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COVER STORY

Commercial Vehicle Powertrains with Lower Emissions and Consumption

DUAL CLUTCH TRANSMISSION
Traxon Dual from ZF for Trucks

MODEL-BASED METHODS
for the Calibration of Modern Internal Combustion Engines

MICRO EMULSIONS
in Diesel Engine Combustion

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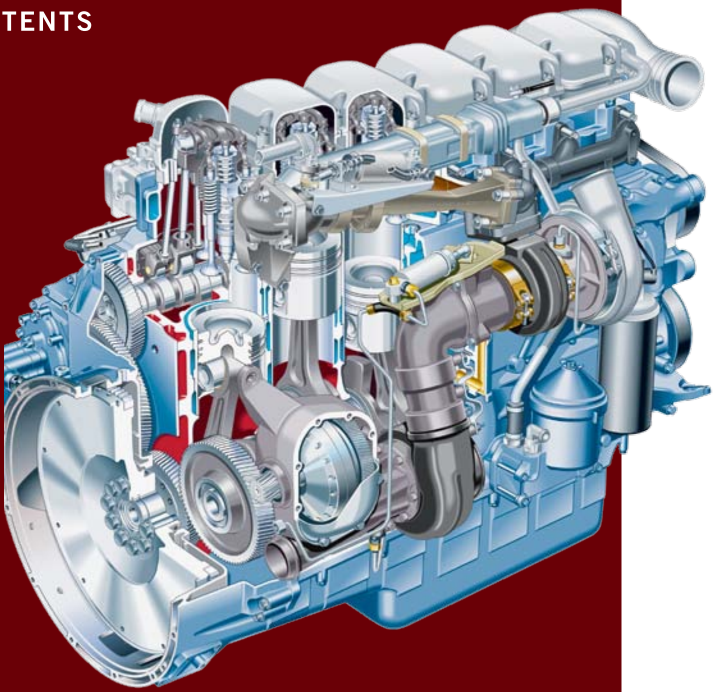

COVER STORY

Commercial Vehicle Powertrains with Lower Emissions and Consumption

The art of developing commercial vehicle engines lies in balancing the individual components in such a way that the overall system offers the optimum in power output and lifetime while at the same time minimising fuel consumption and emissions – and all at an acceptable cost. This calls for the best possible compromise between optimisation strategies that are often mutually conflicting. In order to achieve these ambitious aims, manufacturers are increasingly relying on modular solutions and improved system integration.

COVER FIGURE & FIGURE ABOVE © Scania

Learn from the Diesel

Dear Reader,

what can electrified powertrains learn from the "diesel"? The answer is short and simple: emotionalisation. We might recall that, until the early 1990s, diesel cars were seen as noisy, dirty and slow, used at best as workhorses by cost-conscious business customers such as taxi firms. After all, diesel engines were bought for their fuel economy and not for their driving dynamics. For that reason, diesel-powered cars played only a minor role in Germany. But then came turbocharging and direct injection: two technologies that turned the carthorse into a thoroughbred. The diesel changed from being an engine for cars with the dynamics of a tractor to a premium power unit that left its established gasoline-powered counterparts far behind in many disciplines. One example is its impressive low-end torque, providing plenty of "pulling power" for a comfortable, relaxed driving experience. It was this that made the diesel appealing to a much broader group of car buyers, with the result that diesel cars have accounted for almost 50 % of all new cars registered in recent years.

Whether a diesel engine makes financial sense or not is often of secondary importance in the decision to buy, as customers tend to choose a diesel specifically because they like the way it drives. And this, in my opinion, is precisely what supporters of electrified powertrains should focus on. Simply having good fuel economy was not the reason why the diesel engine made its breakthrough, and low CO_2 emissions alone will not help hybrid and electric drive systems either. What is needed is the targeted emotionalisation of hybrid and electric vehicles. Anyone who has driven a modern electric car will have marvelled at the incredible amount of torque available even from a standstill, and those who have experienced the boost function of a hybrid – either directly at the crankshaft or through an electrically supported compressor – will never want to do without the extra "kick" that it offers. I believe that marketing should do much more to convey these aspects, whether it is in advertising, in conversations with customers at car shows or in the car showroom itself.

Rationality and technical facts about CO_2 reduction are all well and good, but someone who spends a lot of hard-earned cash on a car also wants be emotionally addressed. By the way, this problem can also be transferred 1:1 to discussions concerning CNG. Hardly any other drive concept offers such low costs per kilometre as a natural gas engine, and yet CNG vehicles are little more than a niche product and are seen by the general public as the eccentric choice of boring penny-pinchers – which brings us back to the diesel of the pre-boom era. The solution might be a monovalent system approach which, thanks to a higher level of supercharging, would allow a higher specific power output than a comparable gasoline-powered engine – with driving dynamics and emotion as an added benefit.

Best regards,

Richard Backhaus

Richard Backhaus,
Vice-Editor in Chief
Wiesbaden, 2 February 2015

CO_2 Reduction by Adaptive Raw Emission Control for HD Applications

The demand for low-emission propulsion in the commercial vehicle sector necessitates the use of complex exhaust after treatment such as selective catalytic reduction (SCR) to reduce the diesel engine NO_x emissions. In conventional approaches the engine and the exhaust aftertreatment system are usually controlled independently from each other. This article from the Virtual Vehicle Research Center shows the potential concerning CO_2 reduction by an adaptive raw emission control based on a system-wide view.

NO_X REDUCTION THROUGH EXHAUST AFTERTREATMENT

In commercial vehicles, the diesel engine has been the most commonly used for years. Caused by the principle of the diesel working process high efficiencies are achievable but this also leads to higher raw emission of nitrogen oxides (NO_x).

A technology for reduction of these emissions is exhaust gas recirculation (EGR). For the desired EGR rates, the exhaust gas backpressure needs to be above the boost pressure level. This causes higher pumping losses and therefore a higher fuel consumption. Arguments as discussed in [1] limit the reduction by such internal measures by economical as well

as technical reasons. To meet the limits of newest emission legislations – like 0.46 g/kWh NO_x at Euro VI – an exhaust aftertreatment system is necessary.

On one hand, the use of an aftertreatment system increases the manufacturing costs; on the other hand, it offers the possibility for a functional separation of efficiency- and emission-related measures. In situations where most of the NO_x reduction is done by the aftertreatment system, the engine can be operated with efficiency-optimised settings [2].

RELATIONSHIP BETWEEN SCR EFFICIENCY, FUEL CONSUMPTION AND LOAD PROFILES

The required NO_x reduction by the aftertreatment system to fulfill Euro VI is shown in **FIGURE 1**. If the efficiency of the NO_x aftertreatment system increases from 90 to 95 %, the acceptable raw emission can be increased from 4.5 to 9 g/kWh.

On the one hand, the efficiency while driving depends on the component layout, on the other hand there are also operating and ambient condition dependent parameters affecting the system. Selective catalytic reduction is a common technology for the commercial sector [3]. The temperature is a crucial parameter for this system. For the catalyst used in this project, **FIGURE 1** (right) shows the SCR efficiency versus temperature at constant space velocity. Temperature needs to be beyond 250 °C to achieve efficiencies above 90 %. The temperature itself depends on the load profile and therefore can vary in a broad range. Taking into account the above-mentioned relationships by an adaptive raw emission control a decrease in consumption of operating fluids (diesel fuel and AdBlue) is possible together with emission conformity.

EXPERIMENTAL SETUP

The development and testing of presented adaptive raw emission control was carried out on a heavy duty diesel engine at engine test bed. The basic data of engine and aftertreatment system are listed in **TABLE 1**.

CONTROL STRUCTURE

The study deals with concepts for control of raw emission that are not completely new but the challenge is the large number of considered relationships and interactions. First of all the aim was to ensure emission compliance together with lowest operating fluid consumption even in dynamic operation. An important restriction was the use of state-of-the-art production sensors to ensure the direct usability of the developed solution in a commercial vehicle.

With these given boundary conditions, the team developed the AREC (Adaptive Raw Emission Control) concept that allows a proper control of the complex relationships and therefore an easy han-

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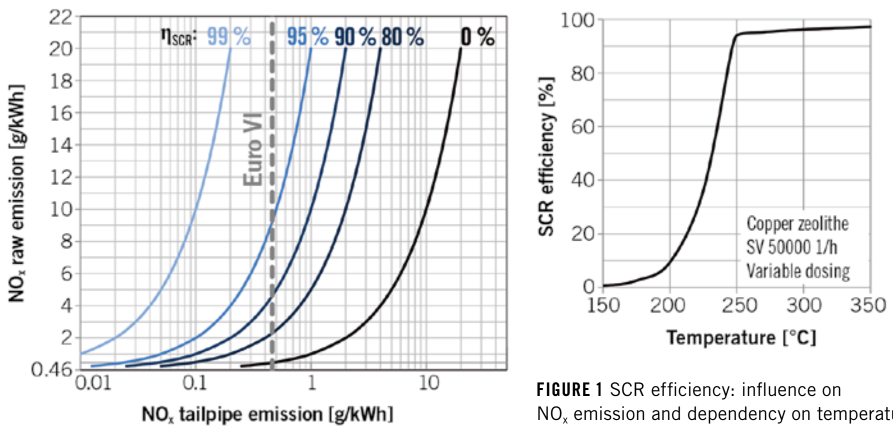

FIGURE 1 SCR efficiency: influence on NO_x emission and dependency on temperature

dling as well as calibration. Additionally to the original engine control unit a rapid prototyping system – where the new functionalities were implemented – was used. **FIGURE 2** shows the basic structure and elements of the control system.

OPERATION MODES AND PHYSICAL MODELS

The basic working principle of AREC is the adaptation of the engine and SCR operation by changing to various operation modes. These represent different calibrations for selected emission levels and engine operating conditions. While the control of the diesel injection path and the air path primarily is derived from map-based strategies, the team developed an after treatment control via a combination of empirical map based models and physical models for all exhaust aftertreatment components. The implemented operating modes are divided into two categories, temperature based thermal management and emission-based operation fluid consumption management.

For the following selected conditions different engine operation modes considered, **FIGURE 2**:
- cold engine
- transient operation
- cold aftertreatment system
- moderate SCR efficiency
- high SCR efficiency.

The algorithm activates heating measures in order to reach or maintain operating temperature in the case of a cold start and to hold the temperature at an appropriate level in the case of low load condition. These measures are implemented in the engine operation modes "warm up" and "heating". With sufficient temperature level the operation modes for moderate or high SCR efficiency are active. AREC controls the transition between these modes by using information provided by the so-called emission integrator. This element monitors continuously the tailpipe emission using values from the NO_x sensor after the SCR system and information from the engine control unit. The calculated tailpipe emission is compared with the target NO_x level and used to select the appropriate engine operation mode. In order to ensure a smooth transition between the different modes the algorithm changes the modes during transient phases.

TABLE 1 Basic data of engine and exhaust aftertreatment system

Number of cylinders	6
Cylinder arrangement	In-line
Displacement	11 l
Compression ratio	17.0:1
Number valves/cylinder	4
Peak cylinder pressure	200 bar
Injection pressure	1800 bar
Charging system / turbocharger	One-stage with VTG
Charge-air cooler	Water/air
High pressure EGR	Cooled with reed valves
Diesel oxidation catalyst	8 l (Pt/Pd)
Diesel particulate filter	22 l (Pt/Pd)
Hydrolysis catalyst	8 l (Pt/Pd)
SCR catalyst	26 l (cooper zeolith)

TEST CONDITIONS AND PROCEDURE

The study consists of two steps for testing and validation. The first step was to ensure the Euro VI compliance. Based on the related test procedure (World Harmonized Transient Cycle (WHTC) and World Harmonized Stationary Cycle (WHSC)) the necessary parameters for the selection of the engine operation modes were determined by calibration. For the given engine concept, minor fuel savings can be achieved within the WHTC. On the one hand this is due to the included cold start, on the other hand by the relatively low load level of the WHTC. A high temporal portion of the operating modes with thermal management measures and low NO_x raw emissions results from this. The modes for consumption-optimised calibration are rarely used.

The second step determined the potential of adaptive raw emission control in different driving cycles at operating temperature. The team conducted experiments on the engine test bed at constant ambient conditions and with the same calibration as used for the Euro VI test procedure. The test sequence covers typical situations like urban, rural and highway traffic. For the most important long-haul traffic situation – the highway operation – three cycles with different vehicle loads and routes were available. Characteristic for the urban and rural cycle are the broad range of engine speed and particularly at the urban cycle the low load level. In all highway cycles the engine is operated in a narrow speed range. Different vehicle loads and road profiles leads to considerable load level differences in the highway cycles.

For better comparability of cycles two parameters are used – the effective average power and the temporal portion of phase with motored or low idle operation, **TABLE 2**.

The amount of motored and low idle operation is given in % of the total cycle duration. The effective average power is defined as the work measured at the clutch per cycle time without phases with motoring and idling.

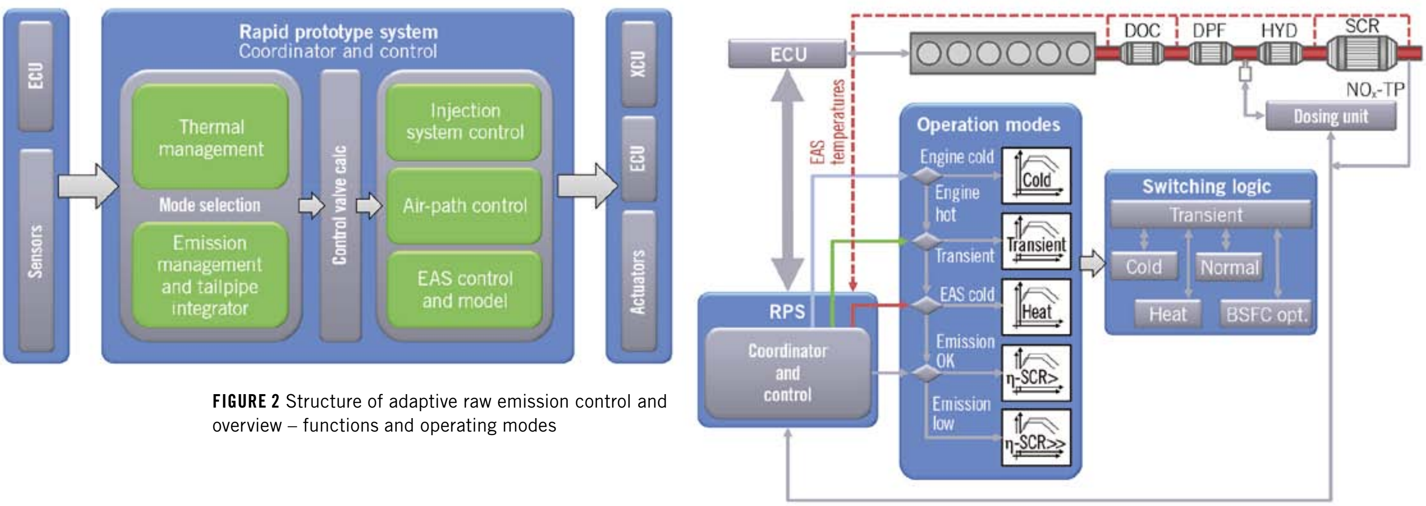

FIGURE 2 Structure of adaptive raw emission control and overview – functions and operating modes

Cycle [-]	Type [-]	Vehicle load [%]	Effective average power [kW]	Percentage low idle and motored operation [%]
A	Urban	75	60	42
B	Rural	75	97	38
C	High-way	75	118	29
D	High-way	100	160	34
E	High-way	50	117	22

TABLE 2 Overview cycles

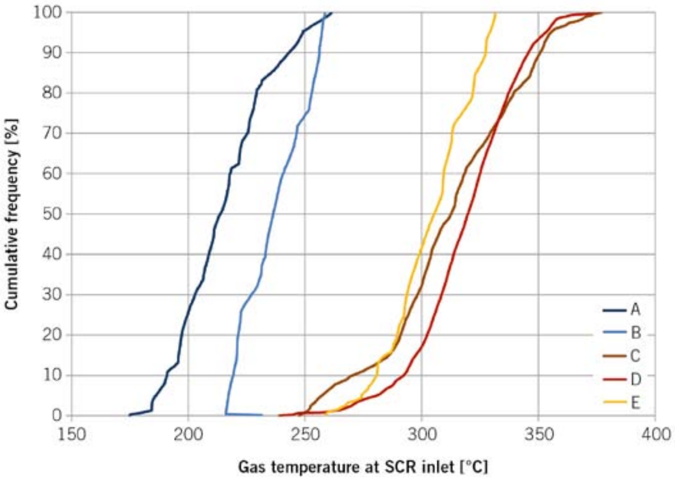

FIGURE 3 Cumulative frequency of the gas temperature at SCR inlet for cycles A to E

Cycle [-]	Difference in CO_2 emission [%]	Difference in operating fluid consumption [%]	NO_x emission [g/kWh]
A	+0.2	+0.2	< 0.4
B	-0.8	-0.2	< 0.4
C	-2.8	-1.5	< 0.4
D	-2.3	-1	< 0.4
E	-1.6	0	< 0.4

TABLE 3 Cycle results

The effective average power is the important parameter in view of fuel consumption since it significantly influences the achieved temperature within the SCR system and thus the SCR efficiency. **FIGURE 3** shows the cumulative frequency of gas temperature at SCR inlet for the cycles A to E. Together with the temperature-dependent SCR efficiency shown in **FIGURE 1** it is obvious that the engine can be operated frequently with efficiency orientated settings.

CO_2 REDUCTION POTENTIAL OF ADAPTIVE RAW EMISSION CONTROL

The lab team performed the above cycles to assess the potential for savings – with activated and deactivated AREC. The differences in CO_2 emissions and operating fluid consumption are shown in **TABLE 3** in relation to the results achieved without AREC. The operating fluid consumption is the sum of diesel fuel and AdBlue consumption. For evaluation, a cost ratio of 1:3 between diesel and AdBlue was assumed. The urban cycle in fact shows a slightly increased consumption for CO_2 emissions and the operating fluid consumption. Using AREC at highway operation CO_2 savings up to 2.8 % and savings in operating fluid consumption up to 1.5 % are achievable.

SUMMARY AND OUTLOOK

Within the work, we compared typical vehicle operation modes like urban, rural and highway traffic beside legal test procedures WHTC and WHSC. The use of adaptive raw emission control ensures compliance to the NO_x emission limits in all conditions. Especially in highway operation with higher temperature the efficient NO_x reduction enables engine operation modes with lower fuel consumption. This leads to a reduction of total fluid consumption of up to 1.5 %.

Euro VI legislation according ISC (In Service Conformity) regulations requires emission measurement in real life operation with PEMs (Portable Emission Measurement) equipment. The measurement takes place on predefined routes with production vehicle on public roads. The result is generated by post-processing with a complex evaluation routine e.g. by AVL PEMs Concerto tool. As the presented algorithm does not cover criteria as the validity of the PEMs windows, which have to be calculated in a one-second interval, AVL is working for quite some time on an extension of the algorithm. Next generation shall cover requirements of PEMs and ISC legislation in detail and shall further on enlarge the fuel reduction potential by introduction of predictive control algorithms.

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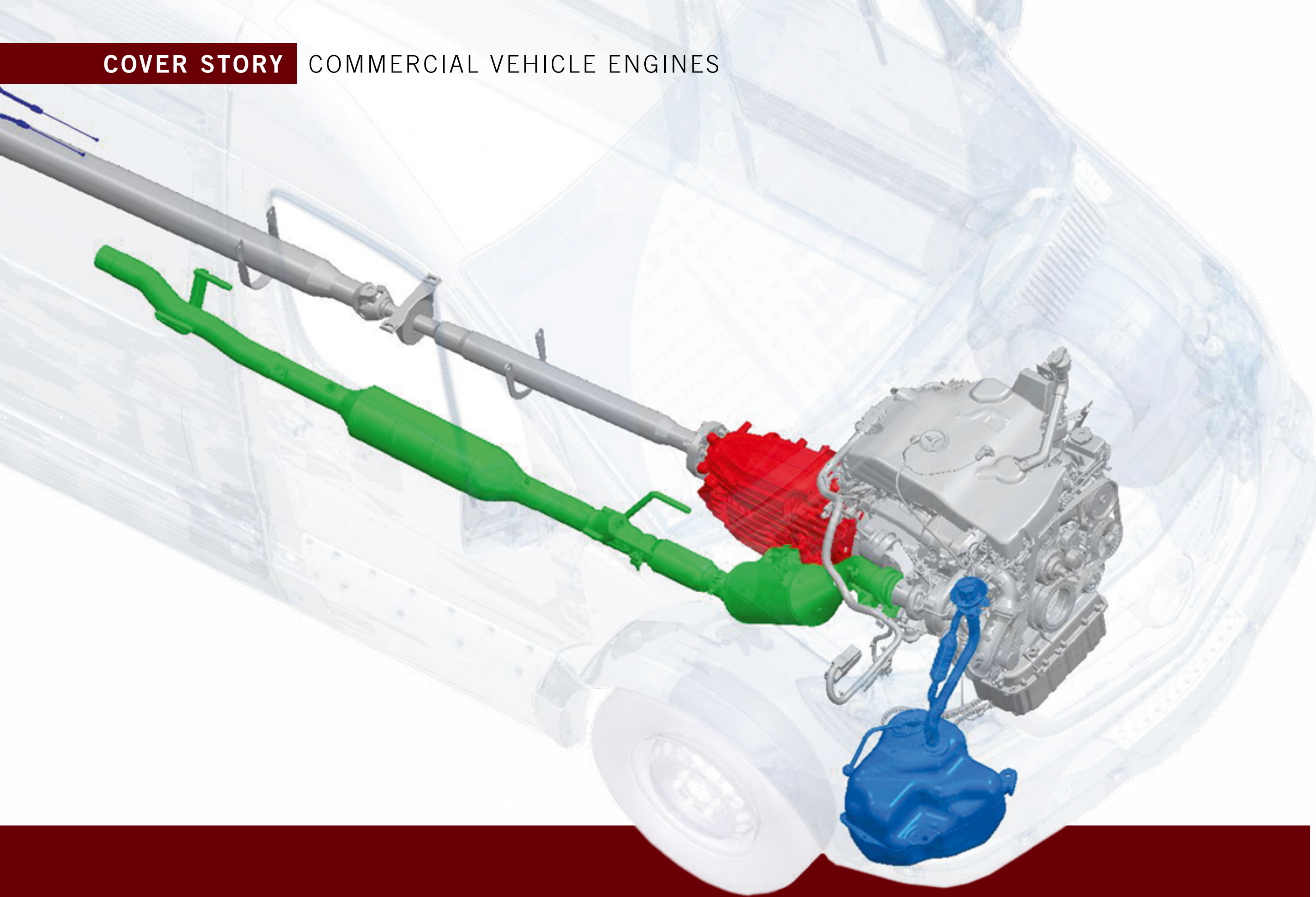

First SULEV Diesel Engine in Mercedes-Benz Sprinter

The new Mercedes-Benz Sprinter for model year 2015 is the first series-production vehicle to be fitted with a diesel engine that already complies with the future LEV III SULEV emissions standards. The following report presents the powertrain, exhaust aftertreatment system and calibration concept of the new transporter for the NAFTA market.

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HISTORY AND MOTIVATION

The name Sprinter is a synonym for the entire van class. The Sprinter is used all over the world in more than 130 countries, mainly sold with diesel engine due to the higher economic efficiency. **FIGURE 1** shows the historical milestones of the Sprinter. Since 2001, Daimler has been selling Sprinter vans in the NAFTA market with diesel engines as Freightliner, from 2003 to 2009 as Chrysler and since 2010 additionally as Mercedes-Benz. In 2007, the V6 3-l diesel engine OM642 was integrated into the Sprinter together with a particulate filter system to meet the NAFTA07 (US2007) emission legislation [1]. In 2010, the goal was to meet the tightened NO_x emissions standards of NAFTA10 (US2010) by implementing the SCR system (BlueTEC) [2]. At the end of 2013, Daimler introduced the 2.1-l in-line four-cylinder OM651 diesel engine together with the seven-speed automatic transmission in NAFTA. The powertrain history of the Sprinter in the U.S. is illustrated with more details in [3].

Considering vehicle improvements like higher cargo volume, drag reduction and extended safety systems it can be summarised that the Sprinter is on its way to more efficiency and cleaner emissions. One further consistent step in this direction is that the Sprinter offers not only the LEV III ULEV version but additionally a LEV III SULEV option with the same OM651 diesel engine powered platform. According to the CARB (California Air Resources Board) emission legislation, the Sprinter is categorised as a Medium Duty Vehicle (MDV). The future LEV III SULEV emission standards of MDVs are to be phased-in from 2018 till 2022. From 2023 onwards the SULEV limits will be mandatory for all newly introduced MDV vehicles.

POWERTRAIN CONCEPTS OF THE NEW SPRINTER

Major powertrain components of the new Sprinter are shown in the cover figure. Besides the common powertrain components like engine OM651, seven-speed automatic transmission, drive shaft, differential and rear axle, the BlueTEC components – AdBlue (DEF: Diesel Exhaust Fluid) dosing system and exhaust system – become more and more critical as it is driven by emission legislation. Therefore, an overall system matching between all powertrain components was necessary to fulfill the emission standards LEV III SULEV and to realise a maximum fuel efficiency and improved driving comfort in the new Sprinter.

The main technology features of the OM651 I4 diesel engine are the two-stage turbocharging system and the common rail 2000 bar piezo injection system. The OM651 is a highly fuel efficient diesel engine for the van segment, even more efficient than the OM642 due to the reduced number of cylinders, lower engine friction loss and the higher level of charging while minimising charge changing losses.

The seven-speed automatic transmission is equipped with a torque converter and a lock-up clutch. In comparison to the forerunner (five-speed automatic), the wider transmission-ratio span with reduced top gear ratio and the closer ratio gear spread (two gears more) in conjunction with an optimised shift program lower engine speeds with its resultant fuel economy improvement and reduced noise and vibration levels.

FIGURE 2 summarises the two powertrain possibilities of the new Sprinter. Both powertrains can be combined with up to three rear axle ratios and offer various gross vehicle weight ratings (GVWR) and four vehicle body variants of the Sprinter. The introduction of the second powertrain variation in NAFTA (OM651 with seven-speed automatic) enables additional possibilities to satisfy specific customer requirements for improved fuel economy and low cost of ownership. New for the model year (MY) 2015 Sprinter is an optional LEV III SULEV version of its OM651 diesel powertrain. Additionally, the OM642 powertrain is completed by an all-wheel drive system in 2015.

FUEL EFFICIENCY MEASURES IN THE NEW SPRINTER

FIGURE 3 gives an overview of the fuel efficiency measures of the new Sprinter. The rear axle has fuel economy and friction optimisation. Additionally, weight, noise and vibration levels are reduced. The energy management consists of battery and power supply management. A new fuel economy optimised generator has been integrated. The Eco power steering pump is electronically controlled depending on the required steering support, while the electronically controlled fuel pump regulates the minimum fuel quantity.

COMBUSTION CALIBRATION

As the Sprinter for the NAFTA region has to fulfill emission test procedures (FTP75, HWFET, US06 Bag 2, UDC, SC03) that differ significantly from those for the European market, the combustion application (i.e. injection timing and quantity, injec-

FIGURE 1 Historical milestones of the Mercedes-Benz Sprinter powertrain development

Engine	Power and torque	Transmission	Drivetrain	Efficiency package	Emissions standard
OM651 I4, 2.1 l	120 kW 360 Nm (265 ft-lb)	Seven-speed automatic (7G-Tronic Plus)	4x2	**BlueEfficiency** Fuel economy rear axle ancillary components	LEV III ULEV **+ LEV III SULEV**
OM642 V6, 3.0 l	140 kW 440 Nm (325 ft-lb)	Five-speed automatic (5G-Tronic)	4x2 **+4x4**	**BlueEfficiency** Fuel economy rear axle	LEV II ULEV

Rear axle ratio	Gross vehicle weight ratio	Vehicle type
3.923	3.88 t / 8550 lbs	Cargo van
4.182	4.53 t / 9990 lbs	Passenger van
4.364	[Canada: 4.49 t / 9900 lbs]	Crew van
	5.0 t / 11,030 lbs	Cab chassis (for body builder)

FIGURE 2 Extended variety of powertrain possibilities for specific customer requirements

tion pressure, boost pressure and EGR) of the OM651 had to be adapted profoundly to maintain high fuel economy within the test cycles and during customer operation.

Due to the wider load and speed range of the US test procedure (compared to that of the European test procedure) the air system adaption strategy had to be modified. Particularly within the US06 Bag2 (which is a reduced variant of the US06 for MDVs) the engineers had to deal with unsuccessful learning attempts. During each learning event the NO_x raw emission increases for a short period of time. Therefore, the adaption strategy was altered so that unsuccessful learning events are minimised even in cases where aggressive driving style is observed.

To achieve state-of-the-art injection accuracy and higher injection pressure, the 1800 bar solenoid valve injection system of the European version was replaced by the 2000 bar piezo injection system. Furthermore the engine features an intake-port shut-off that is applied at low loads and low rpm what allows swirl to be increased. Thus raw particulate emissions were reduced resulting in longer diesel particulate filter regeneration intervals.

Further advantages in fuel economy and raw particulate emissions were gained by matching the combustion application with the optimised performance of the 2nd generation BlueTEC system. The main focus was placed on the warm-up period, as reaching the light-off temperature of the BlueTEC system as fast as possible is crucial for minimising the NO_x tailpipe emissions. This was achieved by adjusting EGR, throttling, injection quantities and timings dependent on the engine temperature.

SECOND GENERATION OF THE SCR SYSTEM

FIGURE 4 illustrates the main components of the BlueTEC system. BlueTEC comprises the exhaust system as well as the AdBlue dosing system with AdBlue tank, filling pipe, dosing pipe, AdBlue dosing module and dosing control unit. Note, the rising requirements of on-board diagnostics (OBD) and its close interaction with engine optimisations are becoming increasingly important.

For the selective catalytic reduction of nitrogen oxides at the SCR catalyst, AdBlue is injected into the exhaust system upstream of the SCR catalyst. Then water vaporises and urea decomposes into ammonia which is stored at the SCR catalyst and reduces the nitrogen oxides of the exhaust gas to harmless nitrogen and water.

Major steps of the 2nd generation were the single unit AdBlue tank with 18 l (4.75 gal.) for all vehicle types, the integration of the standard source unit with volumetric metering and ultrasonic level sensor, the optimised designs of the SCR exhaust system and the dosing control unit. Furthermore, DOC, DPF and SCR catalyst coatings and volumes are optimised to fulfill the requirements of the new emission standard LEV III SULEV.

For a particular economical transportation solution of the new Sprinter, the key challenge of its overall BlueTEC diesel system optimisation was to fulfill the

FIGURE 3 Efficiency features to improve fuel economy and CO_2 emissions

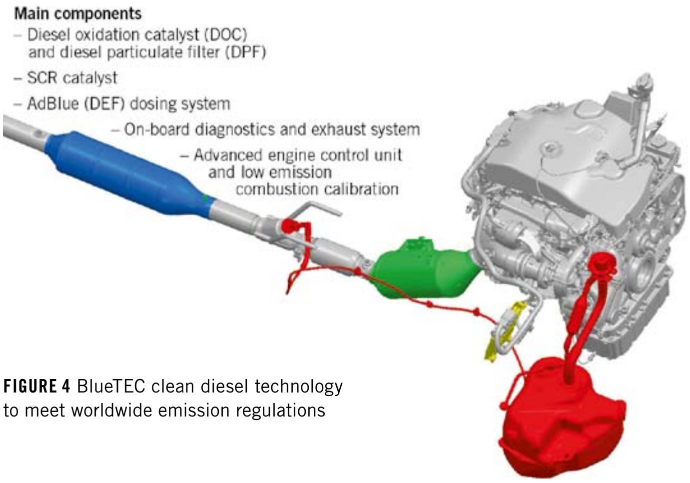


FIGURE 4 BlueTEC clean diesel technology to meet worldwide emission regulations

LEV III SULEV emission standard and simultaneously the aim was to find the best compromise between the highest possible fuel economy at acceptable AdBlue consumptions and a suitable additional weight of the AdBlue tank.

EXHAUST GAS AFTERTREATMENT CALIBRATION

After having fulfilled the ULEV standards in MY14, Mercedes-Benz emphasises its cutting edge position of clean and fuel efficient large vans in MY15 by offering an optional equipment choice that fulfils the SULEV emission standards for all weight categories and body types.

Beforehand, by issuing the LEV III regulation, numerous changes affected the emission category MDV, which covers Mercedes-Benz Sprinter. These changes put various challenges on emission development. Emission compliance needs to be warranted throughout the full useful life of a vehicle's service. LEV III covers a total mileage of 150,000 miles, corresponding to a lifetime extension of 25 % compared to LEV II. In consequence, chemical and thermal aging influences need to be compensated for the extended course. By using up-to-date catalyst technology with optimised design and enhanced gas flow coupled with advanced exhaust gas aftertreatment calibration strategies, Mercedes-Benz reached its goal not to lose any emission performance due to the intensified mileage criteria. As a further enhancement to the LEV II legislation, it is corollary to expect lowered emission limits. The regulation indeed orders accentuations, e.g. regarding non-methane organic gases, oxides of nitrogen and carbon monoxide, shown in **FIGURE 5**.

Within the heavier of the two MDV groups, LEV III requires a tightening of about 29 % for ULEV400 but more than 59 % for SULEV230 in regarding to the respective thresholds of NMOG+NO_x under LEV II ULEV. Simultaneously, an accumulation of these two pollutants was put to law, creating an alternating conflict between heat-up and NO_x raw emissions.

A strong focus is necessary for emission control after engine start, as phase 1 of FTP75 dispenses more than half of the overall weighted tailpipe emission. Due to cumulated pollutant standards on the one and CO_2 fleet average standards on the other hand, the required temperature inside the exhaust system needs to be reached without actions that promote the formation of NMOG or increases fuel consumption too much. By maintaining sufficient exhaust gas temperature and a convertible quantity of raw emissions in phase 2, a low contribution of emissions is reached. This is before restart in phase 3 adds more NO_x emission that needs to undercut the engineering targets in the end.

In addition to the tightening of conventional pollutant types the Greenhouse Gas (GHG) regulation was introduced, which regulates pollutants with a potentially strong impact on climate change. Next to CO_2, GHG covers methane (CH_4) and nitrous oxide (N_2O) emissions by putting both under a cap standard. Among competing goals as the challenging emission standards in conventional emissions, GHG thresholds and the extended full useful lifetime have required a smart choice of catalytic coatings and precious metals without the disregard of production costs. The LEV II certification process for MDV did not allot the SFTP (Supplemental Federal Test Procedure), which passenger cars and LDT had to fulfill additionally.

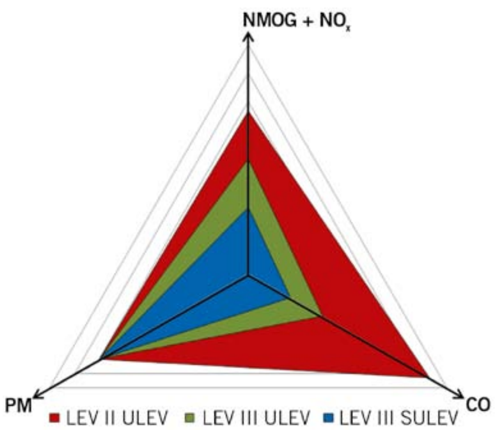


FIGURE 5 Tightening of emission standards since LEV II

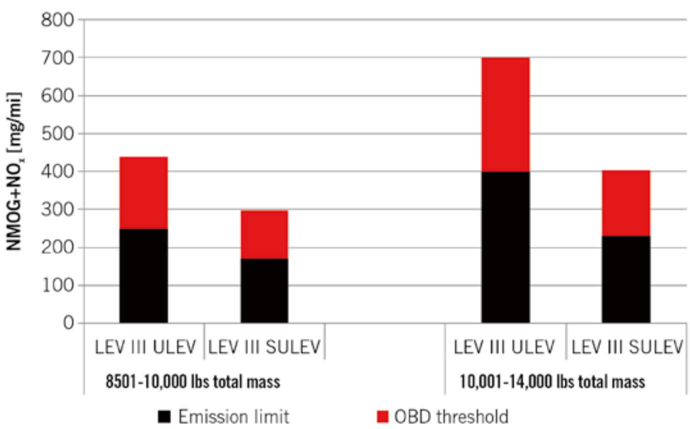
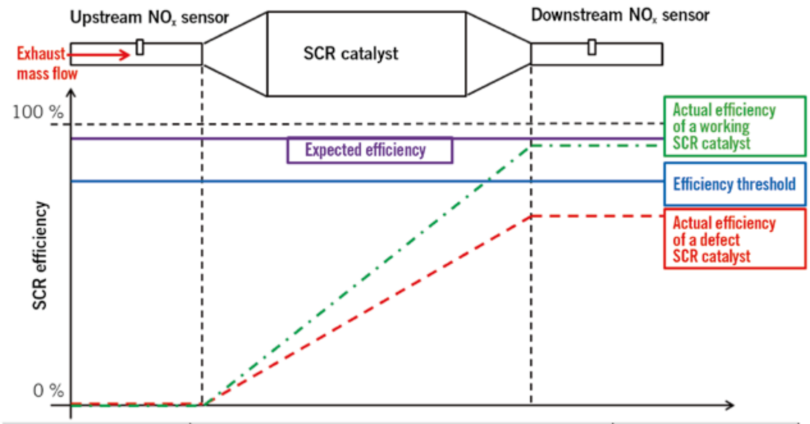


FIGURE 6 Comparison LEV III ULEV and SULEV emission limit (MDV) and related OBD threshold limits (NMOG+NO_x) for SCR catalyst diagnosis



Test case	OBD test procedure				Test results FTP75	
	Pre condition	Pending Mode $07	Confirmed Mode $03 + MIL	Emission test		$NMOG+NO_x$ [mg/mi]
SCR catalyst conversion capability	UDC RGN + FTP72	UDC	UDC	FTP75	OBD threshold	403
					Measured value	316

FIGURE 7 Diagnostic principle and related comparison of SCR efficiency and OBD test results of SCR catalyst efficiency diagnosis for GVWR > 10,000 lbs

Underneath LEV III, MDV now has to show compliance with all FTP- and SFTP-requirements throughout the full useful life of the vehicle. Due to the low power/weight ratio of a Sprinter, the US06 cycle was modified (light weighted MDV) to enable such vehicles to stay within the speed tolerance corridors but no change had to be done for the LA92 (heavy weighted MDV). Nevertheless, a large variety of driving influences needs to be robustly covered by emission calibration, even there are accelerations which require full load. This fact is given for the US06 cycle but also for UDC, which is used for OBD demonstration likewise.

In the past increasing efforts ensuring emission compliance were always going along with steadily rising requirements to monitor comprehensive components of engine and exhaust gas aftertreatment. The challenge is to differ between properly working systems and malfunctions without exceeding legal thresholds by avoiding undesired impacts on the usual system performance.

ON-BOARD DIAGNOSIS CALIBRATION

A major challenge for the calibration of LEV III SULEV MDVs is to meet the OBD regulations passed by the CARB. Low MDV emission limits are directly leading to low OBD threshold limits. The OBD system has to detect OBD threshold limits, e.g. caused by a defective component, and notify the driver by illuminating the malfunction indicator light (MIL) in the instrument cluster. The defective component, indicated by a fault code entry in the relevant control units, must be clearly recognisable for maintenance purposes. For the Sprinter MY15 OBD demonstration, 23 different fault cases with corresponding emission tests have been performed. The validation of these cases required approximately 80 test cycles.

Monitoring of exhaust gas aftertreatment system components is an important task. The correct functionality of the DOC, DPF and SCR catalyst has to be monitored in order to detect emissions to exceed OBD threshold limits. For purposes of detecting a defect SCR catalyst, a best performing unacceptable (BPU) component is generated. This component is damaged so that the emissions are still below the OBD threshold limits but above the emission limits. Within the emissions standards LEV III, clearly tightening OBD threshold limits apply to SULEV compared to ULEV, **FIGURE 6**.

The OBD system must detect the SCR catalyst BPU component. The detection is performed as followed: based on the upstream and downstream NO_x sensor signals, SCR efficiency will be calculated in the engine control unit and compared with the expected efficiency according to the operating mode of the vehicle. The expected SCR catalyst efficiency is calculated by applying a complex function including several input parameters such as temperature, exhaust mass flow, quantity of DEF injection and other values. If the actual SCR efficiency is below the expected efficiency and an efficiency threshold, a fault code will be stored, **FIGURE 7**.

During the OBD demonstration, the fault detection of a SCR catalyst will be performed by three driving cycles after a preconditioning phase. In a first OBD UDC (Unified Driving Cycle) the malfunction must be detected and a pending fault code has to be stored in the engine control unit. In a second UDC the pending fault code has to be confirmed by detecting the malfunction a second time. Subsequently, the end of pipe emission level of the system, including the defective component, needs to be validated in a third driving cycle, FTP75. The level must be below the OBD threshold limit.

All components of the exhaust gas aftertreatment system used for validation of the emission level, except the BPU component, must be aged to the end of the full useful life. This means, according to emission standard LEV III, aged over 150,000 miles. **FIGURE 7** shows the OBD test results: The OBD system is detecting the BPU component in two UDC successfully. The following FTP75 emission cycle proves that the end of pipe emission level is kept below the OBD threshold limit.

EMISSION AND FUEL ECONOMY RESULTS

FIGURE 8 demonstrates the fuel economy and the NO_x emission results of the new OM651 Sprinter in comparison to the still available OM642 Sprinter. NO_x tailpipe emissions are reduced according to the lower emission standards LEV III SULEV. 28.1 mpg at the Highway test cycle and 21.1 mpg at the FTP75 City test cycle are the results of the overall fuel economy optimisation and calibration of the OM651 powertrain and ancillary components. That means for the OM651 Sprinter another 19 % improvement in fuel economy at the FTP75 City cycle and 14 % improvement at the Highway cycle compared with the OM642 Sprinter. Also on the road, the OM651 Sprinter confirms impressively its improvement in fuel economy compared with the OM642: At the AMS fuel economy test cycle, **FIGURE 9**, 26.5 mpg can be reached what means an

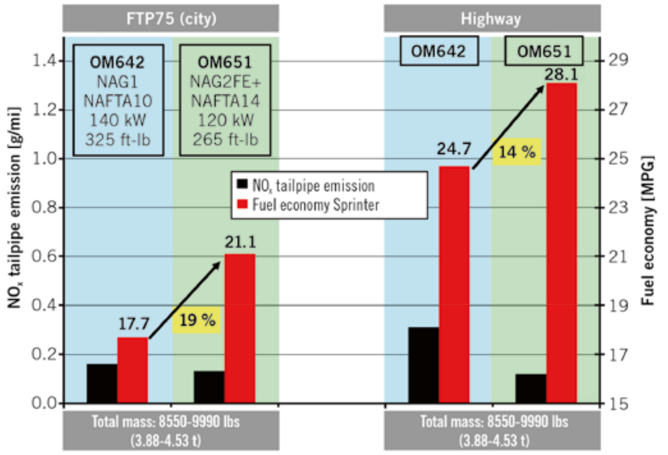


FIGURE 8 Clearly improved fuel economy and NO_x emissions with OM651 Sprinter

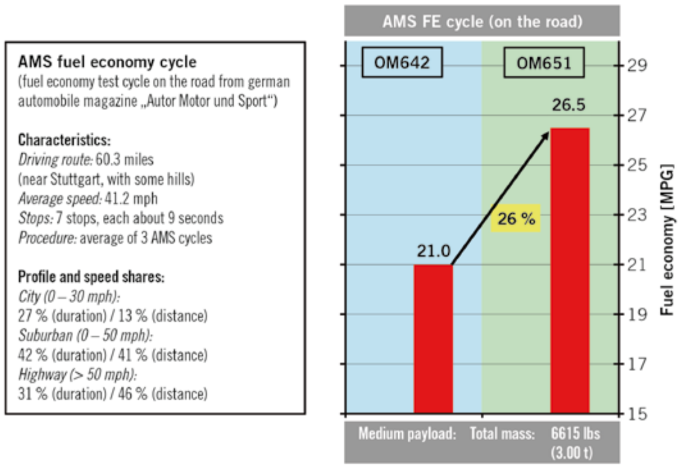


FIGURE 9 Fuel economy results at the AMS FE cycle (real test run on the road near Stuttgart, Germany)

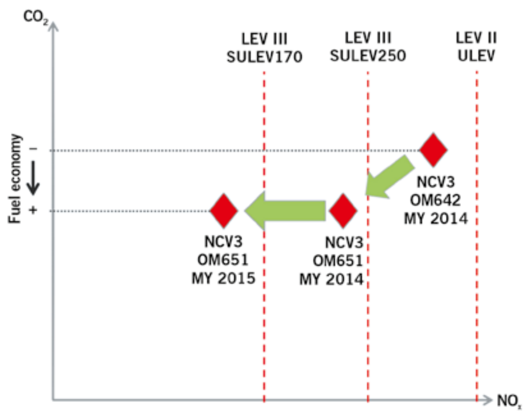


FIGURE 10 Position of SULEV variant in CO_2/NO_x comparison (schematically)

improvement of 26 % in fuel economy in comparison with the OM651 Sprinter.

By meeting the SULEV standards, there has been a strong focus on not exceeding the reached level of fuel economy, neither in city nor in highway use. As shown in **FIGURE 10**, OM651 ULEV shows a significant benefit regarding emissions and fuel economy compared to OM642. With the introduction of a SULEV compliant emission label, the reached level of fuel efficiency of the OM651 could be maintained, nevertheless the end of pipe emission was lowered under all limits of the LEV III SULEV legislation.

CONCLUSION

The new OM651 powertrain offers up to 19 % improvement in fuel economy compared with the OM642 powertrain of the Sprinter. Fuel economy of the Sprinter is benchmark in the large van segment (e.g. Vincentric Best Fleet Value).

Second generation of the BlueTEC clean diesel technology is implemented into the new Sprinter. It reduces the emissions to meet worldwide regulations: The new Mercedes-Benz Sprinter of MY 2015 is the first production vehicle with a diesel engine that already fulfills the future LEV III SULEV emission standards.

Thanks to an improved combustion, emission and OBD calibration the higher requirements of LEV III SULEV have been met and the fuel economy maintains on the same familiar high level of the ULEV version. Consequently, Daimler's Sprinter van remains "best-in-class" regarding fuel economy and has now a further option for lowest emissions. Environmentally conscious van customers now have a serious alternative to gas-powered vehicles (CNG, LPG) by choosing a diesel powered SULEV Sprinter with its common high fuel economy, high torque and fast response. The promissory fulfilment of the SULEV emission standards with a diesel engine demonstrates once again the innovation drive and the green leadership of Daimler's Mercedes-Benz Sprinter.

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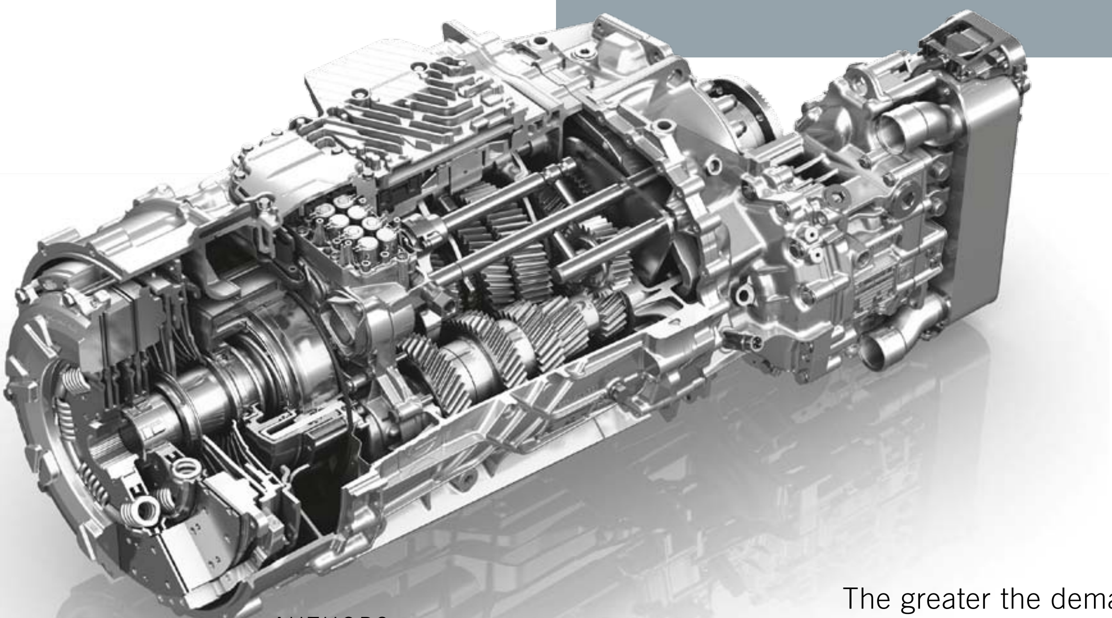
The authors want to thank all colleagues of Daimler AG and development partners, who have contributed to the powertrain development of the new Mercedes-Benz Sprinter. Special thanks go to Markus Paule, Dieter Waeller and Prof. Dr. Joerg Zuern for their sustainable support of this serial development project.

Truck Dual Clutch Transmission Traxon Dual from ZF



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The greater the demand for additional savings in fuel consumption and CO_2 for heavy trucks, the more important are new answers for all areas of the driveline – beginning with the engine through to the axle. With the Traxon Dual automatic dual clutch transmission system, ZF has now developed a key element for this and other requirements.

INITIAL SITUATION AND DEVELOPMENT REQUIREMENTS

The targets of reducing fuel consumption or increasing efficiency are still in the focus of the commercial vehicle industry. However, following the great advancements of the previous years, it is now increasingly more complex to achieve further significant improvements in heavy trucks. Advanced transmission technologies offer a major and easily realisable potential for this. For this reason, the Traxon modular automatic transmission system [1] for heavy trucks is about to enter volume production. Even in the standard design, this system can save 6 % diesel in comparison to the manual ZF-Ecosplit transmission.

Traxon is based on a new basic transmission with twelve or 16 speeds. For its development efficiency-increasing factors were essential, such as a high spread of gear ratios, a good power-to-weight ratio, and the combination with the anticipatory shifting strategy Prevision GPS [2], as well as the noise quality and the suitability for increased input torques. Traxon covers the market demand for versatile transmissions with a modular concept: The basic transmission can be enhanced according to the specific application. This has enabled ZF Friedrichshafen AG, in particular, to design a dual-clutch module also for heavy long-distance haulage trucks for the first time. Along with Traxon and under the name Traxon Dual, this module has, in particular, made possible new axle ratios and additional reductions in fuel consumption.

OEMs have already combined the advances achieved based on automatic truck transmissions for some time with high-geared driven axles: If a gear ratio of about i = 2.85 was still common for these in 2008, it has declined to i approx. 2.53 in the current volume production truck. This trend will continue into the future. It could be accelerated, for example, by EU projects for the statutory regulation of CO_2 emissions in commercial vehicles as well as by potentially rising fuel costs. Up until now, available systems have, however, reached their technical limits. Once again high-geared axles have been able to additionally reduce fuel consumption by significantly reducing engine speed – the keyword here is downspeeding. At the same time, they still only leave a very narrow speed window available for the engine torque required for propulsion, **FIGURE 1**.

According to the results of ZF-internal analyses, extremely high-geared axle ratios of i approx. 2.2 as well as the associated, lower torque reserves increase the shifting frequency by a factor of 2.5. This would lead to restrictions in performance in conjunction with current volume production commercial vehicle transmission systems – whether manual or automatic – since in most cases a gear change also interrupts the tractive force or acceleration respectively. However, the associated losses in comfort have a particularly negative effect: Significantly more gear changes consequently entail more frequent, clearly noticeable load alteration effects in the cabin as well as in the driveline. Against this background, the market acceptance of heavy trucks with strong downspeeding measures would be extremely questionable despite the high potential savings in terms of fuel. ZF's response was the development of the Traxon Dual: A fundamental input end requirement for extremely high-geared axle ratios and downspeeding engine concepts in heavy long-distance haulage trucks with a torque of up to 2500 Nm.

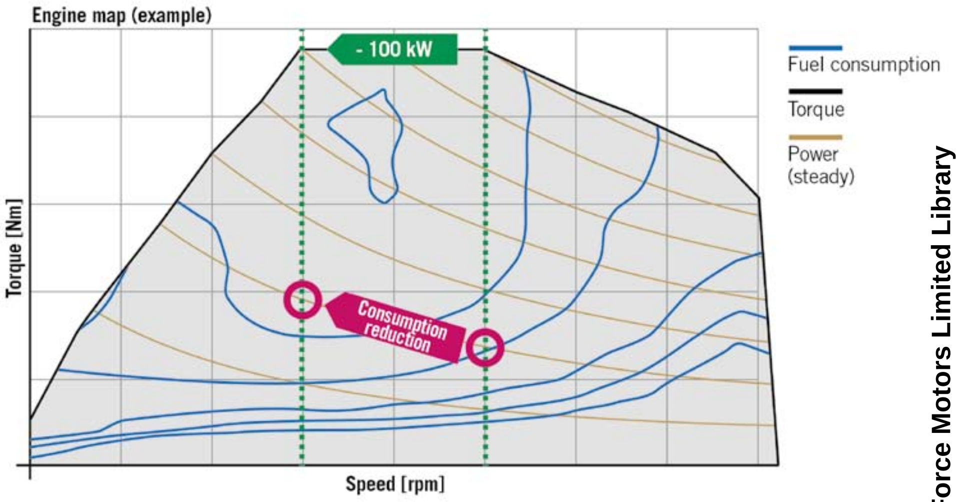

FIGURE 1 If the engine speed is lowered by up to 400 rpm by means of very high-geared axle ratios (green line), this leads to less performance and in particular torque reserves (black line) – even on very slight uphill gradients, the automatic transmission must downshift as a result in order to maintain the current driving speed (schematic view)

TECHNICAL PRINCIPLES OF THE TRUCK DUAL CLUTCH TRANSMISSION

The modular concept of Traxon made possible this dual clutch variant with a very high number of identical parts (80 %). Consequently, this was relatively simply and required only a short development time. It is worth noting that the increase in length and weight with the Traxon Dual, respectively 155 mm and approximately 80 kg, essentially due to the second clutch, was moderate in comparison with the basic transmission. Its advantages of light and compact design could therefore be largely retained, despite the increase of functions.

The constructive modifications to the basic transmission were restricted to a minimum. First of all, owing to the principles involved, Traxon Dual required a second transmission input shaft. This is designed as hollow shaft and drives – in conjunction with the second clutch – the 2nd gear of the three-part splitter group, which is positioned in front of the main transmission, **FIGURE 2**. The solid shaft, however, acts on the gears 1 and 3 of the splitter group via the first clutch. Especially for the dual clutch application, ZF therefore varied the arrangement of the Traxon gearsets: The split up of the number of gears between splitter group, main transmission, and range change group is now resolved with 3-2-2 instead of with 2-3-2 as was previously the case. In addition, ZF adapted individual ratios to the new dual clutch function and its specific fields of application.

The dual-clutch module of Traxon Dual consists of two separate dry clutches according to SAE standard, arranged one behind the other, **FIGURE 3**. As previously described, one is intended for the input-side hollow shaft and one is intended for the solid shaft. The torsional damper integrated in the disks reduces torsional vibrations that increasingly occur especially as a result of downspeeding and downsizing measures. The clutch release mechanism called Dual Conact newly constructed for Traxon Dual actuates both clutches: It is based on two concentrically arranged, pneumatic release cyl-

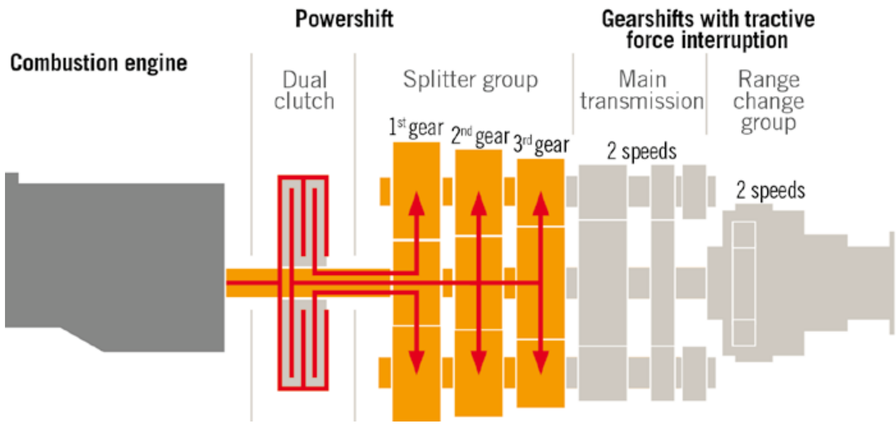

FIGURE 2 Shift concept with dual clutch:
Traxon basic transmission with splitter group, main transmission, and range change group

inders, whose activation is regulated by an electronically controlled solenoid valve for each one. As a result, the control unit of the dual clutch system plays an important role. For Traxon Dual, ZF was able to play to the advantages of the model-based software development, for which an existing modular platform could be used. First of all, the electronics include a specific dual clutch function. During its design, focus was put on Group-internal synergy effects: Each of the two Conact release cylinders is controlled so that the transition between disengaging one clutch (with the current gear) and engaging the other (with the gear held ready) takes place smoothly. The Traxon Dual control unit also features a gearshift strategy specially designed for engines with narrow speed range.

ZF's primary objective was to minimise fuel consumption and it therefore realised the gear changes without tractive force interruption exclusively for the top three gears (10-11-12) in the first development stage of the Traxon Dual. This range has the greatest potential for increasing efficiency since 90 % of all gear changes in long-distance traffic take place there. In a subsequent step, the project team also expanded the gear changes under load – primarily with a view to greater comfort – to the lower gears. That was possible through mere adjustments to the control unit software. As a result, Traxon Dual now opens both clutches simultaneously for the three gear changes between gears 3 and 4, 6 and 7, as well as 9 and 10, **FIGURE 2**.

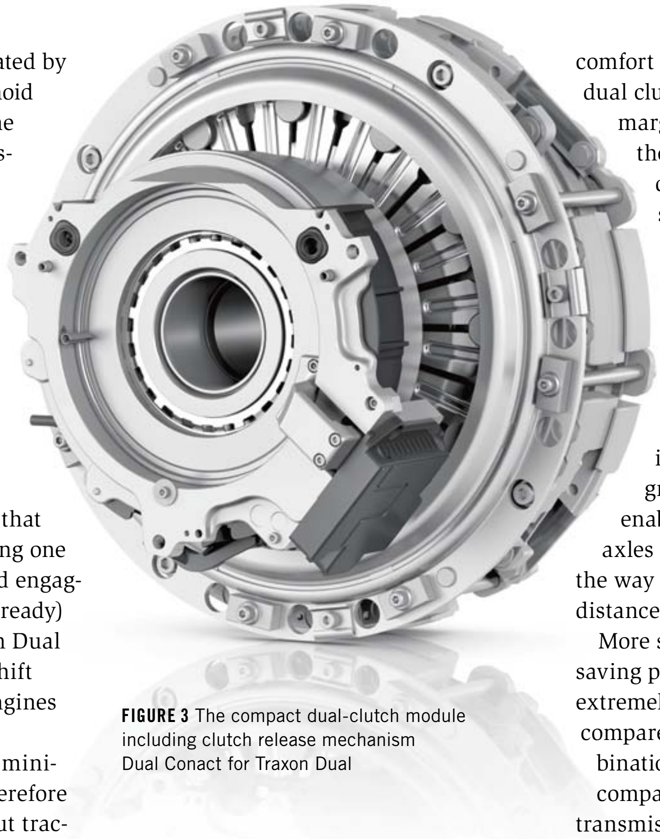

FIGURE 3 The compact dual-clutch module including clutch release mechanism Dual Conact for Traxon Dual

PRACTICAL BENEFITS

The principle assets of the Traxon Dual include one aspect in particular: For the first time and with relatively little additional expense, heavy trucks can also benefit from automated gear changes under load that are hardly noticeable any more while driving. Furthermore, the dual clutch system therefore solves the central problem of acceptance of extremely high-geared axles (i approx. 2.2): As soon as gear changes take place unnoticed, their increased frequency is also irrelevant. As a result, Traxon Dual enables advancements on four levels: Firstly, it increases comfort since the smooth merging of the dual clutch gear changes only trigger marginal cabin movements, and, at the same time, go easy on the driveline components as well as sensitive freight. Secondly, it improves the acceleration or propulsive power: Despite the gear change, driving speed and engine speed do not drop significantly. In a long-distance haulage truck that is advantageous above all when upshifting and downshifting on uphill gradients. Thirdly, Traxon Dual enables the installation of very long axles and fourthly, it therefore paves the way for further fuel savings in long-distance traffic.

More specifically, Traxon Dual's fuel-saving potential in combination with extremely high axle ratios comes to 3 % compared with transmission/axle combinations common today, **FIGURE 4**. In comparison with the Traxon basic transmission, which already enables "smoother" downspeeding concepts thanks to its high spread of gear ratios of 16.69 (twelve-speed version), it produces a reduced consumption of 1.5 %.

Moreover, for specific applications in commercial vehicle transport there is one essential aspect: Thanks to the overall modular concept, Traxon Dual can also be combined with those additional functions that are available for the Traxon basic transmission. That includes, for example, Prevision GPS, the anticipatory shifting strategy [2]: By knowing the route in advance, the dual clutch system can already hold the respective correct gear ready, proactively instead of reactively: Perhaps a lower gear when the

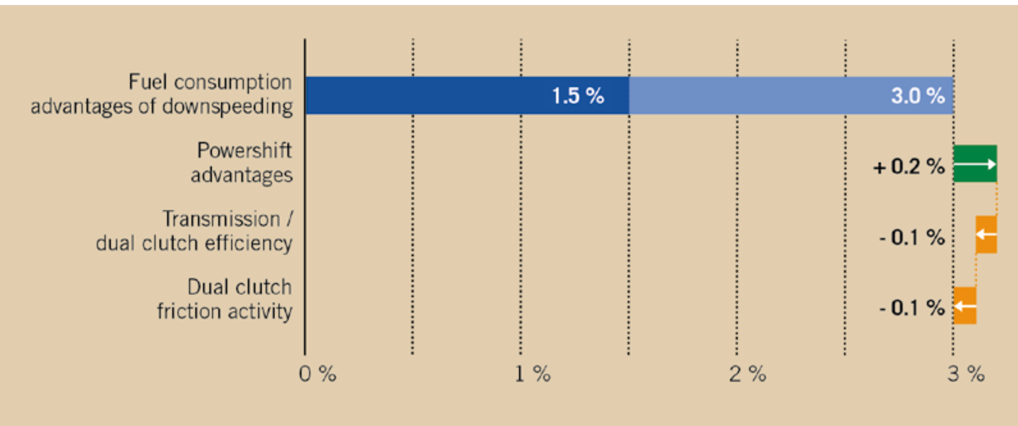

FIGURE 4 Based on the Traxon basic transmission with high spread of gear ratios (dark blue), Traxon Dual enables very high-geared truck axle ratios – and in conjunction with this, a reduction in fuel consumption of altogether 3 %; potential overall transmission losses are compensated caused by the supplemented dual-clutch module for example in terms of friction and transmission efficiency (orange) through the gains in efficiency (green) that arise from the dual clutch function in the daily transport routine

FIGURE 5 Test vehicle from ZF with the Traxon Dual dual clutch transmission system and very high-geared axle ratio being tested in practice

truck approaches a high uphill gradient. In addition, the wear-free transmission brake ZF-Intarder can be integrated in Traxon Dual.

CONCLUSION AND PROSPECTS

With the Traxon Dual dual clutch transmission based on the modular Traxon transmission system for heavy long-distance haulage trucks, ZF provides the prerequisite for OEMs to launch heavy long-distance haulage trucks with extremely high-geared axles (i approx. 2.2) onto the market. The latter ensure robust downspeeding meaning the engine speed is reduced by up to 400 rpm on average. As a result of this, fuel consumption can be reduced by altogether 3 %. Traxon Dual, with its smooth gear changes and no tractive force interruption, enables significantly more comfort and dynamics in addition to these gains in efficiency. At the moment, the dual clutch transmission Traxon Dual is tested in practice, **FIGURE 5**. Investigations are currently being carried out with OEMs as to whether further potential could possibly be found in the dual clutch for the overall truck driveline system – for example in the form of less complex truck engines or charging concepts. In order to exploit all possibilities of this approach, ZF also works directly and very closely together with engine manufacturers as well as axle suppliers. In general, there is a great market interest in Traxon Dual. To meet this demand, ZF is striving to start volume production of the innovative transmission in 2018, this, dependent on specific vehicle projects.

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Optimisation of Crankcase Ventilation for Large Diesel and Gas Engines

IFT has developed a new type of oil separator for crankcase ventilation in high-speed and medium-speed gas and diesel engines. Important development parameters were a high degree of separation, low space requirements, high reliability and low production and maintenance costs.

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PROJECT TARGET

The treatment of the blow-by gas as part of the crankcase ventilation system of large engines usually is not in the focus of important R&D programmes. Nevertheless the impact of a crankcase ventilation system that does not work as efficient as necessary on the profitability of engine operation is often underestimated. Well know problems are either a short service lifetime of the oil mist separator or a low oil separation efficiency which leads to contamination of the engine intake system resulting in reduced power output and engine efficiency. Increased service costs will be the result in both cases.

State-of-the-art filter systems are often based on relatively thin filter mattresses which might be pleated to increase the surface and come in a multi-stage arrangement. With a filter service lifetime usually far below 10,000 h, the filters are designed as inserts in a filter housing which can easily be opened to allow for a quick change. In this case the necessary gaskets can create a problem themselves.

The increase of filter volumes is an established way to improve the separation capability and service lifetime of oil mist separators, combined with pre-separators in the case of high blow-by oil loads. Both solutions will increase the manufacturing cost of the engine and the service expenses for the operator.

Based on the assessment of the actual situation IFT Innovative Filter Technology GmbH started a R&D project based on scientific research and targeted on improving the properties of oil mist separators. The parameters of the project are derived from the requirements of today's supercharged engines, its service intervals and total lifetime:

- oil mist separation rate > 99.98 % or oil concentration of filtered blow by gas < 0.5 mg/m³
- service-free oil mist separator operating time > 20,000 h
- minimised mounting space requirements (specific volume approximately 30 % lower than state-of-the-art)
- reduced manufacturing cost (approximately 20 % lower than today)
- simple and robust design, high reliability of operation.

When comparing these targets with the properties of actual products it quickly became clear that the only way to achieve them is a completely new approach to oil mist separation.

BASIC RESEARCH AND MEASURING TECHNOLOGY

A challenge by itself is the technology of how to measure the oil content of the engine blow-by gas in an exact and reproducible way. This is especially true for the filter outlet gas flow with its extremely low content of oil.

The larger part of basic filter testing is done on a simulation unit which is purpose-built for this project. The reasons are the ability to vary test parameters in a wide range and to limit the test costs. The core part of the test bench is an oil mist generator which is capable of producing a well-defined oil mist by using a two-component injector. By passing the gas flow through a pre-filter it is possible to adjust mainly the droplet size to the desired testing conditions. The test gas flow then can be divided to supply up to five different test objects. This allows performing test runs on different equipment in parallel. After passing through the test objects the gas flow is treated in a fine filter. Then the clean carrying air is compressed in a blower and recirculated to the oil mist generator.

The simulation unit is equipped with an automatic data collection system which continuously registers parameters like volume flow, gas temperature and pressure drop over the test object. Other parameters like the oil content and the droplet size distribution are measured in an intermittent way only due to the very complex measuring process. **FIGURE 1** shows the oil separator test unit.

To determine the oil droplet size distribution a cascade impactor provided by the Technical University of Graz, Austria, is utilised, which is designed to detect three different droplet sizes. The measurement of the total oil mass content is realised by very fine filters which completely segregate the oil mist from the carrying air. The weight increase of the filters due to the accumulated oil is measured with precision scales and related to the measuring time and volume flow. Because of the extremely low oil content of the blow-by flow that comes out of the filter the measuring time needed for accurate results is beyond 50 h.

FIGURE 1 Simulation test bench for separator testing

In addition to the work done on the simulation unit blow-by composition measurements are also taken from field engines. A comparison of results shows that the oil droplet size distribution differs only slightly between real engine operation of different size engines and the simulated blow-by on the in-house test unit. This allows IFT to assume that the translation of the simulation results to real engine operation is within acceptable limits. During field test campaigns the blow-by oil content is analysed by optical means. The so-called Dusttrak unit of TSI Inc. has shown good results.

MATERIAL PROPERTIES

Based on theoretical research IFT has chosen a two-stage depth filtration concept which promises best operational results. The filter packages are compressed at a right angle to the flow direction while the fibers mostly are arranged in parallel to the gas flow. These particular properties, among others, are protected by patents.

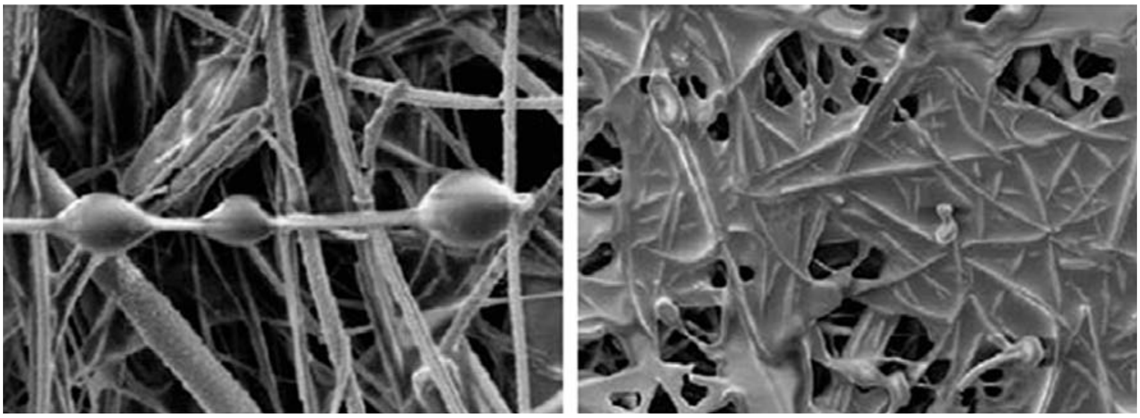
In the next step different fiber and fleece configurations were investigated based on the principal design parameters to determine their relevant technical characteristics. Among others, they consist of the specific pressure drop, the separation capability for finely dispersed oil droplets, the saturation and drainage behaviour, the mechanical and chemical stability and the elasticity of the material under influence of oil mist and engine operation. The materials investigated comprise two basic types of fibers, the first one (type-1) consisting of approximately 90 % SiO_2 with CaO and ZrO, the second one (type-2) consisting of approximately 40 % CaO, 20 % MgO together with ZrO_2 und SiO_2. Variation parameters are the fiber thickness and the compression of the test objects. While the type-1 fiber is showing very good characteristics for being used as the entry stage of the filter, the type-2 fiber is especially suitable for the following fine filter stages.

The following step was intended to examine more deeply the long time operational behaviour of the different materials. The most interesting aspect is the initial build-up of oil within the fabric and the ensuing stationary characteristics of accumulation and discharge of the oil.

Within a new filter the process starts with accumulation of oil in the fiber fabric. This reduces the free flow area and leads to an increased pressure drop. At a certain point of oil load (hold-up) the oil which is under influence of gravity starts to flow in a downward direction. Another important aspect which has to be taken into account is the different load condition in blow-by flow direction. After start-up the separator will finally reach a balanced operating condition which is a function of fiber material, oil load and oil properties. The oil load under stationary conditions determines the pressure drop and the separation behaviour of the fiber media. The REM photos in **FIGURE 2** show the fiber condition with different oil loads.

Another good indication of the oil saturation status is the comparison of the filter weight between new and stationary operating condition. Pressure drop

FIGURE 2 REM photo of fiber fabric with different oil load



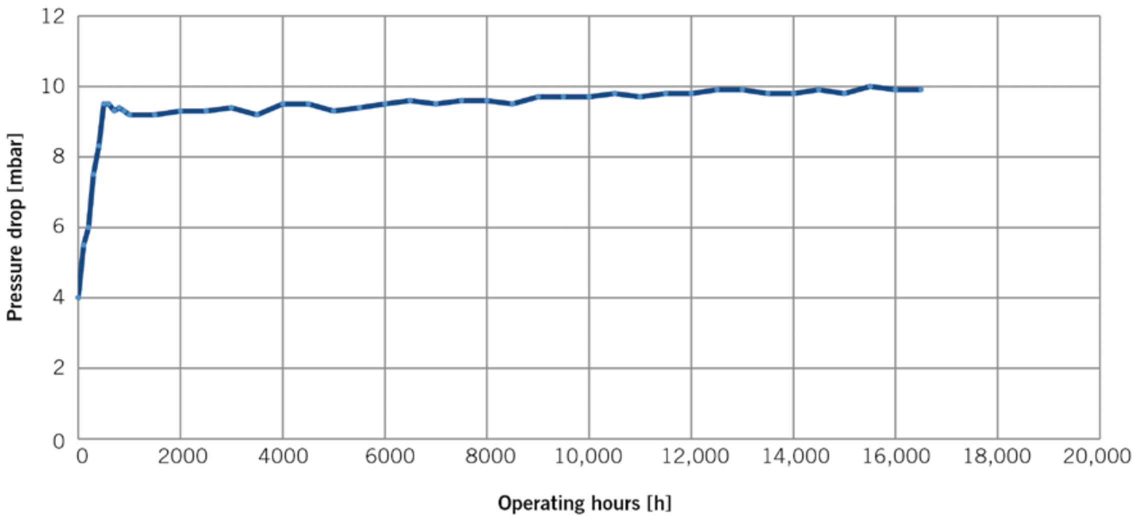


FIGURE 3 Pressure drop trend analysis of test separator over more than 16,000 operating hours

and separator weight are regularly monitored and recorded on the in-house test bench. **FIGURE 3** shows the development of the test filter pressure drop over an operating time of more than 16,000 h.

As a means to check the separation function and the material condition of the separator after a long operating period of 15,000 h, a test separator which in principle was still operational has been dismantled and diagnosed in detail. In principle, the results are as expected. On one hand the oil load is continuously reduced from the inlet to the outlet side. On the other hand there is also the predicted increase in oil load from the top to the bottom of the fiber package. In this case, the fabric close to the bottom of the separator is completely saturated with oil and there is no more blow-by gas flow in this area.

When taking a closer look one can observe an inhomogeneity in the oil load, **FIGURE 4**. There is a formation of "arteries" or "channels", which are locations with high oil loads where the oil is discharged down to the filter bottom. The formation of this structure seems to be typical for fleece layers with fibers oriented in parallel with the main gas flow. It could be observed on all dismantled separators so far and develops already soon after the separator start-up.

All filters checked where at an equilibrium condition. This means that the amount of oil separated from the blow-by flow is the same as the amount discharged back to the oil reservoir.

SEPARATOR HOUSING DESIGN

In addition to the efficient separation of the blow-by oil content by the fiber fabric the other important feature of a long-life separator is the reliable sealing of the fleece package along the housing walls. Based on the separator service life target of 20,000 h the housing is realised in a tightly welded design. After reaching the end of the service interval, the complete separator will be exchanged with a new one. The old part can be disassembled and partially recycled or disposed of in a proper manner as required by local regulations. Apart from offering a low-cost solution this design is intended to avoid any internal or external sealing issues.

The laterally compressed fleece layers are properly inserted into the housing with the required pre-stressing. This preload assures that the fiber material rests against the housing wall in a permanently tight way. But another type of sealing is necessary in the top and bottom areas of the body.

Based on a number of test series IFT has decided to glue the fiber packages to the top and bottom wall of the separator. This bonding is realised with a liquid sealing compound which flows into the fiber package for about 10 mm and is allowed to solidify there. After that step the top and bottom parts of the housing are permanently welded to its prismatic middle section. Choosing the sealant and performing the sealing operation have to be done with extraordinary care. **FIGURE 5** shows the suboptimal result of one sealing test in the first row and the target configuration in the second row.

Apart from the basic functional criteria the housing design can be widely adjusted to the customer requirements. This allows meeting the connection and

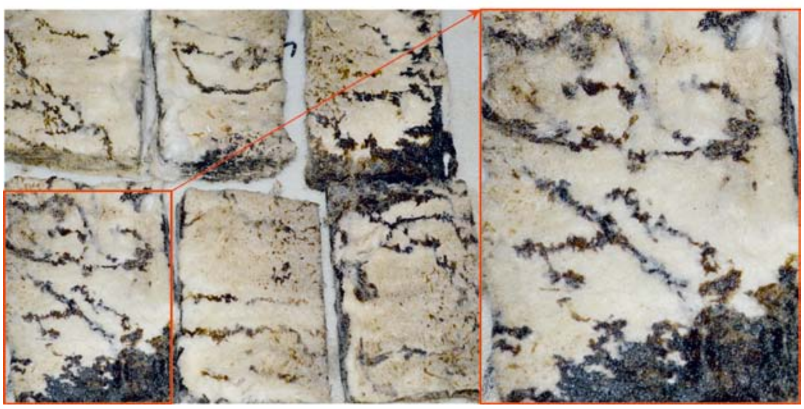


FIGURE 4 Oil saturation behaviour and formation of oil discharge „channels“ in fleece fabric

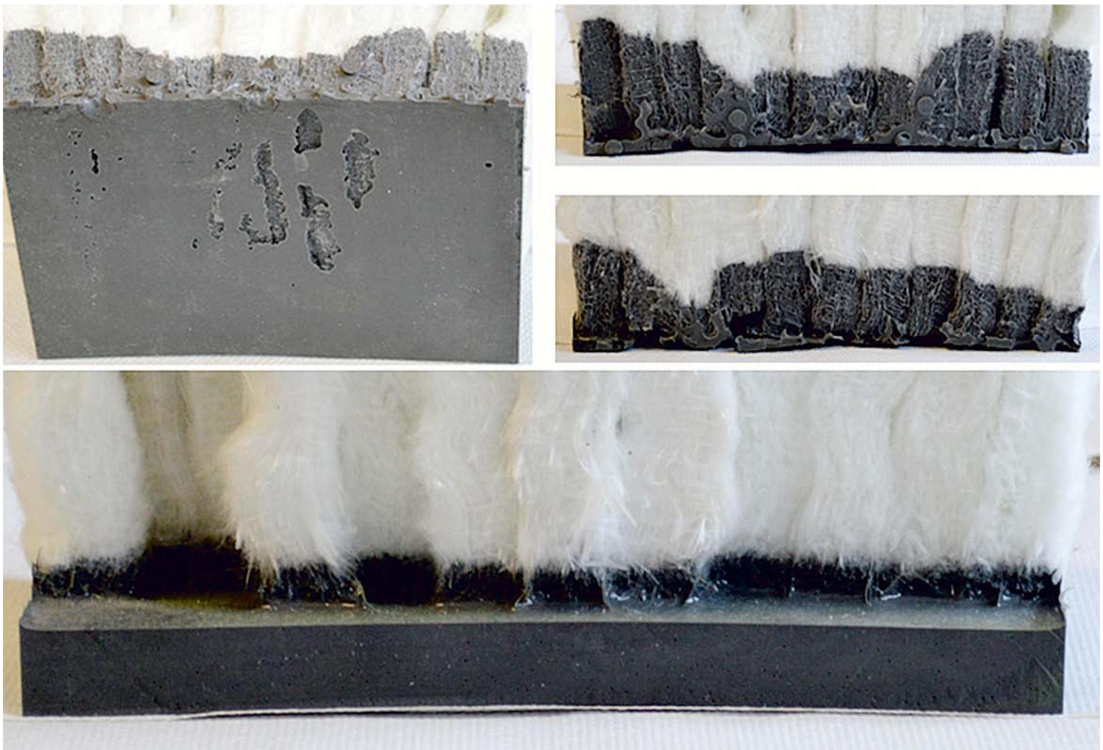


FIGURE 5 Examples of bonding tests

Test engine	Power output class	Start-up of filter	Operating time 02/15 (hours)
Natural gas engine	300 kW	From 02/2013	Approx. 16,000
Biogas engine	600 kW	From 08/2013	Approx. 12,000
Natural gas engine	600 kW	From 08/2013	Approx. 6000
Diesel engine	1000 kW	10/2013	Approx. 7000
Natural gas engine	1000 kW	From 02/2014	Approx. 6000
Natural gas engine	1500 kW	From 08/2013	Approx. 8000
Natural gas engine	3000 kW	From 01/2014	Approx. 7000

TABLE 1 Status of field testing with prototype filters

mounting specification of the engine and the application respectively.

FIELD TESTING

In addition to intensive short-term and durability testing on the in-house simulation test bench there are numerous prototypes which are deployed on different field engines. IFT carefully chooses these field test engines to cover a broad spectrum of power output, fuel type and engine application.

When it comes to assuring a reliable operation of the crankcase ventilation system, it is not only the blow-by oil separator but also the integration into the complete engine system that has to be taken into account. Important points are the blow-by extraction point in the crankcase, the re-circulation of the cleaned gas flow into the engine intake system, the available pressure difference between extraction and re-circulation and the sound and safe return of the separated oil into the oil pan. Additionally one must observe the ambient temperature and the vibration level at the mounting position. IFT offers a consultancy service and support for the dimensioning and design of the complete crankcase ventilation system. **TABLE 1** gives an overview about IFT filter on the field, which have reached a multiple of the lifetime of competitor products.

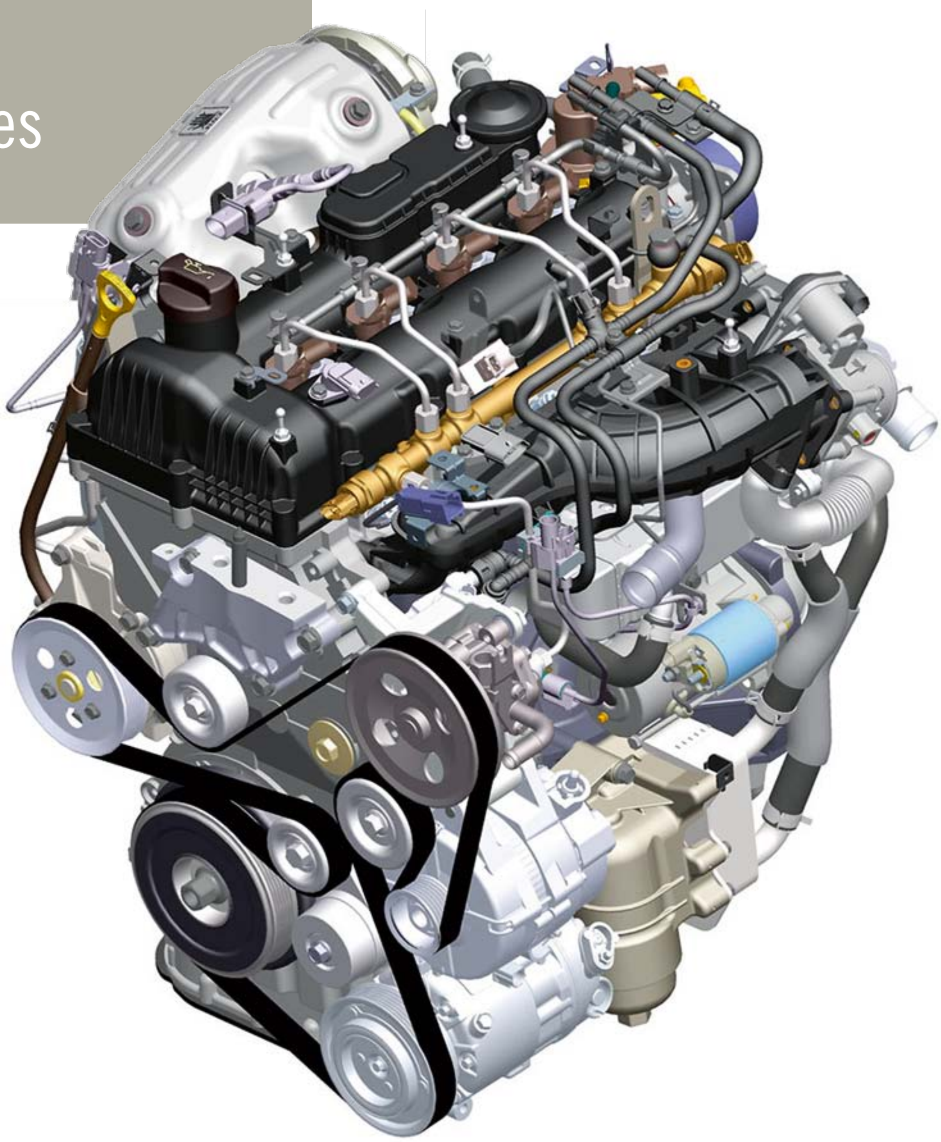
The field tests and real conditions confirmed the forecast of the simulation and the knowledge of the pilot plant. The unlimited series release was given in 2014, based on the throughout positive experience.

SUMMARY

This article is presenting a new oil mist separator that has been developed for the crankcase ventilation system of high-speed and medium-speed gas and diesel engines. The R&D project consists of comprehensive research work to determine the properties of the blow-by gas and to optimise the choice of material and the design parameters of the separator. In addition to the theoretical analysis and in-house testing on a purpose-build simulation unit the results are verified by numerous prototypes continuously running on field engine of different size and application. Test separators are now in operation at important engine manufacturers.

Model-based Methods for the Calibration of Modern Internal Combustion Engines

Design of Experiments and model-based parameter optimisation are the keys to mastering complex engine management systems. In the following report, Hyundai and Etas show how model-based development methods can sensibly support the calibration of modern internal combustion engines.



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INCREASE IN THE CALIBRATION PARAMETERS

With CO_2 and exhaust gas emissions limits getting tougher all the time, engine management systems are becoming increasingly complex in response. The result is a constant increase in the calibration parameters – such as characteristic curves and engine maps – that need to be optimised in the overall system. At the same time, strong competition is forcing manufacturers to shorten development cycles and cut development costs. To be able to carry out engine calibrations that ensure maximum ride comfort, high dynamics, and low emissions under these circumstances, there is a need for new computer-assisted calibration methods to complement conventional ones [1].

Engineers at the Hyundai Motor Europe Technical Center GmbH (HMETC) in Rüsselsheim, Germany were quick to recognise this need: in powertrain development, they have been making greater use of Design of Experiment (DoE) and model-based optimisation methods on top of increased automation levels since 2005. Acceptance of the initial solutions was severely hampered by their lack of user-friendliness and the fact that they did not cover all engine development process steps.

However, the introduction of the Etas Ascmo [2] software resolved this situation: in addition to a program structure and user interface tailored to model-based ECU calibration, the software provides helpful functions to support inexperienced users. Examples include the cleverly structured "ExpeDes" test plan editor, automatic model generation, and on-the-fly extrapolation of target variables for any driving cycle.

As an example, the following sections describe the use of this new solution in a preproduction engine project at the HMETC Powertrain Division. The description focuses on new methodical approaches and benefits.

PROJECT SCENARIO

The test candidate was a 2.0-l, four-cylinder diesel engine with preproduction engine hardware and ECU software, **COVER FIGURE**. Fuel was supplied to the engine by means of a 1600 bar common-rail system with solenoid valve injectors. Fresh air was directed to the engine intake via a variable nozzle turbocharger (VNT) with downstream intercooler. In addition to a cooled exhaust gas return (EGR) line, a similarly cooled low-pressure EGR loop was installed. The engine delivered a maximum of 110 kW at 4000 rpm.

At the beginning of the tests, the existing calibration already complied with the Euro 5 emissions standard. The objective was to use the DoE software to further reduce the engine's fuel consumption. To do this, it was important to find the optimal balance for the following calibration parameters:

- air mass/EGR rate (mAir)
- start of injection (SOI)
- swirl flap position (swirl)
- exhaust back pressure flap position for low-pressure EGR control (LP_ExFl)
- boost pressure (p_{Boost})
- rail pressure (p_{Rail}).

The relevant target variables are listed below:

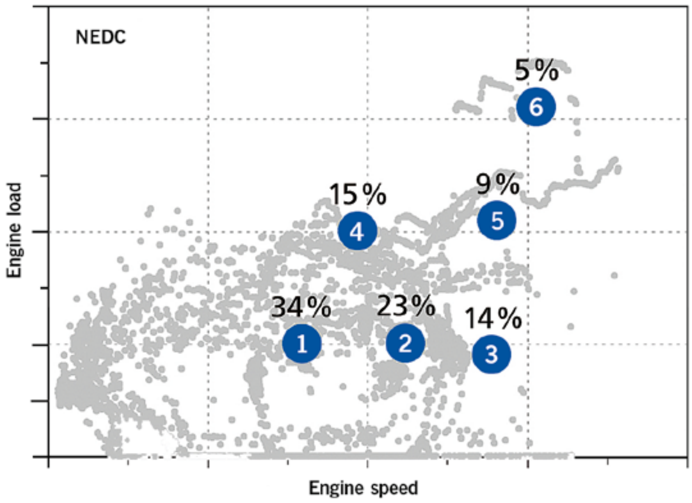
- fuel consumption (CO_2)
- particulate mass (soot)
- nitrous oxides (NO_x)
- hydrocarbons (HC)
- carbon monoxide (CO)
- combustion acoustics (dBA).

All tests were conducted on the engine test bench, with subsequent in-vehicle verification on the emissions chassis dynamometer.

During the basic measurement run, the test vehicle, a Hyundai ix35, attained the Euro 5 limit values in the New European Driving Cycle (NEDC). This provided the basis for determining the CO_2 value to be used as a reference for the optimisation. As shown in **FIGURE 1**, the relevant operating points for optimisation on the engine test bench were supplied by the dwell times of rpm and load in the NEDC.

PLANNING DATA ACQUISITION ON THE TEST BENCH

The DoE module used for test planning divides the workflow into eight user-friendly steps that can be processed sequentially or iteratively. A useful function facilitates the compression, and therefore optimum distribution, of measuring points via selected input variables, **FIGURE 2**. In the case at hand, the measuring points were compressed in the vicinity of small air masses, because in addition to greater measuring inaccuracy, less smooth physical dependency was also expected in this area due to high EGR rates.



No.	Engine speed [rpm]	Load [Nm]
1	1545	32
2	1875	30
3	2140	23
4	1720	90
5	2155	100
6	2280	154

FIGURE 1 Distribution and weighting of operating points in the NEDC test

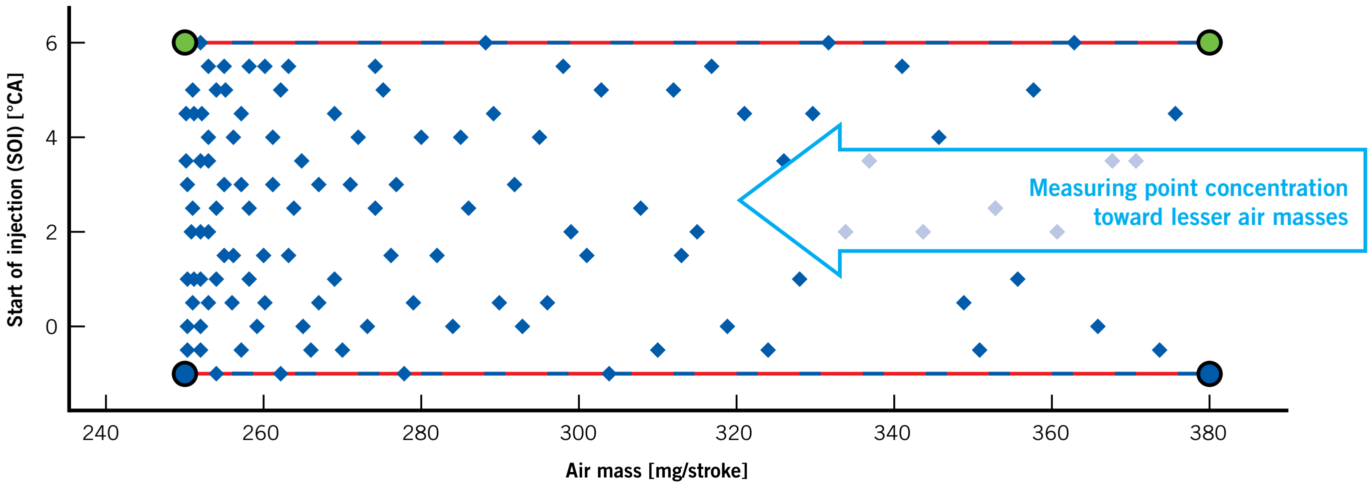


FIGURE 2 Experiment planning using local measuring point concentration

Another function allows users to divide the test plan into a variable number of sections ("blocks"). Given a sufficient number of measuring points, each individual block offers optimum distribution for modelling. Furthermore, all blocks are complementary, meaning that no overlaps occur when they are combined. During live measurement on the test bench, it is therefore possible to quickly determine after each block has been run whether the requisite model quality has been achieved and the test run can be completed early. This can significantly reduce the amount of time and effort required for measuring. In this case, seven blocks of equal size were formed. As an example, **FIGURE 3** plots modelling accuracy for the smoke number as a function of the number of measuring points used for model generation.

KEY ELEMENT: RAW DATA ANALYSIS

Once the measurement data has been gathered, the next phase is raw data analysis. This often proves to be the most important data evaluation step. As well as identifying faulty measurements and drifts, it also provides insight into optimisation potential. The DoE software supports this process very efficiently: interactive diagrams allow users to display calibration parameters and/or target variables in relation to each other and, for example, to isolate the areas in which target variables display their optimal values. Different isolated areas are highlighted in different colours and can be limited to the intersection of the selected points. This facilitates the effective visual evaluation of measurement data and the identification of good parameter combinations. In this sample project, the calibration data limited in this way served as the starting point for optimisation.

AUTOMATED MODELLING

The core of Etas Ascmo is its user-friendly modelling function, which is largely automated. Unlike the model-based calibration tools available on the market until now, users are not required to select a specific type of model from a large number of options. Instead, the tool suggests a single, particularly flexible and powerful model type for them based on Gaussian processes (GP). In GP models, specific optimisation methods are used to automatically determine, from a complete family of curves, the function that with the highest degree of probability correctly maps the system behaviour represented by the measurement data [3]. This approach makes it possible to model even highly nonlinear behaviour by very complex systems to a high degree of accuracy and without overfitting. To do this, users do not have to parameterise the model. For purposes of comparison and

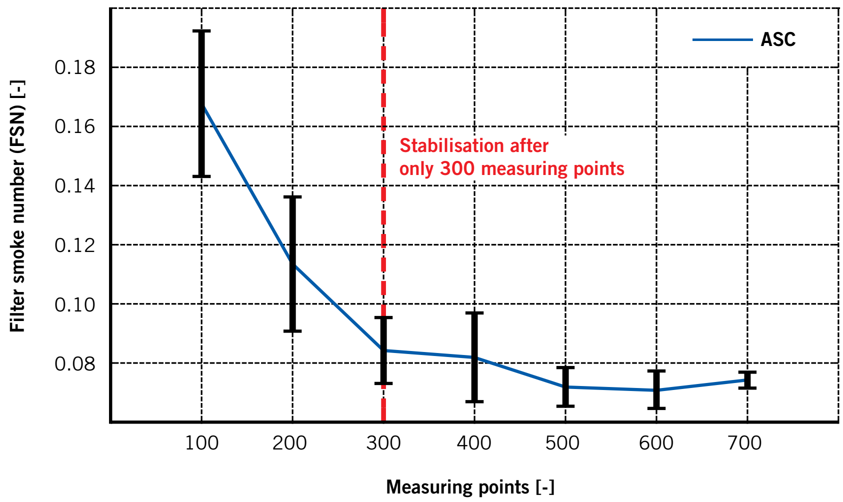


FIGURE 3 Model accuracy of Etas Ascmo model (ASC) versus data record size: mean error of global smoke number model (determined by means of verification measurements, error bar = standard deviation obtained with five repeat measurements)

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	sNO_x [g/kWh]	sCO [g/kWh]	Smoke (FSN) [-]	CO_2 [%]	Combustion noise (dbA)	be [g/kWh]
Model range	0.4-2.5	0.7-16	0-6	4-13	75-95	210-460
Root-mean-square error (RMSE)	0.058	0.57	0.089	0.059	0.23	5.32
R^2	0.97	0.98	0.97	0.99	0.98	0.99

TABLE 1 Quality of global model within the limits of definition range established by verification measurements

special cases, the full range of classic polynomial models is also available.

In comparison to the extrapolation behaviour of polynomials, which tend to strongly diverge outside the measured range, the extrapolation characteristics of the GP models are very lenient: within the extrapolation range, they slowly converge to the mean value of the measured data.

If users suspect that a target variable possesses a basic linear tendency, they can alternatively select a linear extrapolation behaviour. In this example, this permitted high-quality target variable forecasts in the extrapolation area, which was close to the model's definition range.

A critical issue for GP models is often the computing times and memory capacities required for processing large measuring ranges. However, the efficient GP implementation allows to generate models from tens of thousands of measuring points even on a standard PC in an acceptable time.

The high flexibility of the GP models also enables users to create global engine models with rpm and load as additional input variables. As a result, system behaviour can be forecast even at operating points that have not been measured. In order to assess the maximum attainable quality in our sample project, the measurement data of the six operating points was used to create a global model in addition to local models. In both instances, the quality of the models was satisfactory and the modelling of physical dependencies was largely correct. In some cases, the global model provided even better characteristics than its local counterparts. Only the modelling of CO emissions, with a value range of up to 16 g/kWh and a standard deviation of 0.58 g/kWh, remains somewhat too inaccurate. **TABLE 1** shows the statistical quality levels of the global models based on verification measurements.

OPTIMISATION RESULTS

While Etas Ascmo's range of functions for local optimisation is comparable with that of other commercial tools, its strength lies in its global modelling and evaluation capabilities, which enable it to automatically optimise entire engine maps with respect to drive cycles. This is accomplished by directly linking the characteristic maps for the calibration parameters with the global model. Then, based on a list of weighted operating points, a current cycle prognosis is calculated online for each change of the characteristic maps. This means that the effect of manual modification of the calibration data on the result of the driving cycle can be assessed quickly and easily. Above all, however, it means that a pow-

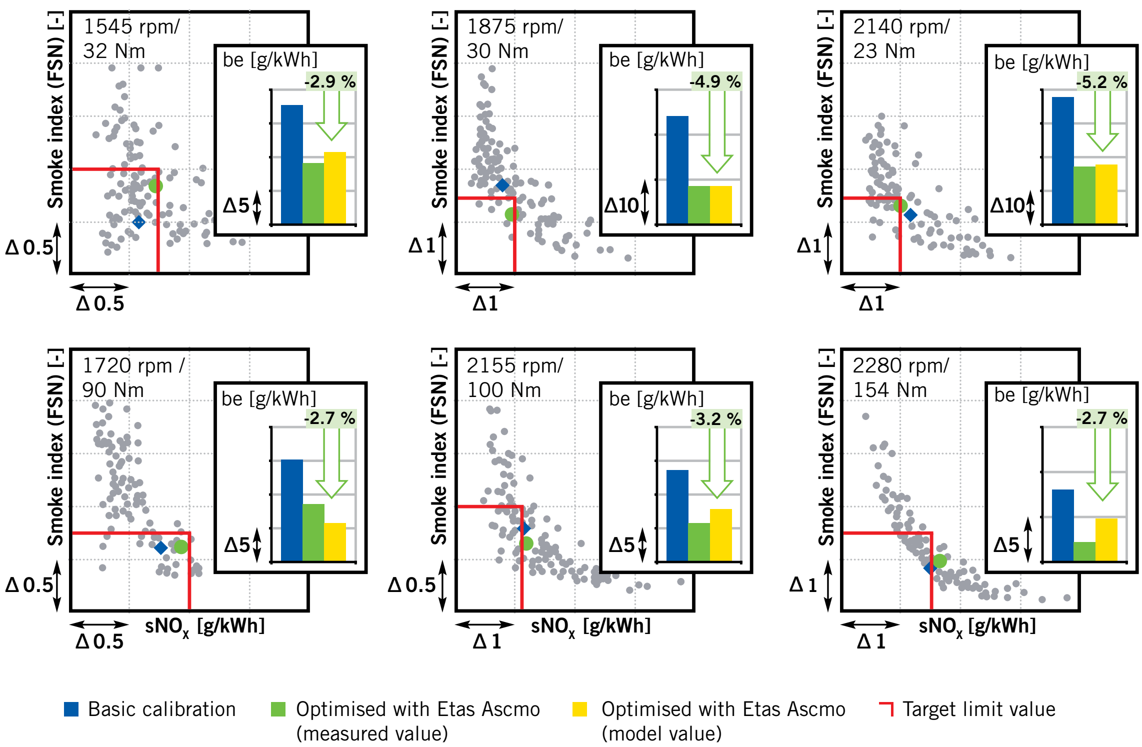
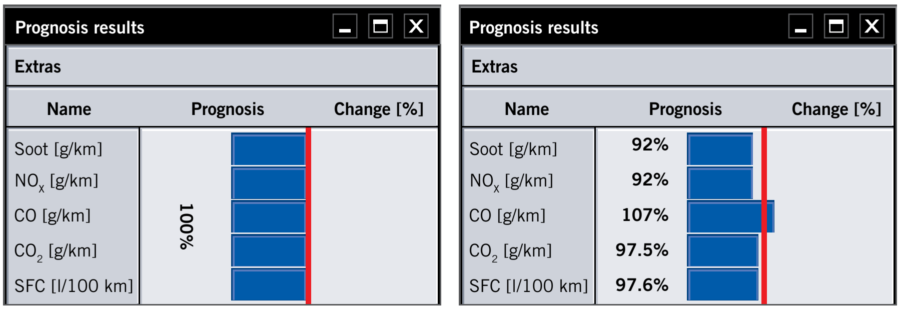


FIGURE 4 Optimisation results based on measurements taken at six operating points on the engine test bench

Pre-optimisation prognosis
Values adjusted as per results of dynamometer pre-testing

Post-optimisation prognosis
Forecast based on applied weightings and adjustment factors



	NO_x [g/km]	CO [g/km]	Soot [g/km]	CO_2 [g/km]	FC [l/100 km]
Deviation Etas Ascmo vs. in-vehicle measurement	-3.7 %	-4.0 %	-12.5 %	+0.2 %	+0.3 %
Tendency	+	+	-	++	++

FIGURE 5 Prognosis based on cycle extrapolations before and after optimisation (partly screenshot)

erful optimiser can be used to automatically generate calibration data, which achieves minimal fuel consumption while staying within the cycle's limit values and respecting local limit values and map smoothness. The optimisation results achieved in this way based on the analyses are summarised in **FIGURE 4**.

As in the software named Inca from Etas, the GUI can be toggled between a working page and a reference page, each with its own set of engine map data. Extending this to accommodate additional pages for managing alternative data sets would be a useful extra feature.

During verification on the dynamometer, the vehicle with optimised calibration achieved a 2.5 % reduction in fuel consumption compared to the base data, accompanied by slightly reduced smoke and NO_x emissions. When we consider that the base data version was mature to start with, we can see these increases for the impressive achievement they are. Moreover, the value measured is very close to the DoE model forecast. **FIGURE 5** shows the results of pre- and post-optimisation cycle extrapolations. Using a variety of factors, the calculations were adjusted so as to fully correlate the output values with the results of the basic, pre-optimisation measurements obtained on the dynamometer.

SUMMARY

Overall, the evaluation had a very positive outcome. Particularly in the area of engine calibration, the tool quickly achieved a high degree of acceptance among calibration engineers on account of its advanced task-centered functionality and its user-friendliness. Whereas many publications on model-based optimisation have tended to emphasise the time and cost savings it delivers, the focus for HMETC was more on the measurable increase in quality and the improved documentation of calibration results.

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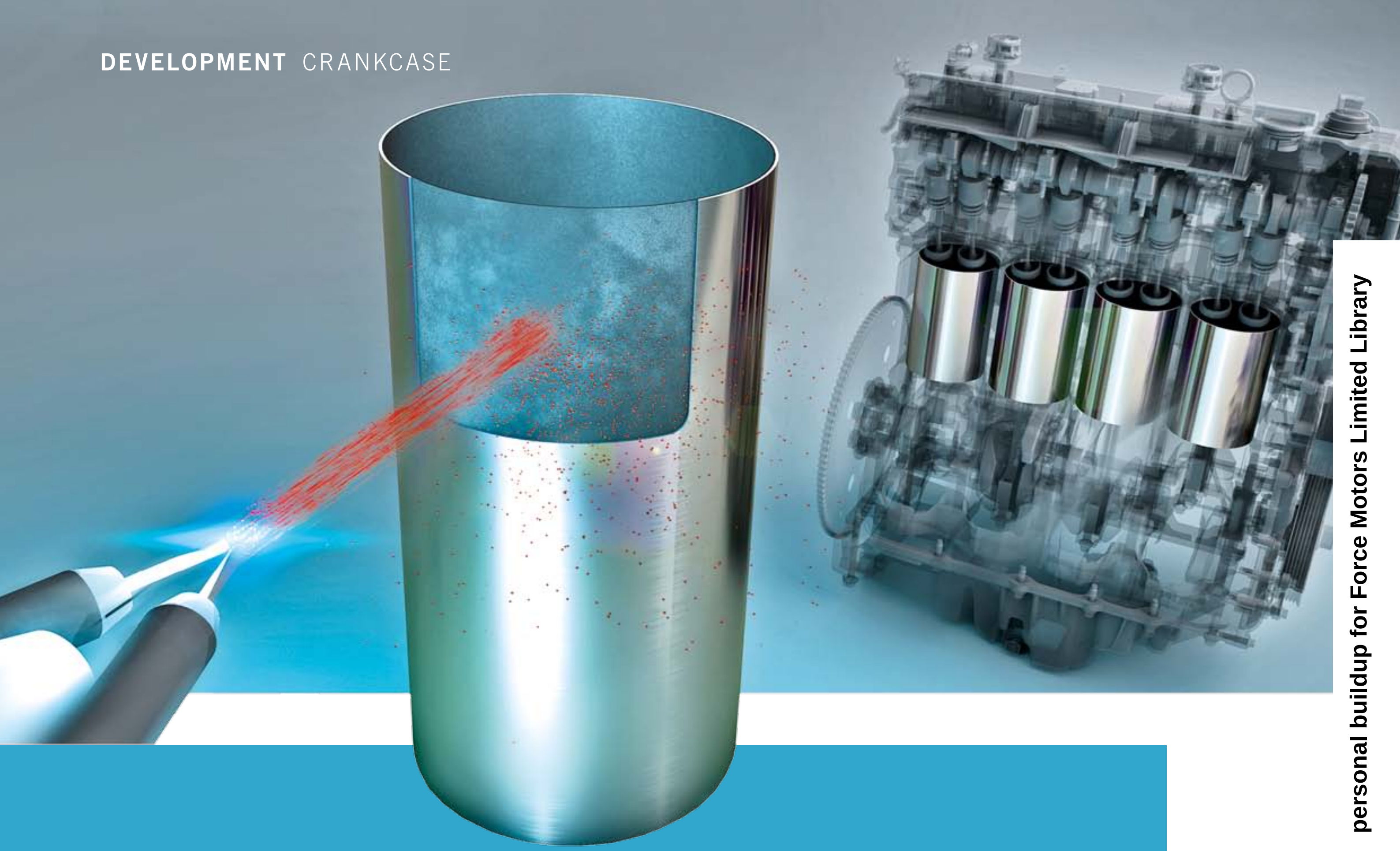

Sprayed Fe-Al Cylinder Liner with Optimised Thermal Conductivity

Federal-Mogul's thin-walled, thermally sprayed Sprayfit cylinder liners are a newly developed solution for providing a ferrous cylinder bore surface in aluminium alloy engine blocks. This liner design achieves, for the first time, heat transfer equal to that of a thermally sprayed direct bore coating.

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PROVEN IN MOTORCYCLE APPLICATIONS

Aluminium (Al) continues to gain in importance as a material for lightweight engine designs. To ensure the optimum tribology in an Al crankcase, the material combination used for the piston ring pack and the running surface of the cylinder typically includes either a cylinder liner of a suitable type, or a coating which is applied directly to the cylinder bores in the block. In the case of liners, homogeneous outer diameter (OD) machined grey cast iron liners with varying wall thickness concepts (cast-in or press-fitted) are just as widely used as OD machined and additionally coated liners (hybrid liners) with an outer layer of AlSi12 that serves to improve cohesion to the Al crankcase.

Federal-Mogul utilises thermal spraying to manufacture fully sprayed cylinder liners consisting of a Fe/Al composite. The ferrous running surface of this design is similar to a thermally sprayed direct bore coating and offers at least the same tribological benefits. In addition, the outer Al layer of the liner establishes a thermal conductivity that is almost equal to a direct bore coating. These properties are combined with the industry-wide fitting process for Slip Fit Liners, allowing the Sprayfit technology to offer an optimal mix of robustness and innovation by combining the advantages of an inserted liner with the functional properties of a direct bore coating.

The new Sprayfit liners are already proven in motorcycle racing and in road bike applications and have also proven their beneficial functional properties in gasoline and diesel PC engines during engine test bed trials. The first series applications are planned for the end of 2016.

SPRAYFIT CYLINDER LINER DESIGN

The newly developed Sprayfit cylinder liners are mechanically or thermally fitted in the engine block. As they have a wall thickness in the region of just 1.0 mm, they are ideally suited to lightweight engine designs that seek to reduce the ferrous content to the absolute minimum required for tribological purposes. Sprayfit liners consist of a carbon steel layer 0.4 to 0.6 mm thick, built up by arc wire spraying, and an Al layer 0.4 to 0.7 mm thick, sprayed directly onto the inner layer using the same procedure. After honing of the inner diameter, the final wall thickness is around 1.0 mm, with the remaining ferrous cylinder running surface now contributing only 0.1 to 0.3 mm to the total. Sprayfit liners can be manufactured with bore diameters of between 20 and 300 mm. **FIGURE 1** provides an overview of common running surface technologies for

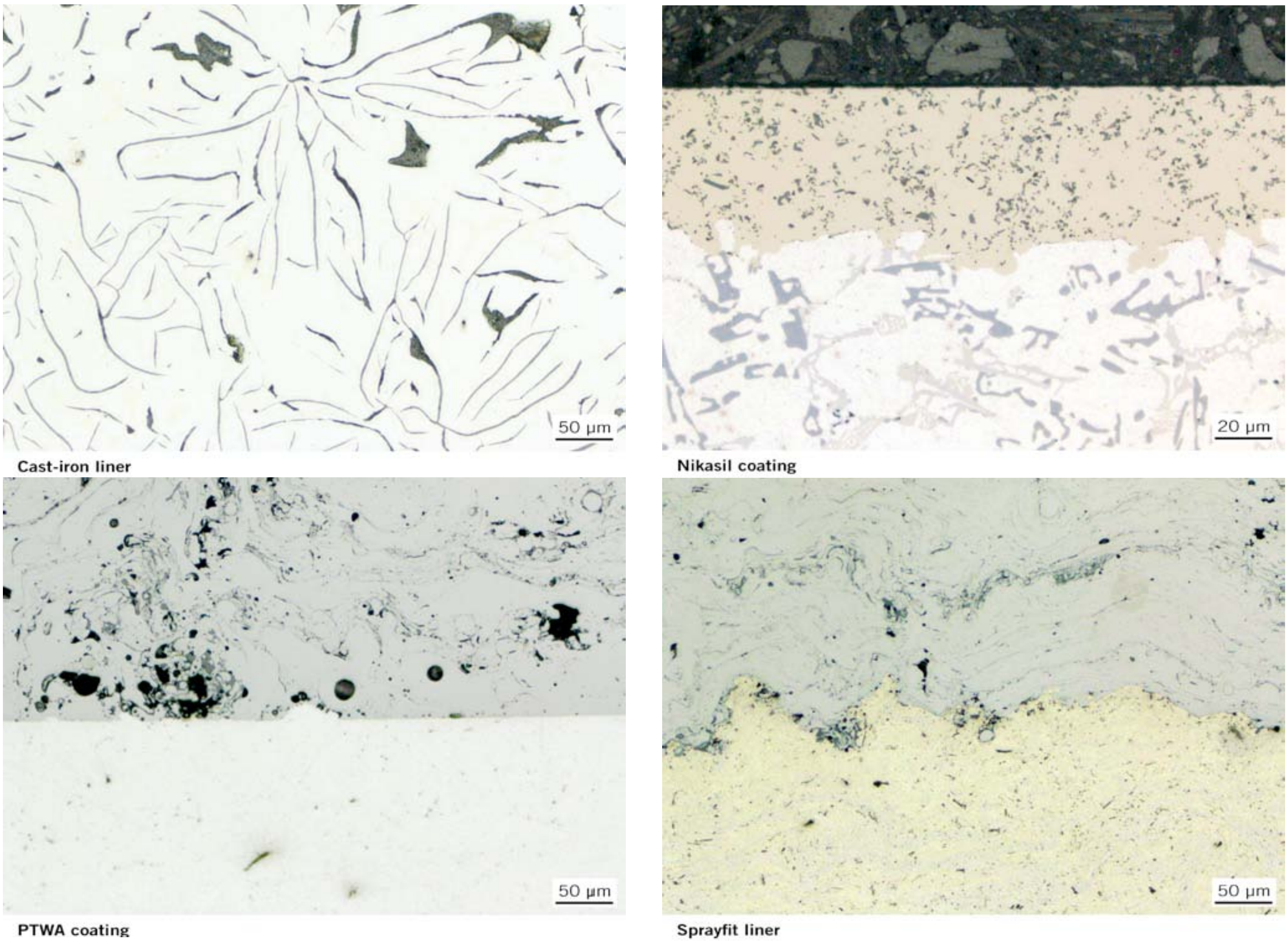

FIGURE 1 Overview of common running surface technologies for lightweight engines

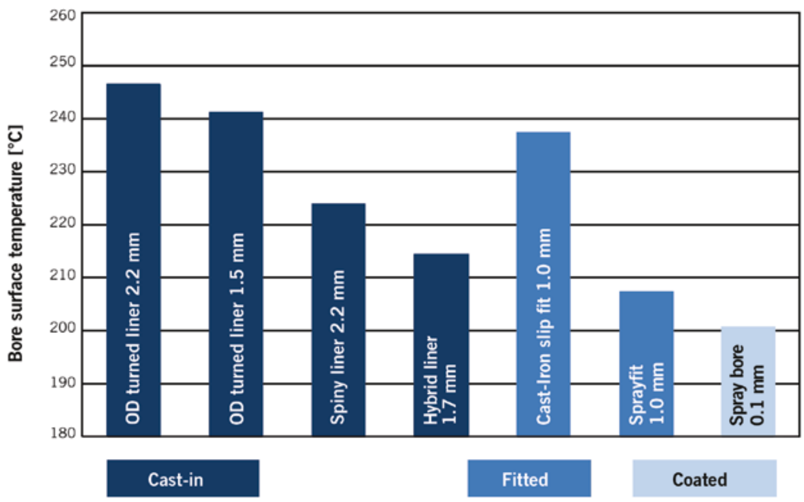

FIGURE 2 FEA model of the cylinder wall temperatures of different liner technologies

lightweight engines: in addition to a grey cast iron surface and the Sprayfit liner, a Nikasil coating and a thermally sprayed bore coating (named PTWA [1]) are shown.

GENERAL PROPERTIES OF SPRAYFIT CYLINDER LINERS

Because of their low ferrous content, Sprayfit liners offer significant weight benefits over cast-in liners; in a state-of-the-art four cylinder engine this can be as much as 2 kg. Even when compared to a weight-optimised, thermally fitted grey cast liner, the Sprayfit liner offers an additional weight benefit of 0.5 kg per crankcase.

In addition to the weight benefit, optimised heat transfer from the combustion chamber to the crankcase is another advantage, enabled by the small thickness of the ferrous layer. In this respect, the Sprayfit technology is almost on par with the level of a thermally sprayed direct bore coating. **FIGURE 2** shows an FEA model of the cylinder wall temperatures of different liner technologies under otherwise identical boundary conditions. Measurement of the thermal conductivity of fitted liners confirms these results. The ferrous layer thickness can be flexibly controlled during thermal spraying. This level of freedom can be utilised to define exactly the heat transfer from the combustion chamber, even limiting it if required.

Another issue for modern engines is the thermal management of the complete engine block which becomes more challenging as ignition pressures continue to increase while land width is being gradually reduced. As a result, despite the ever smaller land widths, cooling ducts need to be drilled to avoid thermal overload of the land and a subsequent loss of strength. The thermal expansion of the cylinder liner acts as a limiting factor in this situation: if a grey cast iron liner is fitted, the Al engine block shows a greater expansion than the liner when the engine is running. To ensure a sufficient press-fit between liner and block in the hot state, the pre-load at room temperature needs to be high enough, which can result in a high mechanical load during the cold state. **TABLE 1** highlights the constant press-fit of a Sprayfit liner over the temperature range although the liner is inserted with little pre-load. Also the mechanical load on the land is reduced in comparison to the grey cast iron liner.

Tribologically, Sprayfit liners – and ferrous direct bore coatings – are superior in comparison to cast-in grey iron liners. Numerous publications on this topic [2] describe improved values for friction and wear under certain conditions. However, it is always advisable to consider all components in order to achieve an optimal interaction of all moving parts in the combustion chamber. Among the influential factors are the choice of honing, the ring coating, the ring design, and the piston design.

In terms of corrosion there is no significant difference between grey cast iron liners or a ferrous thermal direct bore coating. However, depending on the requirements, the process of thermal spraying can be used to manufacture highly corrosion-resistant layers (e.g. with a chromium content greater than 13 %).

EXPERIENCE WITH MOTORCYCLE APPLICATIONS

Sprayfit liners have demonstrated a much longer durability (service time of more than ten races) in motorcycle racing than the electroplated NiSi Carbide coating previously commonplace. A comparison between two types of coating used in racing (52 kW, 500 ccm, 8500 rpm, air-cooled single-cylinder engine) reveals a pronounced difference after only a few races, **FIGURE 3**. The electroplated coating already shows serious wear, namely vertical grooves. The hard SiC particles (depicted in dark grey) do not withstand the piston ring's mechanical load. Instead the particles are pressed into the soft Ni matrix (depicted in light grey). The initial surface structure achieved by honing is

	Block stress in MPa		
Liner (interference)	Cold	Warm	Fired
Iron 1 mm (150 µm)	50.1	23.1	24.4
Sprayfit 1 mm (150 µm)	38.5	40.5	47.3
Sprayfit 1 mm (75 µm)	16.8	23.1	29.9

TABLE 1 Constant press-fit of a Sprayfit liner over the temperature graph

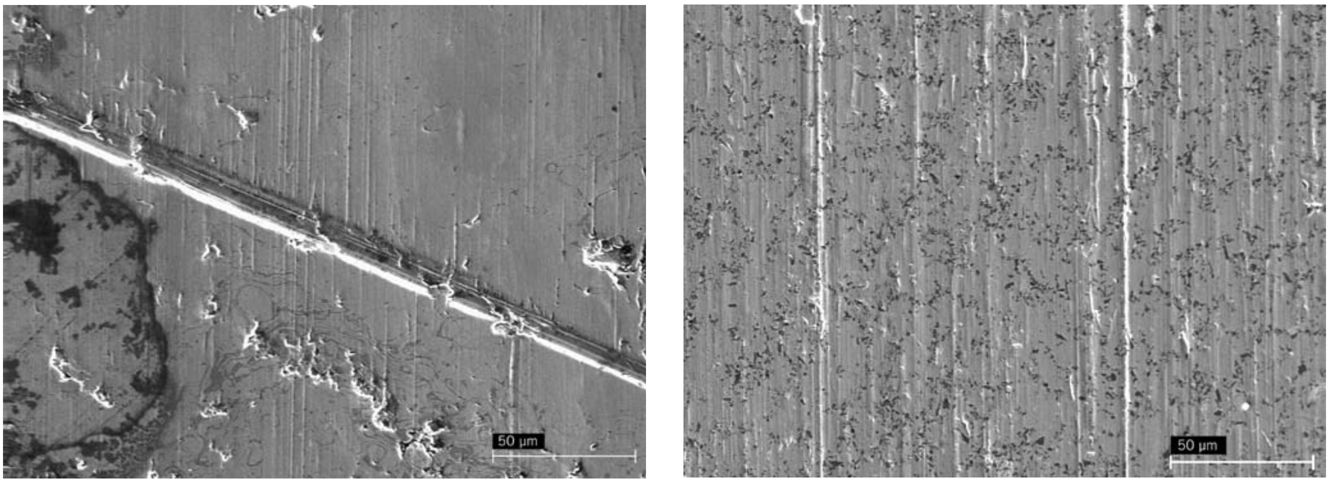

FIGURE 3 Comparison of an electroplated NiSi carbide coating and a Sprayfit liner after several motorcycle races: Sprayfit (left), Nikasil (right)

completely destroyed. In contrast, the homogeneously hard ferrous running surface of the Sprayfit liner shows hardly any surface wear and the honing structure is still intact.

Compared to a direct bore coating, Sprayfit liners offer an additional advantage in that the liner can be exchanged in a simple and cost-effective process if necessary, making application of a new direct bore coating unnecessary.

CAR ENGINE TESTS

After the successful initial use in motorcycle engines, Sprayfit technology was also integrated into passenger car engines and was examined on the test bed in direct comparison with common systems. Sprayfit liners were tested in a state-of-the-art diesel engine with a specific output of more than 63 kW/l and an ignition pressure greater than 180 bar. The engine was run at rated power at 4000 rpm. In production specification the engine is fitted with grey cast iron liners and demonstrated oil consumption of 9 to 11 g/h at rated power. The test results of the diesel engine fitted with Sprayfit liners showed a slightly lower oil consumption, of between 6 and 9 g/h. All other functional engine parameters showed no difference. A further comparison between the Sprayfit liner and a direct bore coating of the cylinder running surface showed a much lower share of ferrous residues in the engine oil, a finding which points toward the high layer quality of the Sprayfit running surface.

A gasoline engine test was also carried out. The application of the Sprayfit liner showed no irregularities in the functional properties when used in a 1.8-l displacement engine with 175 kW output.

COMPARISON TO DIRECT BORE COATING

Thermal spraying processes are also used in direct bore coating of Al cylinder running surfaces for car engines.

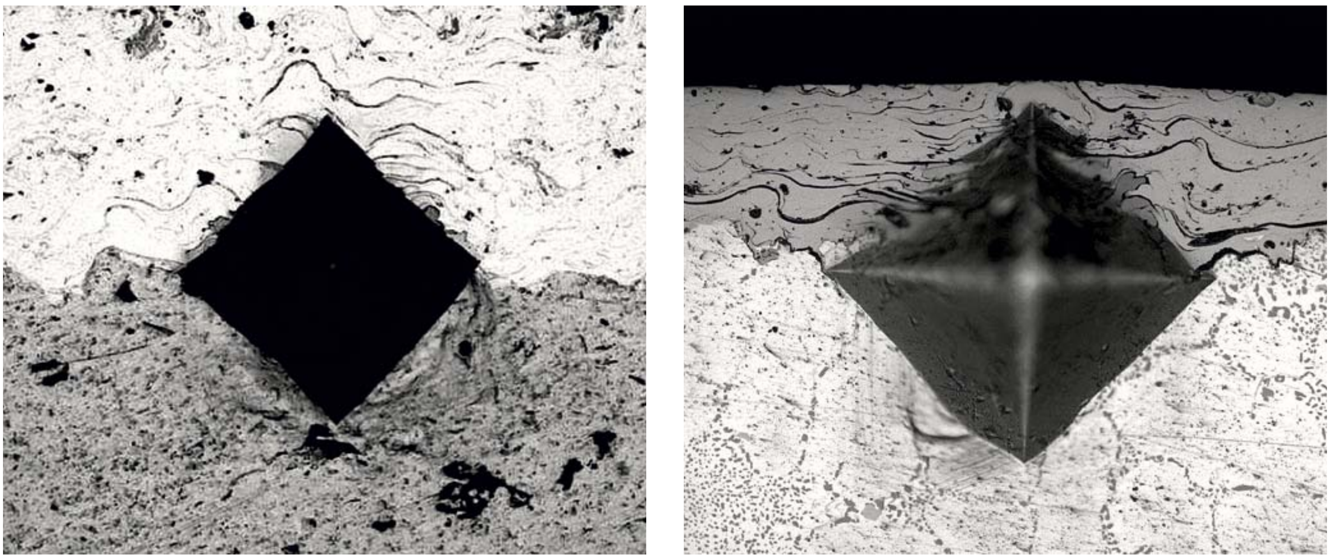

FIGURE 4 Result of adhesion test using an indenter: Sprayfit liner (left) versus arc-sprayed direct bore coating (right)

However, due to the boundary conditions, direct bore coating is a complex and costly process, for the following reasons:

- When cast, the Al crankcase has to be free from surface-breaking defects, which places the toughest demands on the casting process [3, 4].
- The cast surface needs to be activated, i.e. roughened, and subsequently cleaned and dried.
- To avoid an overly high scrap rate, the surface quality of the cylinder running surfaces is checked during an extra work step after activating.
- The lower bore end has to be closed to prevent an uncontrolled material spray into the crankcase.
- During the direct bore coating the tool has to plunge into the bore and rotate in it. This is not practical if the cylinder diameter is smaller than 70 mm, limiting the application range, particularly for engines of smaller displacement.
- If anything goes wrong during the direct bore coating, the complete crankcase will be scrap. The Sprayfit liner quality, however, is checked after manufacturing and process-related rejects are immediately removed instead of fitting to the crankcase. As the fitting process is identical with that established for grey cast iron liners, the insertion technology is already available for mass production.

Due to the manufacturing process of the Sprayfit liners, which are sprayed onto a rotating cylindrical mandrel from the outside, complete freedom exists to produce an optimum layer structure. Core benefits are the high kinetic energy of the process and the extraction of particle overspray. The effects even become visible in a direct comparison with a direct bore coating that is already known to work well in the engine. The high kinetic energy in the spray process results in a microstructure that is both very dense and internally well-bonded. Even an extreme load situation, such as the one depicted in **FIGURE 4**, will not result in a separation of the Sprayfit layer composite. The direct bore coatings that were applied to a narrow bore show much less cohesion, although the level is still sufficient for current engines.

Beyond these generic quality differences, which are characteristic of a layer that is externally sprayed, the Sprayfit liner manufacturing process offers additional levels of freedom to optimise the running surface. As an example, the functional properties could be further improved in the future by dispersing hard particles in the structure or by using materials with a high chromium content to produce corrosion-resistant running surfaces.

SUMMARY

The thermal spray process is an enabling technology to manufacture cylinder liners that combine the benefits of a direct bore coating with the economics and mechanical durability of a fitted cylinder liner. With the Sprayfit liner, the critical layer interface is transferred from the crankcase to the liner. There is a high level of freedom in defining the liner measurements, both in the manufacturing process and in the choice of materials. In short, Sprayfit offers an innovative, economic and proven solution to meeting the core requirements of lightweight engine design and effective thermal management.

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Influence of Micro Emulsions on Diesel Engine Combustion

Micro emulsions made of diesel fuel and water provide the possibility to greatly reduce the particle emission of the diesel engine without negatively influencing additional emissions or efficiency. Micro emulsions were developed and their application possibilities analysed by cooperation of the University of Cologne, the RWTH Aachen University and the Trier University of Applied Science within a FVV research project. Beside combustion and emission behaviour of premixed micro emulsions with varying composition also strategies for load-dependent mixtures were tested. Consequently, the micro emulsion was optimised regarding the resulted requirements.

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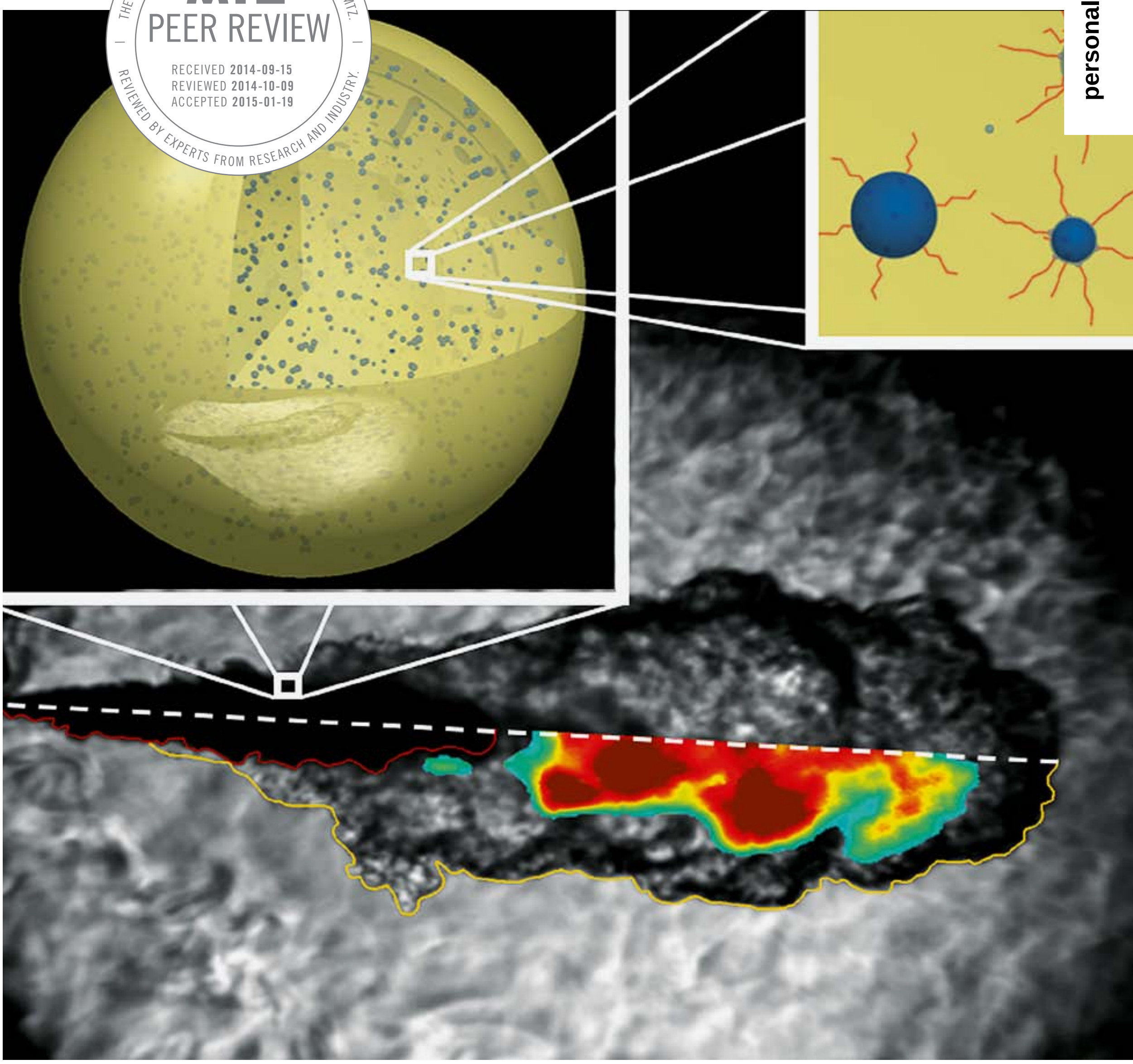

1 MOTIVATION

Due to its high efficiency and wide performance spectrum the diesel engine will still be important in the future in many application areas like goods traffic and decentralised energy supply. Despite the comparably low emission level of modern diesel engine combustion processes, particularly with regard to emissions of particles and nitrogen oxides as well as fuel consumption, further development work is necessary. A possibility for a targeted reduction of diesel engine emissions is feeding water into the combustion. This can be realised by humidifying charge air, direct or layered injection or emulsified fuel. The substantial effect of air humidification or direct water injection is the reduction of the combustion temperature and therefore, the inhibition of nitric oxide formation. Thereby, the influence on fuel consumption and particles is low [1]. On the contrary, by jointly injecting water and fuel, as it is the case for emulsified fuels, the influence on combustion and soot formation is significant. Known macro emulsion fuels were not able to establish in motoric operations due to mostly negative properties. In particular, the fast decomposition of diesel and water and as a consequence thereof variations in the combustion of emulsions impeded the use of emulsions. In this context the micro emulsions being analysed could offer distinct advantages.

2 DEVELOPMENT AND PROPERTIES OF THE MICRO EMULSIONS

Already in 1979 Feuerman [2] formulated the first emulsion of water, gasoline and non-ionic surfactants, which gained a reduction of environmentally hazardous exhaust gases during the combustion process. Criteria like stability and temperature invariance cannot be realised by an emulsified fuel in shape of a macro emulsion. Basically, two types of emulsions can be distinguished, **FIGURE 1**. The previously mentioned macro emulsions are macroscopically inhomogeneous and turbid. Moreover, the emulsifying process needs an energy input; the domain size is located in the range of 10 µm. A crucial disadvantage is the thermodynamic instability, which leads to phase separation. Whereas micro emulsions are thermodynamic stable, therefore develop spontaneous and are long-term stable [3]. The sizes of water domains are smaller than the wavelength of visible light and make the micro emulsions look clear and transparent. Additionally, they have an inner structure, which prevents the contact of water with metal parts of the engine.

With the aid of emulsifiers or surfactants it is possible to produce emulsions. Surfactants are technical, surface active solubiliser. They exhibit an amphiphile (Greece: loving both) character. The molecules have a hydrophilic (water-loving) head and a lipophilic (fat-loving) tail. An amphiphile film develops if water is admixed to oil. Hence, the interfacial tension is minimised. Caused by the lower energy input small water droplets in micro metre size can be dispersed in oil (and vice versa). By means of a suitable selection and combination of ionic and non-ionic surfactants it is possible to generate temperature invariant micro emulsions [4]. The combination chosen here is a consequence of the commercial ionic surfactants containing elements like phosphorus or sulphur, which lead to hazardous emissions during combustion. Thus, the ionic surfactant (oleic acid) and the neutralising base (monoethanolamine) are self-generated in an in-situ neutralisation reaction.

The real systems depicted in **FIGURE 2** distinctly differ from models in their phase behaviour. The steep curve of the phase boundaries accompanied by temperature invariance cannot be reached by the combination of ionic and non-ionic surfactants. Meso-

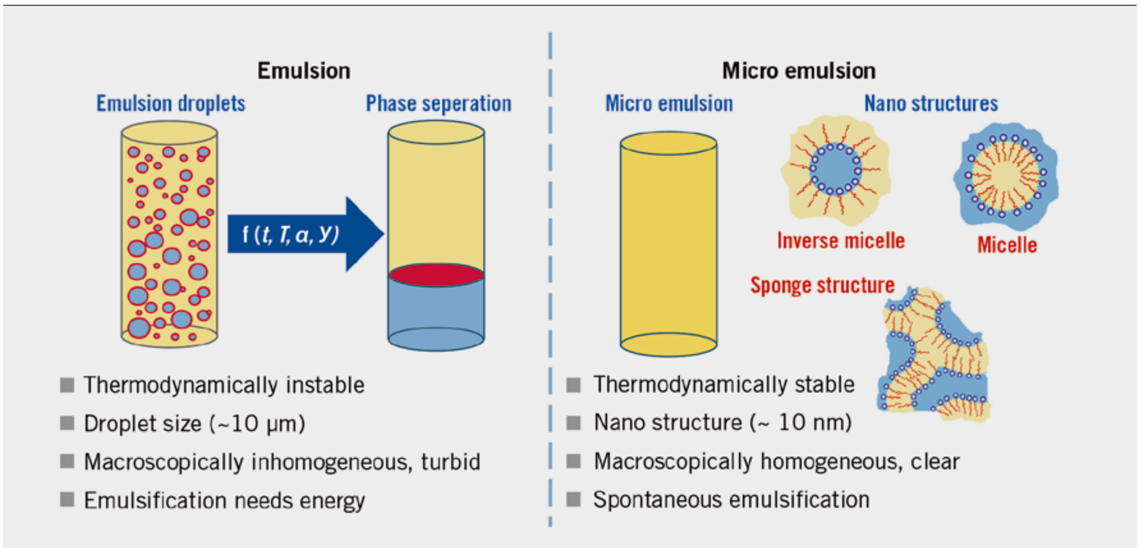

FIGURE 1 Comparison emulsion and micro emulsion

phases occurring in the single phase area are suppressed by the targeted application of ethanol and the lyotropic salt ammonium nitrate to prevent the associated increased viscosity. In the three phase diagrams with varying water contents it can be recognised that the needed amount of surfactants for temperature invariant phase behaviour increases with the water content. A micro emulsion with a water content of 24 % requires at least 20 % surfactants. The content of surfactants for micro emulsions with 16 % and 8 % water could be reduced to 15 % respective 13 %. The shape of the fuel matrix in **TABLE 1** results from the consideration that the influence of water can be regarded separately from that of the surfactants. The determination of the most important physical properties of the micro emulsions resulted in the finding that these are mostly located in the tolerance range of diesel fuels. The density as well as the heat value varying in relation to water content; the influence of the amount of surfactants is low due to the heat value similar to the one of diesel. The tribological behaviour of the micro emulsions was in the HFRR test (High Frequency Reciprocating Rig) throughout more favourable than that of diesel fuel and therefore obviously under the limit value. A reason for this is the boundary layer formation of the surfactants as well as the increased viscosity of the micro emulsions.

3 INJECTION, IGNITION AND COMBUSTION

The component tests with solenoid valve injectors of the type Bosch CRIN2 showed that no abrasive or corrosive wear of the injection system is to be expected. However, a substantial cavitative attrition occurs in the area of the pilot valve, which under normal boundary conditions leads to a failure of the injector already after 60 operating hours with 24 % water content in the micro emulsion. Increasing the leakage counter pressure yields notable improvements for the tested injector. For adapting to the reduced heat value, the nozzles cross sections were appropriately adjusted. The thereby corresponding increase of injection mass as well as delayed evaporation, caused by high enthalpy of evaporation, are reflected in the optically accessible high-pressure chamber by a significant increase of the liquid penetration length and the amount of liquid phase in the spray. This deteriorated mixture formation in combination with temperature setback results in an extended ignition delay with increasing water content. Hence, the position of ignition is located closer to the liquid phase and therefore a worse stoichiometry in the premixed reaction zone can be expected. Due to these influences stable combustion cannot be guaranteed, particularly in lower load points with high water content. In order to determine the influence on the combustion the single-cylinder engine specified in **TABLE 2** was used. For the tests a full load point (C100) and two part load points (B50 and A25) were chosen out of the ESC cycle, since they are representative for the operating range in the target application. As an example the results for the operating point B50 are depicted in **FIGURE 3**. The testing of the micro emulsion was conducted at constant engine conditions (rail pressure, exhaust gas recirculation rate, air to fuel ratio) in a variation of the position of fractional mass burn by adapting the injection start. Herein a declining NO_x level appears at constant position of fractional mass burn and constant exhaust gas recirculation rate. Moreover, the influence on the fuel consumption is very low. The NO_x-particulate-trade-off proofs that with the aid of micro emulsions a significant improve-

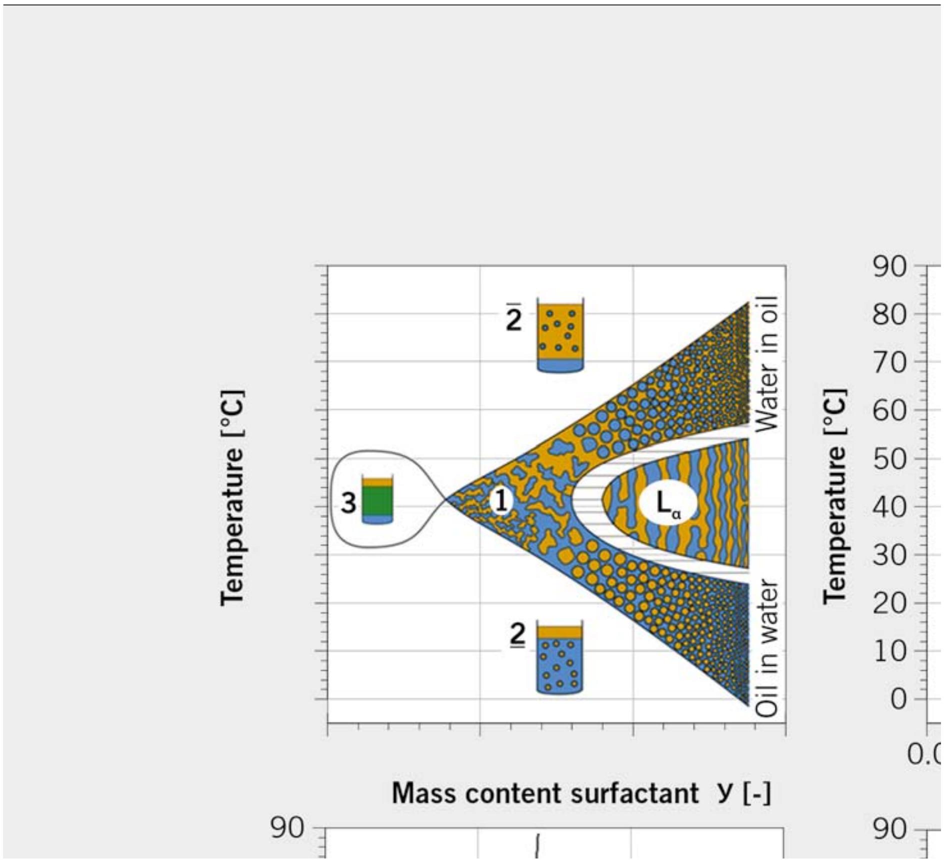

FIGURE 2 Phase behaviour of micro emulsions (schematic (upper left): with 8 %, 16 % and 24 % water; composition of real systems: Water/Ethanol/Ammonianitrate – Diesel – Oleic acid/Monoethanolamin/Wallamid OD4 with variable contents

TABLE 1 Fuel matrix

Micro emulsions		Mass fraction water [%]	Mass fraction surfactant [%]	Heat value [MJ/Kg]	Density (20 °C) [kg/m³]	Viscosity (20 °C) [mm²/s]	Viscosity (80 °C) [mm²/s]	Wear scar diameter in HFRR [µm]
Surfactant constant	DWME0020	0	20	41.14	843	6.54	1.99	153
	DWME0820	8	20	37.26	862	20.48	4.07	238
	DWME1620	16	20	33.17	874.5	34.65	6.37	–
	DWME2420	24	20	28.9	890	58.36	8.13	257
Surfactant minimised	DWME0813	8	13	37.4	857	20.19	3.85	264
	DWME1615	16	15	–	–	–	–	–

TABLE 2 Tested engines

	Displacement [l]	Stroke [mm]	Bore [mm]	Compression ratio [-]	Ratet power [kW]	Maximum torque [Nm]	Maximum injection pressure [bar]
Single-cylinder engine	2.136	160	128	16.9:1	69	358	1800
Multi-cylinder engine	4.04	126	101	18:1	113	657	1400

ment can be realised. Already with a water content of 8 % one can reach a considerable reduction of particle emissions, which can be even more improved with increasing water content. For a micro emulsion with a water content of 24 % a measurement of particle emissions by measuring filter smoke number value is hardly feasible. By adapting the nozzles to the fuel heating value of the micro emulsions a constant injection duration could be yielded, but with this the combustion duration changed. As it can be seen in 10 % of the fractional mass burn in **FIGURE 3** the ignition delay, increasing with the water content, causes, a slightly delay of the first combustion phase. The significant delay of 90 % of the fractional mass burn at constant centre of combustion indicates a faster combustion with increasing water content. Therefore, a minimal influence of the combustion duration even at an operating-point-dependent mixture can be expected if no nozzle adaptation is possible.

Besides measuring the filter smoke number value, also filters were loaded to enable a detailed analysis of particle mass and composition in all operating points. **FIGURE 4** shows that with gravimetric particle measurement a significant reduction of the particles with increasing water content can be shown. However, the influence of the micro emulsion structure by varying surfactant content could not be completely resolved. The particle samples were analysed regarding their composition. Therefore, the parts of organically soluble (SOF), water soluble (WSF) and insoluble (NSF) components were determined, whereas the latter mainly consists of carbon. The analyses depicted for operating point B50 in **FIGURE 4** show that micro emulsions mainly reduce the carbon-rich components of the insoluble fraction. Whereas the mass of SOF and WSF is only slightly influenced; therefore, these fractions are the largest components by mass.

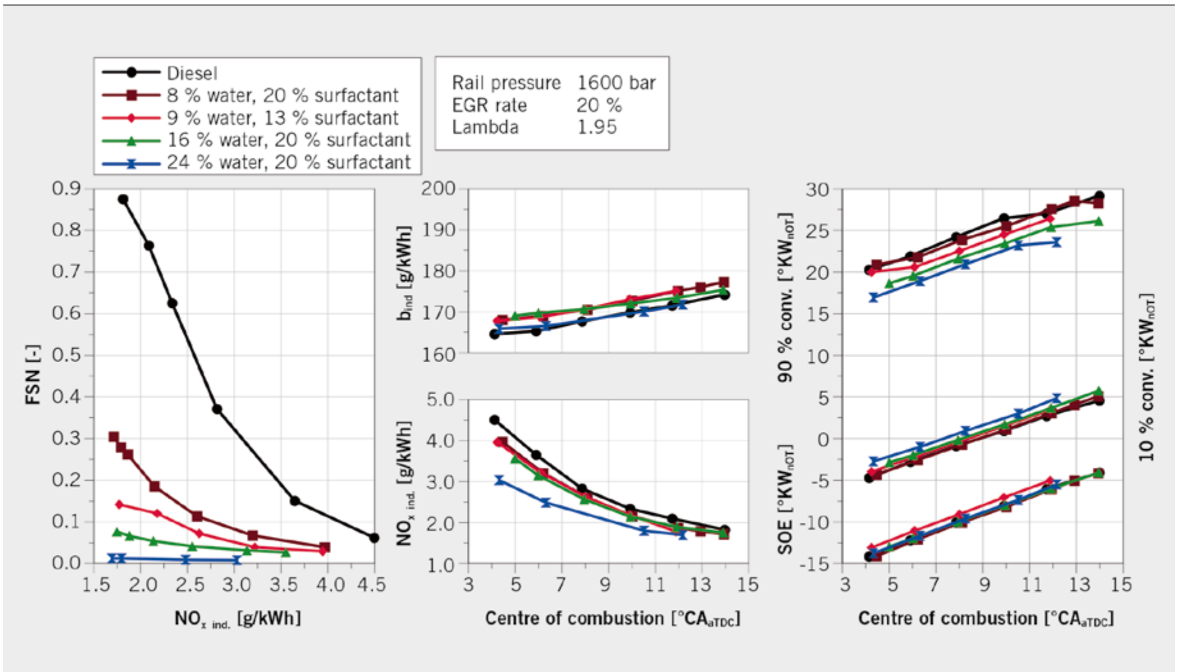

FIGURE 3 Comparison of premixed micro emulsions at part load point B50

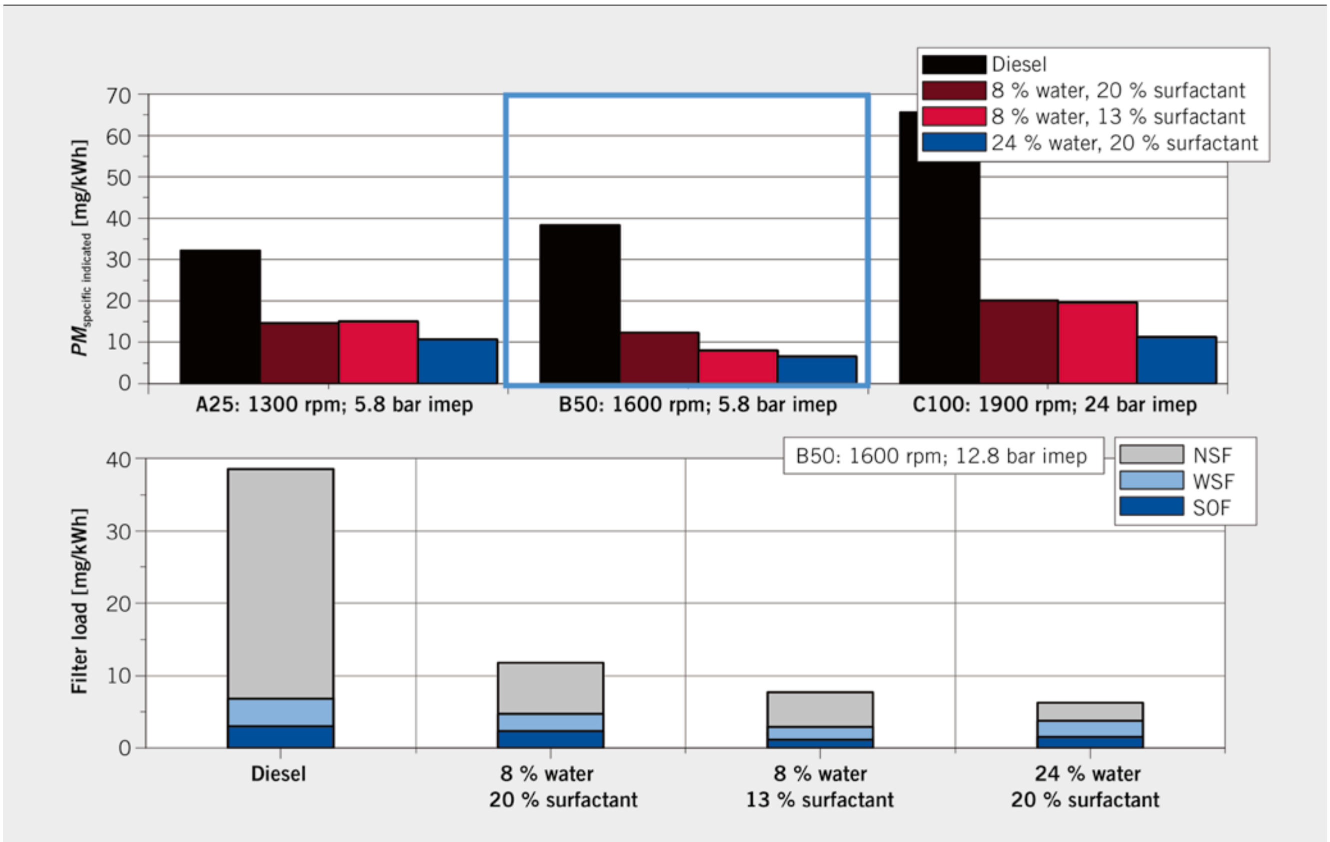

FIGURE 4 Particulate mass in examined load points and particulate composition exemplary in load point B50

4 LOAD-POINT-DEPENDENT MIXTURE OF MICRO EMULSIONS

FIGURE 5 describes the mixing ratio of diesel and water in a model mixing chamber; in the upper part of the picture without the admixture of an emulsifier. The intermixing of diesel and water does not happen here and the water flows as a streamline through the mixing chamber. Adding surfactants of a micro emulsion to diesel or water already after a flow path of approximately 2 cm a homogeneous mixture of diesel and water emerges (lower part of the picture).This and further tests [5] proved that by adding suitable surfactants already after a short time a homogeneous diesel/water-mixture forms, taht is to say micro emulsions have very high formation kinetics. The measurements of the formation kinetic for the micro emulsion with a surfactant content of 20 % were conducted in a stopped-flow spectroscopy. For this purpose, the oleic phase with the surfactants and the aqueous phase with the additives were directed at a point in time of $t = 0$ s into a measuring cuvette and the transmission of light was recorded with a helium-neon laser. With the aid of an exponential adjustment of the intensity the time constant τ as a measure for the needed time until reaching the homogeneous mixture was determined. In **FIGURE 6** the results for the micro emulsions with 8 % and 24 % water at 20 °C as well as a measurement at 60 °C are depicted to estimate the influence of temperature. As expected the time constant grows with increasing water content. So, the micro emulsion with 8 % water needs 1.8 s to form, while the monophase micro emulsion with 24 % water needs approximately 7.6 s. Another measurement at 60 °C illustrates a strong influence of temperature on formation kinetics. The difference in temperature of 40 °C causes a significant reduction of the formation duration to 0.8 s resulting from the lower viscosity of the micro emulsion. Contrary to pre-mixed micro emulsions on-board and on-injector blending offer the possibility to produce the diesel/water-mixing ratio in a needs-oriented way, and therefore operating point optimal, directly at the engine.

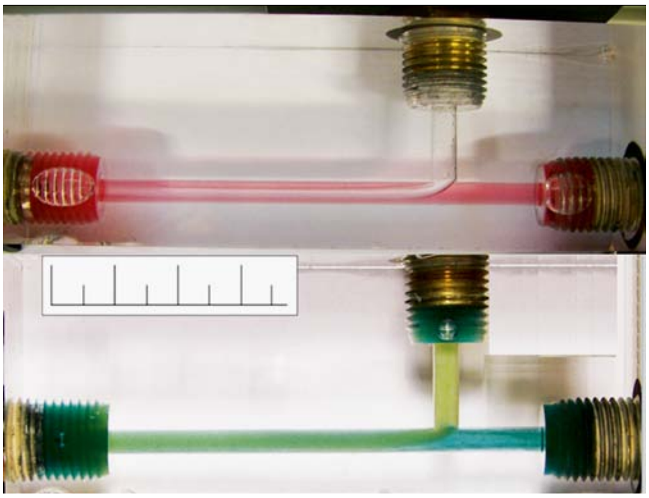

FIGURE 5 Diesel-water mixing behaviour without (top) and with (bottom) surfactant

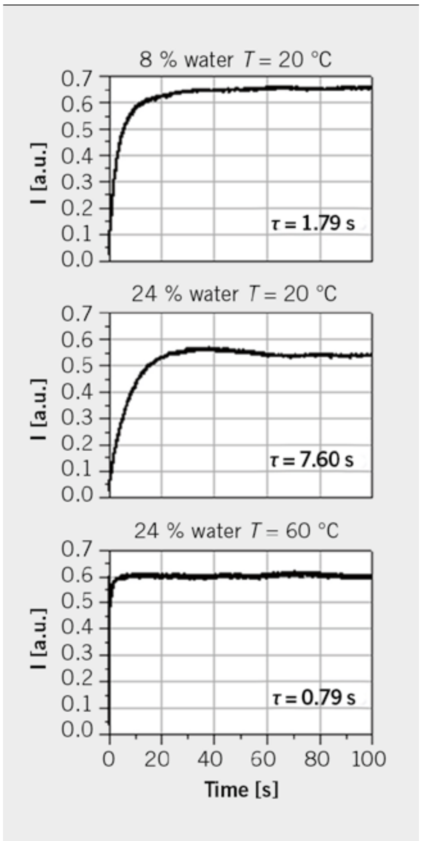

FIGURE 6 Stopped-flow measurement of mixture kinetics

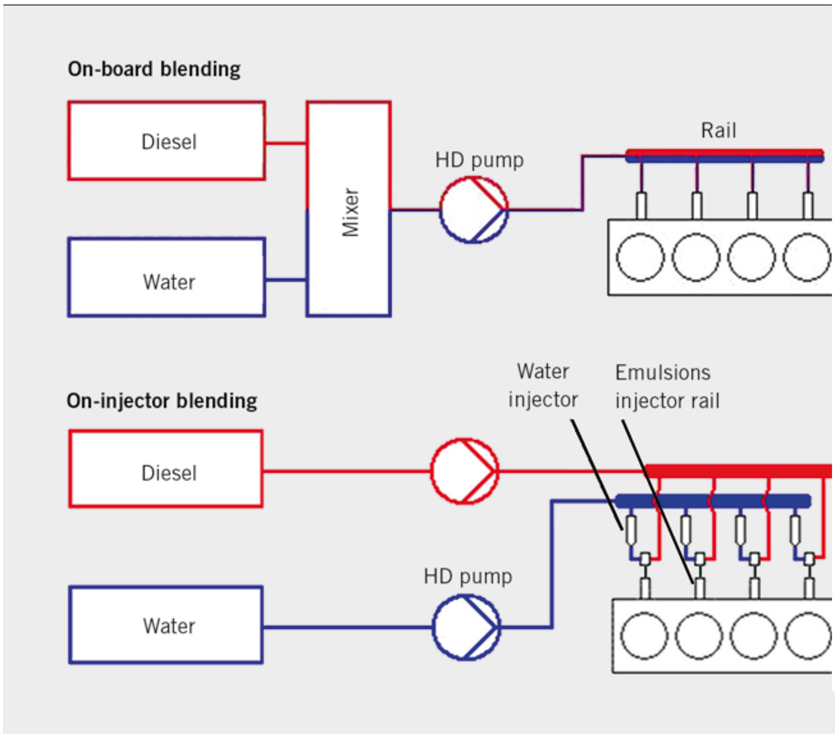

FIGURE 7 Schematic setup of on-board and on-injector blending systems

Additionally, the admixing of water to diesel during the operational phases critical to the emulsion operation, like the engine warm-up, can be completely prevented. **FIGURE 7** illustrates the schematic construction of the two systems. At the on-board blending system a mixing unit is separately fed with diesel and water. In this mixing unit they are mixed to a homogeneous diesel-water micro emulsion and afterwards forwarded into the common-rail by a high-pressure pump. For the engine used in the tests, **TABLE 2**, the injection into the combustion chamber is conducted with solenoid valve injectors. The relatively simple construction and mixing of micro emulsions in the low pressure range are advantages of this mixing system. Disadvantageous is the relatively long reaction time to changes of the water content in the diesel-water-mixture, **FIGURE 8**, that is to say during load changes. In contrast, a substantially better mixing behaviour promises the on-injector blending system. There the micro emulsion is formed directly at the injector. Due to a relatively low dead volume the mixing times are much shorter in adaptation the water amount to the given operating point. The disadvantages of this system are the relatively complex construction and the additionally needed water injection system. Because of the short mixing length the formation of a homogeneous mixture needs a fast-working emulsifier. The effect of the water dosage on performance and exhaust emission behaviour are depicted in **FIGURE 10**. In this test the engine was firstly adjusted without water dosage in the operating point B50 (pme = 10 bar; n = 1565 rpm). Then the water supply was activated. Already after approximately 2 s a sharp decrease of the mean effective pressure could be observed as a consequence of the water admixing to the diesel fuel. The response time of 2 s from the moment of water activation to the first visible decrease of pme results from the length of the fuel line from the mixing chamber to the main injector. After a short transitional period of approximately 5 s the stationary emulsion operation of the engine is finally reached. In this experiment an adjustment of the injection mass for load regulation was deliberately relinquished to use the engine load as an evaluation criterion for the temporal progress of water dosage. The simultaneous decrease of NO_x and soot emissions show the influence on the combustion; but due to the reduction of load it can only be evaluated qualitatively. At this, it should just be described how spontaneously the water in the fuel reacts to exhaust values. Considering this background the fact that only due to the load change an influence on the decrease of NO_x concentration and opacity is given has no great importance. With increasing engine load the transitional time decreases to less than 5 s because of higher fuel volume flows as well at activation as at deactivation of the water; this was proved in the C100 operation point (pme = 18 bar; n = 1853 rpm).

5 SUMMARY

Micro emulsions offer a very good possibility to insert water into the combustion chamber. In contrast to macro emulsions they only need a minimal energy input to form and are thermodynamic stable so that they are storable and phase separation will not emerge. Additionally, the spontaneous formation enables a mixture of the micro emulsions in a high-pressure circuit to realise transient

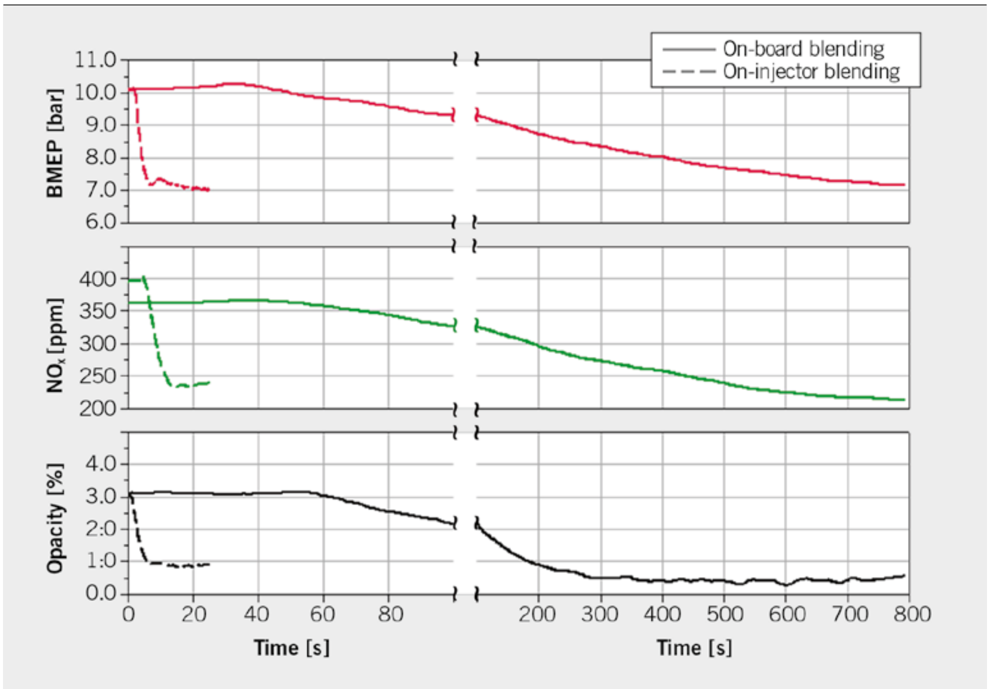

FIGURE 8 Temporal change of pme, NO_x und opacity with water mixture activation

engine conditions during the micro emulsion operation, too. Depending on the used water content the need for surfactants increases to produce a temperature invariant, storable micro emulsion. An optimisation of the micro emulsion to the mixture directly at the engine significantly reduces the need for surfactants while thermodynamic properties of the emulsion stay comparable. It could be depicted that micro emulsions have a very good lubricity and do not evoke any signs of corrosion. However, an enhanced occurrence of cavitation could not be prevented with the usage of solenoid valve injectors. Adapting the injection system to a mixture in the injector in combination with modern leakage-free systems might lead to notable improvements. The positive influence of micro emulsions on the emission of NO_X and particles could be proven in all tested operating points. At constant operating conditions and comparable NO_x levels and fuel consumption, a significant reduction of the particles could be realised. Considering stable combustion and to realise the full potential a load-dependent dosage of water should be targeted. For realising this aim two mixing systems were examined. Evaluating on-board blending and on-injection blending in respect of the engine response during a load change of the engine results in the superiority of the on-injector blending compared to the on-board blending. While at on-board blending a stationary operating condition after a load change of the engine can only be reached after several minutes, the on-injector blending gains a stationary operation condition after a few seconds.

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