some time before the admixture of ethanol in various levels of concentration will be replaced. Flex Fuel vehicles offer a viable solution here for both car manufacturers and consumers. The vehicle automatically determines the ethanol content in the fuel and adjusts the engine control system accordingly.

The "chemical analysis" of the tank content can be performed in two different ways: with a separate ethanol sensor, or with model-based ethanol determination, i. e. analysis of the existing sensor signals.

2 Ethanol Content Measurement

2.1 Flex Fuel Sensor

The separate ethanol sensor is the most expensive, but the most precise measurement method, and therefore a very reasonable solution. In the fuel line two electrodes are immersed in the fuel flow immediately before the injection, **Figure 1**. The fuel flow becomes an electrical element as part of an electric circuit. The liquid's resistance and dielectricity change depending on the amount of ethanol in the fuel. These values are measured, and the resulting calculated ethanol content is available to the engine control unit, which now adjusts the variable dimensions, such as injection period and ignition time.

The residual volume of used fuel between sensor and injection nozzles necessitates a time delay. The "new" measured value only applies when the residual volume is used up in line with the calculated injection amount. In practice, however, this measurement is not as easy as it might sound. The electrical values very much depend on the fuel temperature; the sensor therefore requires a separate temperature recorder for the fuel temperature. The measurement can also be impaired by low quality fuel (e. g. water content, dirt, unintended admixtures).

2.2 Model-Based Ethanol Detection

2.2.1 Detection with Lambda Sensors

The lambda sensor measures the ratio of existing to required oxygen in the exhaust gas (for stoichiometric combustion, Λ=1) [1]. With a high ethanol content the required amount of air per injected fuel

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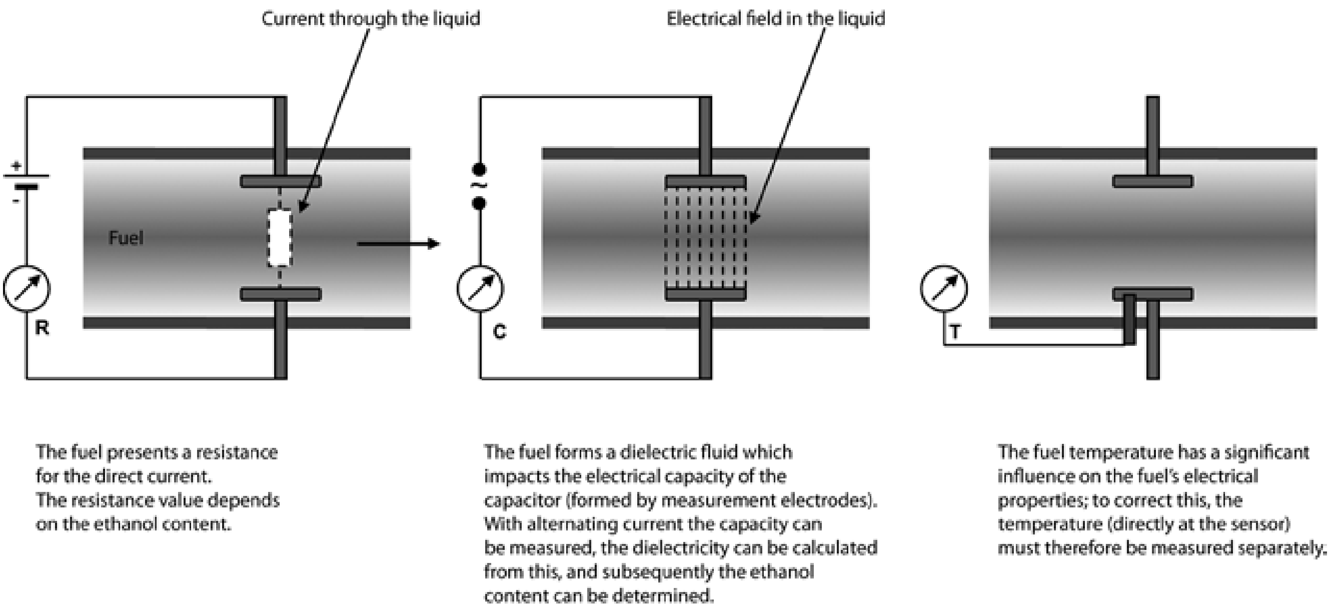

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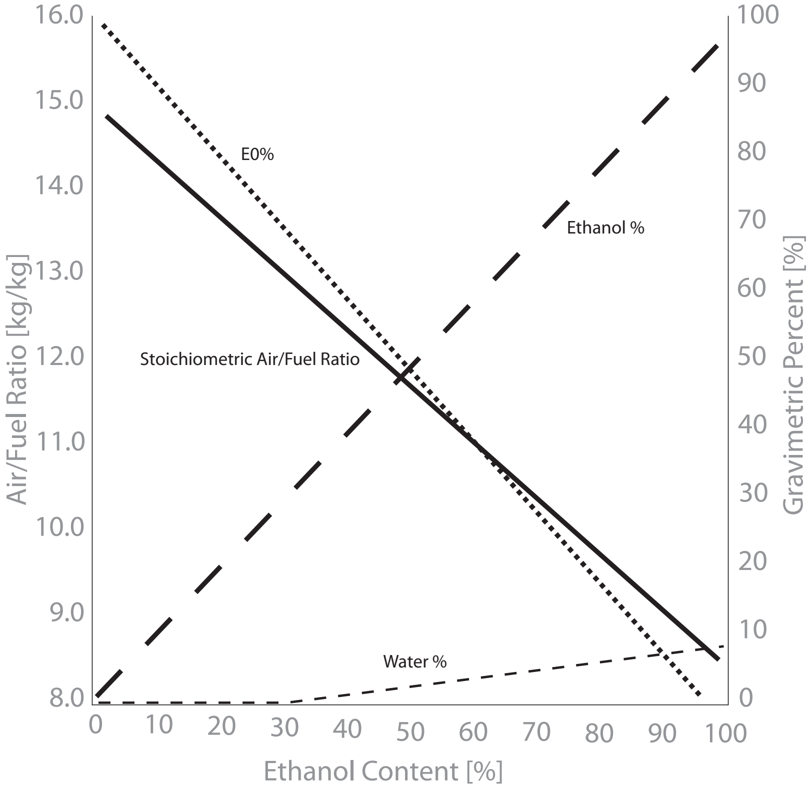

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volume is lower; more residual oxygen remains in the exhaust gas, the lambda value increases In **Figure 2** the course of the stoichiometric ratio is plotted via the ethanol content. The rising water content from E20 onwards is due to the fact that the ethanol used in Brazil (E100) has a 7 % water content. Slight fluctuations in running engines are normal and are intercepted by the lambda regulator; a delta-Λ above the normal level is interpreted as changed ethanol content (and vice versa), and the model calculates the fuel composition accordingly. One disadvantage of this method is that mixture errors (e. g. caused by production tolerances and the aging of the components in the fuel system), can also result in changes in the air-fuel ratio. **Figure 3** shows the effect of a ±5 % flow-through error of an injector on the calculated ethanol content.

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With this procedure several cylinders are operated with different mixtures (rich/lean). The engine's temporal torque values then become measurably irregular – even if the driver can scarcely notice it. With the deviating oxygen requirements with ethanol/petrol, the ethanol content can be determined from the resulting "engine roughness". This procedure is, however, not used very much, as it is too inaccurate in practice.

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Modern engine control systems also include the electronics for the fuel-level indicator. These electronics detect a refueling process because of the sudden increase in the tank's content. As a change in the ethanol content can only occur when filling the tank, any other kinds of fluctuations must be caused by faults. Furthermore the tanked up fuel amount (of still unknown ethanol content) limits the maximum ethanol increase. If, for example, a tank has 20 liters of E50 in it, and 20 liters are pumped in (which can range from E5 to E85), a new mixture ratio of E27.5 to E67.5 is possible. A mixture ratio above E68 or below E27 is then implausible and must be considered a fault, as must a sudden mixture change without tanking.

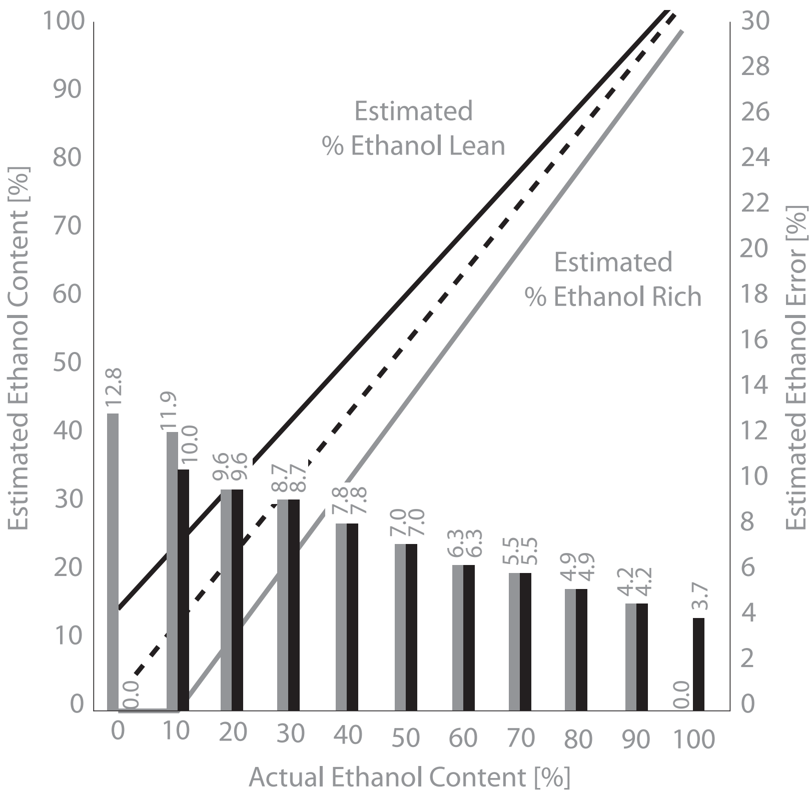

Figure 3: Effect of a ±5% flow-through error of the injectors on determining the ethanol content [2]

3 Conclusion and Outlook

In conclusion it can be said that there are several ways to determine the ethanol content in a fuel. As the results are emission-relevant, legislators are already demanding at least a plausibility monitoring with a second measured value (buzzword: "On Board Diagnosis (OBD)"). Of course false values can also damage the engine. It must also be pointed out that the impairments referred to here, caused by low quality fuel or mixture errors, require validation. Typically the separate ethanol sensor is chosen as the most precise instrument, and its value is aligned with the lambda model.

If faults (i. e. unacceptable deviations) occur, various strategies are pursued – within the framework of legally applicable specifications: during a certain time one of the values can be dispensed with. When three values are at hand (e. g. sensor plus lambda plus fuel level), one of them can be "overruled". If a credible ethanol value is not provided, a safety strategy must be pursued.

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With the model-based detection there can be no doubt that full potential has not yet been exhausted – quantitative improvements (at the least), can still be expected here. Flex Fuel sensors are also continually further developed; and they doubtlessly promise the highest potential: the additional costs for the manufacturer continue to be as tangible as they were before. Nor should we forget that the higher unit numbers will probably bring prices down, with robustness (e. g. against impurities) also increasing with new developments.

In view of the highly advanced development and increasing suitability for daily use, it can also be expected that here in Germany, the percentage of approved Flex Fuel vehicles will also increase significantly in the foreseeable future. Naturally enough, this will also depend very much on the future economical and legal framework conditions surrounding ethanol.

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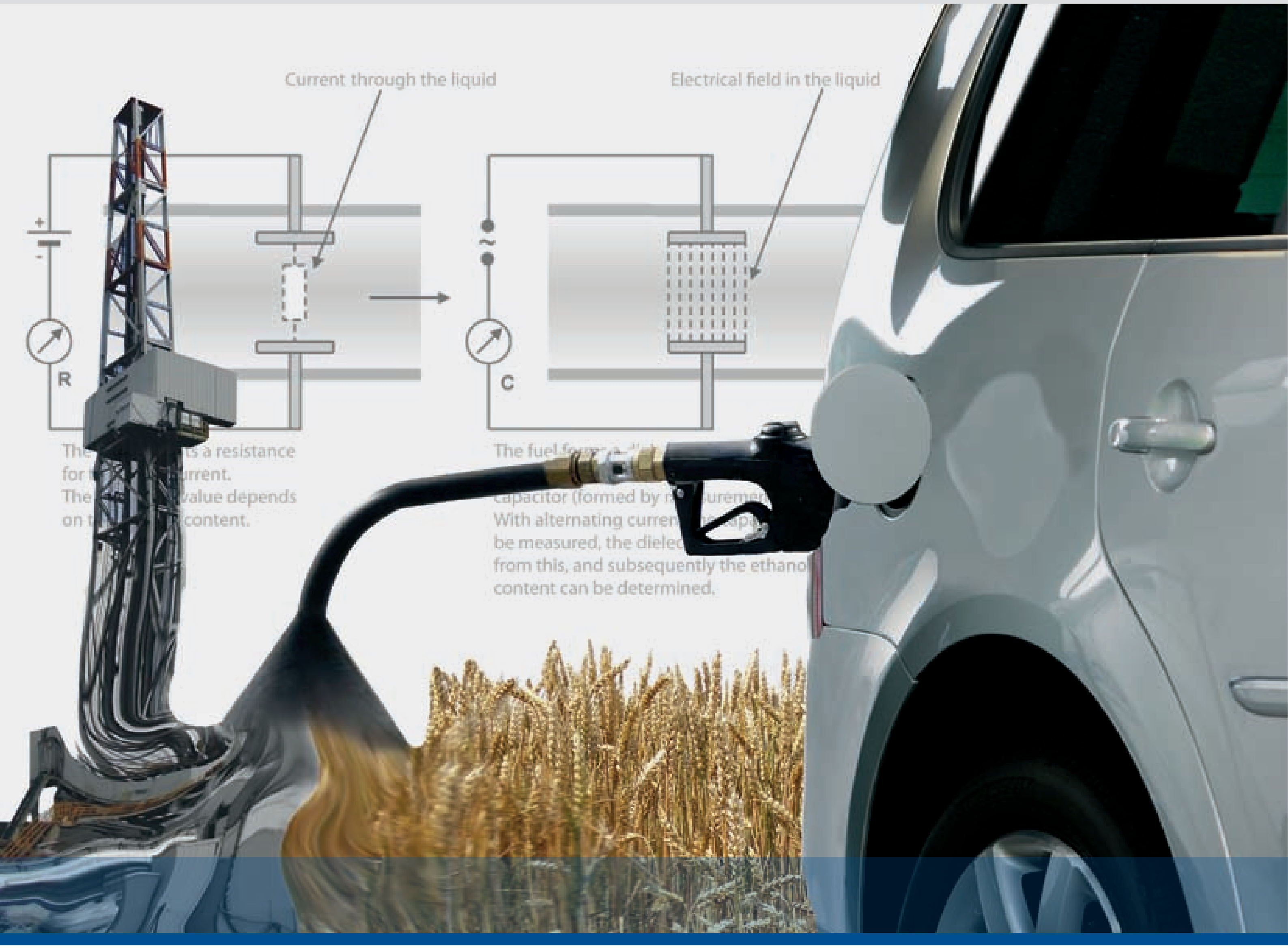

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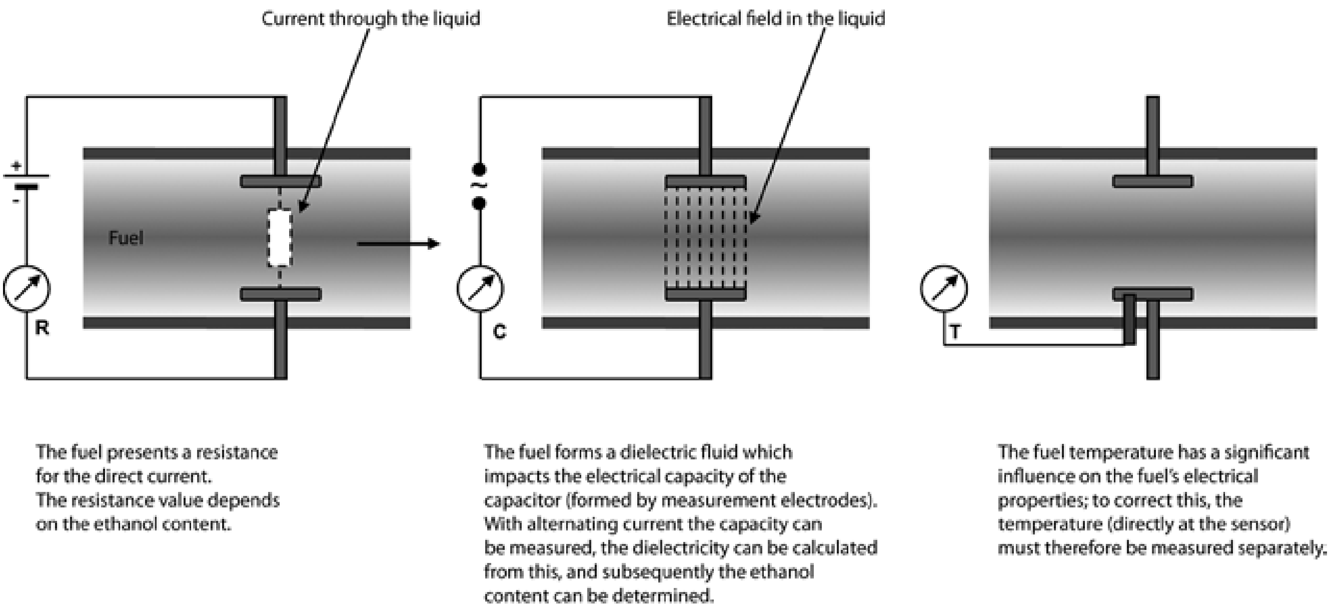

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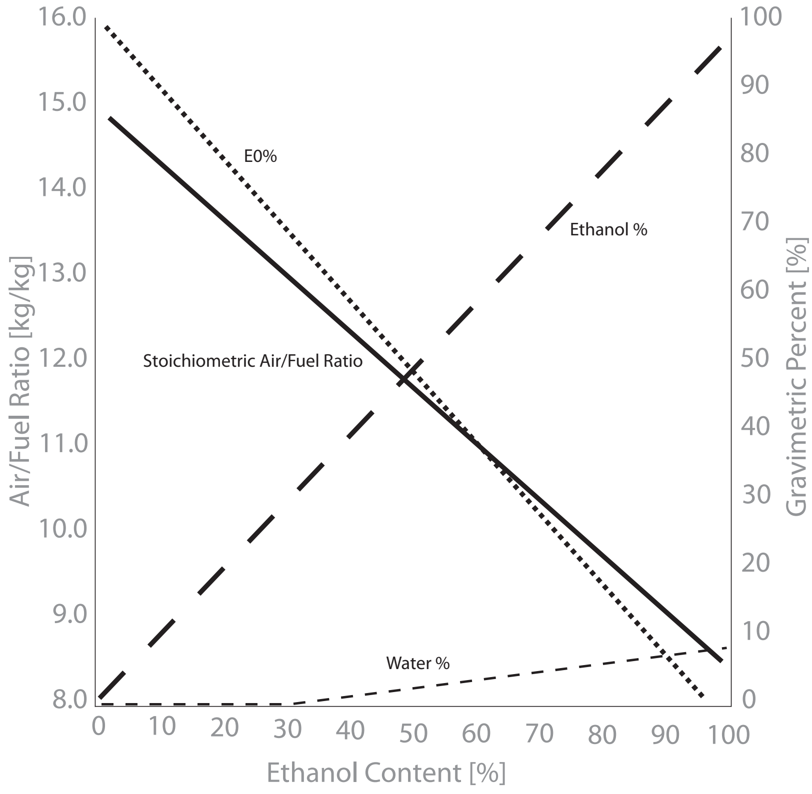

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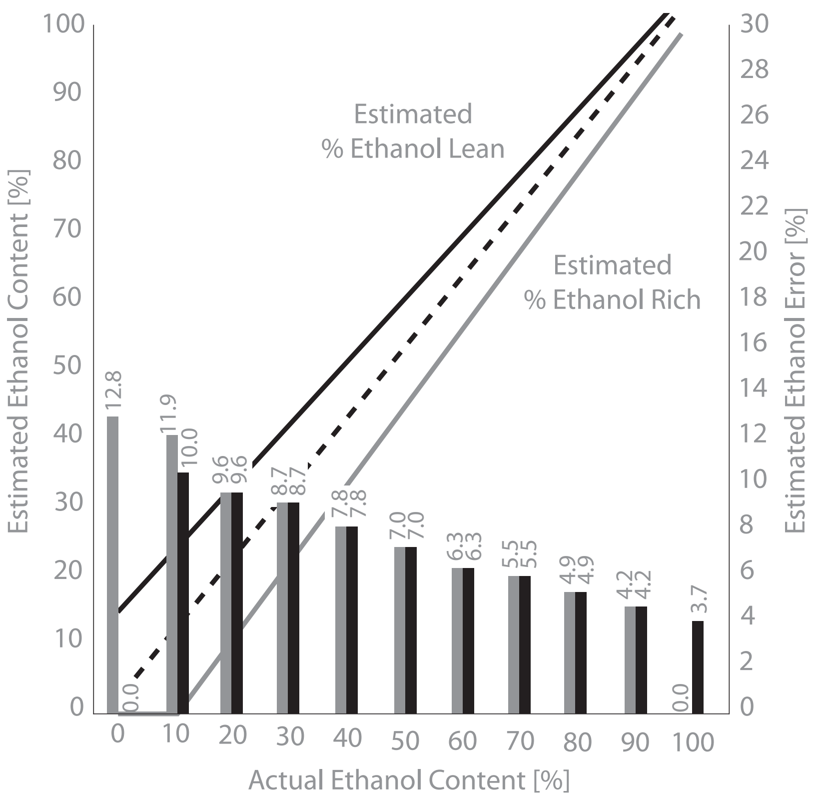

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Fuel Filters with Integrated Diesel Water Separation and Discharge for Modern Diesel Engines

Due to continuous development and increased introduction of Common Rail Systems in truck and car applications, the effective protection of the injection system and associated components against various kinds of wear has become more and more important [1]. As well as providing more effective protection against wear, the supply of water-free fuel to the injection system and automatic water management inside the fuel filter are required. Mann+Hummel is in a position to offer an "Enabling Technology" for modern diesel injection systems by diesel fuel filters with integrated diesel water separation anddischarge.

1 Water in Diesel Fuel

The removal of undissolved water from diesel fuel has become an essential prerequisite for the use of modern Common Rail Systems. DIN EN 590 specifies a maximum water content of 200 ppm for diesel fuel. This level is normally not exceeded in Western Europe, but in regions with inadequate fuel qualities and wide variations in temperature, there will be more water in the diesel fuel. In countries with high humidity, water in the saturated air enters the tank by the ventilation system during the day and condenses during the night at lower temperatures. Eventually, condensed water enters the fuel system highly dispersed by the supply pump. As well as to system failure due to inadequate lubrication, abrasion and corrosion in the fuel system can finally lead to complete system break down [2].

2 Removal of Undissolved Water by Current Systems

Many of the present fuel filters systems for diesel water separation use the hydrophobic properties of the filter medium or of its impregnation to repel free water droplets. Free water droplets in the diesel fuel are blocked by a water repellent surface of the filter medium.

There they coalesce with other droplets and sink due to the density difference between water and diesel. At the bottom of the filter the separated water is collected in a water collection bowl [3].

Due to the rapid development and subsequent refinement of injection systems for diesel engines over the last decade, the demands on fuel quality and thus also on fuel filtration have changed drastically [2]. The continuous increase of the flow rates, for example, leads to higher shear rates in the supply system. Thus water droplets enter the fuel filter further dispersed and with high velocity.

3 Fuel Filters with Biodiesel

Further challenges for the diesel fuel filtration have arisen through increasing demand for first and second generation diesel biofuels [5]. The climate change debate as well as increasing costs for fossil fuels has led to more widespread use of new biofuels. Vegetable methyl esters are commonly referred to as biodiesel. Also known as FAME (fatty acid methyl esters), FAMEs are available in different qualities, from recycled waste fat to pure RME (rapeseed methyl ester). Biodiesel fuel differs from common diesel fuel in its chemical constitution, the higher content of organic particles and particularly in its higher affinity to water [4]. Thus the effective protection of modern diesel injection systems becomes more and more important when biodiesel fuel is used. Due to the reduced surface tension between biodiesel and water, compared with conventional diesel fuel, current diesel fuel filter systems are expected to have reduced water separation efficiency [6].

4 Multi-Stage Concepts for Diesel Water Separation Under Severe Conditions

For applications with increased demands on the fuel filter system Mann+Hummel offers a four-stage concept for best possible protection against wear and free water [7]. In a first stage a coalescer element concentrates finely dispersed water droplets in the inflowing diesel fuel. The droplets coalesce and grow in size and are separated in a second stage by a hydrophobic filter media layer. Depending

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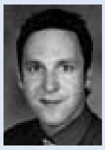

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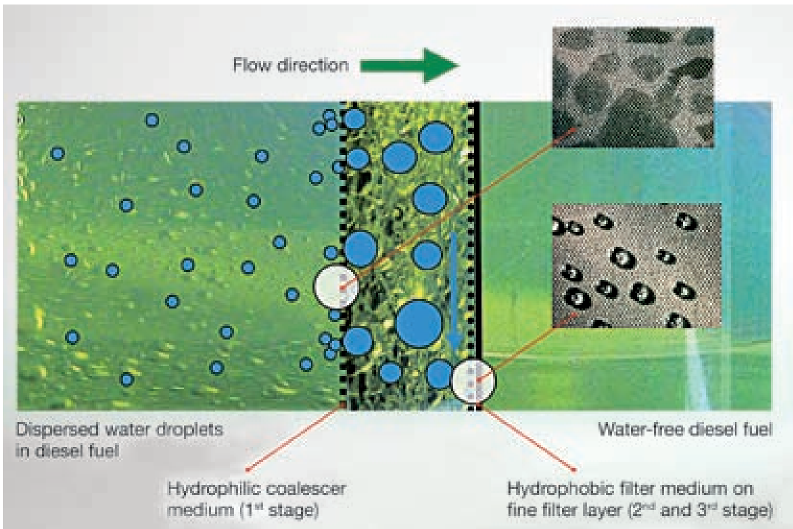

Figure 1: Functional principle of a multi-stage concept for diesel water separation at high flow rates and low droplet sizes

on the application, the hydrophobic second stage is combined with a fine or pre filter media forming the third stage for effective water and particle separation, **Figure 1**. Finally, the separated diesel water is drained and purified automatically in the fourth stage, the Mann+Hummel HydroCleaner.

In a first stage the finely dispersed water droplets reach the influent face of a hydrophilic coalescer medium where they are concentrated. The droplets formed on the influent face are than carried inside the fibrous bed where their transport is limited by the inner structure and they attach to fibres. Following droplets then collide and agglomerate with these fixed droplets which grow in size and are detached by the flowing fuel stream. On the effluent face of the medium the enlarged droplets reach the surface where they remain attached and grow until they are again detached. This is triggered by hydrodynamic forces of the fuel flow or due to gravitational forces. Either these enlarged droplets will sink directly in a water collection bowl of the filter unit or they will reach the second stage formed by the hydrophobic filter medium. Here the droplets are blocked by the hydrophobic surface and sink to the bottom of the filter [8].

5 Discharge of Separated Diesel Water

Current diesel fuel filter or filter systems require regular monitoring of the level of separated diesel water in the collecting bowl. Depending on the design the diesel water level has to be checked visually by the driver or can be detected by conductivity sensors. The collected diesel water is removed by manually activated drain screws or by suction devices [2].

In practice, the system is drained until diesel is visibly flowing out of the drainage. Hence drained diesel water is contaminated by high amounts of free diesel droplets and dissolved hydrocarbons and has to be treated as pollutant substance. Direct discharge into the environment is prohibited. Thus, all existing diesel fuel filter systems depend on the care and knowledge of its operators. By the development of the Mann+Hummel HydroCleaner a fully automatic system is available which for the first time integrates drainage, purification and environmentally friendly disposal of separated diesel water, **Figure 2**. In the fourth phase, dissolved hydrocarbons in the diesel water are blocked by ultra filtration membrane technology.

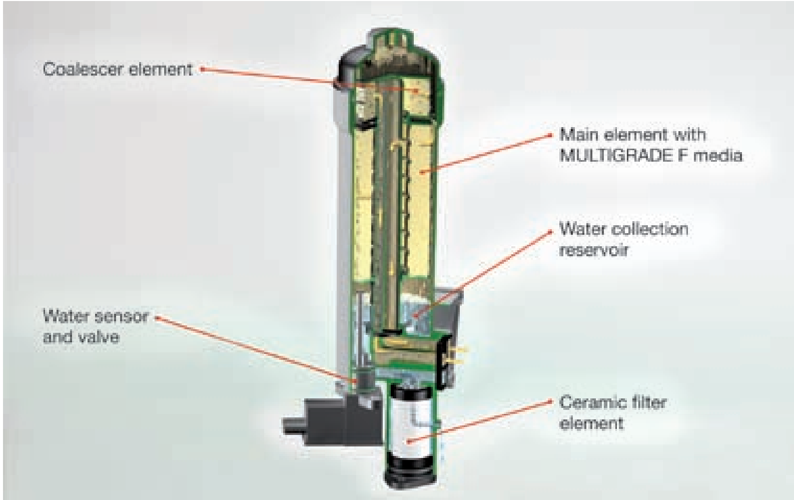

Figure 2: Fuel filter module with integrated four-stage-concept: Coalescer, hydrophobic layer and fine filter element, HydroCleaner

6 Filtration of Diesel Water

During the basic development, particular value was set on the special driving conditions during operation of the vehicle and the corresponding behaviour of the membrane in terms of permeability (flow) and selectivity (cut-off) of the membrane. For example particles, alternating pressure, peak temperatures above 100 °C and high amounts of water (1 % and above) must be expected within the lifetime operation of the membrane element.

The precipitated diesel water contents a large proportion of dissolved and strongly emulsified hydrocarbons. This means that an additional task for the system is the decomposition of the diesel water emulsion as well as the reliable rejection of hydrocarbons according to environmental laws. The micro-emulsion with very fine fuel droplets is effected by the fuel pump and its high shearing forces. In-house investigations have shown that especially these emulsions are quite critical for any adsorption technology. In this case, there is a risk of rapid breakthrough of hydrocarbons.

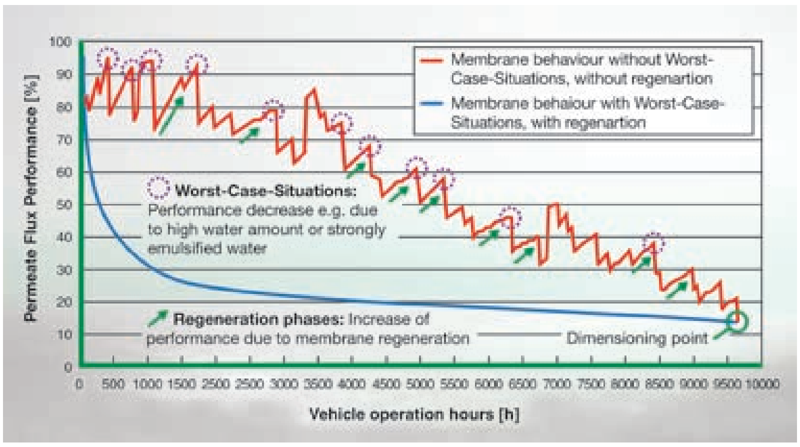

Figure 3: Membrane behaviour under worst case conditions and regeneration

Within the operation period a hydrocarbon layer is formed on top of the separation layer due to fouling and concentration polarisation effects. This hydrocarbon layer constantly decreases flux performance. Due to the positioning of the membrane in the vehicle, typical cleaning procedures like (back) flushing, cross-flow operation or chemical treatment are neither feasible nor cost effective.

On the other hand, rejected hydrocarbons are able to detach again of the membrane channels by difussion effects and car vibrations within non-operation periods of the membrane. The hydrocarbons can then be flushed back to the fuel tank through a special vent system (hydrocarbon recirculation). As a result, no unwanted decrease of permeability and selectivity has to be feared within lifetime operation and hydrocarbon limits can strictly adhered. Actual hydrocarbon limits, e.g. Anhang 45 of the German "Abwasserverordnung" can be reached.

The blue line in **Figure 3** shows the theoretical long-term flux behaviour without special worst case situations and car specific influences. The red line marks a schematic long term behaviour with worst case situations (e.g. high emulsified water and/or high amount of water > 1 %).

7 Design and Functionality

The design of a ceramic membrane is typically of asymmetric structure, **Figure 4**. This means that on top of a porous support with high mechanical and chemical resistance two or more ceramic interface layers are applied before the final separation layer is coated inside the membrane channels. Depending on the desired application pore size of < 1 nm (nanofiltration), 1-50 nm (ultrafiltration) or 50-1,000 nm (microfiltration) is used.

8 Membrane Material and Filtration Mechanism

The choice of a suitable material is oriented to lifetime operation with strongly chemical, thermal and mechanical treatment of the membrane element. After intensive material screening with different polymeric materials it could be seen that these conditions are quite critical, especially regarding to temperature peaks and frost. No suitable material could be specified. On the other hand, ceramic membranes are very stable and successfully used also under critical long term conditions like solvent filtration and temperatures above 200 °C. Porous oxide ceramics with alumina oxide as support structure and interface layers, and titanium- or zirconium dioxide as the separation layer is successfully used since many years.

The components are rejected by size exclusion and, with decreasing pore size, also through chemical interactions between surface and feed components. Driving force for the filtration is the pressure and concentration gradient between raw side (feed) and clean side (permeate) of the membrane. The necessary feed pressure is given by the fuel pressure of the system.

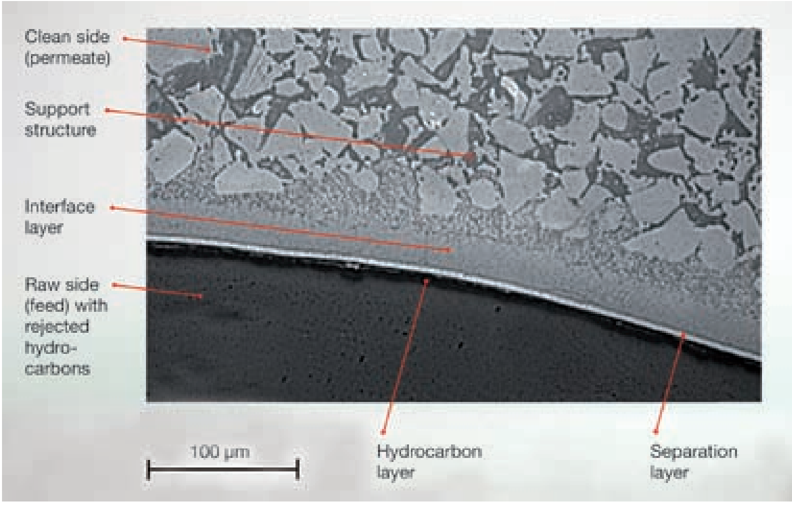

Figure 4: Cross-section of a ceramic ultrafiltration membrane

9 High Flexible Membrane Service

An essential part of the Mann+Hummel water discharge system HydroCleaner is an intelligent 2 level water sensor/valve combination together with special discharge logic. The sensor can detect the operation status of the ceramic membrane element individually for every vehicle and condition (fuel quality, environmental conditions, temperature etc.). A warning signal is given to the driver and/or to the electronic control unit which can be read out at the next service. Additionally other safety features and alarm functions are integrated, for example detection of membrane defects and sealing leakage.

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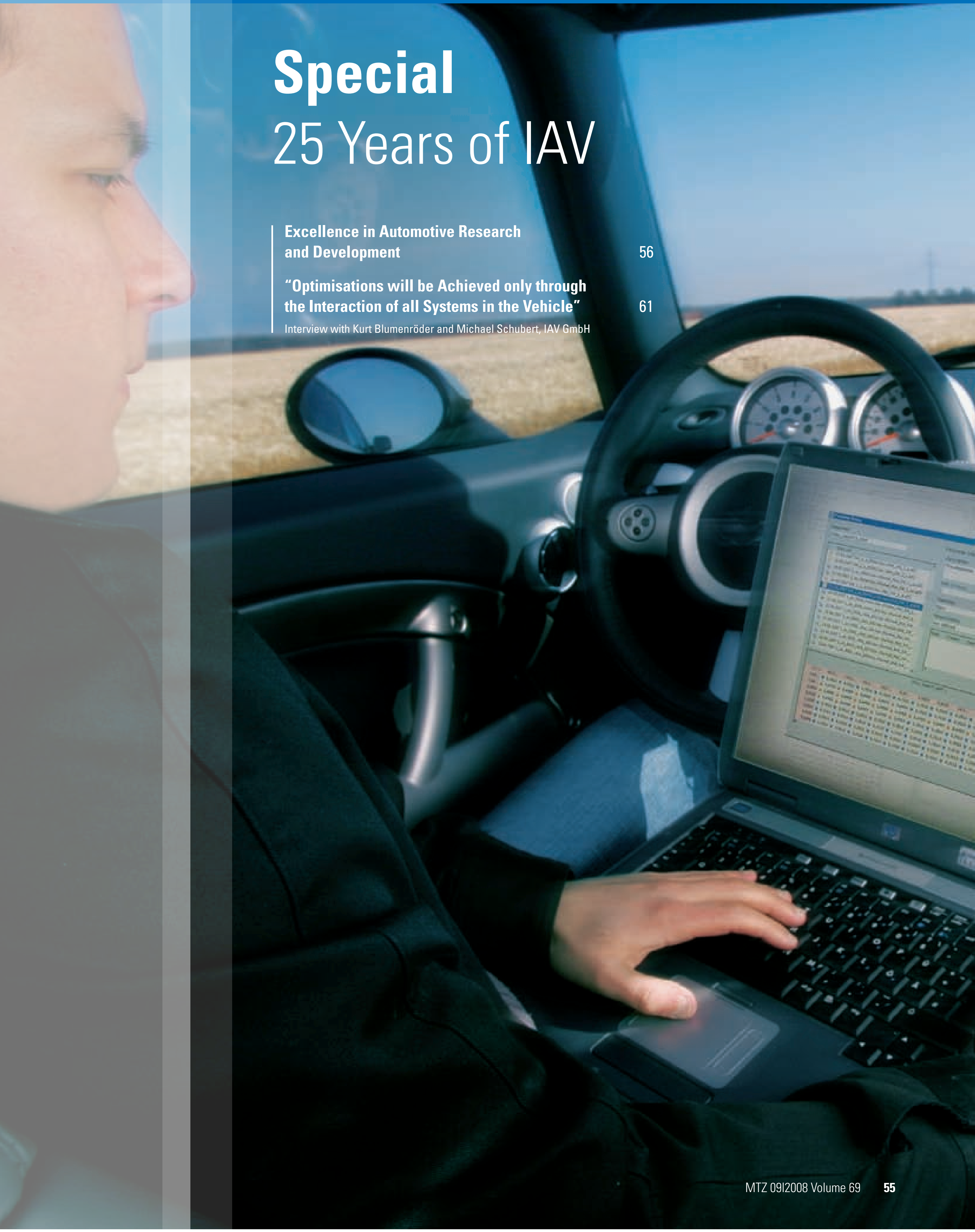

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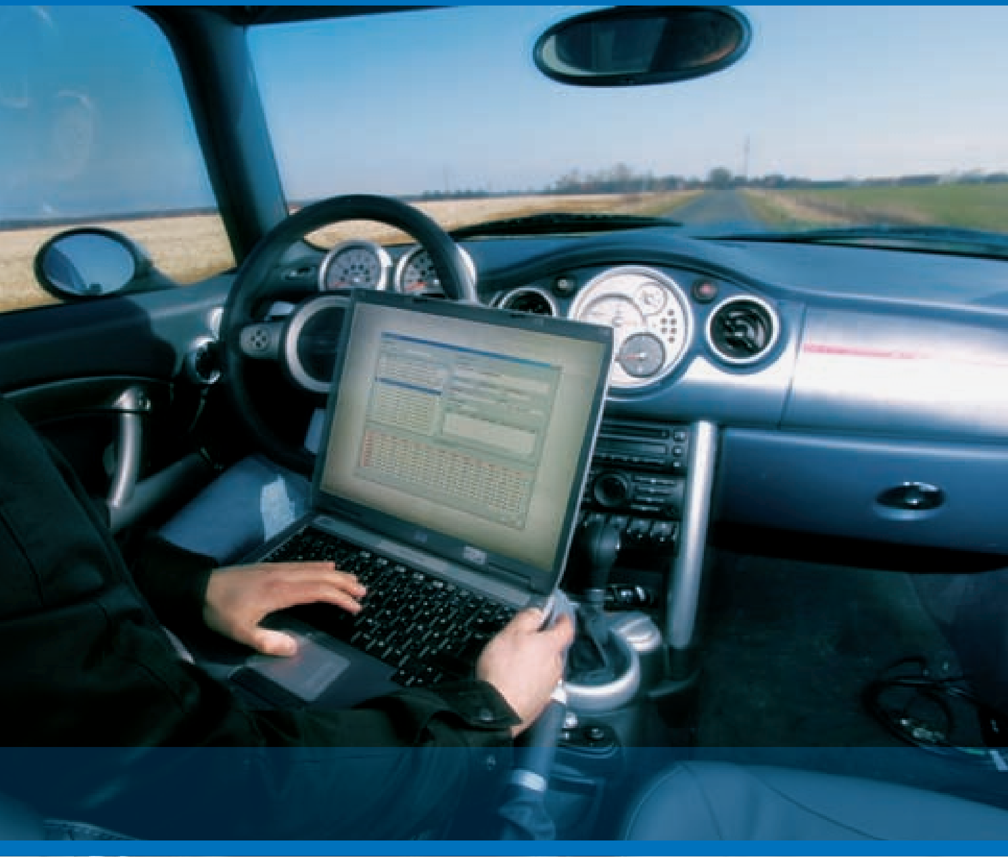

Excellence in Automotive Research and Development

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1 Innovation in Series Production

The tight mesh between cutting-edge research and industrial application was handed to IAV when it was founded in 1983 and still known as "Institute für Aggregatetechnik und Verkehrsfahrzeuge". This was the doing of both main players – Professor Hermann Appel from Berlin University of Technology on the one hand and Europe's largest automobile manufacturer, Volkswagen AG, on the other. The young company attracted attention very early on through first-class engineering achievements – such as in 1986 with the "Eco Polo" whose two-cylinder, direct-injection diesel engine ran on 138.6 miles to the US gallon (1.7 l/100 km). IAV's experts also played a major part in developing the first three-liter car, the Lupo 3 l TDI. The vehicle was launched in 1999.

The company constantly works on improving the way in which driving fun is combined with protecting the environment. For example, by developing ever more fuel-efficient Spark Ignition (SI) and diesel engines. And needless to say, IAV is also right at the fore in the future trends stakes – in the various Homogeneous Charge Compression Ignition (HCCI) concepts and low-emission Compressed Natural Gas (CNG) engines as well as in the big issue of hybrid drives. It is particularly here that the company can apply its broad-ranging expertise in the whole vehicle because in combining electric motor and combustion engine, all disciplines must work hand in hand and look for completely new solutions.

And it is precisely for this that IAV is best qualified because its experts are familiar with all aspects of the automobile:

Today, IAV GmbH is one of the world's leading development partners to the automotive industry. A glance at the company's list of clients, which reads like a who's who of all prominent vehicle manufacturers from across the globe, proves this claim. But what makes IAV so attractive to the well-known names from Europe, Asia and the US? One explanation for its success is doubtlessly the high level of expectations set by IAV's engineers in terms of developing technically outstanding and, at the same time, truly production-ready solutions for all areas of the automobile.

Whether vehicle or powertrain development and mechatronics, electronic or infotainment – the company has practiced contact persons and experienced developers for every field. Cutting-edge testing facilities and realistic testing environments ensure a high level of quality throughout the entire process – from design and testing right through to the construction of prototypes and production planning.

IAV's greatest asset is its proficient staff: Meanwhile, over 3,000 men and women, the majority of them engineers, work for the company – researching worldwide for better cars. Alongside the German development centers in Berlin, Gifhorn and Chemnitz, IAV also has operations in other European countries, in the US, South America and in Asia, **Figure 1**. "The automotive industry is a global busi-

Figure 1: IAV is globally positioned and directly on its clients' doorstep

Figure 2: IAV offers end-to-end safety concepts on a one-stop shop basis: simulations as well as crash test results go directly into the design of tested components

ness – so is IAV", say Presidents Kurt Blumenröder and Michael Schubert.

"Ingenieurgesellschaft Auto und Verkehr" can look back on 25 years of constant growth in which its shareholders – meanwhile on board alongside Volkswagen AG are Continental Automotive GmbH, ArvinMeritor GmbH, Freudenberg & Co. KG as well as SABIC Innovative Plastics B.V. – have always persistently supported the company's development and will continue to do so in the future as well.

2 With a Constant Eye for the Entire Vehicle

Engineering expertise in the entire vehicle – that is IAV's hallmark. The Berlin-based automotive experts are in full command of all disciplines in modern automotive engineering and can provide OEMs with turn-key solutions.

Only an eye for the whole picture guarantees solutions that qualify for use in mass production – that is why, in its 25-year history, IAV has deliberately built up extensive competencies in all fields of vehicle development: Its engineers are experts in powertrain development, powertrain mechatronics, vehicle development and vehicle electronics alike, and the growing number of turn-key projects demonstrate that this "engineering expertise in the entire vehicle" is an attractive line-up for automobile manufacturers and system suppliers.

2.1 Vehicle Development: Link Between Idea and Production Readiness

At the same time, IAV can take care of all stages in the development process on the client's behalf – starting from concept and design, the construction of prototypes and component testing right through to developing entire vehicles and meeting vehicle requirements. To do this, for example, we conduct crash tests and test drives. In the vehicle body field, IAV's experts develop all components from the bumper bar to the complete chassis. Numerous test benches are used for examining the load-carrying capacity of a design and for conducting endurance tests where parts are rigorously put through their paces – whether in assessing body stiffness, the lifespan of the power window system or vehicle behaviour under various climatic conditions.

IAV's Vehicle Development business area covers the full portfolio of activities involved in vehicle development, from design and concept formulation, suspension, vehicle exterior, vehicle interior, simulation, crash testing, competitor analysis right through to overall vehicle testing and production planning, Figure 2.

2.2 Engine Development: Less Carbon Dioxide and Lower Costs

On the engine-development side, IAV's experts are currently being confronted with two mega-trends: Carbon dioxide emissions need to be reduced, yet at the same time costs are also to be brought down. "Unfortunately, these two demands are very hard to reconcile – they often lead to a trade-off situation", says Dr. Gerhard Maas, Head of Powertrain Development at IAV, describing the problem. "Although fuel consumption can be reduced with the aid of measures to lessen friction in the engine, this often costs extra money." The same also goes for other innovations, such as the use of demand-controlled auxiliaries, Figure 3.

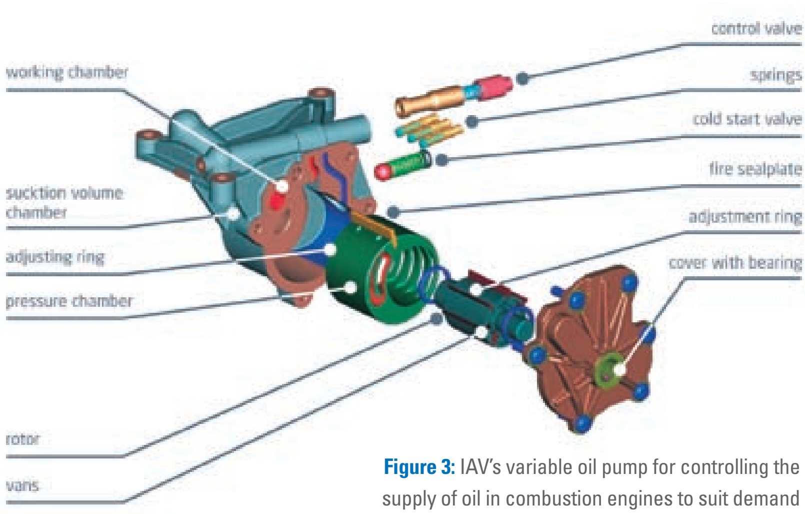

Figure 3: IAV's variable oil pump for controlling the supply of oil in combustion engines to suit demand

To satisfy both demands, the engineers are often stretched to the bounds of what is technically feasible. "Component stress is getting closer to reaching its limit", Maas states. Take downsizing for example: The engines are made to deliver more and more power and torque, leading to a higher power density. Although new materials – such as in crankshaft plain bearings – are capable of handling these demands, they also lead to higher costs.

Without the support of the computer, it would not at all be possible any more to develop the eco-friendly hi-tech engines in modern-day vehicles. "In the meantime, virtual product development plays an important part at IAV", says Maas. "Advanced simulation methods are making it possible to take component load as close to the limit as possible." These days, engineers are able to calculate thermomechanical load – on the cylinder heads for example – and the thermodynamic processes in the combustion chamber with extreme accuracy. The experts are aiming to improve these simulations further and use them to an even greater extent in product development. This will provide the means to speed up developments without compromising on quality.

This being so, it will also be possible in future to optimise existing engine concepts even further – in the diesel engine through higher injection pressures of over 2000 bar, in the SI engine through higher injection pressures and second-generation spray-guided direct injection capable of generating precise stratified charges in the cylinder. Although new concepts, such as HCCI processes also promise better fuel economy while reducing pollutant emissions, they also lead to higher costs on account of the complex technology involved.

Although consideration has always been given to the cost of powertrain components and systems at the development stage, the application of target costing approaches will put a stronger focus on them in future. Here, development and production will move even closer together.

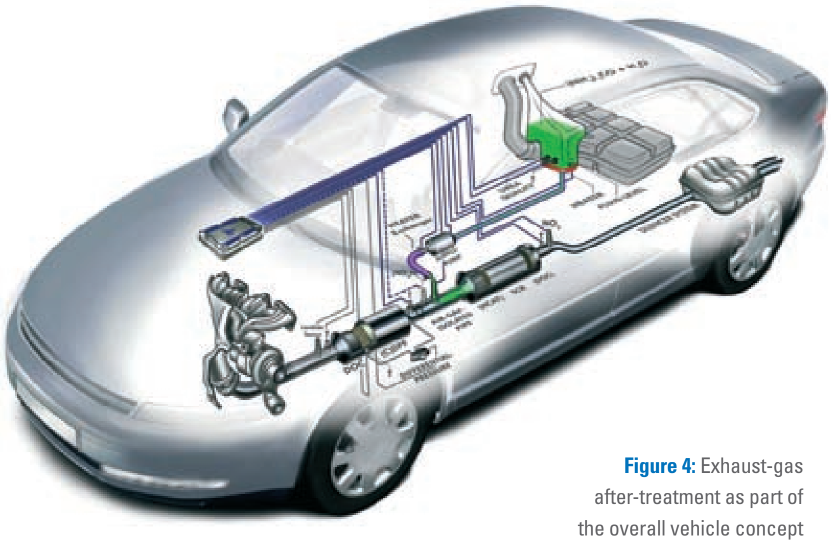

Figure 4: Exhaust-gas after-treatment as part of the overall vehicle concept

2.3 Exhaust Aftertreatment: New Engines Are Demanding New Answers

The new approaches, however, are not without side-effects – they are not making the business of exhaust aftertreatment any easier. Resolving the trade-off in reducing emissions while bringing down consumption and cutting costs, therefore, demands a complex approach to the overall system, **Figure 4**. This applies both to the diesel as well as to the gasoline engine, with the technologies for optimising both combustion processes being seen to converge all the time. For instance, diesel-typical "lean-burn operation", that is combustion with excess air in the cylinder, is also being taken forward on to direct injection gasoline engines: Although this reduces consumption and, with this, CO_2 emissions, it also results in the three-way catalytic converter no longer being able to reduce nitrogen oxide from the exhaust gas. Consequently, approaches from diesel-engine development are being applied in exhaust aftertreatment here too. Given the ever-tighter emission standards as well as the large number of vehicle and engine models, developers are faced with the task of balancing the competing options for reducing nitrogen oxide emissions one against the other.

It is in this field that IAV's engineers are working on two alternative technologies in particular: enhanced NO_x adsorption catalysts and SCR catalysts (Selective Catalytic Reduction), both of which come with quite specific benefits and drawbacks. "NO_x adsorption catalysts produce a small increase in fuel consumption and

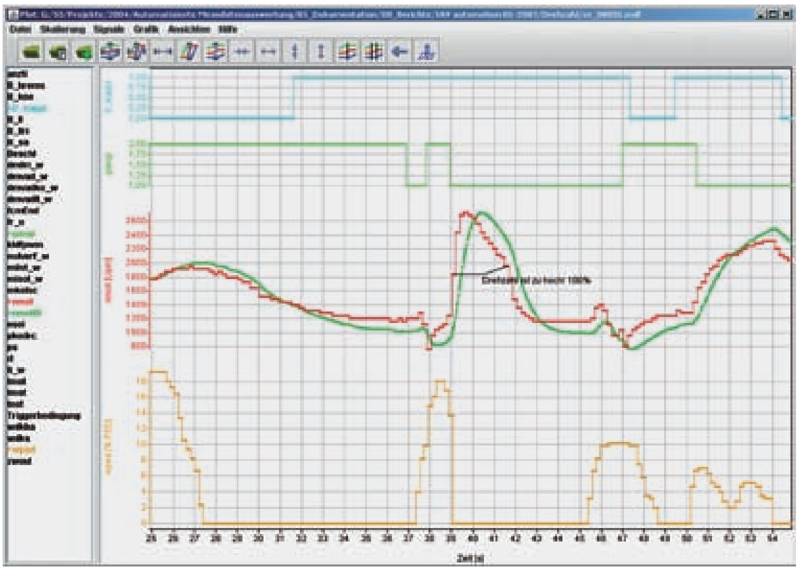

Figure 5: Diagram of measuring signals

are slightly more complicated to calibrate because every now and again the engine needs to run on a rich mixture to regenerate them (that is with air deficiency) and the catalyst requires active desulfation", explains Dr. Lutz Krämer, head of the department Diesel Aftertreatment, business area Powerterain Mechatronics at IAV. A further problem lies in the higher costs for the precious metals, particularly rhodium, which has shot up in price over recent years.

In the SCR process, the exhaust gas is mixed with ammonia which reduces the nitrogen oxides to water and nitrogen. Although the system offers a good level of efficiency, it requires a complex metering system and an additional tank for the ammonia – adding weight and costing space in the vehicle. "This is why the SCR catalyst is more suited to large cars whereas the NO_x adsorption catalyst is better for integrating in small models", Krämer says. Today, emission specialists also find out which system is best suited to a vehicle in extensive simulations.

In future it will be possible to improve both systems still further: Today, for example, the ammonia for the SCR catalyst is provided in an aqueous solution. The extra tank could undergo a significant reduction in volume if it were possible for the vehicle owner to refill it on a regular basis or the reducing agent were, for example, to be stored as gas in crystalls. Improved particulate filters are also on the agenda for IAV's engineers: Whereas numerous concepts use the robust but relatively expensive silicon carbide, the ceramic cordierite is expected to see a revival. At one time, the material had to contend with a short lifespan, but thanks to improved control strategies and modern injection systems, the low cost alternative to silicon carbide is facing rosy times again, particularly when the aim is set on integrating catalyst functions and particulate filtration. To assess different technologies better, IAV is currently setting up a new laboratory for exhaust aftertreatment in Berlin. "From 2009 on, this will provide us with one of the most modern testing sites for diesel engines and their exhaust aftertreatment well beyond Germany", Lutz Krämer says.

2.4 AMeDA also Answers Questions Before they Are Asked

The tremendous significance of cutting-edge measurement technology is also demonstrated in IAV's own in-house AMeDA (Automatic Measurement Data evaluation) development project. "On the one hand, the number of interconnected systems in modern cars is growing all the time, on the other, cost is reducing the number of mules for testing the smooth interaction between components", says Michael Papendieck, IAV Vice President Systems in the Infotainment Electronics division. This is why, in vehicle development, it is important to make optimum use of every test mile driven. It is precisely at this point that AMeDA comes in: In the network of systems, it provides the capability of conveniently filling and managing all vehicle communication data recorded during a test drive and also of evaluating them automatically, **Figure 5**. The results from these data, which are evaluated on the basis of various, user-programmed search patterns, are filed in a database and can be retrieved by the customer at any time through a Web browser.

As the recorded data are available for an unlimited period, it is possible to carry out any chosen evaluation, also after the event. "This way, we can use the data from today to answer the questions of tomorrow", Papendieck explains. In addition, the system is a constantly growing fount of knowledge: "When we are on the trail of a specific problem, we get together with the relevant experts and define which pattern we need to use for searching the measurement data. We then use such patterns to create templates. With the help of these templates, we can then search for such patterns in the stored volumes of data automatically – expert knowledge is now available to many developers." In future, the AMeDA system could not only be used in developing vehicles but also in quality assurance, in production and even in service.

2.5 Autonomous Assistance Ensures Greater Safety

Modern measurement technology will also change the experience of driving itself: Even today, numerous systems work as assistants to the driver – some delivering information to support the driver (for example navigation systems, others automatically taking action in the vehicle (for example ABS and ESP). But this is only just the beginning: Ambient sensors – such as radar or imaging systems – observe the traffic on the road, identify critical situations and help to avoid accidents. "The car of the future will recognise and predict traffic situations and, if a collision threatens, for example, apply the brakes in good time – this can save several metres of braking distance", says Rene Zschoppe, head of IAV's Driver Assistance and Comfort Electronics department.

Figure 6: Assistance systems, such as lane-keeping, support the driver and help to minimize the risk of accidents – IAV has been working in this field since 1995

"Optimisations will be Achieved only through the Interaction of all Systems in the Vehicle"

In an interview, Kurt Blumenröder, President and CEO, and Michael Schubert, President and CFO, of IAV GmbH spoke about the opportunities of globalisation for a development partner in the automotive industry and the potentials offered by alternative powertrain concepts.

Dipl.-Ing. Kurt Blumenröder (left) is President and CEO of IAV GmbH in Berlin;
Dipl.-Ing. Michael Schubert (right) is President and CFO of IAV GmbH in Berlin

Question *IAV works for all of the world's prominent automobile manufacturers - what makes you so attractive as a development partner?*

Schubert *IAV has comprehensive expertise in powertrains, electrics and electronics. Added to this is vehicle development. Our attractiveness lies in the depth and breadth of the competencies we offer our clients. We always have our eye focused on the entire vehicle. Our staff members of staff know all the necessary interfaces, possess the understanding necessary for intermeshing the disciplines involved in development work and are in command both of the components as well as the modules, and know about the effect they have in the overall vehicle. The combination of our staff's comprehensive know-how and our expertise in the entire vehicle sets us apart from the competition and makes us a reliable partner to automobile manufacturers.*

Question *Many new concepts in automotive engineering are being discussed at present to make traffic greener. Where do you see the greatest potential, and how is IAV involved in this development?*

Blumenröder *The overarching subject concerning virtually all areas of automotive development is the necessity to reduce CO_2 emissions. Whether downsizing, hybrid, safety, lightweight construction or interconnection of electrical systems – each individual discipline makes its contribution because it is only through the interaction of all systems in the vehicle that optimisations will be achieved. IAV, with its extensive portfolio, is extremely well positioned in this regard and works in all fields on future solutions for its clients.*

Question *Your company is growing at a fast pace – you took on 450 new members of staff in 2007 alone. What makes you so attractive as an employer?*

Schubert *In addition to shallow hierarchies, we are distinguished in particular by our ability to find answers to exciting, innovative and widely varying challenges. As an independent development partner, we work for virtually all automobile manufacturers and component suppliers, that is in an international environment. We have to familiarise ourselves with technical contexts as well as with different corporate structures. This makes our work complex, but also extremely varied. Our high level of acceptance on the market and the wide-ranging skills of our staff are further aspects that make us an attractive employer.*

Question *Attractive new markets are emerging – such as in China and India. Will it become increasingly necessary for developers and manufacturers to set up operations there too? And what are IAV's plans in these regions?*

Blumenröder *This process is underway as we speak. European manufacturers and suppliers have already opened production plants in Asia, for example. IAV has been active for some time now with locally based subsidiaries in China, Japan and also India. We support both the local automobile manufacturers and Tier-1 there as well as our German rsp. European clients. The local manufacturers are trying to serve their domestic market but, in a second step, they also want to export. To do this, they need to catch up on the technical side and this is where we can give them support: We have crash test facilities, experts in passive safety and we know how to bring vehicle emissions into line with the market requirements here. Europe's OEMs, on the other hand, want to adjust their high standards to the local level in the emerging markets and – with our help – technically downgrade vehicles.*

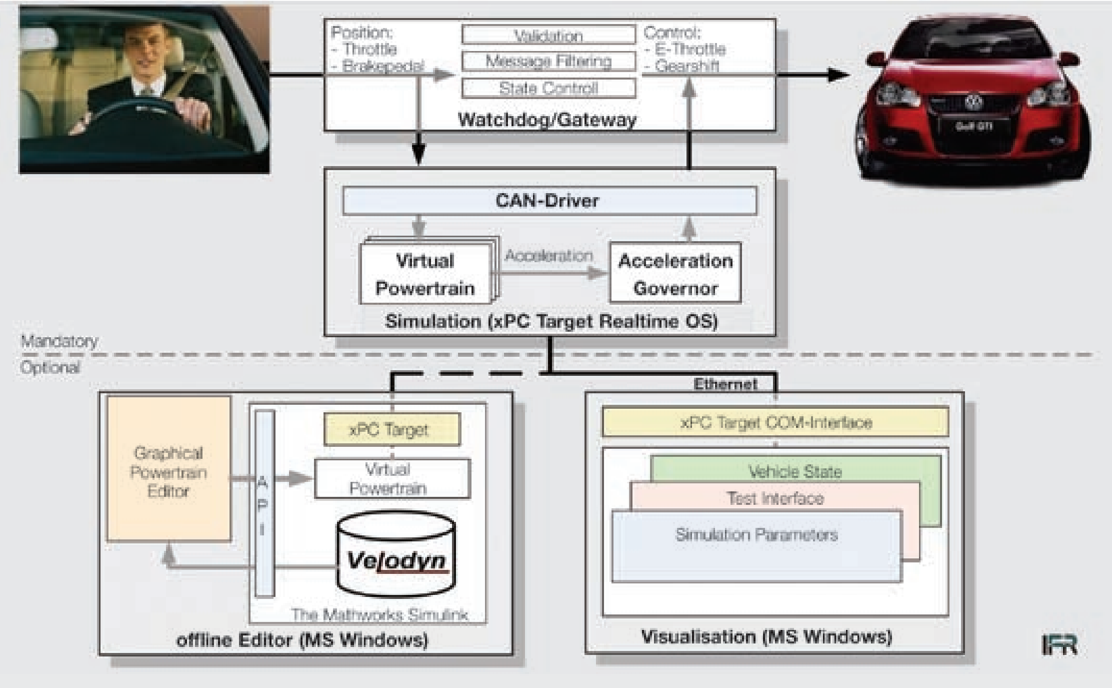

Figure 7: Structure of IAV's InDrive-Simulator simulation system

IAV has been working in this field since 1995 – in research, advance engineering and development for mass production. He likes to compare these „clever vehicles" with an example from history: "It is not very easy to steer a horse-drawn carriage into a tree – because there is an intelligent creature harnessed to the front of it." And in just the same way as a horse is capable of getting its rider home safely and in one piece all by itself, so too could autonomous cars one day drive along our roads without any human intervention whatsoever. "The sensors necessary to do this are, in principle, already available today", Zschoppe adds. "But there is still a great deal of research to be done in this field." Before sufficient progress has been made, we will see further vehicle assistants making decisions for themselves – for instance, work on developing systems for autonomous emergency braking has been going on for some time now, **Figure 6**.

2.6 Hybrid Development Calls for a New Way of Thinking

"Expertise in the whole vehicle" is a particular advantage to IAV clients in the field of hybrid development – among engineers, this is regarded as the "royal league of system integration": It is where all disciplines must be included in optimisation. Only with efficient communication between the various specialist departments involved is it possible to make hybrid projects a success. This is why, in hybrid development, IAV brings together its expertise in project teams made up of experts from all the classic departments.

And they face major challenges: Even as early as selecting the operating strategy, consideration must be given to the planned type of vehicle application and the overall system – comprising environment, traffic, driver, vehicle and powertrain. There are different topologies to choose from, such as the serial, parallel and power-split hybrid which, in turn, also provide many degrees of freedom. And finally, allowance must be made for the interactions between the classic combustion-engine powered drive system and the additional electrical components.

To optimise the system early on in the development process, IAV uses a variety of simulation tools that are coupled together by means of "co-simulation" – for example for the mechanics, hydraulics or thermal balance. And, hand in hand with Braunschweig University of Technology, IAV is already working on a further aid, the "InDrive Simulator": The model of a new drive concept is employed in a real-life vehicle and tested "online" in real-life traffic, **Figure 7**. The idea for this came from research aircraft that allow the characteristic properties of airplanes to be simulated in flight before they are actually constructed.

For Wilfried Nietschke, head of the Powertrain Mechatronics (Gasoline Engines) business area and chief hybrid developer at IAV, the debate currently surrounding lower emissions is having positive effects on the entire automotive industry. "This discussion has triggered something extremely positive in research and development: For the first time, thought is now being given to finding ways of optimising the entire vehicle's energy balance across all technical disciplines. This is why the hybrid is the jump-start for new technologies and an opportunity for lateral thinkers."

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Therefore, since the second quarter of 2008, ATZ and MTZ have the status of refereed publications. The German association "WKM Wissenschaftliche Gesellschaft für Kraftfahrzeug- und Motorentechnik" supports the magazines in the introduction and execution of the peer review process. The WKM has also applied to the German Research Foundation (DFG) for the magazines to be included in the "Impact Factor" (IF) list.

In the ATZ/MTZ Peer Review Process, once the editorial staff has received an article, it is reviewed by two experts from the Advisory Board. If these experts do not reach a unanimous agreement, a member of the Steering Committee acts as an arbitrator. Following the experts' recommended corrections and subsequent editing by the author, the article is accepted. *rei*

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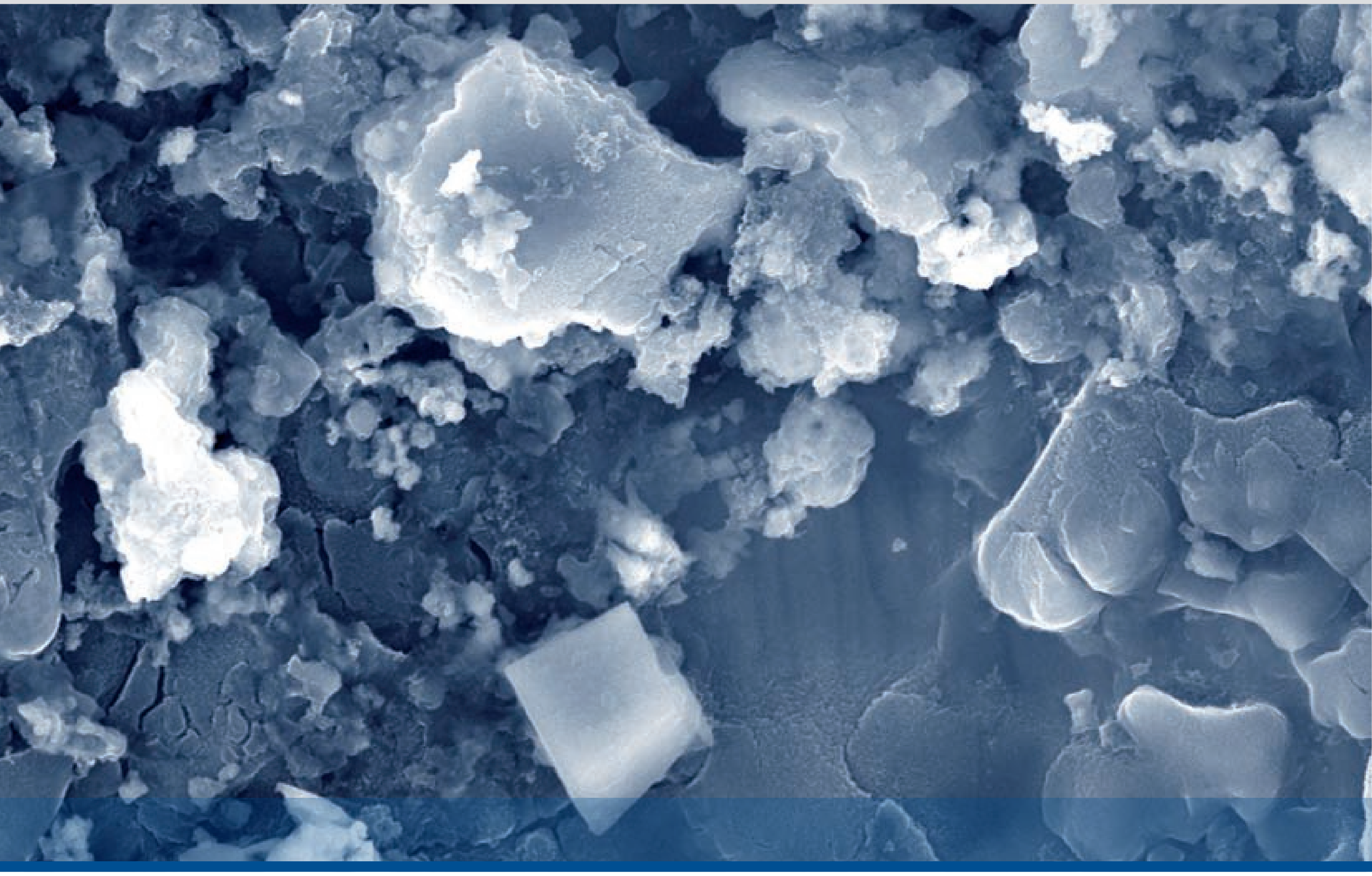

Influences on the Formation of Deposits on Injection Nozzles in Direct-Injection Diesel Engines

Striving towards a robust design of direct-injection diesel combustion processes poses a major challenge for long-term reliability of engine performance. The long range reduction of power is a result predominantly forced by deposits in the nozzle orifice. In the scope of the FVV research project "Coking", No. 14567 N, the Institute of Combustion Engines in Aachen (VKA) of RWTH Aachen University (Germany) determined important parameters accelerating or decelerating the formation of nozzle fouling. The objectives of a 6.3-l truck engine were the realization and detailed characterization of measures to impede or even inhibit nozzle coking.

1 Introduction

In recent years great improvements in the design of new diesel engines have been achieved particularly by the advances in the technology of diesel injection (DI) systems [7]. Downsizing of nozzle orifice diameters by enhanced spray characteristics in combination with increased injection pressures offered the opportunity to carry out engines featuring high power density along with minimized emissions. However, these modern nozzle geometries tend towards a critical formation of deposits [8]. Commonly this nozzle fouling results in a reduction of power output observed at engines during test cell investigations [12] as well as in prototypes, **Figure 1**.

The purpose of the underlying FVV research project was the identification of the parameters dominantly influencing the formation of deposits in injection nozzles of DI diesel engines. Test cell investigations utilizing a diesel engine were conducted in combination with coupled CFD/FEM simulations, which calculated time-based local thermodynamic conditions within the nozzle.

In the following the findings of the test cell investigations are presented. A medium-duty in-line six cylinder engine with 6.3 l displacement is utilized, which has implemented a unit pump (UP) system for fuel injection. The medium-duty truck engine certified with Euro IV displayed a repeatable tendency towards nozzle fouling during former tests. Prerequisite was the operation with various flow-optimized nozzle geometries in a defined coking cycle. The combustion system is calibrated for application of a SCR system. Therefore, exhaust gas recirculation is not necessary leading to relative lean air-fuel mixtures.

2 Definition of the Coking Level

A direct rating of the formation of deposits respectively nozzle fouling or coking was done through the relative alteration of power output and fuel flow at rated power. Only alterations of higher than 2 % were defined as significant taken into account repeatability of the boundary conditions and the duration of the coking cycles. The coking level is explicitly not time specific, since the monitored formation rates indicated a declining behavior and the time length of the coking cycles varied. Thus a quantifiable rating is only allowed for identical coking cycles. On the other hand, as non-significant estimated nozzle fouling (coking level < 2 %) always stabilized prior to the end of the test.

3 Temperature Measurement at the Nozzle Tip

The first task of the test cell investigations was the measurement of temperatures directly at the nozzle tip. The results represented boundary conditions for later simulations. For this a thermo couple was fixed near the nozzle hole outlets. The signal wire extended through the valve cover and the nozzle holder along the nozzle to its tip, **Figure 2**. Thereby the temperature is directly measured in the combustion chamber without affecting the combustion process.

The measurements depicted the conditions at the surface, Figure 2. Temperature follows engine power output approximately linear. Two factors compromised the interpretation of the results: First, the nozzle remained as is to guarantee full durability; therefore, the cover of the thermo couple was welded to the nozzle tip. Through this

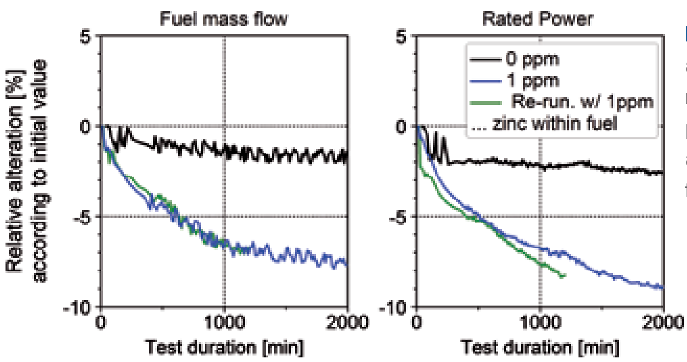

Figure 1: Relative alteration of fuel mass flow and engine power output in % as result of nozzle fouling in ppm

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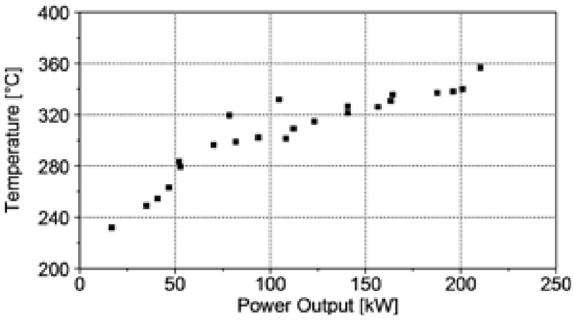


Figure 2: Correlation of temperature at the nozzle tip and engine power output

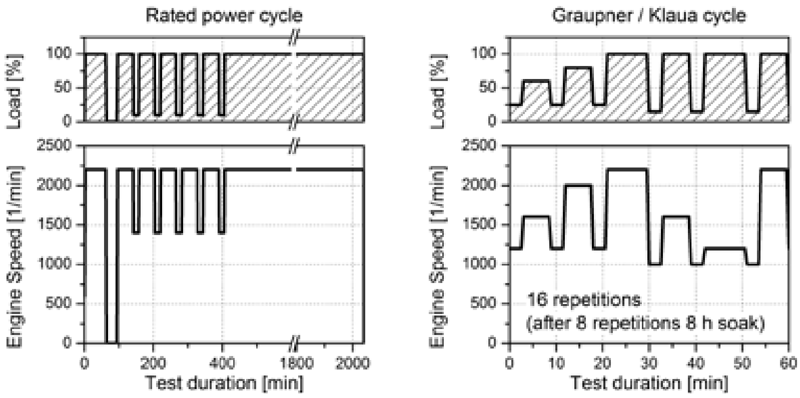


Figure 3: Coking cycle for test cell investigations

the thermo couple was fully exposed to heat radiation (for example intrinsic soot incandescence) during combustion.

Secondly, a strong dependency of the temperature on the position due to the complex flow regime was supposed. The measured temperature level was verified by slightly varying boundary conditions and a sensitivity analysis during the simulations. The fixed thermo couple was entered in an oven chamber before assembling to the engine and checked for functionality.

4 Varied Parameter

The experiments were based upon the already gathered experience and publications regarding the formation of deposits in internal combustion engines. Three fundamental parameters were varied. First, the temperature in the nozzle hole poses an important criterion in the findings of Lepperhoff and Houben [1]. Furthermore, the flow regime has a major influence, as postulated for example by Caprotti et al. [5], which is altered by the geometry of the nozzle orifice. Due to the high effort necessary to perform the test cell investigation, the number of coking cycles respectively test runs was limited. The variation of chemical properties of fuel and lubricant thus focused on bio diesel share and zinc concentration.

5 Execution of the Experiments

Each test run was conducted with new nozzles (and holder combinations) to assure identical start conditions and the possibility to analyze the deposits subsequently. For reference rated power was measured prior and post the coking cycle with constant conditioned boundaries. Fuel flow and power output were monitored to determine the course of the coking level during the test run. Prerequisite for measurement during the conduction of the coking cycle were stabilized steady-state behavior of the engine and rated conditions.

5.1 Influence of Temperature

In the course of the experiments two different coking cycles were applied. Graupner et al. [2] developed an injector deposit test for modern diesel engines for the field of passenger car applications which was altered for the use in the test engine. The cycle operated in a wide load and engine speed range with a relative long soaking duration in the middle of the test time. On the other hand, the second cycle consisted to a large extent of rated power, **Figure 3**. Referring to the temperature measurements it is validated that the rated power cycle results in higher temperatures as the adapted Graupner/Klaua cycle.

Furthermore, another possibility of variation offered the fuel temperature at engine inlet. The standard value of 40 °C was enhanced in compliance with the safety instructions at the test bench by 33 K. Both measures increasing the temperature at the nozzle tip resulted in an accelerated formation of deposits, **Figure 4**.

5.2 Influence of Cavitation

The consequences of different flow regimes were investigated during the test cell experiments by the comparison of the basic highly rounded flow optimized nozzle hole geometry with variants. The chosen boundary conditions (zinc-doped fuel and increased fuel temperature) built up significant deposits applying the base nozzle geometry. To give an example, the test run with the base nozzle is compared to another in which the nozzle was not flow optimized respectively the hole inlet rounded, **Figure 5**. Both nozzles featured cylindrical hole geometries and the same hydraulic flow rate (HFR). The absence of rounding implies a discrete transition of sac hole to nozzle orifice and thus enforces tendency towards cavitation.

The nozzle without rounding resulted in a considerably decreased coking level. Conclusion is that the nozzle coking is reduced with rising tendency towards cavitation. The formation of deposits is diminished or the removal of existing deposits is accelerated. A very similar result is observed by Argueyrolles et al. [3].

5.3 Influence of Increased Bio Diesel Share

The variation of bio diesel share displayed no significant influence on the formation of deposits, left hand side of **Figure 6**.

6 Enhancement of Deposit Formation by Zinc

As published by for instance Caprotti et al. [5], zinc is supposed to enhance the for-

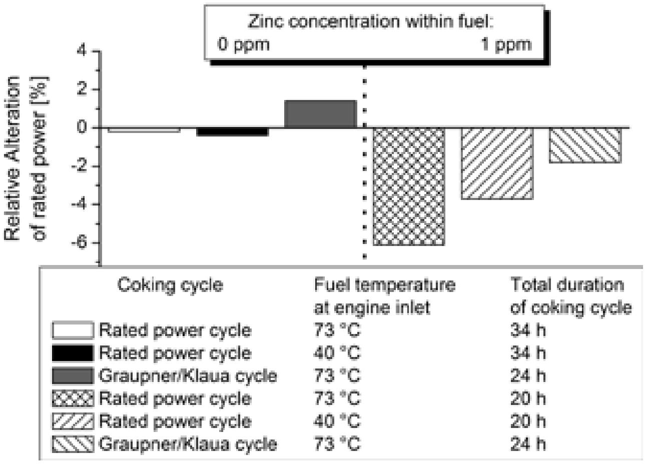


Figure 4: Influence of temperature on formation of deposits

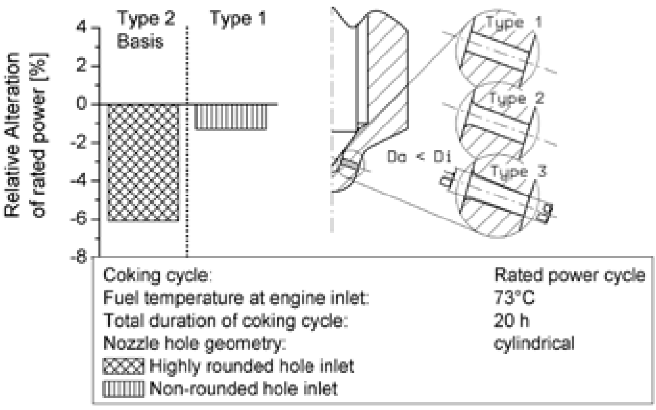


Figure 5: Effect of different nozzle hole geometries on formation of deposits

mation of critical deposits in nozzle orifices. Contemporary diesel fuels contain generally no zinc in any functional component. Therefore, the metal represents a form of contamination within diesel fuel. The doping happens rarely during transport and distribution. More frequently it is dissolved out of zinc-containing components of the low-pressure fuel circuit (for example zinc-coated housing of a fuel filter). Additional sources of zinc compounds pose additives (for example ZDDP) of modern engine lubricants. These additives improve the dry-running behavior of the engine [9] and are part of nearly all available engine oil products. Published measurements rule out the introduction of zinc into internal combustion engines through environmental air [6].

6.1 Mechanisms of Transport

The transport of zinc into the nozzle orifice is done by means of fuel or by means of burned mixture gas. The injection system of the test engine utilizes unit pumps. Their plunger is driven by an oil-lubricated cam shaft. Oil and fuel are separated by rubber sealing rings. Caused by the concept, the transport of oil into fuel cannot be excluded.

Besides, lubricant is transported via the piston rings or eventually via the turbocharger into the combustion chamber. Various zinc compounds are gaseous in the chamber during and after the combustion. These compounds may reach the location of deposit formation, when the flow in the nozzle is reversed at the end of injection and burned gas enters the orifice. The flow reversal is caused by the rapid closing of the needle. The mass inertia of the injected fuel lowers the absolute pressure downstream the needle seat forcing a backflow which includes gas out of the chamber. Thermophoretic forces [4] in the nozzle hole lead to the flow of particles in gas towards a temperature depression. As the nozzle is colder as the introduced gas, the concentration of particles respectively zinc compounds at the wall is increased [1] advancing the formation of deposits.

6.2 Controlled Doping of Zinc

In this research project the quantity of zinc which flows through the nozzle had to be accurately documented to assess the impact of zinc on nozzle coking. The base fuel and the utilized lubricant were zinc-free. Additionally, zinc-containing components in the fuel circuit were exchanged by zinc-free components if possible to minimize undesired zinc sources. The defined doping of zinc was then realized by additives which were added to the base fuel.

6.3 Influence of the Parameter Zinc

During the complete test cell investigations no significant coking level was reached in the absence of a critical zinc concentration. The threshold for critical concentration belongs to the engine configuration. For the basic test engine configuration the limit was within 0.6 and 1 ppm, Figure 6. The formation rate of deposits features a decreasing behavior, Figure 1. Stabilization was not observed for significant coking levels due to the limited duration of the coking cycles.

Two further coking cycles were planed to enhance the knowledge about the zinc-based formation of deposits. The subsequent tests utilized the same nozzles

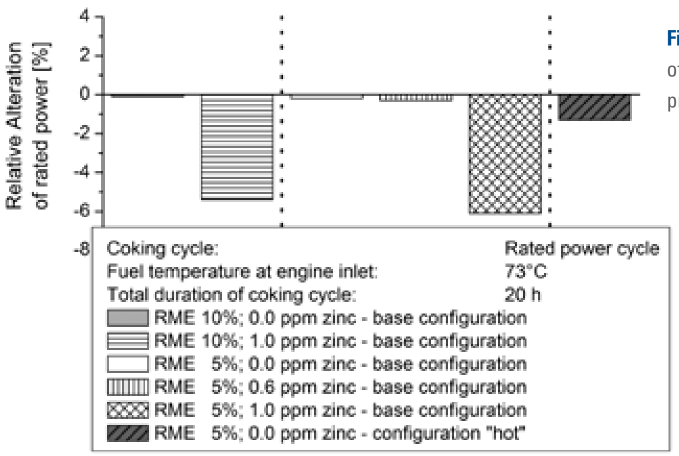


Figure 6: Impact of varied fuel properties

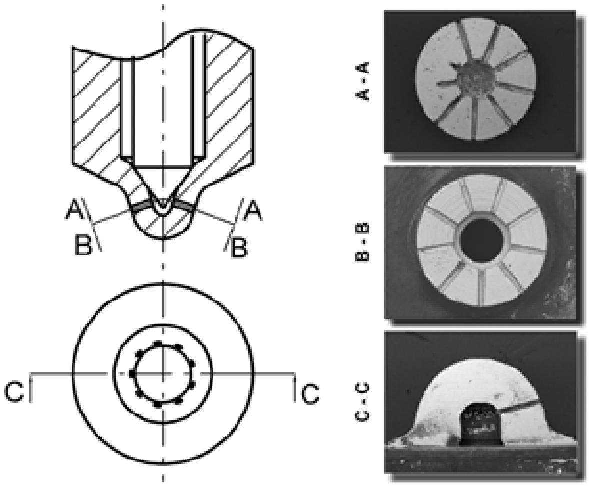
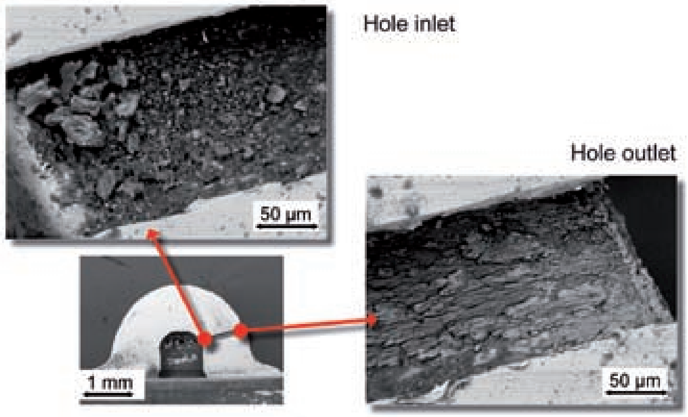


Figure 7: Nozzle preparation for deposit analysis

Figure 8: Deposit morphology dependent on the position in the nozzle hole (cut C-C)



without any modification. The outcome was intended to determine the potential of reversibility. In the first test run the diesel fuel contained 1 ppm zinc, which results in a significant coking cycle according to the previous tests. The adjacent coking cycle was operated with identical boundary conditions except for the fuel which was now zinc-free. The nozzle fouling created in the presence of zinc diminished during the second test run. A subsequent electron microscopic spectral analysis of the deposits displayed zinc at the limit of detection. It was concluded that at least a partial reversibility of zinc-based nozzle fouling occurs during zinc-free operation.

7 Analysis of Deposits by Means of Electron Microscopy

To gain a deeper understanding the deposits were analyzed subsequently to testing. Methods of electron microscopy, as scanning electron microscope (SEM) and energy dispersive X-ray (EDX) [10, 11], offer a good opportunity but require optical access to the examined location. Therefore, an extensive preparation of the inner nozzle area was necessary.

By means of three different procedures the ability to prepare all locations inside the nozzle orifices and the sac hole for microscopic analysis were obtained, **Figure 7**. In the first procedure the lower nozzle tip was cut at the cone defined by a rotation of the center lines of the nozzle holes. Thereby the view onto the upper side of the nozzle holes was achieved (A-A). An evolution of this technique succeeded in the possibility to view the lower side of the orifices inclusively the sac hole (B-B). A milling of the nozzle tip normal to the needle center line completed the optical access of the inner nozzle (C-C).

7.1 Macroscopic Deposit Analysis

In a first step an appraisal was conducted by observing one nozzle of each set utilizing low magnifications. Herewith relevant elements were detected within the deposits, and a first appearance of the coking was acquired.

Acknowledgements

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The nozzle operated with the Graupner/Klaua cycle [2] in combination with low fuel temperature and zinc-free fuel and lubricant displayed a virtually clean orifice. A cleaning in an ultrasonic bath removed approximately the complete nozzle fouling. On the other hand, even nozzles with non-significant coking level presented a visible deposit coating. Downstream the nozzle hole the concentration of elements increased which are definitely lubricant-based. The strength of the coking varied. Besides, the comparison hole-to-hole indicated a high deviation which was excluded to be based on the nozzle preparation. Furthermore, the correlation of the deposit coating and the orientation of the nozzle hole in the combustion chamber were ruled out.

Zinc-based deposits included particles which were detected already at relative small magnifications. The size of these grains abated in direction of the fuel flow, **Figure 8**. These characteristical components were subject to further inquiries.

7.2 Microscopic Deposit Analysis

The detailed observation of the surface respectively the morphology of the deposits revealed differences based on various boundary conditions. Those of nozzles operated with zinc-free conditions demonstrated a fine but crispy texture. Similar microstructures originated out of test runs conducted in the presence of zinc. Moreover, zinc-derived deposit coatings displayed more than one layer. The additional layer presented glassy surfaces enclosing the already mentioned particles,

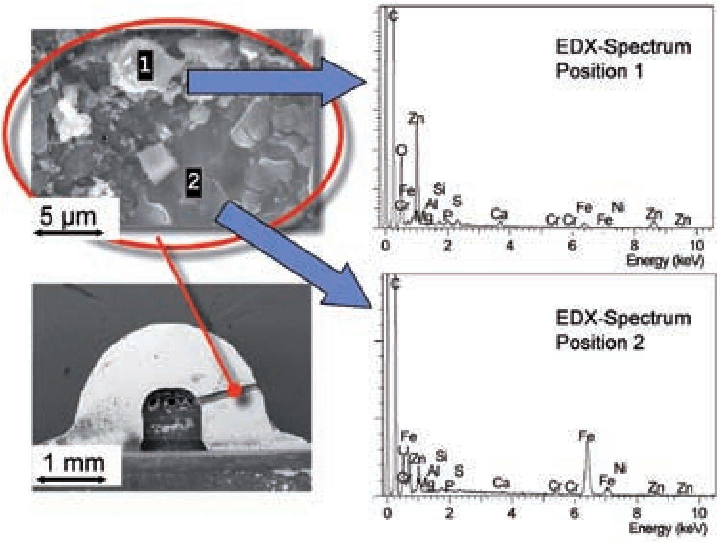

Figure 9: Elemental analysis of deposits via EDX

Figure 9. These grains posses an increased zinc concentration in comparison to the surrounding glassy surfaces. Their existence throughout the total nozzle hole suggested an overall inhomogeneous distribution of zinc in the deposit.

8 Measures Against a Nozzle Fouling

Finally, on the basis of the evaluated results of the investigations the minimization of the presence of zinc in the nozzle orifice is advised. A zinc-free diesel fuel is highly recommended. In case of the need to utilize zinc-containing lubricant, the transport into the fuel circuit has to be avoided. The decrease of fuel temperatures is indicated to reduce the formation rate of deposits. Enhanced cavitation is supposed to lead to a reduced tendency towards nozzle fouling. Since actual trends in the development of combustion processes point towards flow optimized nozzle hole geometries, a careful consideration is suggested regarding the demands of the combustion process development and robust engine performance. First approaches are presented by Argueyrolles et al. [3].

9 Outlook

In the course of the research project important parameters influencing the formation and removal of deposits were identified. In the future inquiries regarding the mechanism of the coking in the presence of zinc is desirable. A first step could be a more detailed analysis of the deposits utilizing greater magnifications. A mass spectroscopy of the deposits appears critical according to the immense chemical complexity. Additionally, the potential of further metals regarding nozzle fouling is interesting, which may be components of lubricant additives or contaminations inside the diesel fuel. Especially agricultural products as feedstock of bio diesel contain salts which constitute partly of those metals. Hence, further detailed investigations are indicated.

Currently the development of injection systems focuses on common rail technology including multiple injection events per stroke. An assessment of the effects on nozzle coking is proposed.

10 Summary

In the scope of the FVV research project "Coking", No. 14567 N, of the Institute of Combustion Engines in Aachen (VKA) of RWTH Aachen University (Germany) several coking cycles with varying boundary conditions were conducted utilizing a 6.3-l truck diesel engine to investigate the mechanisms and important parameters regarding the formation of deposits in injection nozzles.

The analysis revealed an increase of nozzle fouling by increased temperature inside the nozzle hole. The metal zinc was identified as a major parameter affecting the build-up of deposit coatings. Above a critical zinc concentration a significant level of coking is achieved within a relative short period. These deposits are partly reversible by zinc-free operation. Manufacturing processes were utilized and advanced to gain optical access to all positions inside the nozzle. The detailed analysis by means of electron microscopy revealed that zinc-derived deposits contain an inhomogeneous distribution of the metal. Nozzle fouling based on zinc-free operation displayed morphology with a finer texture in comparison to the coking developed in the presence of zinc. Cavitation is the major parameter to inhibit the formation of deposits.

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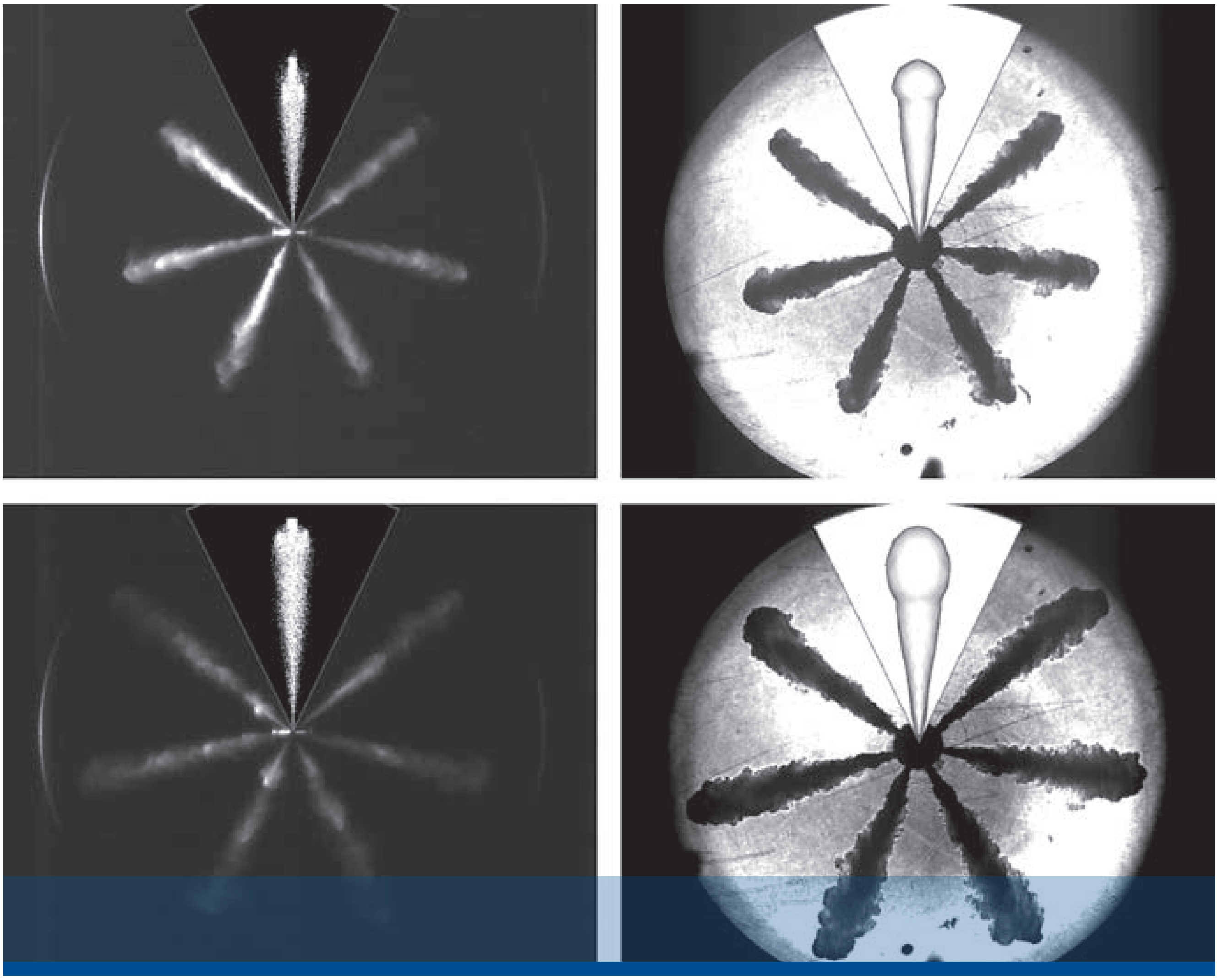

Gasoline Evaporation as a CFD Model for Spark-ignition Engines

Within a research program covered by the Deutsche Forschungsgemeinschaft (DFG) a multi-component evaporation model for gasoline was developed at the Institut für Technische Verbrennung (ITV) of the University of Hannover. This model for a spark-ignition engine respects the properties of real gasoline as a multi-component mixture and describes its evaporation behaviour by a statistical approach. The program was accompanied by experimental investigations at the Institut für Kolbenmaschinen (IFKM) of the University of Karlsruhe. The PhD thesis containing this work was awarded with the Award “Hermann Appel Preis 2007“ of the IAV GmbH.

1 Introduction

The knowledge of the spray breakup and evaporation process in GDI engines, homogenous as well as stratified, is essential for an accurate prediction of the of the injection event. In most cases in experiments and simulation the gasoline is substituted by a simplified surrogate fuel. The properties of this single component species diverge from gasoline which consists of more than 200 components of different hydrocarbon groups. The transfer is not valid in all cases.

This work presents the development of a multi-component evaporation model, which describes the composition of the surrogate fuel consisting of an infinite number of components by a statistical approach. The composition effects in the liquid and gaseous phase and their influence on the following processes can be discussed and evaluated.

2 Multi-component Evaporation

Modelling of evaporation processes in engines impacts directly the forthcoming processes like ignition, combustion and emission formation. Uncertainties in the calculation of one model will be assigned into the next model and grow steadily. So it is visible that the choice of the fuel model has big influence on the quality and portability of the simulation results. In “state-of-the-art“ simulation software, the fuel is mostly described by a single component surrogate, which ability to describe the global properties of gasoline is insufficient in several cases.

This attempt is motivated by the necessity to minimise the complexity and computational effort because, even by neglecting the interaction of the single species, for each component one mass and energy equation has to be solved additionally. The multi-component evaporation-model presented in this paper applies the principals of the continuous thermodynamics on the evaporation process and avoids the dilemma of rising computational time with increasing accuracy.

2.1 Modelling the Evaporation

The continuous evaporation model is based on a statistical approach and is well known in the literature for describing the behaviour of multi-component mixtures like crude oil and polymer solutions [2, 6]. Instead of using a small finite number of species the composition of a continuous mixture is described via a probability density function. **Figure 1** explains the basic difference between a discrete and a continuous mixture. Both diagrams in Figure 1 show the fractions of individual components of the mixture printed over a macroscopic property (for example the boiling temperature). The

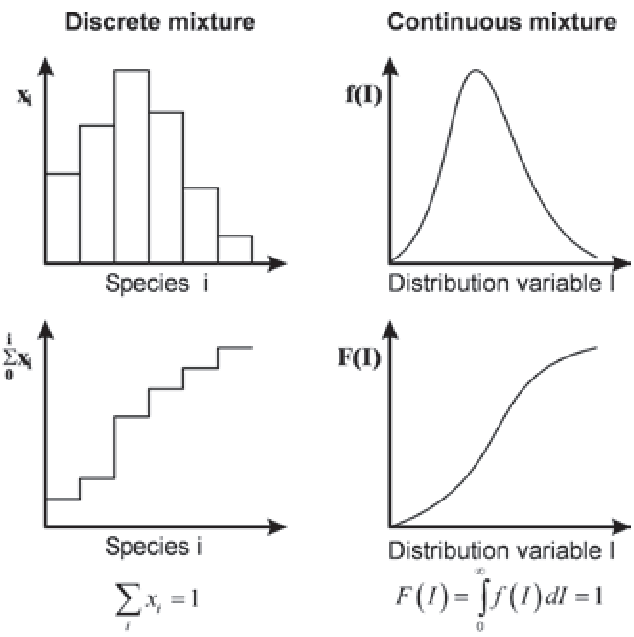

Figure 1: Discrete and continuous multi-component mixture

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Table 1: Distribution parameters known from literature and form parameters used in this work

Parameter	Diesel (49)	Gasoline (79)	Gasoline new
Θ in g/mol	185	86	96
σ in g/mol	43	36	24
α	19	6	16
β	10	15	6
γ	0	0	0

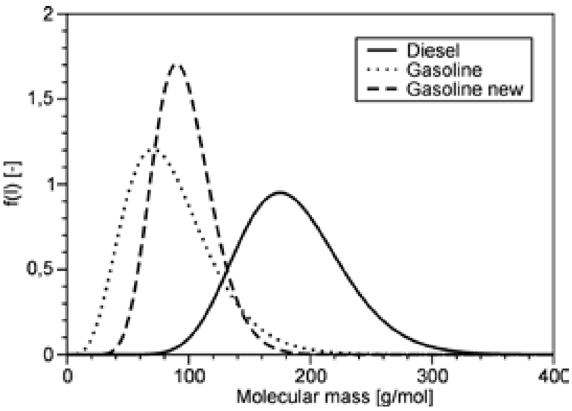

Figure 2: Distribution functions based on parameters shown in Table 1

area beneath both curves must be equal to unity 100 %. In one case this will be reached by a summation, in the other case by integration.

While in the „classic“ approach the mixture is described by discrete species, in the continuous case the assumption of an infinite number of components is made. The distribution function on the right hand side of Figure 1 defines the composition of the mixture by a distribution variable. In this case the molecular mass was chosen. The form of the distribution function directly influences the evaporation behaviour. In the literature beneath the Gaussian distribution mainly the gamma function, Eq. (1), is known to describe the composition of crude oil derivates [10]. The distribution function represents one homologous group of the mixture, for example alkanes.

In Eq. (1) I is the distribution variable and α, β and γ are the form parameters which define the shape of the distribution function. The form parameters will be set (in this work) in such a way that the boiling curve and the vapour pressure of the model fluid match to the real fuel. Integrating the distribution function leads to the zeroth moment oh the distribution Eq. (2). Weighting the function with I respective I^2 and integrating leads to the first and the second moment of the distribution function. The first moment Θ represents the mean and the second moment Ψ is a measure for the variance [7]. Eq. (3) and Eq. (4) describe the relation between the mean and the variance molecular mass of the fuel dependent on the form parameters of the distribution function. **Table 1** shows values for the form parameters for gasoline and diesel known from the literature.

In this work a new distribution function was introduced (gasoline new), based on current investigations concerning the composition of gasoline [4]. This function allows a better correspondence with steam pressure and boiling line of gasoline.

Figure 2 shows the different distribution functions according to Table 1. The fundamental difference between diesel and gasoline can be seen very clear, as well as the difference between the function for gasoline known from the literature and used in this work. The smaller width of the curve represents a smaller variance of components, but with a higher fraction in the mixture. Since gasoline consists not only of the group of alkanes, but also of other hydrocarbon groups which influence the evaporation behaviour, an extended formulation has to be developed to include this instance into the model.

In the literature many approaches are known to enable the description of another species group (for example aromatic compounds). Beside the semi-continuous approach presented in this work, approaches with two distribution functions [5] as well as distribution functions with more than one distribution variable ($f(I, J)$) [3] were implemented successfully. Due to the fact that the aromatics contained in the gasoline consists only of a few different aromatic hydrocarbons with similar properties, the semi-continuous approach seems most suitable and thus was chosen for the work presented here. Following the approach of the semi-continuous thermodynamic, in analogy to [12], the distribution function in Eq. (1) will be superposed with a discrete aromatic species. The mathematical relation is shown in Eq. (5).

Examining the transport processes in the liquid phase and linking it with the gaseous phase, leads to conservation equation for the mean and the variance of the distribution function, which describe the constantly changing composition of the liquid phase due to evaporation. A detailed formulation can be found in [8].

The model presented in this work was implemented in the CFD-code Kiva-3V. The Reynolds-averaged Navier-Stokes equations (RANS) are solved with a conventional k-ε-turbulence model. The interior of the rapid compression machine was modelled with a non-spray adapted, block structured sector-grid with a side-lengths of approx. 2 mm in the relevant spray region. Due to the symmetry of the geometry the grid was reduced to the volume around one spray plume. The fluid exiting the nozzle is described by so called blobs. These are fuel ligaments with the size of the nozzle orifice. The primary and the secondary breakup droplets are described by the Rayleigh-Taylor and Kevin-Helmholtz breakup models.

2.1.1 Single Droplet Investigations

In order to demonstrate the functionality of the model, detailed single droplet investigation were performed to analyse and validate the potential of the model under isolated boundary conditions. **Table 2** contains the boundary conditions used for the investigations.

Toluene was used in this work to represent the aromatic species, since it is the aromatic species with the highest fraction and its properties differ only slightly to the rest of the aromatic species [4]. Beside the toluene fraction in the mixture all other parameters were held constant, in order to evaluate the influence of the aromatic species. **Figure 3** and **Figure 4** show the impact of the toluene fraction on the temporal progression of the droplet size and temperature. An increasing of the fraction slows down the evaporation process. The degression of the droplet's surface area increases nearly linear with the toluene fraction, what can be seen in the progression of the temperature as well. Starting with a „pure" multi-component mixture, it shows a nearly linear increase in temperature after a short heat up. The increasing toluene fraction leads to decrease in temperature rise forming a temperature plateau at a fraction of 60 %. At the end of the droplet's lifetime a fast asymptotic increase of the temperature can be observed.

Due to a higher heat capacity and so resulting higher boiling temperature of the mixture by rising toluene fraction, more energy is needed to support the evaporation. The discussed plateau, known as adiabatic droplet temperature in evaporation processes, is formed after reaching the boiling temperature of the toluene (383.4 K [11]). This variation shows plausible the influence of aromatic fractions in fuels on the evaporation process of single droplets and forms the expectation to observe this effects in complete sprays as well.

Table 2: Boundary conditions for single droplet investigations

Feature	Number	Unit
Ambient temperature	500	K
Ambient pressure	5	bar
Droplet diameter	20	µm
Droplet temperature	300	K
added species	0, 20, 40 and 60 % toluene	–

Table 3: Boundary conditions for the rapid compression machine investigations

Feature	Number	Unit
Injection pressure	300	bar
System pressure at start of injection	20	bar
Gas temperature	630	K
Fuel	Premium gasoline (RON 100)	–
Injection mass	approx. 17 and 2 x 8,5	mg

2.2 Experimental Investigations in a Rapid Compression Machine and in a Pressure Chamber

The validation of the semi-continuous evaporation model for gasoline sprays was performed in a rapid compression machine (RCM) and in a temperature and pressure chamber. Different optical measurement techniques were used in these two setups. While on the RCM a Mie/Scattering and Schlieren setup was realised, a laser induced fluorescence technique (LIF technique) was applied on the pressure chamber. The different techniques used ensure a detailed examination of the injection and evaporation process. The Mie and Schlieren technique enables to investigate the liquid and gaseous phase of the spray, while the LIF technique visualises the fuel fraction in the gas phase qualitatively. For the investigations in the RCM a piezo research injection system with a direct actuated needle was used, developed by the Institut für Technische Verbrennung in Hannover. The nozzle was derived from a production common rail system with a 7-hole configuration with a diameter of 160 µm for each hole. **Table 3** explains the boundary conditions for the RCM measurements.

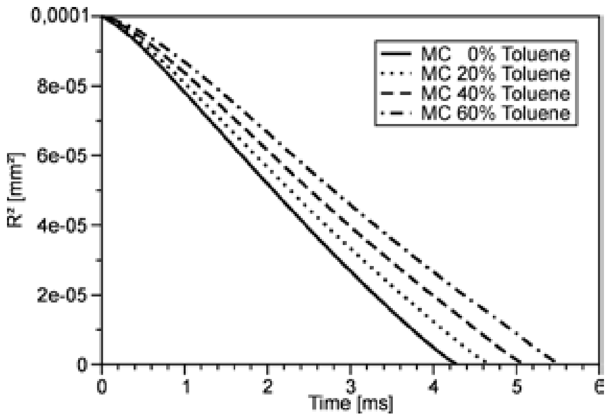

Figure 3: Digression of the radius squares R^2 for toluene mixtures of 0, 20, 40 and 60 %

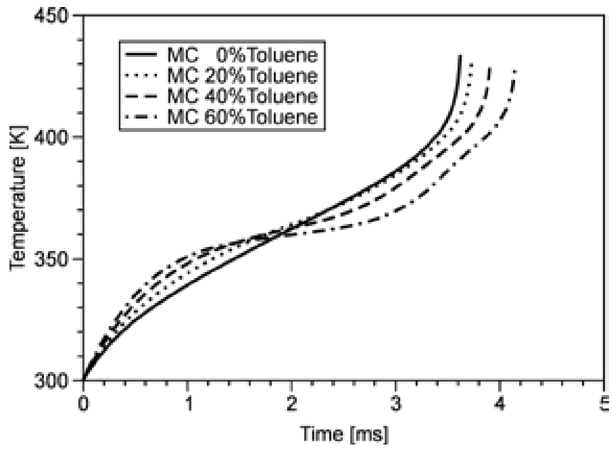

Figure 4: Temperature plots for toluene mixtures of 0, 20, 40 and 60 %

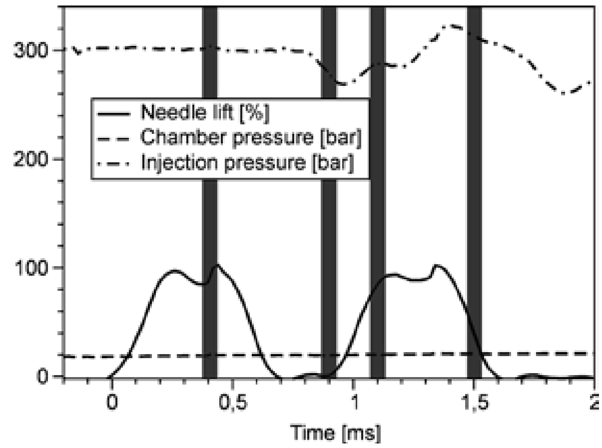

Figure 5: Signal plots for needle lift, chamber pressure and injection pressure for the two-pulse injection

For the validation of the simulation a two-pulse injection was chosen with a dwell time of approximately 0.2 ms between the two pulses. The measured signals for rail pressure, needle lift and cylinder pressure are presented in **Figure 5**. Grey lines in Figure 5 indicate the exposure times during the investigations and correlate temporal to the pictures in Figure 6.

In **Figure 6** both simulation and experimental results are shown, while the calculated results appear on the ("12 o'clock") position and superpose the RCM pictures. Figure 6 the left hand side shows the Mie/Scattering pictures and in the right hand side the Schlieren pictures. The liquid phase is presented in the simulation by droplet clusters (finite number of drops with equal properties) and the gas phase by an iso-surface that separates the gaseous fuel from the air. 900 µs after start of injection the first injection ends, while the second injection did not begin so far. The mixture cloud lifts of from the nozzle, visible by lifted iso-surface. The second pulse breaks through the mixture cloud from the first injection and rises its penetration depths. In both phases a good congruency between simulation and experiment is visible in all presented pictures. Most macroscopic aspects of the spray, like penetration and break-up angle, match between simulation and experiment.

The so far used optical measurement techniques allow only an integral investigation of the spray phenomena. An investigation of the spray's interior is not possible with these techniques. Therefore further experimental investigations were conducted in the temperature and pressure chamber using the LIF technique. These measurements were performed at the IFKM in Karlsruhe. The optical fuel (free of aromatic components) used for the investigation was mixed with 3-Pentanone, which gains fluorescence properties when activated with a laser beam with appropriate wave

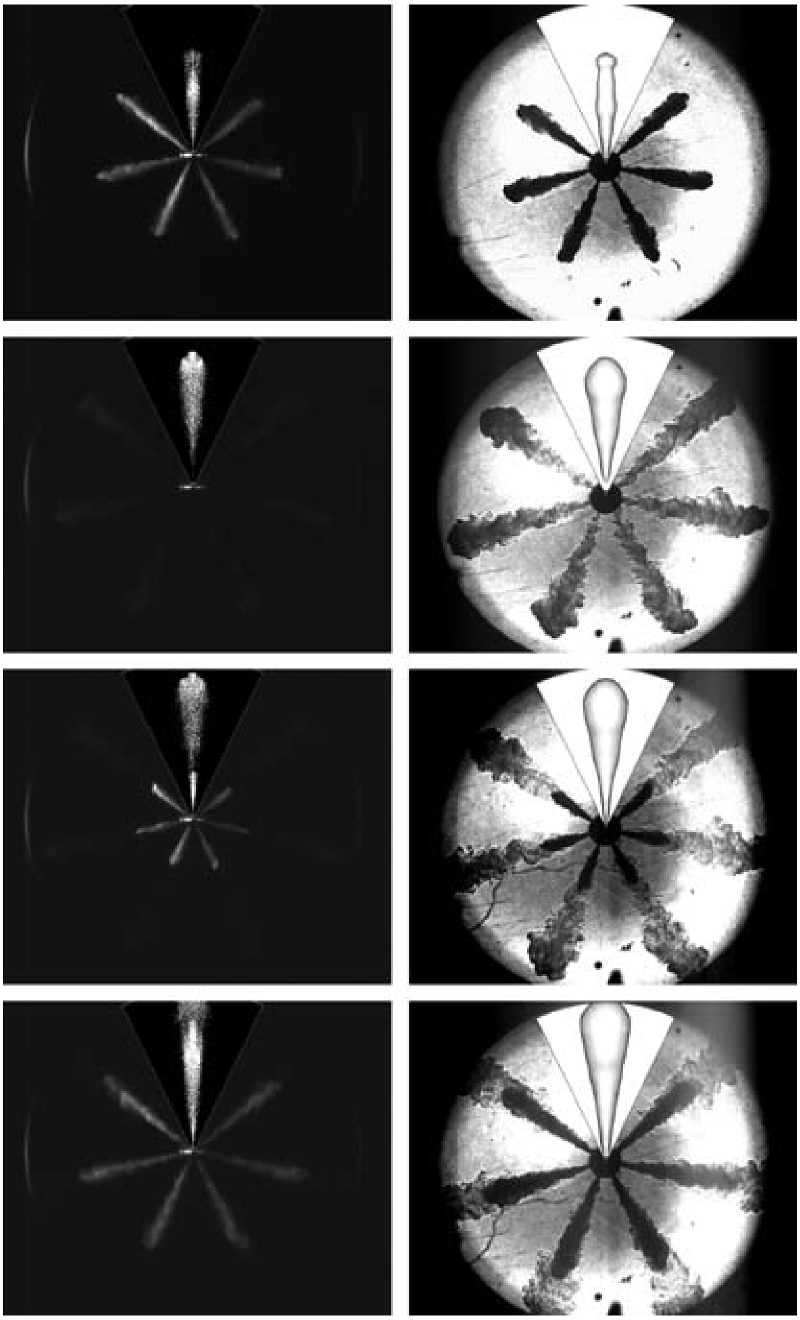

Figure 6: Comparison of Mie (left) and Schlieren (right) experiments and simulation results for the two-pulse injection at 400, 900, 1100 and 1500 µs after start of injection; the framed sector presents results from the simulation

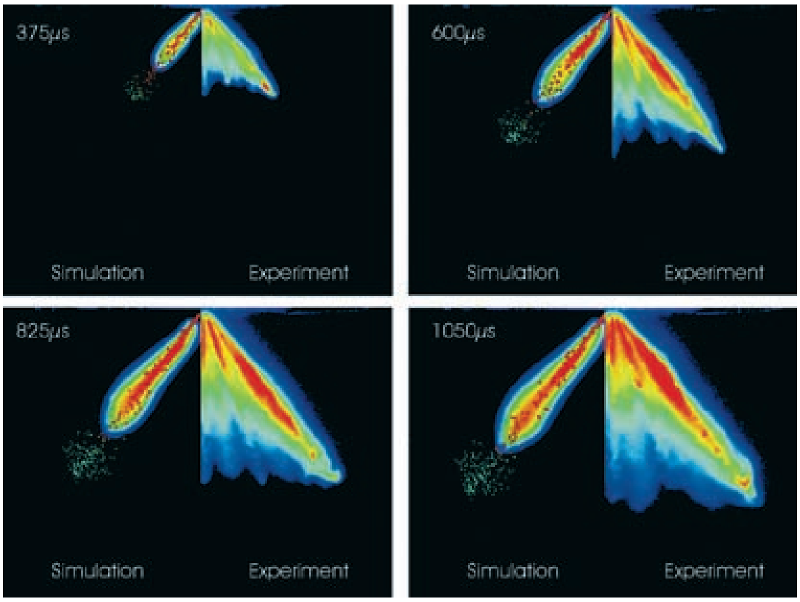

Figure 7: Comparison of results from simulation calculations (left) and LIF measurements (right) at 8 bar backpressure and a chamber temperature of 25 °C at 375, 600, 825 and 1050 µs after start of injection (from top left to bottom right)

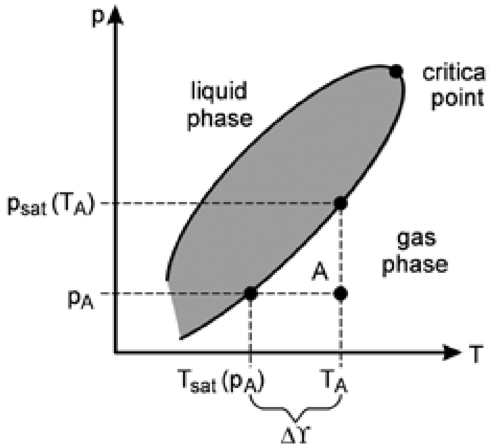

Figure 8: Schematic presentation for the degree of superheat

lengths. The nozzle is a 12-hole nozzle for stratified gasoline combustion. The optical set-up uses a laser, which beam was spread out and formed an approximately 1 mm thick central cut trough one of the 12 sprays.

The operating point shown here was at 8 bar backpressure and 25 °C of temperature. The rail pressure was set to 200 bar. **Figure 7** shows the results of the optical measurements and the calculated results at different times after start of injection. The LIF pictures are on the right hand side, and the calculated results are on the left hand side of Figure 7. Since the LIF technique is just qualitative, the colour scale in both investigations was adopted in this way, that the maximum intensities in simulation and experiment match the same colour.

For the analysis it has to be noted, that the tracer is present in both liquid and gas phase, but the fluorescence intensity from the liquid is very poor due to refraction at the surface of the droplets. The parcels in the simulation were scaled at coloured analogous, in order to take this effect into account. On the right side of the picture a fluorescence intensity can be observed outside of the spray plume cut by the light sheet. This is caused by the partial reflection of the laser beam on the droplet's surface and another reflection on the spray plume behind the light sheet. This should be kept in mind and should not be taken into account. Qualitatively a good correlation can be seen between calculated and measured results. Both intensity/fuel fraction as well as the macroscopic form of the gas phase show same behaviour. Especially the temporal and local maximum of intensity inside the spray is calculated correct; the same observation is valid for the fuel concentration rectangular to the spray axis. At 825 µs after start of injection can be observed, that the high intensity at the spray tip is mainly related to the liquid phase. Gaseous fuel is also present in the simulation, but the concentration is too low and cannot be seen in Figure 7 due to the chosen scaling.

As a summary it can be observed, that the model presented in this paper is capable to describe the evaporation process in gasoline engines. Both optical measurements techniques enable a detailed validation of the model with a high qualitative accuracy. The total potential of the model can be observed especially at boundary condition typical for gasoline engines, where low temperatures and pressures are present and the fuel properties, like vapour pressure, are dominating the processes.

3 Flash Boiling

Flash boiling is known to be the sudden phase change in nozzles during injection, when the local pressure falls beneath the vapour pressure due to high fuel temperatures and/or low backpressures (comparable to cavitation in diesel nozzles). The most dominant parameter for the flash-boiling intensity is the degree of superheat, **Figure 8**.

Starting at a point *A*, a fuel is in a superheated state defined by the temperature T_A and the pressure p_A. The pressure $p_{sat}(T_A)$ is the saturated vapour pressure at temperature T_A. $T_{sat}(p_A)$ is the saturated temperature at the pressure p_A. The difference T_A-$T_{sat}(p_A)$ is called degree of superheat, here $\Delta\Upsilon$. A fluid existing in a superheated state with $\Delta\Upsilon > 0$ will evaporate very fast (and isobar) until the equilibrium with $\Delta\Upsilon = 0$ is reached.

3.1 Continuous Flash-Boiling Model

The continuous flash-boiling model consists mainly of four separated, but temporal competing processes:

1. nuclei formation
2. bubble growth
3. bubble disruption
4. evaporation.

First vapour nuclei are formed inside the nozzle and transported in the liquid phase to the nozzle's exit. The number of nuclei formed is described by Eq. (6). The most influencing parameter for this process is again the degree of superheat $\Delta\Upsilon$. The bubbles formed inside of the nozzle are enclosed in fuel ligaments

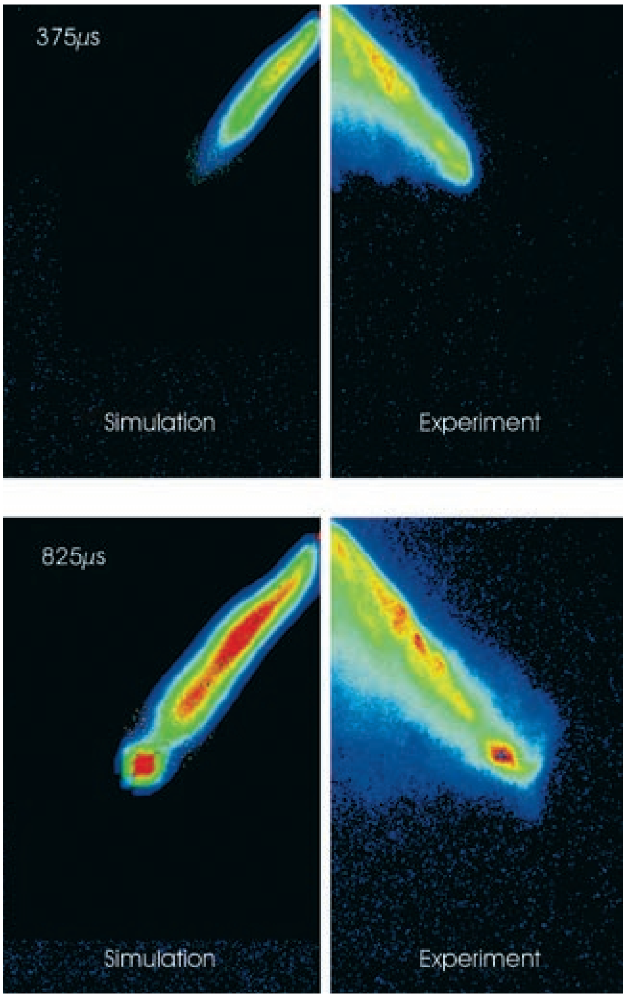

Figure 9: Comparison of results from simulation calculations (left) and LIF measurements (right) at 1 bar backpressure and a chamber temperature of 100 °C at 375 and 600 µs after start of injection (from top left to bottom right)

and can grow or collapse depending on the surrounding conditions in the gas phase. The fundamental description of the bubble dynamic is based on [9] and is known as the Rayleigh-Plesset equation, Eq. (7). This non-linear second order differential equation in the Flash-Boiling sub model of the CFD-code is solved for each drop in each computational time step, representing the highest computational effort of the whole model. When the bubbles grow, the vapour fraction inside the droplets grows also, Eq. (8). If a critical value for the vapour fraction is reached (ε_{krit} = 0.45-0.53 [1]), the droplet becomes unstable due to the vapour enclosed inside of the drops and breaks up into a very high number of droplets, which is twice as high as the number of bubbles before. This mechanism is responsible for the fast atomisation associated with flash-boiling. These three processes are all the accompanied by the conventional diffusive evaporation process.

The mass transport linked to the evaporation is described by Eq. (9), while the heat transfer coefficient α_{fb} is dependent on the degree of superheat [1]. The here described sub-processes are not separated in the real case, but temporally and locally linked. In the real flash-boiling case evaporation and bubble disruption are competing processes with equivalent time scales. The model shown here contains only a simplified description of the flash-boiling breakup by not-including the different directions and speeds of the very small droplets after the breakup and so leading to uncertainties especially at conditions with high degree of superheat. Due to this fact the validation of this model is limited to very small degrees of superheat.

From a basic point of view the here presented model is not dependent on the fuel used in the simulation. But evaluating the presented sub-processes it is visible, that the properties of the fuel, especially the vapour pressure and the degree of superheat, are dominating the process. The continuous flash-boiling model is able to describe these features and enables so a more detailed formulation of the processes, due to the multi-component aspect of the model fuel.

3.2 Experimental Investigation in the Pressure Chamber

The developed flash-boiling model was validated on the same optical LIF setup in the pressure chamber presented in Chapter 2.2 before. The operating point for the injection system is the same like in the prior investigations. The pressure and temperature was set to 1 bar and 80 °C resulting in a degree of superheat of $\Delta\Upsilon$ = 4 K according to Figure 8. **Figure 9** compares the calculated results (left) to the experimental results (right). For both shown time steps a very good congruency between the results can be observed. The shape of the spray and the distribution of the fuel concentration matches very good between simulation and experiment. Especially to mention the concentration lift off at the spray tip.

Figure 10 presents the droplet size distribution throughout the spray for the described operating point. The liquid

Formulae

$$f(I) = \frac{(I-\gamma)^{\alpha-1}}{\beta^{\alpha}\Gamma(\alpha)} \exp\left[-\left(\frac{I-\gamma}{\beta}\right)\right] \quad \text{Eq. (1)}$$

$$F(I) = \int_0^\infty f(I)dI = 1 \quad \text{Eq. (2)}$$

$$\int_0^\infty f(I)IdI = \Theta \quad \text{Eq. (3)}$$

$$\int_0^\infty f(I)I^2dI = \Psi = \Theta^2 + \sigma^2 \quad \text{Eq. (4)}$$

$$g(I) = x_{mol} f(I) + x_{dis}\delta(I - I_{dis}) \quad \text{Eq. (5)}$$

$$N = C \cdot \exp\left(\frac{-5.28}{\Delta\Upsilon}\right) \quad \text{Eq. (6)}$$

$$R\ddot{R} + \frac{3}{2}\dot{R}^2 = \frac{\Delta p}{\rho} \quad \text{Eq. (7)}$$

$$dM_{fb} = \frac{\alpha_{fb} \cdot \Delta\Upsilon \cdot A}{h_{fg}} dt \quad \text{Eq. (8)}$$

$$\varepsilon = \frac{V_{Bl}}{V_{Bl} + V_{fl}} \quad \text{Eq. (9)}$$

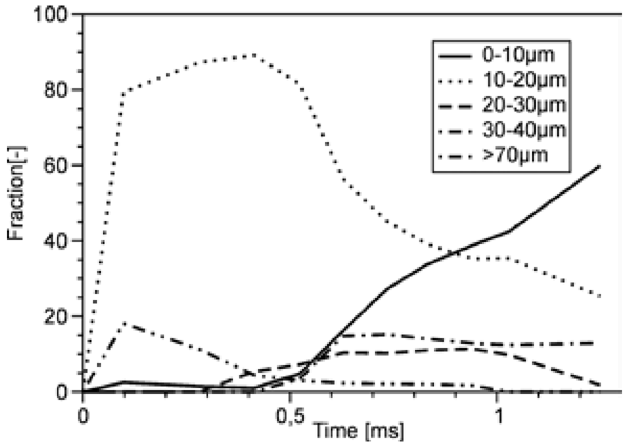

Figure 10: Fractions of drop size classes calculated with the continuous flash-boiling model

ligaments are exiting the nozzle as so called blobs with a diameter equal to the nozzle diameter (here 120 µm). Shortly after start of injection the droplet class of 10-20 µm drops reaches a high fraction of more than 80 % of the total spectrum. The other size classes show significantly lower values. At 0.5 ms after start of injection the fraction of the smallest size class increases up to a value of over 60 %. The fraction of the 10-20 µm class falls in the same time frame beneath a value of 30 %.

The fractions of the other classes remain nearly unaffected. Only the class of the biggest drops is not existing after 1 ms after start of injection. In these values the impact of the flash-boiling breakup can be observed clearly. In contrary to the conventional breakup mechanism the breakup occurs very fast and immediate after exiting the nozzle, forming droplets with a small diameter of less than 20 µm. According the conventional mechanism the breakup would occur within a cascade, forming intermediate droplet size classes. Caused by low absolute pressure in relation to the vapour pressure of the liquid, the bubbles inside the droplets can grow very fast until breakup occurs, resulting in a high number of small droplets of the 10-20 µm class observed.

Acknowledgements

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After the end of injection ($t > 0.5$ ms) the evaporation of these drops is present, leading to their smaller participation, the part of the next smaller class in sum is increasing.

4 Summary and Conclusions

The presented work at University of Hanover explains the uncertainties that are unavoidable by using surrogate fuels in conventional formulation of the fuel gasoline for spark-ignition engines, and shows the possibilities gained with the continuous evaporation model. The experimental validation of the model in a rapid compression machine and in the pressure chamber demonstrates the functionality and potential. Decomposition effects in the liquid phase and their impact on the evaporation can be described with a slight increase in computation time.

Superposing the continuous model with a discrete aromatic species can progress the development of more detailed ignition and combustion models than before that are highly dependent on the concentration of aromatic species. The flash-boiling model is limited to operating points with a low degree of superheat, but shows clearly which dominant effect is applied by the fuel properties and how important a problem related formulation of this processes can be. An extension of the model with a more complex bubble-growth caused breakup and validations at higher degrees of superheat are logical and necessary development steps for this approach.

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Dr. Akihiko Saito
FISITA President

"FISITA is an Engineering Congress – Not a Lobby Group"

In the run-up to the FISITA World Congress from September 14 to September 19, 2008 in Munich (Germany) MTZ and the outgoing FISITA President Dr. Akihiko Saito, talked about his personal goals, the experiences he has made being President of the world body for automotive engineers and the 32. FISITA World Congress. Also the challenges the automotible industry has to face were discussed by Saito, who is Senior Technical Executive at Toyota as well.

MTZ In your opinion, what is the main function of FISITA?

Saito FISITA's mission is to share knowledge among the world's automotive engineers and by doing so, contribute to the development of automotive technology.

MTZ What would you say are the biggest challenges that the automotive industry has to face nowadays?

Saito Our biggest challenge is to develop a new approach to product development which is more forward-looking and proactive. Toyota's vision for research and development can be summed up in two words: "Zeronise" and "Maximise". "Zeronise" means we are striving for zero negative impacts on our environment, zero accidents and zero traffic congestion. At the same time, we are striving for maximum positive impact on personal enrichment through comfort and fun. This philosophy is key to gaining widespread consumer acceptance of more sustainable technologies. Take as an example Toyota's petrol hybrid vehicle Prius, which gained worldwide popularity when the second generation was launched in 2003. It provides outstanding environmental performance, and at the same time, uplifting driving performance. This is exactly the realisation of both "Zeronise" and "Maximise".

MTZ What are the goals, the expectations for the FISITA World Congress 2008?

Saito Our first goal is always to have excellent quality technical information and lively exchange between engineers from industry and academia. We will also have an innovation at the Munich congress that we are calling "Islands of Excellence". We had a competition in which universities from around the world applied for the chance to present their automotive research projects.

> "Our biggest challenge is to develop a new approach to product development which is more forward-looking and proactive"

MTZ What impacts will the congress have, politically and on the automotive industry?

Saito We have to keep in mind that FISITA is an engineering congress – not a lobby group. But we know that today, people everywhere are more concerned about the social and environmental impact of the automobile than ever before. Politicians have to react to that concern, but to do this well, they must understand the facts. I worry that, too often, the message from some of our leaders seems to be "don't worry, technology will come along and solve everything". The present concern over oil prices and future alternatives to fossil fuels is a prime example. As engineers and scientists, we accept that the world is looking to us to find solutions to sustainable mobility. And every day we are making progress. But we are not magicians. We know that sustainable mobility calls for a partnership between players in the automotive industry, but also energy companies, policy makers and consumers. Cleaner, safer vehicles also need better infrastructure, improved urban planning, better driver education and the right fiscal incentives. If we can use the FISITA Congress to help convey the technical reality behind the choices facing governments around the world, then I believe we can contribute to a climate where mobility policy decisions are based on the best scientific knowledge, and not just on short-term political expediency.

"Our first goal is always to have excellent quality technical information and lively exchange between engineers from industry and academia"

MTZ If you look back, with which goals and ambitions did you start as FISITA President?

Saito When I was a young engineer starting my career with Toyota, it was considered a great honour to be asked to present a paper at the FISITA Congress. Today the industry is much more global and the pace of technical development is faster. There are many more competing conferences and sources of information on offer for engineers. So when I had the privilege to be elected FISITA President, my ambition was to strengthen our association so that we can respond more rapidly to the changing needs of our members and protect FISITA's reputation as the world body for automotive engineers.

MTZ What did you achieve, in which concerns do you think you have brought FISITA forward?

Saito We have expanded our links with industry. Today, most of the leading global vehicle manufacturers and top suppliers support FISITA, which is vital to our future development. Just as important is the fact that now, among our FISITA Honorary Committee we also have the three leading energy companies, along with leading companies from the electronics and Intelligent Transportation Systems worlds. This is important for us because we have to build links between engineers working in the vehicle world – our traditional members – and those colleagues working in related and increasingly important areas like fuels and lubricants, electronics and telematics. We have also welcomed our first member companies from India and China – the two fastest-growing automotive economies, where FISITA has to be more involved. In 2012 we will have the FISITA Congress in Beijing, hosted by SAE China. I think we can be proud of the progress we have made in strengthening FISITA's organisation over recent years, but there is still much work to be done.

MTZ Which personal conclusion can you draw from your presidentship?

Saito The activities we support in the education area, such as our sponsorship of the FISITA Formula SAE World Cup, has given me the chance to meet students from around the world as they take part in exciting, highly successful competitions organised by our member societies. This has led me to conclude two things: First, that the work of FISITA and our 38 societies makes a real difference. And second, that the next generation of automotive engineers absolutely possess the skills and the talent needed to solve the challenges we now face.

MTZ Dr. Saito, it has been a pleasure talking to you.

Interview conducted by Roland Schedel.

Dr. Akihiko Saito

is FISITA's President for the 2006 to 2008 term. He was born in 1940 and joined Toyota in 1968 after earning a Doctorate in Mechanical Engineering at Nagoya University (Japan). In 1980, Dr. Saito became a member of the team that handled product planning work for the Corolla, served as Chief Engineer and became a General Manager in the Product Planning Division in 1987. Dr. Saito was named to the Board of Directors in 1991, became Managing Director in 1996 and Senior Managing Director in 1998. In 2001 Dr. Saito assumed the position of Executive Vice President. In July 2005 he became Senior Technical Executive in Toyota and also Vice Chairman of Denso. He was named Chairman of Denso in June 2007.

Dr. Akihiko Saito, FISITA President, Senior Technical Executive Toyota Motor Corporation