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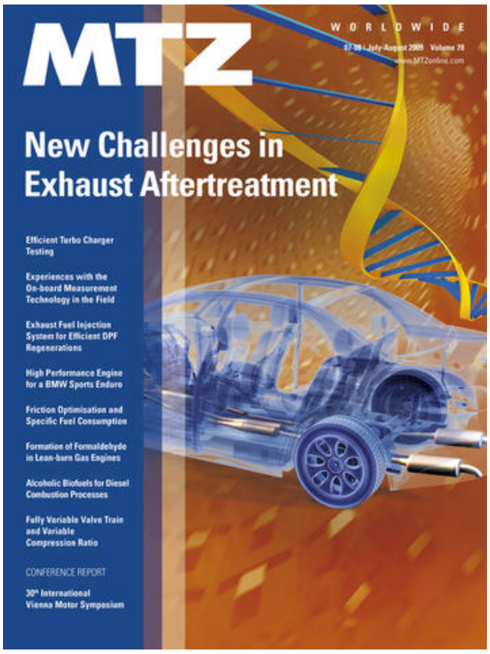

MTZ worldwide 8/2009, as epaper released on 16.07.2009
http://www.mtz-worldwide.com

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MTZ WORLDWIDE
07-08 | July-August 2009 Volume 70
www.MTZonline.com

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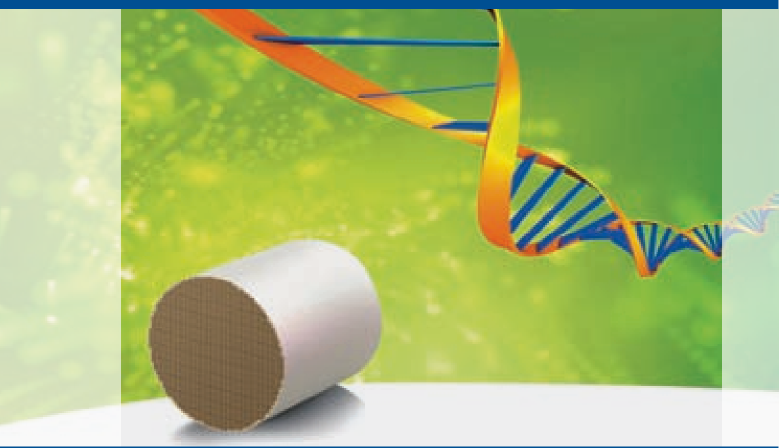
COVER STORY

New Challenges in Exhaust Aftertreatment



4
Modern molecular biology provides the tools to design surfaces on the nanometer scale and opens the way to optimize many industrial processes. In a proof-of-concept study, Namos GmbH successfully reduce the amount of precious metals required for a **Diesel Oxidation Catalyst**.

COVER STORY

DEVELOPMENT

RESEARCH

CONFERENCE REPORT

RUBRICS

Pause for Thought

Dear Reader,

At the moment, the search for the perfect compromise between the optimisation of the internal combustion engine and its electrification resembles an orienteering race through uncharted territory. What the most important thing for finding one's way when the going gets tough? A perfect navigation system? Or, if you prefer not to rely on the possibilities offered by electronics, a compass and a detailed map?

No, the most important thing is to stop, determine exactly where you are and then explore your surroundings. It is not enough merely to pursue a vision. Many great engineers have failed because, even though they kept their eyes firmly fixed on their destination, they were unable to proceed when the path became too difficult.

The upcoming holiday period offers a good opportunity for walking. Even the ancient Greeks were aware of the beneficial effect of this kind of motion. The pupils of Aristotle, the Peripatetics, would often go for a long walk in order to find enlightenment. For us, it is less important to discover absolute meaning but rather to find the most meaningful ways to discover the powertrain of the future. Nevertheless, our minds become much clearer when we are able to get away from our everyday business and pause for thought in the mountains or at the seaside.

The future is in our hands! I wish you a relaxing holiday, but also a creative one.

Johannes Winterhagen
Bratislava, 29 June 2009

Johannes Winterhagen
Editor-in-Chief

MTZ WORLDWIDE
07-08|2009

Founded 1939 by
Prof. Dr.-Ing. E. h. Heinrich Buschmann and
Dr.-Ing. E. h. Prosper L'Orange

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

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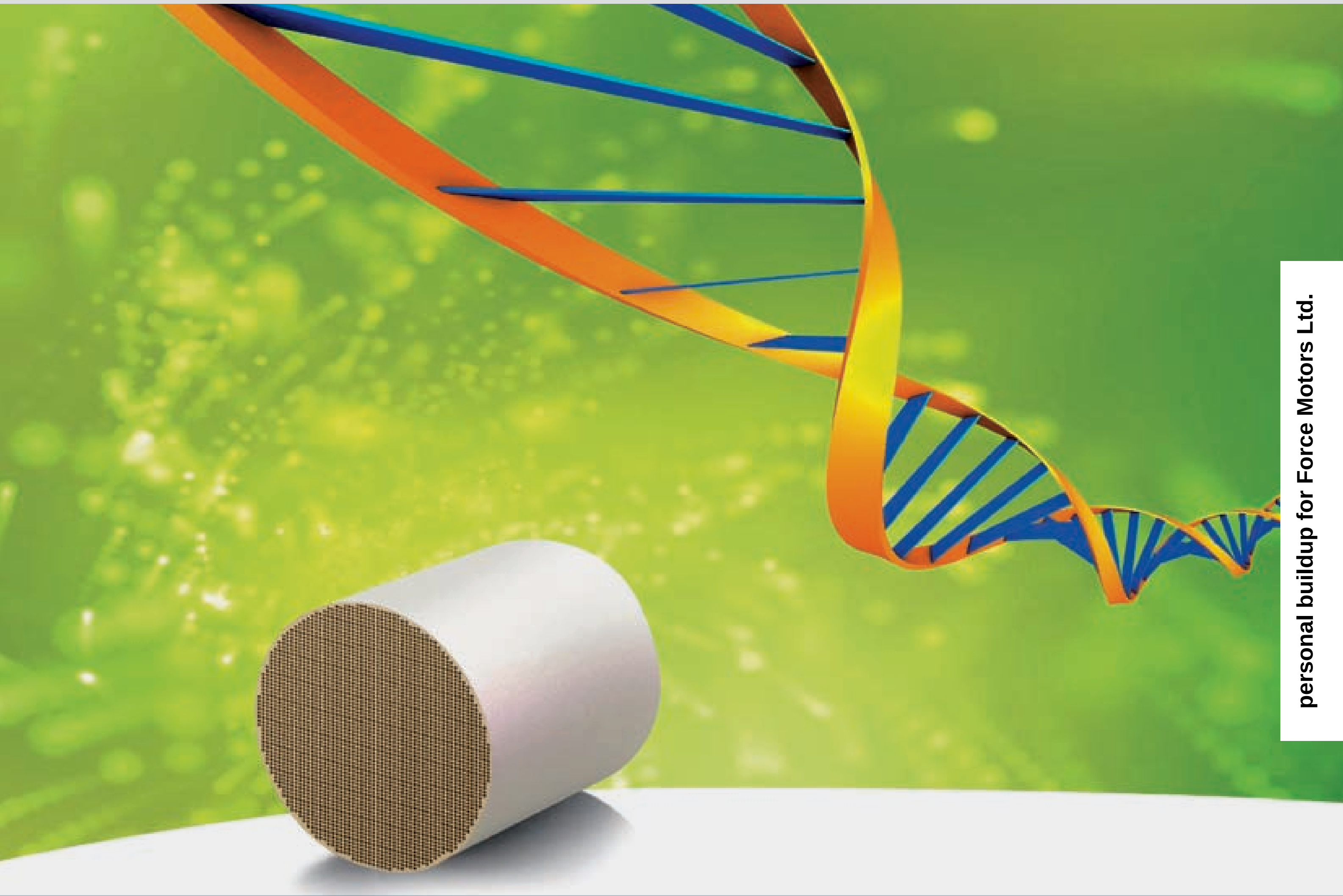

Industrial Biotemplating Saves Precious Metals in Catalysts

Modern molecular biology provides the tools to design surfaces on the nanometer scale. This opens the way to a breakthrough innovation, which can optimize many industrial processes. In a proof-of-concept study, scientists were able to successfully reduce the amount of precious metals required for a diesel oxidation catalyst. This was the first successful application, and right now the biotemplating technology awaits further development for other applications involving catalytic processes or specifically designed surfaces for industrial processes.

1 Introduction

Like most raw materials, platinum prices have dropped to an all-time low in the preceding months. However, this price drop is arguably based on upheavals in the world financial system. Platinum is predominately minded in only a few places in South Africa and Russia. This and the fact that the demand is tied to the demand for cars [1] explain the general volatility in the platinum market. Medium and long-term prices for platinum are however expected to rise based mainly on bullish demand expectations.

Platinum is the active component in catalysts, which reduce polluting car emissions from diesel or gasoline. Such catalysts are the three-way catalyst, the diesel oxidation catalyst, carbon-particulate filters and NO_x storage catalysts. The required platinum significantly increases the prices of these catalysts.

The operating temperature is so far the one significant difference in the exhaust treatment of cars running either on diesel or gasoline. In ageing simulations, three-way catalysts are routinely subjected to temperatures exceeding 800 °C. The trend is to increase this temperature up to 850 °C in an effort to further reduce the carbon-particle and carbon oxide emissions from modern diesel engines in compliance with Euro 5 and Euro 6 standards. This is due to the proximity of the catalyst to the engine and also to the high temperature cycles required for the active soot combustion in the particle filter. Such high temperatures are also generated in the NO_x storage catalyst for the conversion of nitrogen oxides during the rich fuel/air mixture cycles.

At such high temperatures, the precious metal particles tend to coarsen (sinter) much faster, and a large portion does not retain significant catalytic properties for any length of time. Although exhaust gas catalysts have been consequently improved since their introduction in cars, the potential to save on the use of precious metals is still high.

Engineers usually introduce new materials or optimize the size and distribution of the metal particles on the carrier substrate to reduce the required amount of precious metal. However, basic conditions limit the choices of materials, and for the most part the initially optimized distribution does not outlast the high temperatures. Furthermore, the new and often complex processes can be expected to significantly raise the overall costs.

2 The Use of Biological Templates to Design Precious Metal Surfaces

There is overwhelming scientific indication [2] that ageing will diminish the catalytic properties of all but a small portion of the initially present precious metal. Few metal particles will be exposed or have retained the right size. A selection has obviously taken place between catalytically active and inactive precious metal. If this selection could be accomplished upfront when the catalyst is manufactured, most of the precious metal could be saved without reduction in catalytic activity.

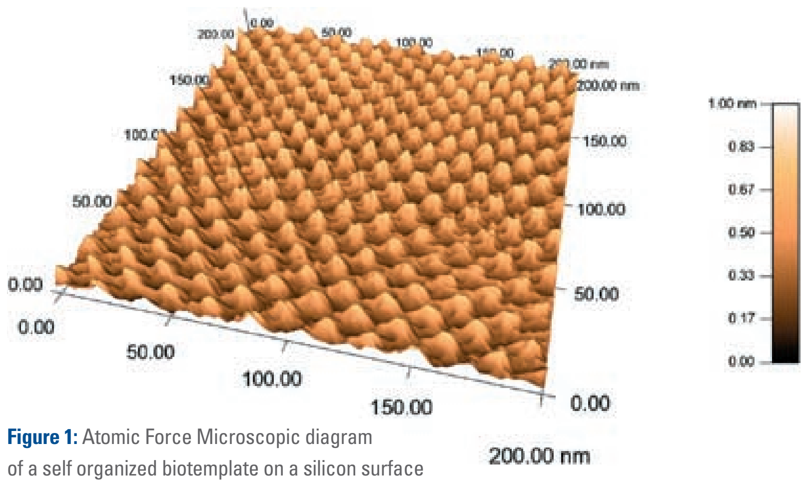

Figure 1: Atomic Force Microscopic diagram of a self organized biotemplate on a silicon surface

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To accomplish this we have combined platinum, palladium and specific biological molecules as biotemplates in different ratios and have used the resulting compounds to coat aluminum oxide particles in a conventional coating process. It is important to note that the biological templates have been used in the process only temporarily as scaffolds to control the configuration of the precious metal particles. Biological structures are well suited to serve as such scaffolding materials based on their ability to self-organize to regular nano-sized structures on surfaces. Other suitable features are their defined size of only a few nanometers and their affinity to metals, **Figure 1**.

The strong binding to the biotemplate is required to control the configuration of the metal deposits. Biological templates can therefore temporarily serve as a matrix to determine the configuration of precious metals in the manufacturing process.

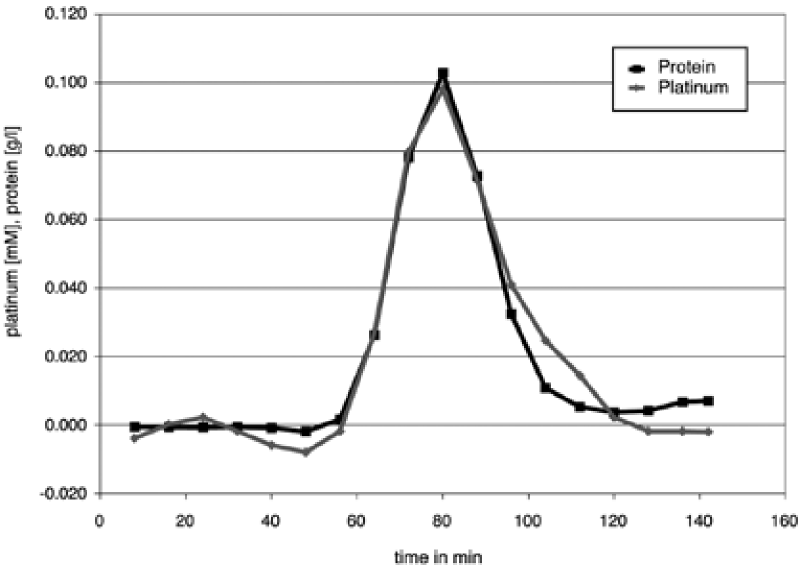


Figure 2: Retention diagram of a Platinum-Biotemplate solution – the accordance of time of maximum concentrations at the column outlet gives indicates bindings between both components

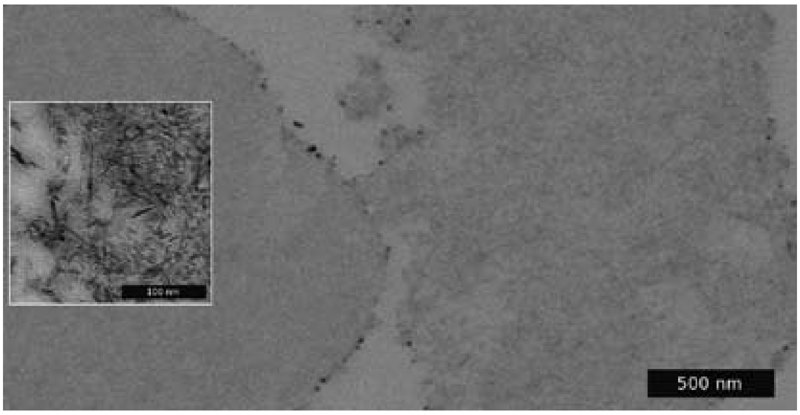


Figure 3: Gamma aluminum oxide particles with platinum clusters, TEM micrograph of a micro section

We used column chromatography to demonstrate the binding of metal and biotemplate. **Figure 2** shows a retention diagram of a platinum-protein mixture. The mixture was applied to a Sepharose 6B column, which separates molecules according to their size. Sepharose 6B matrix has a defined system of differently sized pores. Dependent on their size, molecules have to pass through different volumes during their passage through this column. Large molecules do not fit into the smaller pores in the Sepharose column material and therefore pass quicker through the column. Small molecules move slower because they have to move through the smaller Sepharose pores as well. Continuous sampling of the column eluate yields fractions of the chromatographed mixtures by particle size. We analyzed these fractions and determined their composition.

Based on their size difference, the original platinum precursor and the protein separate easily on the Sepharose column provided there are no interactions between the two components. The two almost coinciding peaks in Figure 2 strongly suggest that almost all the platinum in the mixture is bound to biotemplates. Only the eluate collected between 96 and 120 minutes shows a trace of platinum, which has not bound to the protein.

A simple use of this affinity of biological templates to metals is the controlled pore selection when coating the carrier substrate with the catalyst. All commercial carrier substrates used for exhaust gas catalysts consist of highly porous particles and their sizes fall in the micro meter range. These particles in turn consist of crystallites measuring a few nanometers. This accounts for the distinct hierarchy of pore systems. The size of the pores and the temperature determine the overall catalytic conversion. The accessibility of micro and meso pores for exhaust gases drops precipitously due to the changing diffusion mechanism (Knudsen diffusion). Based on their design and the presence of precious metal with sufficiently large pore systems, the efficiency of exhaust gas catalysts rises from zero to 100 % within a small temperature range. Therefore the precious metal in micro and meso pores no longer contributes to the catalytic conversion.

Using standard methods to deposit precious metals, carrier substances like aluminum oxide are incubated with precious metal salt solutions. A selection of different pore systems does not take place. However, the affinity of the metal salts to carrier substrate leads to a preference for smaller pores (capillary effect). Inside the aluminum particles (where the precious metal scarcely contributes to the catalytic conversion) the metal salts are adsorbed, deposited and then reduced to very small metal clusters. The use of biotemplates prevents this as the analysis of carrier substrate micro-sections under the transmission electron microscope (TEM) proves.

For this analysis we incubated pure Al_2O_3 powder with a platinum salt-biotemplate solution and allowed it to dry. The coated Al_2O_3 powder was embedded in epoxy resin and hardened at 60 °C. The epoxy block was then trimmed until the particles were appropriately placed for the cutting step. Using an ultra microtome Leica EM UC6, the embedded sample was cut into about 70 nm thin sections. The sections were then fixed on a TEM grid for analysis.

Figure 3 shows a TEM micrograph of an appropriately prepared section at an acceleration voltage of 80 kV. The precious metal clusters appear as dark areas in the micrograph. They are exclusively located at the boundaries of the aluminum oxide particles. There were no precious metal deposits inside the lower level micro and meso pore systems of the individual γ-Al_2O_3 crystallites.

3 Results for the Development of Diesel Oxidation Catalysts

We measured the catalytic conversion of synthesis gas on cores to investigate the behavior of catalysts, which were prepared using biological templates. The temperature required for 50 % conversion served as the reference parameter for the catalytic activity (T_{50} Temperature). Logically, the smallest gas molecules are most likely to diffuse into small pores. In typical diesel engine emissions this is carbon monoxide. We have therefore limited our measurements to carbon monoxide. Our control samples consisted of materials, which we prepared like the investigated samples save the presence of the biotemplates. Our measurements therefore showed the influence of the biotemplate component.

For the preparation of samples we used cores provided by the company Interkat GmbH in Königswinter, Germany, **Table 1**. The cores were three inches long and had a diameter of 1 inch. They were derived from a solid body, which was commercially coated with pure gamma-aluminum oxide. All these cores were coated with platinum by dipping them in precious metal precursor solutions. We applied this procedure to coat three control samples and used different platinum concentrations between 0.3 and 1.14 grams per liter in the process. We prepared another sample using the same process but with added biotemplate. The platinum concentration for this latter sample was 0.46 grams per liter. We measured the precious metal contents by way of mass spectroscopy in the remaining precursor solutions.

The reduction in a hydrogen stream followed an internal standard procedure to guarantee high activities in the fresh state, **Table 2**. We simulated the load caused by the high temperature cycles for the carbon particle oxidation in the carbon particle filter by aging the samples for 24 hours at 850 °C. This treatment should simulate driving a distance between 100,000 and 200,000 km, dependent on the driving habits.

Table 1: Cordierit core coated with a pure aluminum oxide washcoat by Interkat GmbH, Königswinter (Germany)

Load Al_2O_3	110 g/l
Water Adsorption Al_2O_3	0.5 l/kg
Volume	0.037 l
Channel Densitiy	36 cm^{-2}
Diameter	25 mm
Length	74.5 mm

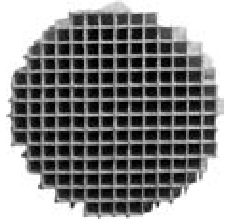


Table 2: Pretreatment of samples and conditions of measurements

Reduction	with H_2, T = 200 °C, t = 3 h; V = 22.5 l/h
Conditioning	in air, T = 450 °C, t = 6 h; V = 22.5 l/h
Ageing	in air, T = 850 °C, t = 24 h
Model Gas	500 ppm CO, 5 % O_2, rest N_2, VVH = 50,000 h^{-1}

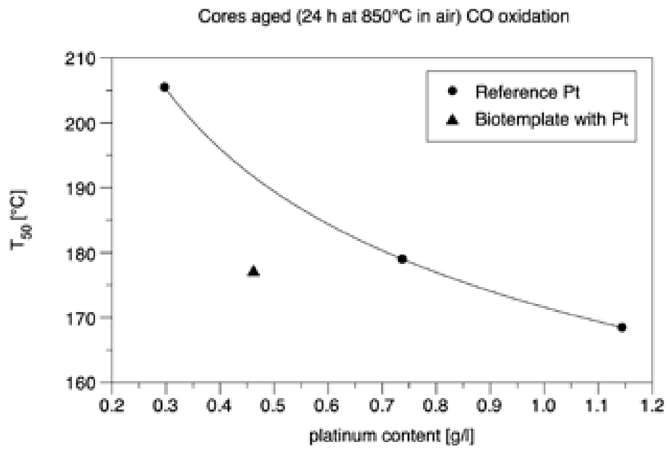


Figure 4: Comparison of light-off-temperatures of an impregnated core using biotemplates with reference samples without biotemplates

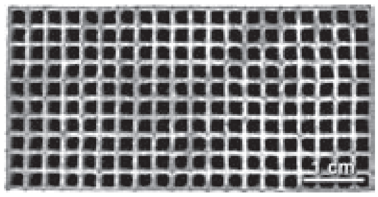
The catalytic activity was measured using a Horiba SIGU Reactor at the Queen's University of Belfast, School of Mechanical & Aerospace Engineering. Figure 5 lists the experimental conditions. The carbon monoxide oxidation was measured using an infrared spectrometer (FTIR) and the data used to determine the T_{50} temperatures. We plotted the T_{50} temperatures against the measured precious metal concentrations, **Figure 4** shows the results. The curve fitting for the control values was done using the formula T_{50} = a/log((x-b)/c) with x denoting platinum concentration. We determined the numeric values for a, b and c in the above formula and found a very good fit for the curves. With regard to the T_{50} temperature the comparison between the biotemplate sample and the control sample shows that the biotemplate sample contains 42 % less precious metal at the same light-off temperature.

To ascertain the reproducibility we measured the carbon monoxide oxidation catalyzed by the biotemplate samples using the "Multi Channel Monolith Reactor" [3, 4] at the University of Technology in Darmstadt, Germany, Institute for Technical Chemistry II [3, 4]. This high throughput technology allows it to measure different precursor formulations within a single honeycomb body. The individual channels of the honeycomb body are impermeable to water and every individual channel can therefore be impregnated and used as a flow-through reactor for catalytic measurements (Cordierit 410 from Inocermic, **Table 3**.

Five channels were coated with a biotemplate-containing precursor and the catalytic conversion was measured as a function of the pH. The control samples were identical to the samples except that the biotemplate was omitted. They were also prepared in the same honeycomb body. As described above for the core pretreatment, the honeycomb body was subjected to a reducing hydrogen stream and then conditioned and aged before measuring the catalytic carbon monoxide oxidation for each channel, Table 2. To deduce how much platinum was saved we generated fitted curves for three control samples, which were prepared using different platinum concentrations. (The curve fitting procedure is described above for Figure 4. We compared the parameters for the biotemplate samples and the control samples and calculated the precious metal required to achieve the same catalytic conversion. **Figure 5** shows how much precious metal was saved as a function of the pH of the precursor solutions. In the pH range between 8 and 10 about 50 % precious metal was saved. The results show a high degree of reproducibility.

Table 3: Monolith coated with pure aluminum oxide used for multi-channel-screening

Load Al_2O_3	105 g/l
Water Adsorption Al_2O_3	0.5 l/kg
Volume	0.135 l
Channel Density	11 cm^{-2}
Length	75 mm



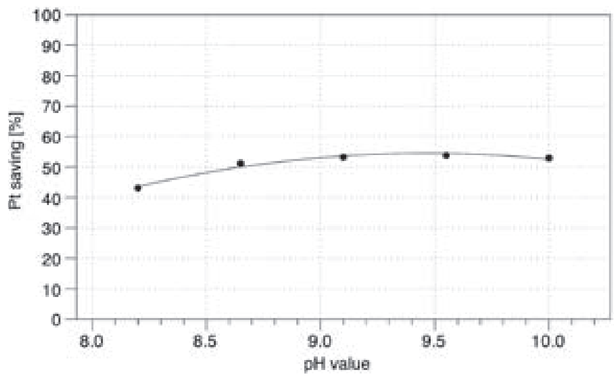


Figure 5: Platinum savings of channels prepared with precursors using biotemplates with regard to reference channels prepared without biotemplates (after aging at 850 °C for 24 hours in air)

4 Feasibility to Use Biotemplates in the Production

Like for any new process additional questions must be answered after the successful proof-of-concept and before biotemplates can be commercially used to reduce the amount of precious metal in catalytic converters for car emissions. Aspects requiring further positive answers are the risks, including the risks after long-time use, the proper reproducibility, the assured availability of all raw materials and the cost-effective implementation.

Even though not all analyses have been done yet, biotemplating shows a lot of promise to meet all requirements. Biotemplates are only used temporarily. Except for traces, the biotemplates are absent from the final product. Therefore there is little risk of influencing other components and a negative long-term impact is highly unlikely. While borrowing the method of modern molecular biology and biotechnology to configure industrial surfaces is a completely new concept, the procedures themselves have been long established even for commercial application. As scaffolds to trigger the depositing of precious metals in the right configuration, the amount required per batch is no more than a few grams. Different from the precious metals, the raw materials for the production of biotemplates are readily available and cheap. The procedures draw on already established methods, and a significant rise in general costs is therefore unlikely. In the long run there are even prospects of reducing costs because the complex nature of biopolymers and their properties may lend themselves to the design of more streamlined processes.

5 Summary

Bio-molecules can be used in the manufacture of catalytic converters for car emissions as interim scaffolding agents to improve the proper catalytic configuration of precious metals in the nano scale dimension. Right now, the effect of these biological templates is based mostly on the pore selection, which prevents the penetration of precious metals into micro and meso pores. Due to their high activation energy, these pores do not participate in the catalytic conversion. Micrographs of carrier substrate sections have shown the positive effect on the pore selection. Measurements of the catalytic conversions achieved with platinum-coated cores have shown that about 50 % platinum can be saved through the use of biotemplates. We used identical cores prepared without biotemplates as controls.

So far, little attention has been given to reduce the amount of platinum needed for the production of catalysts. The presented method introduces the commercial use of biotemplates to reduce this amount of platinum. We have reason to expect few if any increases in system costs when the new technology is implemented, since the innovation uses established procedures and can be applied to other serial productions of catalysts. The savings in platinum are however significant.

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personal buildup for Force Motors Ltd.

Carbon Monoxide

The New Emission Challenge for Diesel Passenger Cars

The intensive efforts being undertaken to reduce consumption against the backdrop of tighter NO_x limits are driving the carbon monoxide (CO) component in diesel emissions to levels which demand increasing attention from developers of diesel engines. IAV GmbH analyzed the various influences on CO tailpipe emissions and, on the basis of simulation and engine testing, describes here a number of approaches that employ targeted calibration measures.

1 Introduction

Given the principle on which diesel engines operate, their CO raw emissions are very low. The use of diesel oxidation catalysts (DOC) has so far made it possible to remain below emission limit values. However, current trends, as shown in **Figure 1**, lead to a rise in CO emissions and reduce the efficiency of catalytic converters in the New European Driving Cycle (NEDC).

Although introduction of the Euro-5 standard leaves CO limit values unchanged at 0.5 g/km, these trends, coupled with the tighter demands placed on the durability of emission-reducing devices, are posing a growing challenge.

2 Formation of CO in the Engine

The formation of CO is an intermediate step in hydrocarbon combustion. Subsequent oxidation to CO_2 in the course of combustion largely takes place by means of the OH radicals formed in the combustion zone. In addition to a sufficient supply of oxygen, this process also demands high temperatures (T > 1200 to 1500 K).

In zones of local oxygen deficiency at air/fuel mixture $\lambda < 1$ (for example from incomplete mixture formation in conjunction with high EGR rates) and as a result of CO oxidation "freezing" in low-temperature zones (for example where combustion takes place in areas close to the wall), the formed CO cannot be further oxidized [3]. However, charge motion into lean zones reduces the concentration of CO as stoichiometric conditions permit completion of the CO oxidation process.

3 Exhaust-gas Aftertreatment with Diesel Oxidization Catalysts

The DOC is employed as standard in diesel passenger cars for reducing CO and HC emissions using exhaust-gas residual oxygen. The catalytically active precious metals, primarily from the platinum group (PGM), that are chemically fixed on a porous substrate, are crucial for this oxidation process. Particular requirements for DOC formulations are high specific conversion rates and early light-off behavior at low exhaust emission temperatures (T < 180 °C).

This is mainly influenced by precious metals loading and dispersion. Here, pressure of cost on manufacturers from high prices for precious metals is leading to a growing need for new developments and optimizations of current technologies. Thermal aging from sintering and contamination of the active sites, for example as a result of sulfation, reduce DOC activity during real-world vehicle usage.

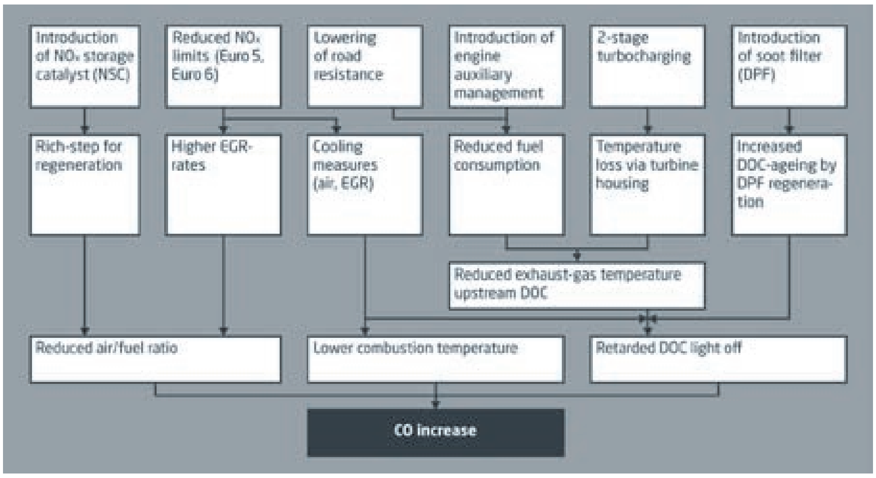

Figure 1: What causes CO to rise

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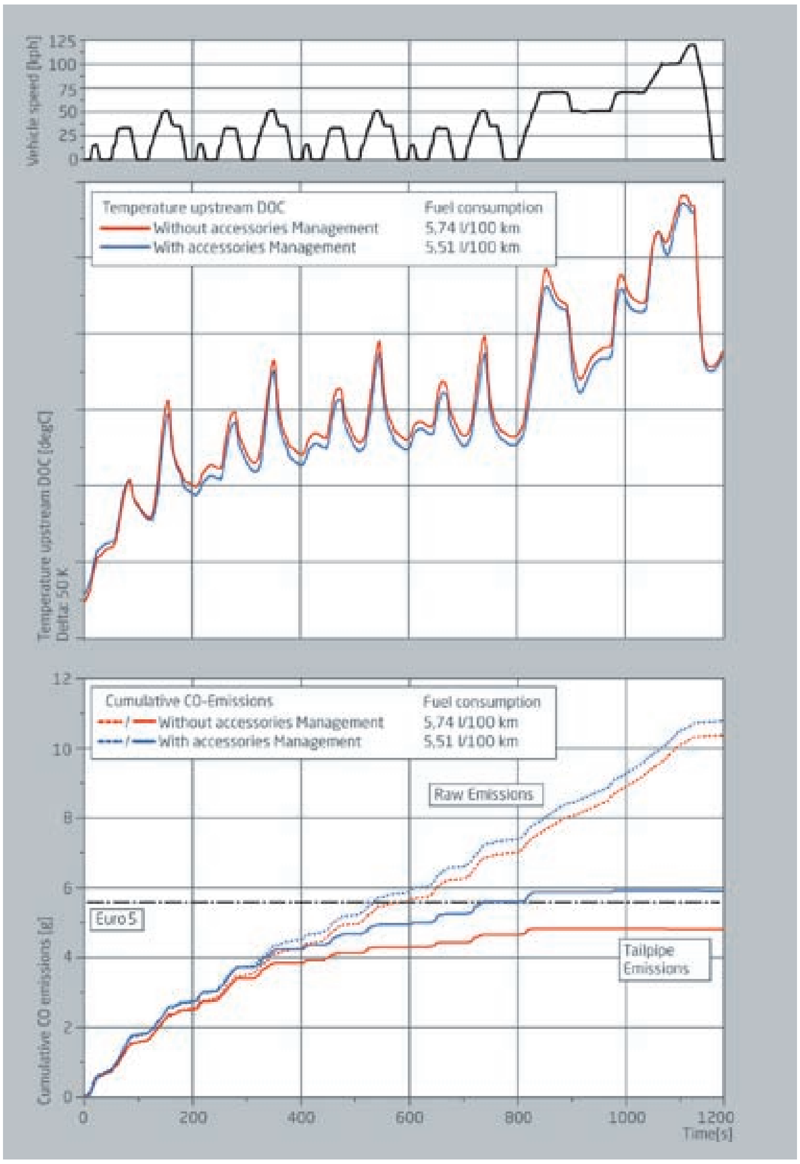

Figure 2: Influence of auxiliaries management on CO emissions and exhaust-gas temperature upstream of the DOC

4 Test Setup and Baseline Measurement

The basis for this study is a common-rail diesel engine of the 2-l displacement category, equipped with two-stage turbocharging and Euro-5 emission control technologies. It is applied to a vehicle with an inertia weight class of 3500 lbs that uses an auxiliaries management system to reduce fuel consumption. The exhaust system used is a hot-end „close-coupled“ system consisting of a DOC and a diesel particulate filter (DPF).

The auxiliaries management produces an appreciable cut in consumption of 3 to 5 % in the NEDC. However, this leads to a massive 25 % increase in CO tailpipe emissions, taking them beyond the Euro-5 CO limit values, due to the small drop in exhaust gas temperature of 5 to 10 K in the phase prior to DOC light-off. **Figure 2** shows the relevant temperatures upstream of the DOC and the cumulated CO emissions upstream and downstream of the DOC for production calibration with an aged exhaust emission system. For the version with auxiliaries management system (basic) used in the comparison, the DOC shows an CO conversion of only 45 %; the presented comparison variant without auxiliaries management reaches 54 %.

The following sets out to demonstrate approaches to reduce CO emissions in the NEDC without affecting the CO_2-reducing measures of the auxiliaries management system. These approaches are based on engine calibration methods, which in turn are supported by simulation of the exhaust system, and are neutral with respect to fuel consumption. In order to achieve a high level of test comparability, free from influences from either driver, vehicle or boundary conditions, the NEDC is realized on a dynamic engine test bench using an engine speed-torque profile. This methodology enables as many as eight NEDC measurements a day with very good repeatability [1].

5 Calibration-based Measures to Reduce CO

In general it is possible to distinguish between two paths for reducing CO with nine measures:

1. Reduction of CO raw emissions by:
   - 1.1 Optimization of pilot injection quantities
   - 1.2 Shortening of pilot injection intervals
   - 1.3 Optimization of rail pressure
   - 1.4 Increasing of air mass
2. Increasing exhaust-gas temperature to improve CO conversion by:
   - 2.1 Retarding of main injection
   - 2.2 Reduction of rail pressure
   - 2.3 Splitting of main injection into two injection events (so-called split main)
   - 2.4 Activation of the throttle valve while coasting
   - 2.5 Activation of exhaust-gas recirculation (EGR) while coasting (including the necessary throttle valve control).

5.1 Reduction of CO Raw Emissions

Proceeding from a Design of Experiments (DoE) model, **Figure 3** shows the various influences on CO raw emissions for two different coolant water temperatures (T_{CW}) during steady-state driving at 32 km/h. In particular, the quantity and the interval of the first pilot injection (PI2) are seen to have a major influence on CO raw emissions, whereas the influence of rail pressure, start of main injection and EGR rate is less significant.

Strikingly, the effect of reducing the EGR rate on CO raw emissions is not as pronounced for the cold engine as it is for the hot engine. This is related to the EGR cooler bypassing in the warm-up phase, enhancing CO oxidation.

Boundary conditions for Euro 5, noise etc. severely restrict the potential of the options given in the measures 1.1 to 1.4 and 2.1 to 2.2. The parameters of pilot injection quantity and interval are already optimized in the range of low CO emissions.

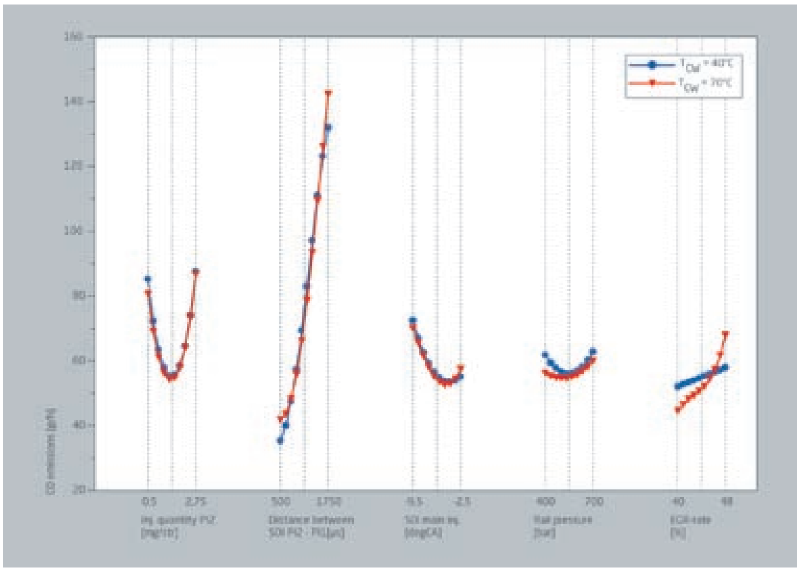

Figure 3: DoE modeling of the various influences on CO raw emissions

5.2 Increasing Exhaust-gas Temperature

Initially, simulation can be used to illustrate the influence exhaust-gas temperature has on DOC efficiency and pre-select measures suitable for heating up the system. To this end, a 1-D simulation is used to establish the simulation baseline by modeling the NEDC measurement with auxiliaries management using the AxiSuite simulation tool [2]. Further-reaching simulation studies proceed from the demonstrated model accuracy of $\Delta T = \pm 10$ K in relation to temperature response, and ±5 % for the cumulated emissions.

Figure 4 (left) shows an analysis of potentials for the NEDC. As the computations demonstrate, even small rises in mean exhaust-gas temperature of 20 – 30 K are capable of improving CO conversion by as much as 70 % in the overall cycle. The heavy impact of temperature on conversion rate is attributable to operation in the light-off temperature range. This is why, in addition to ensure early heat-up, it is necessary to keep DOC temperature above the light-off temperature. If the flow distribution at the DOC inlet is not homogeneous, three-dimensional DOC simulation coupled with CFD may be introduced to improve light-off prediction.

In a subsequent step, a heating strategy is defined that aims to increase temperature without adversely affecting consumption or NO_x emissions. This is presented in Figure 4 (right) and clearly shows that heating in the coasting and idling phases combined with

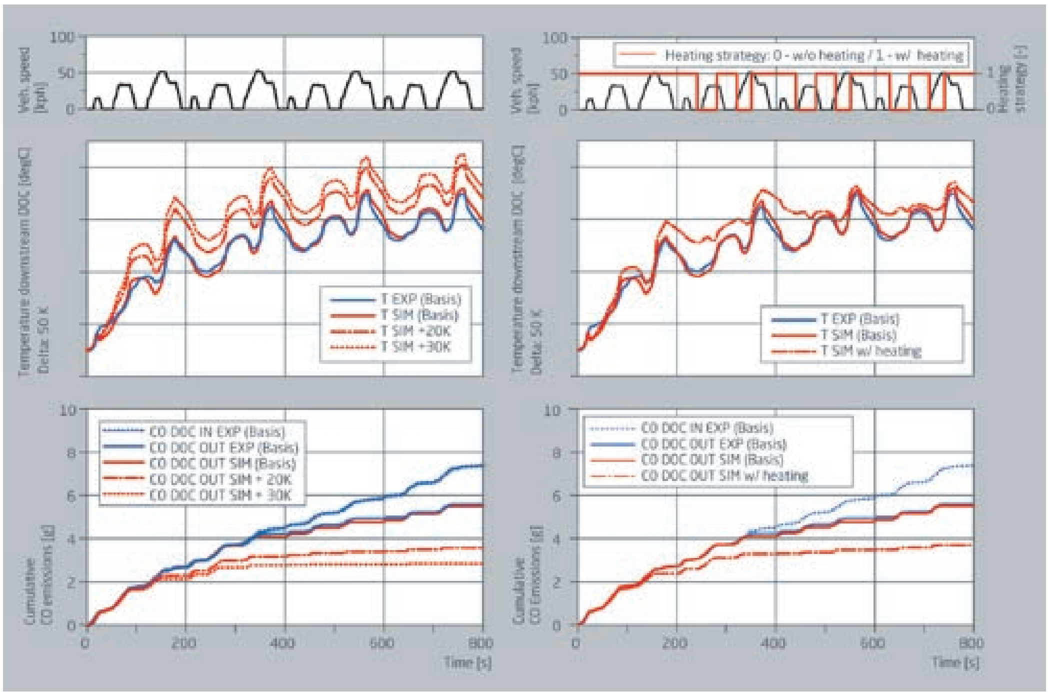

Figure 4: Examination of the influence of temperature on CO tailpipe emissions in the NEDC by means of simulation

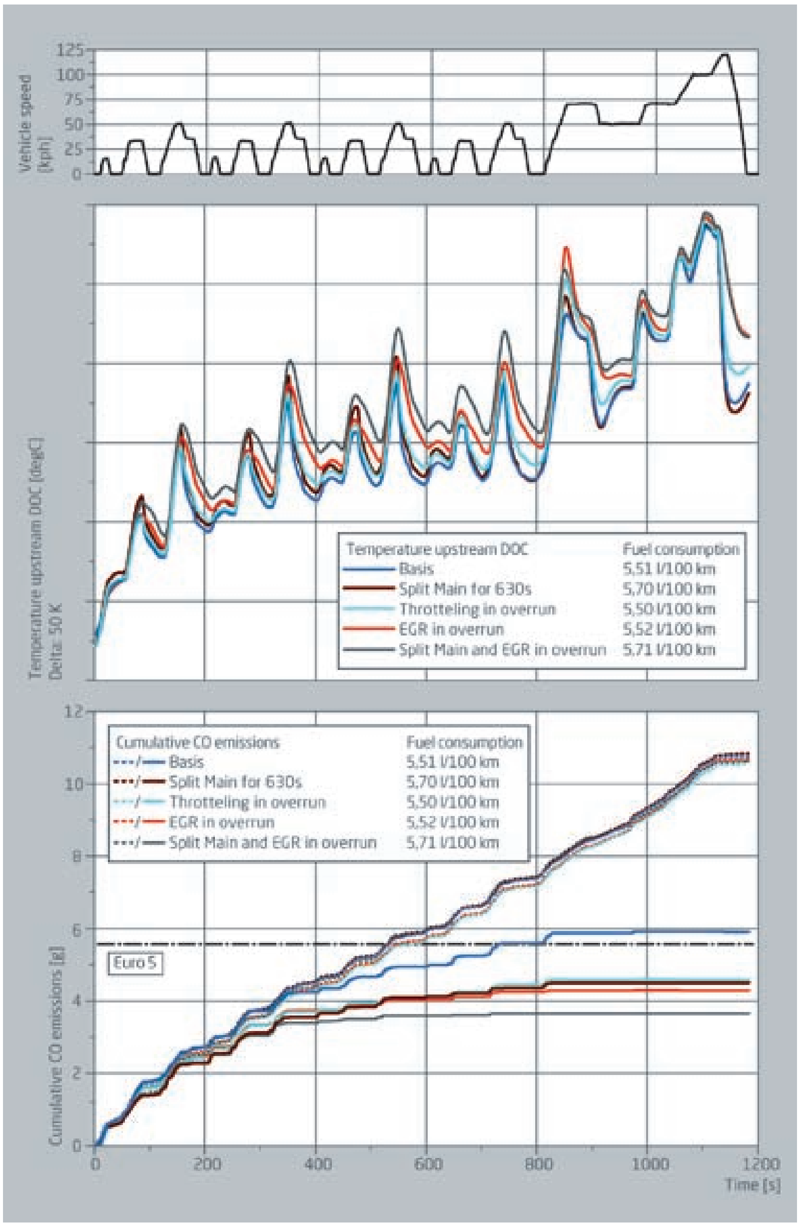

Figure 5: Calibration-based measures for reducing CO tailpipe emissions

rapid system heat-up does produce the desired rise in temperature. However, although higher peak temperatures are produced by a heating strategy that only affects the acceleration phases, it fails to prevent the DOC from cooling down.

5.3 Implementing Measures by Way of Calibration

In the following, the heating strategy proposed by simulation is implemented into the calibration using measures 2.3 to 2.5. The suitability of the measures is assessed for vehicle operation, **Figure 5**.

5.3.1 Measure 2.3, 2.4 and 2.5

Splitting main injection into two injection events in measure 2.3 is capable of increasing exhaust-gas temperatures by 10 – 15 K. To minimize the anticipated higher level of consumption, this measure is restricted to the light-off range (split main deactivated after 630 s). At constant NO_x emissions, CO conversion increases to 57 %. The resultant 3 % rise in consumption, however, it not acceptable.

By activating the throttle valve while coasting in measure 2.4, the DOC temperature rises by 10 to 15 K. With raw emissions remaining unchanged, CO conversion increases to 56 % which is below the legal limit. In terms of CO_2 emissions, this measure shows no deterioration compared to the basic measurement. Throttling intake air, however, leads to a drop in system pressure, which might be limited as a result of component constraints.

As mentioned in measure 2.5, opening the EGR valve and partially closing the throttle valve (through the EGR control strategy) while coasting causes temperatures in the DOC to rise by 20 – 25 K. As a result, CO conversion increases to 60 %. This measure is neutral in its effect on fuel consumption and therefore on CO_2 emissions.

5.3.2 Combination of Measures 2.3 and 2.5

If both measures 2.3 and 2.5 are combined, it is possible to estimate the potential for maximizing the rise in temperature in the DOC (the higher fuel consumption is being ignored for this specific assessment). Splitting main injection into two injection events in combination with EGR and throttle valve while coasting produces a temperature increase of 30 K; CO conversion rises to 66 %.

Activating EGR or throttling while coasting can reduce CO tailpipe emissions to a sufficient extent. As these measures are neutral with respect to fuel consumption, per this assessment, they are the preferred means. However, it needs to be borne in mind that further steps are necessary to maintain drivability, response behavior and controllability of EGR. Any negative effects need to be minimized by appropriate calibration and developing the requisite algorithms.

6 Summary and Outlook

IAV GmbH's development process uses methods of calibration and simulation to identify and implement measures for diesel engines suitable for reducing CO while leaving consumption unchanged, **Figure 6**. This article illustrates various approaches that prevent exhaust-gas temperatures falling as a result of measures to reduce consumption while at the same time en-

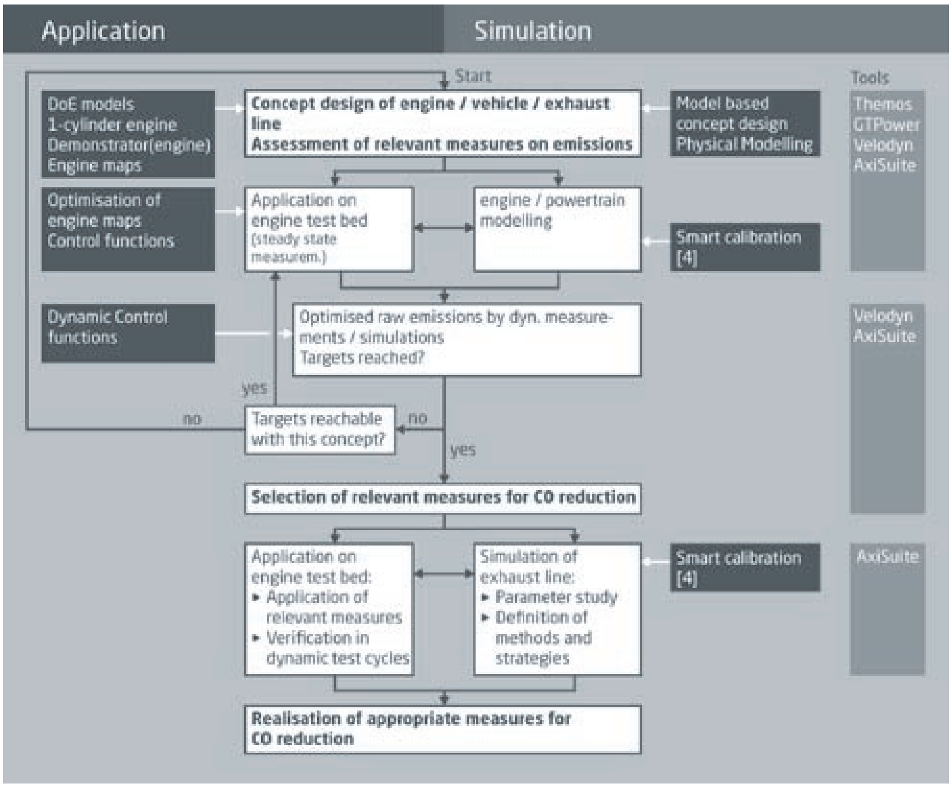

Figure 6: Interaction of calibration and simulation in IAV's development process

suring that Euro-5 limits are met. When applying measures that "reportedly" do not affect calibration, it must be remembered that even a slight temperature drop in the diesel oxidation catalysts' (DOC's) light-off range can lead to a significant rise in CO tailpipe emissions. The development process detailed is also used for Euro-6 calibration for which the described trade-off is even more pronounced.

To ensure an efficient usage of the DOC system throughout the vehicle life time and, in particular, detect aging-induced changes in the DOC's conversion performance, IAV GmbH is also pursuing approaches to identify light-off during actual vehicle operation and realize them in open and closed-loop control strategies.

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Efficient Turbo Charger Testing

Charging/boosting is one of the most important measures to improve engine efficiency, and turbochargers are the state-of-the-art charging device today. That is why AVL widens its portfolio and introduces such test systems to the market. In the following, these systems by AVL for Saab/GM are introduced. Since OEM's focus on charged engines, Saab needs similar testing equipment to the turbocharger manufacturers. This is required to develop and maintain the knowledge to specify characteristics and to drive the technology forward.

1 Introduction

The pressure for permanently increasing efficiency of combustion engines resulted – among many other measures – in the rising importance of engine charging. Forecasts say; within a decade only few combustion engines in automotive use will be naturally aspirated. A share of nearly 74 % of all automotive engines is predicted to be turbo charged by 2014 [1].

Saab/GM is one of the pioneers in gasoline engine turbocharging. More than 30 years ago Saab introduced turbocharged engines to the family performance car market; forming the "rightsizing concept" (Saab 99 Turbo). The growing availability of high-octane alternative fuels (such as E85 and bio-gas which both have octane numbers much higher than premium gasoline) together with the popularity of the diesel cycle engines, further strengthens the turbochargers popularity on the market. Within GM Europe the Trollhattan development site is responsible for charging predevelopment and turbocharger component testing. In 2006 Saab/GM decided to improve their testing facilities and searched for a turbocharger test system. At the same time AVL Instrumentation and Test Systems started developing such a system due to the evident engine trends. Saab/GM in Trollhattan is a pilot customer for this test system and closely escorted the development process from the start until today. Sourcing took place based on the good experiences with the „GM One Lab" approach, where AVL was chosen as system supplier for all GM testing centers world wide.

To meet the Saab/GM requirements AVL decided to establish an exclusive co-operation with Swissauto Wenko AG in Switzerland. The Swiss charging specialists contributed with thermodynamic experience, everyday testing knowledge and a lot of technical expertise; hence building the know-how backbone from the testing & application side. Before that Swissauto had been used by Saab/GM as an independent supplier of third-party measurement data of turbochargers; the simulation methods for engine process simulation for turbomatching are based on those measurements. In parallel the AVL Schrick Boosting Competence Centre emerged in Remscheid. Services for development, thermodynamic simulation and prototyping for turbochargers and their components are offered. The turbocharger test rig of the same specification as discussed in the text below with a thermo-shock device completes the portfolio for qualified measurement and mapping services.

2 Reasons for OEM to Operate a Test Rig

Saab/GM targets to purchase a turbo charger test rig are:

- acquire reliable and traceable data for the virtual development of power-train systems
- understanding actual technology and investigating in innovation-steps
- research for improvement of charger and engine simulation models
- avoiding noise in finished products
- benchmarking of supplies and products in the market
- verifying suppliers quality
- having a tool to generate specifying-knowledge enough to discuss turbocharger design with the suppliers on a high level
- reproduction of vehicle fleet problems or failures of charging systems isolated from the combustion engine
- easy and standardized exchange of data to existing test facilities and simulation environments.

This leads to the main requirements for a turbocharger test system:

- high accuracy and excellent long term repeatability of measurement results
- full control of how turbocharger data was acquired and the related accuracy
- high stability of operating points for shortest possible measurement times
- upgradeability with functional options beyond standard mapping features
- possibility to develop methods to measure compressor maps to cover the entire speed range
- wide range of mass flow to cover the required flow range of the turbine
- possibility to develop measurement methods beyond standard mapping, like for instance measurement of two-stage and other combined charging systems
- state-of-the-art Automation System and testing application for intuitive use
- high availability through well performing maintenance and spares services

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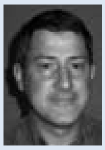

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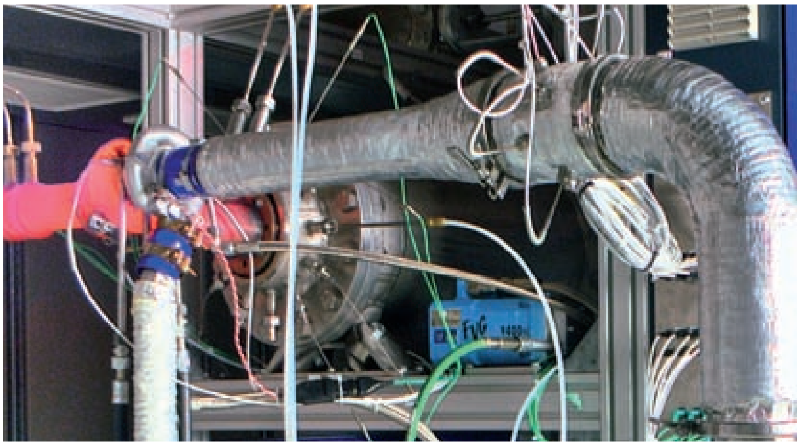

Figure 1: Measurement pipes; V-Clamp quick-connectors

– the freedom of a powerful and versatile hot gas flow rig for tests of various flow components such as EGR-coolers, charge-air coolers, duct systems etc.

3 The Test Rig

The hot gas turbocharger test rig is considered as a state-of-the-art testing device; consisting of a pressurized air supply, the combustion gas source, pipes, conditioning systems, sensors, controllers and actuators. Adding the automation and data acquisition system, the testing application and data evaluation features (test algorithms, formulas, characteristic maps, limits and subsequent systems reactions) make the test system complete. The testing efficiency of such systems is defined by the ratio of overall test bed hours vs. productive hours; meaning the time of testing operation producing exploitable measurement results. Aside from the general targets for a test system, like accuracy and dependability, versatility was asked for by Saab/GM. This means a system with open source code, as well as a modular design to enable results of methodology development to be built into the rig's hard- and software.

Saab/GM has significant experience in working with turbocharger data originating from different sources, and knows how this makes good back-to-back comparisons difficult. Thus the first project in the rig is to develop a robust method to measure turbo maps that can be reproduced in other rigs, and that give the relevant data for component evaluation and process simulation. The results of this must be possible to program and build into the rig after delivery. To be sufficiently versatile the rig must also be able to actuate the turbocharger (most common the VGT settings) via signals which are common in vehicle use (e.g. PWM). Furthermore the rig has its own highly precise electrical actuator that can replace the turbochargers actuator which enables many different types of tests to be executed.

To assure the investment the rig helps Saab/GM to solve other tasks in addition to turbocharger characterization; such as flow component testing (EGR-coolers, cylinder head ports, piping etc.), research projects where the engine environment is not precise enough like pulsating flow, as well as improved hot gas cleanliness to facilitate detailed optical flow measurements. Furthermore the rig facilitates enough space around the test unit in order to run it with adjacent piping and components to measure their impact in the units' ideal performance.

As a result of the huge variety of geometrical charger and manifold interfaces, charger testing will stay a challenging task and requires detailed planning, design and manufacturing of the interfacing parts. Because of this, a highly flexible set of piping with industry standard V-clamped flanges is used on the test rig. These are capable of being tightened manually in order to reduce the mechanical rigging effort but never the less offer the required geometrical degrees of freedom, **Figure 1**. The positioning of the sensors (T, p) is realized by placing standardized measurement pipes within the gas path without dis- or relocating sensors from the pipe or from their electrical connectors. This significantly reduces parameterization efforts and increases result quality by eliminating erroneous readings or damaged sensors as root causes of incomplete measurements.

Figure 2 shows the main GUI (Puma Open Graphical User Interface) screen of the test rig as part of the turbocharger application based on „Puma Open". All relevant charger and test system parameters are visualized online and ensure the safe and reliable operation of the test rig and

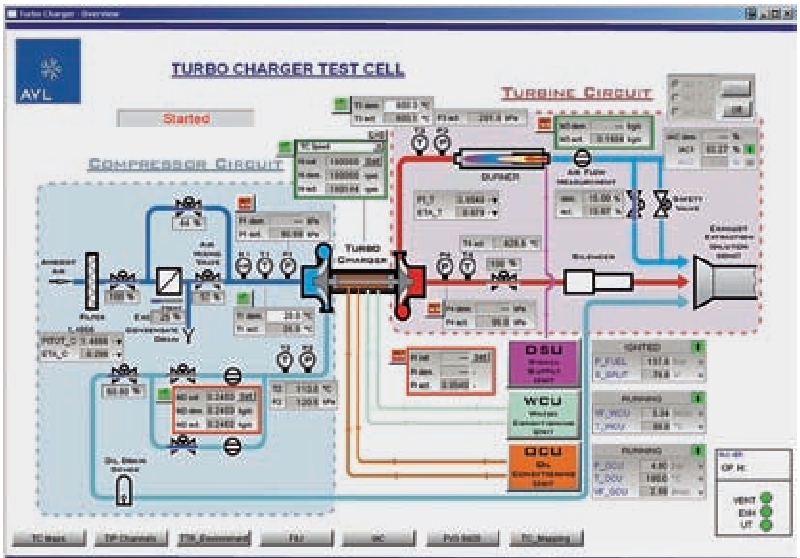

Figure 2: AVL Puma Open – turbocharger application GUI

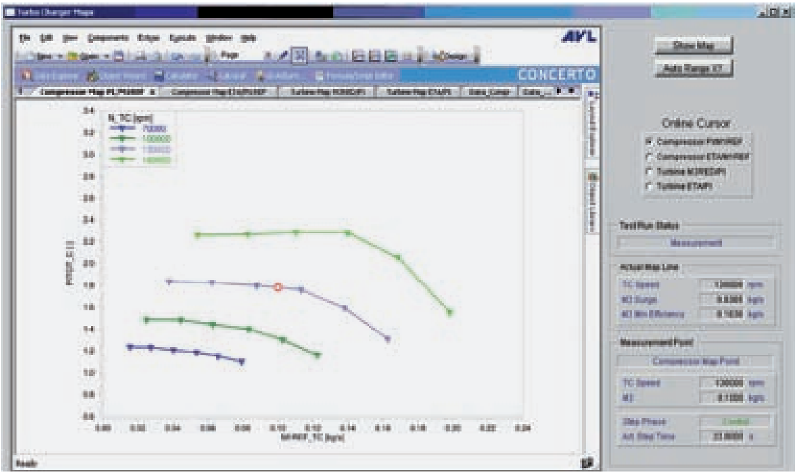

Figure 3: Online map with indicator of actual operating point

the charger. The test rig is equipped with an application package comprising of the turbocharger test systems control, test libraries, data evaluation and result evaluation layouts. The flexibility of the AVL solution allows Saab/GM to customize all graphical layouts of result data and user interface.

The online status indicator in the maps enables users to identify in which state the turbo charger is operated in, **Figure 3**. This helps in creating operators´ trust into the test system from the first day and improves the result quality because online comparison between the testing targets and the actual results is possible. Experienced test bed operators adding comments or new result processing methods to the saved data files, generate a deeper insight and knowledge to the results and thus generate further value to the entire test operation. This know how can be embedded into the test system. For instance, during mapping, the testing progress can be verified online as the predefined map layouts get displayed online while "growing" on the screen along with the real measurements.

Experienced users find all degrees of freedom provided by the Puma Open application. Test methods prepared in the office save valuable time at the test rig. Dynamic reloading of automatic testing procedures together with the possibility to modify settings during runtime allows a maximum of effective test rig operation and test execution. All relevant thermodynamic calculations are included in the testing application as well as standardized routines for online and offline data processing based on „AVL Concerto".

Several result evaluation layouts (e.g map styles) are supplied as part of the standard application and can easily be selected. Saab/GM is free to define further layouts in case of future demand.

A new feature in turbocharger testing is auto mapping. This function enables the operator to automatically create a map for any turbocharger. The operator uploads or defines safety limits (such as max allowed speed and temperature, wheel size and number of blades) already with the definition of the turbocharger parameters prior to testing. For mapping purposes the number of speed lines and measurement points per speed line will be typed in a dedicated dialog window of the application screen. The test system automatically checks out the operating limits of the charger, defines the target maps and finally executes the mapping line by line. The stabilization criteria defines how a measurement point is recognized to be stable; band widths and gradients can be defined and edited according to the charger specification. This feature enables the operator to balance the accuracy requirements versus the expected testing time.

One of the key components of any turbo charger test rig is the burner system. Saab/GM first requested CNG firing due to cleanliness requirements for optical studies. The current diesel burner can fulfill even the tough soot requirements for the optical measurements Saab/GM does in the hot gas flow. **Figure 4** shows an example of the filter smoke numbers vs. temperature, several further opacity measurements have proved the very low values. The diesel burner reduces the facility requirements and eases the working place safety measures significantly. Gas compressors and gas handling devices are not required leading to a beneficial reduction of the utility cost. Diesel or gasoil is easily available in any testing location all over the world.

The fast control response of this burner type results in optimal temperature stability and enables economic testing operation by reducing stabilization time to the minimum. The burner allows high flow transients e.g. for cyclic speed profiles at unchanged T3; a stability of T3 better than +/- 6 °C has been realized at speed cycles with 5000 rps.

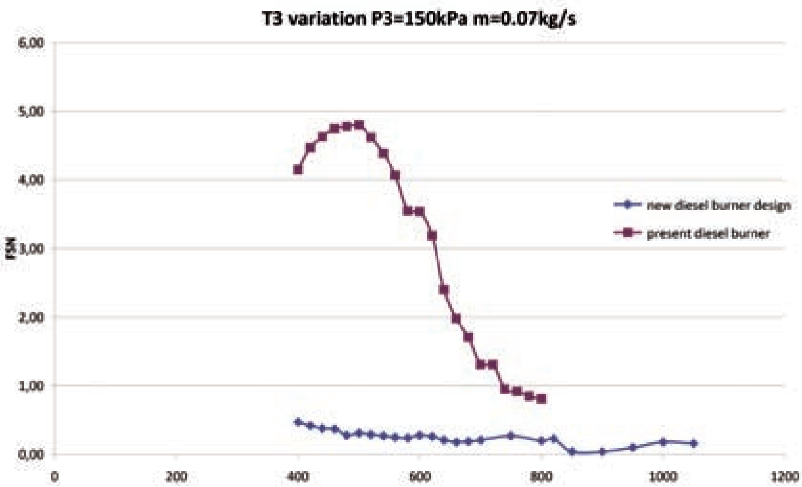

Figure 4: FSN comparison versus temperature

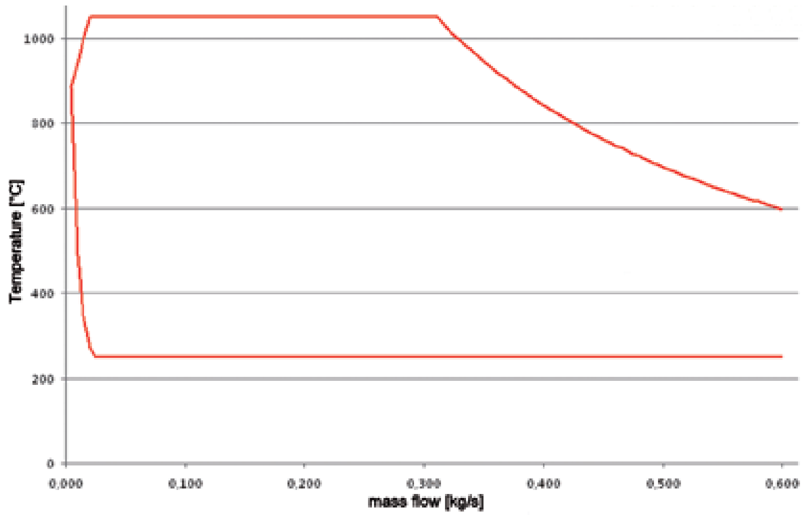

Figure 5: Diesel burner; temperature versus mass flow chart

Figure 6: Example for online monitoring for validity of sensor readings

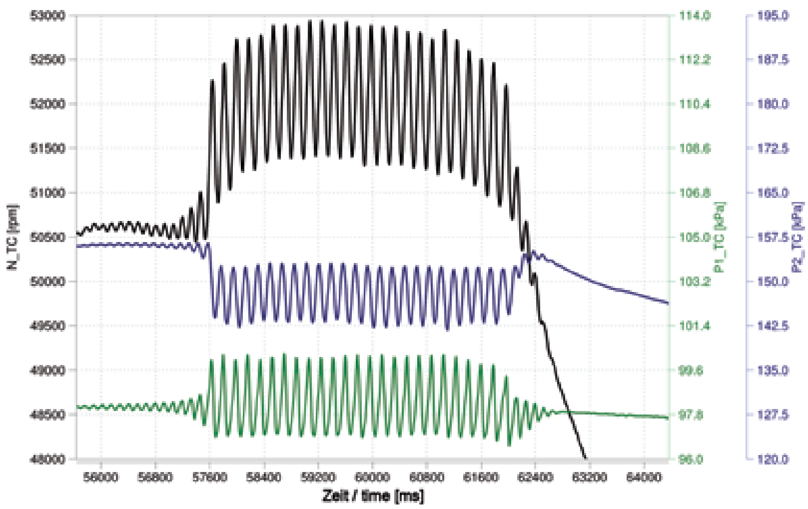

Figure 7: Sensor reading example during surge

The power range of the diesel burner, shown in **Figure 5**, exceeds actual CNG fired burners giving the widest range of use with a hot gas test rig. This is possible due to the use of a high range of fuel injection pressure.

It was decided to include the industrial air compressor directly as actuator into the burner control loop. Very low pressure pulsation and very good mean pressure stability at turbine entry are the gains, resulting in extraordinary speed stability of the settled turbine. At some measurement points the stability criteria might not be reached. In **Figure 6** the temperature distribution window can be seen representing the detection of an inappropriate measurement point. Subsequently the testing application re-approaches the target point in another iteration, takes the measurement and flags the measurement point if not sufficiently stabilized. The number of approaches can be defined by the user, giving another chance to balance the cost vs. accuracy conflict to the actual needs. This allows finishing an unmanned mapping processes even when stabilization is not possible in specific operating points. Users who want to follow different operating strategies can define their rules completely on themselves or verify the targeted points after the processing of the map in manual mode.

This freedom is found in the surge determination routines as well. In **Figure 7** an example diagram under compressor surge is shown. The actual surge determination method can be edited or completely defined by the operator. This meets a very important requirement from Saab/GM that the definitions and limits of surge and choke should be chosen freely; however counting on the current method as a reliable and experienced base. This is of high importance because neither a standard recognition procedure nor standardized measurement methods have been defined so far, even though versatile and stable methods have been proposed [2].

The systems´ compressor path starts with an intake air conditioning stage. This efficiently removes the variation of e.g. surge coming from temperature fluctuations which can not be corrected by calculation. This ensures a very good reproducibility of compressor maps over a long term even under changing environmental conditions.

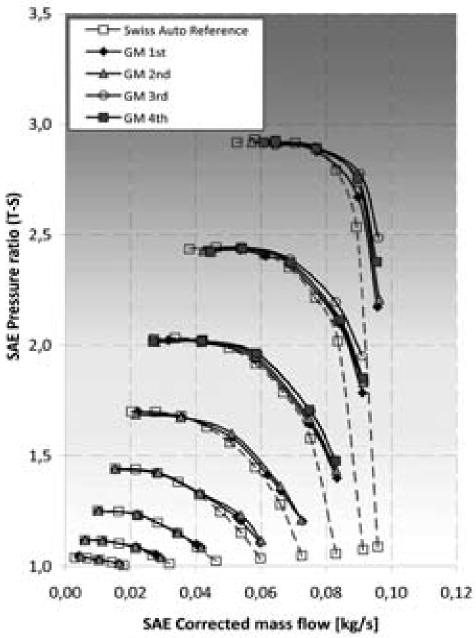

Figure 8: Good correlation example of repeated runs for the pressure ratio on Saab/GM and Swissauto test rigs

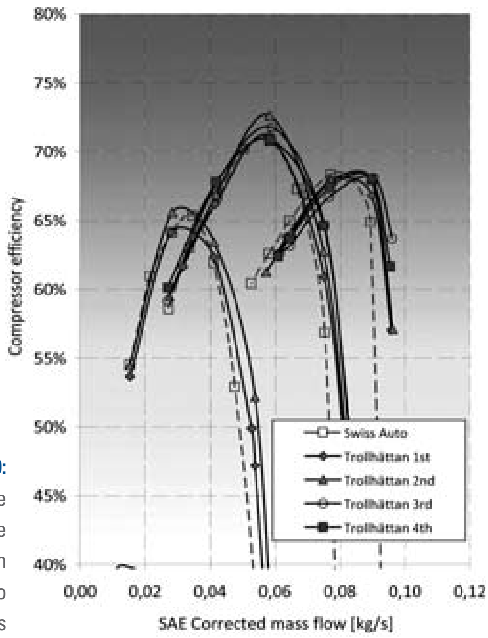

Figure 9: Good correlation example of repeated runs for the compressor efficiency on Saab/GM and Swissauto test rigs

Experienced testers know about the limited repeatability of hot film anemometers. Inevitable residues build up and require frequent calibration which is expensive and reduces the availability of the test system significantly. The measurement uncertainty if not calibrated according to the tough requirements is a matter of fact. AVL decided to use measurement devices named V-Cone [3] after the compressor exit as the differential-pressure based measurement shows a very good repeatability combined with a high tolerance against soiling. This results in very few calibration efforts and outstanding long term stability as shown in **Figure 8** and **Figure 9**. For covering the flow range of 0.005 kg/s to 0.6 kg/s the unequally paired measurement systems are used in a parallel arrangement and managed by the testing application. One-line monitoring of relevant conditions (e.g. mach numbers within the measurement tubes) enable the operator to monitor the measurement accuracy.

Efficiency and responsiveness of engine charging systems is undergoing permanent improvement. Mapping a single charger on a hot gas test rig gives quite good information about the characterization of the single component under standardized environmental conditions. Unfortunately this brings only half the truth when – like Saab/GM requested – the characterization of sequential charging systems is considered. Isolated measurement of the single chargers in single housings will require much simulation work to accurately predict the combined charging systems behavior. Thus, the complex charger systems require measurements as a complete system to more clearly define the effects of the subsystems on each other. The Saab/GM test rig can cope with this requirement since provisions are made and verified to completely control and measure two-stage chargers as a complete integrated system.

4 Outlook

To fully comply with the Saab/GM requirements the test trig will be equipped with further functions for extending the measurement possibilities in the current phase two of the project. These are a closed loop compressor circuit, a pulsating flow device and a thermo shock device.

For extending the turbine mapping range a closed loop compressor circuit will be introduced and completely fitted into the system. The test rig is already designed to completely contain this extension reducing the space requirements of the facility significantly. The high integration level is represented by very short process times changing between the closed and open loop measurements.

To understand the influence of pulsing flows and optimizing simulation models beyond characterization of chargers under steady flow conditions, Saab/GM required the possibility to test chargers operated in a pulsating flow environment. The test rig will be extended with a pulsation device providing flow and pressure alterations to the compressor, the turbine or both at the same time. Saab/GM needs this for improving the detailed understanding of dynamics to improve the simulation modeling quality as well as the performance of the turbomachinery under unsteady conditions. Currently most turbochargers are optimized for steady flow even though the turbine rarely operates in such manner. Smaller displacements and lower cylinder counts will stress this issue more in the future.

Thermo mechanical endurance testing with a twin charger arrangement will be included to the test rig. Fast transient turbine temperature alterations can be induced to the chargers to verify the effects of thermo mechanic stress to the life time of the turbine components like e.g. evolutes or blades.

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Experiences with the On-board Measurement Technology in the Field

The OBM technology is mor far reaching approach than the OBD technology. It allows direct monitoring of the exhaust gas emissions through a suitable measurement system in the motor vehicle. The pollutants CO, HC, NO and PM are measured quasi-continuously and the actual values are compared with stored values for each individual motor vehicle. Unlike the OBD, the emissions are monitored directly and exhaust gas deteriorations through unforeseeable errors or error combinations are recorded.

1 Application of On-board Measurement Technology

The EU has two guidelines which enable direct measurements at the engine and at the exhaust gas after teatment system, parallel to the on-board diagnosis system. These guidelines are 98/69/EG [1] and 99/96/EG [2].

In the guideline 98/69/EG paragraph 14, the European Parliament and the Council describe: "Further new provisions for on-board diagnosis systems (OBD) should be introduced with which a malfunction of the emission-reducing device of the motor vehicle can be recognized immediately and by which the originally emission level of motor vehicles being in service can be significantly maintained better through regular and random controls. The installation of an on-board measurement system (OBM) or of another system, which by measurement of the individual pollutant components of the emission recognize error functions, is permissible provided that the OBD system integrity is maintained."

This technology can be used successfully for a variety of tasks, e.g.:

- for reducing of fuel consumption through permanent monitoring of the combustion output at the engine
- for controlling emissions while driving, through measurement of the pollutant concentrations in the exhaust gas after treatment under natural conditions
- for the enhancement of the efficiency of eco-driving training measures, through visualisation of the effects of real driving behaviour
- for the allocation of ecological fees connected to the actual emissions to improve the environment
- for the installation of reduced pollution zones because of the actual output of the individual motor vehicles in traffic [3].

2 Sensor Technology

The current sensors function according to the following measuring procedures:

- change of the electrical resistance in a semiconductor, e.g. with a metal oxide
- change of the current flow through an electrochemical cell
- change of the heat conductivity in a pellistor
- change of the voltage at the gate of a mosfets
- change of the oscillation frequency in a resonator
- change of the absorption of light, infrared beam or microwaves [4].

The most important sensor procedure today for measuring exhaust gas in motor vehicles is the electrochemical technology with the λ-sensor for the measurement of O_2 and the similarly constructed NO_x sensor for nitrogen oxides [5, 6, 7].

The practical testing of on-board measuring technology for particles still needs to be done, particularly regarding the recording of fine particles [8].

3 Problem of Contamination of OBM Systems in the Field

The long term stability of OBM systems in the field is one of the most important conditions for their practical use. The automotive industry requires stability up to 100,000 km in private automobiles, and up to 400,000 km for commercial motor vehicles.

Most of the contamination forms a moist grey film of soot in the gas leading parts, which usually leads to a loss of sensitivity of the measurement signal and to a blockage of the free gas streaming. **Figure 1** shows the contaminated components of the analyser of an OBM system which was tested in the field in a long term test.

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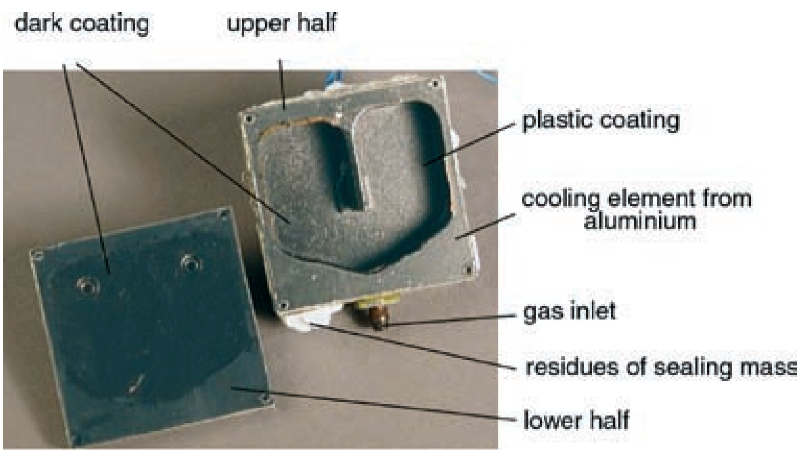

Figure 1: Contaminated micro gas cooler

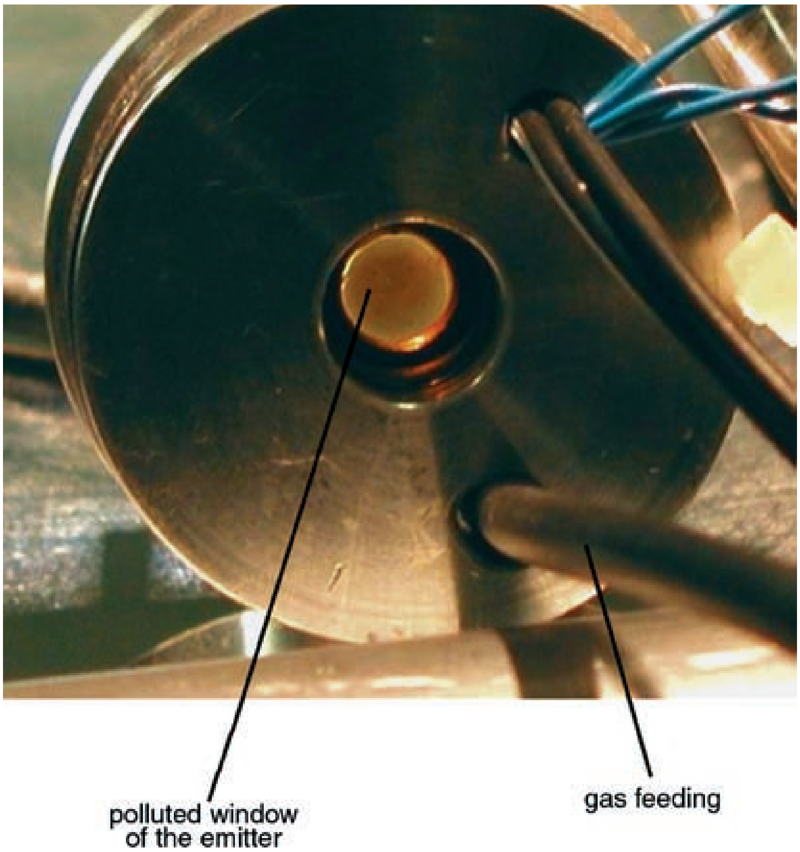

Figure 2: Contaminated emitter window

The contamination of the optical components leads to an especially strong influence on the signal quality. With the deterioration of the infrared window, resulting from normal driving conditions, the measuring signal constantly drops by an exponential function. Indeed at the beginning this loss of sensitivity can still be corrected by a zero phase calibration with ambient air. Eventually the device loses so much sensitivity, that the measurement must be stopped and the system has to be cleaned, **Figure 2**.

Automation has to take over this task, because the sensitivity and the zero point cannot be calibrated while the device is in use.

A suitable measure for the protection of the analyser is the installation of a particle filter into the exhaust gas flow, which is being measured. Manually exchangeable rough or fine filters may be used. It is also possible to use automated controlled, self-regulating filters.

The installation of a regenerative filter system in a light commercial motor vehicle is shown in **Figure 3**.

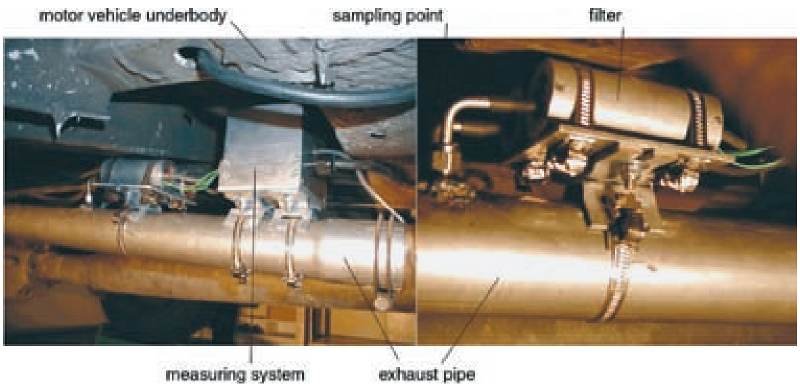

Figure 3: Installation of a regenerative filter

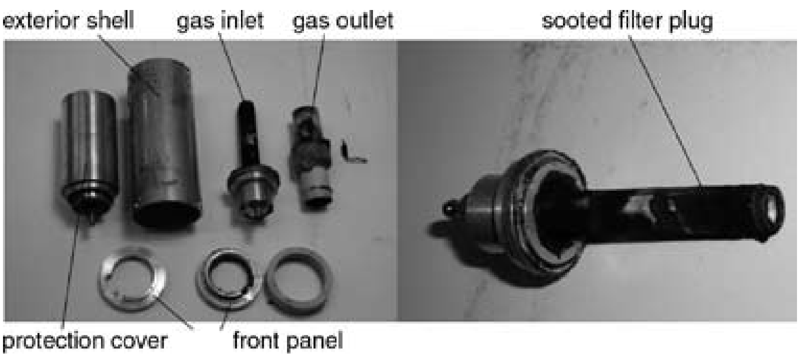

Figure 4: Soot loading of a ceramic filter cartridge before heating

Regenerative filters achieve a practically unlimited operation by switching and heating of the filter components and by the backwashing of the particle residues. The amount of particles in the gas flow determines the frequency of the automated regeneration of the filters through burning at approximately 800 °C, coupled with the back flushing of the particles into the exhaust gas stream. In a spark ignition engine, which meets the exhaust standards Euro 3 and 4, it is sufficient to regerate the filter every 2 to 4 weeks or after a few thousand km. In an older compression ignition engine the cleaning must be done every 2nd or 3rd day. In older ship diesel eniges it must be done hourly.

Figure 4 shows the soot loading of a regenerative ceramic filter with a medium pore size of 10 μm.

The regenerative filtering of exhaust particles is vital for the long term stability of OBM systems, even when the measurement of the exhaust gas quality is not done continuously, but only at intervals. Without regenerative filter components, OBM technology cannot fulfil the expectations of the automotive industry. All devices,

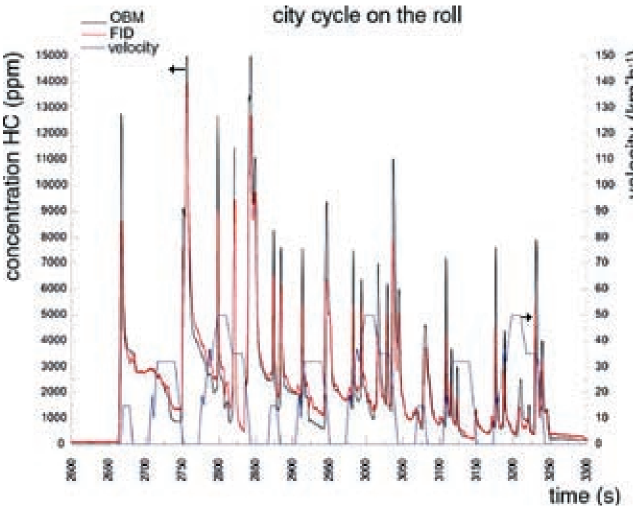

Figure 5: Comparison of the hydrocarbon concentrations with certified large analysers and with the OBM system

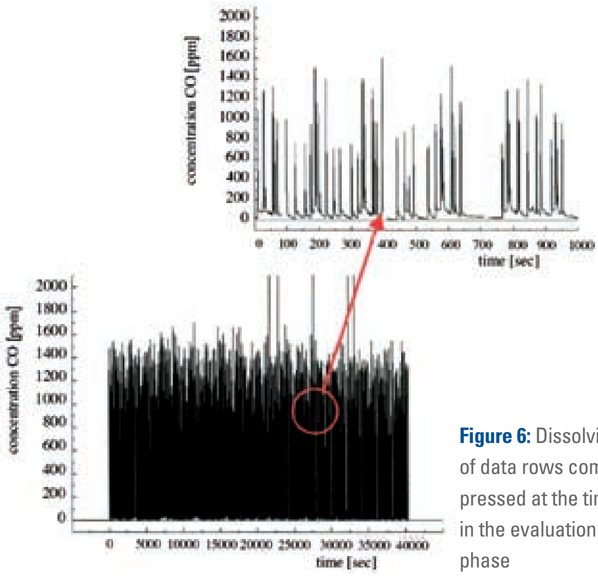

Figure 6: Dissolving of data rows compressed at the time in the evaluation phase

which function without renewable filters can be used for only short time applications and in a strongly restricted way, independent of the quality of the downstream measurement technology.

An exception is measurement done at high temperatures. In this area filtering is not needed because soot particle are automaticely burned. However, all other gases must be taken with a far more complex testing procedure. The use of micro-gas preparation systems is nearly always necessary for such sensor types.

4 Experiences with the OBM System

The use of an appropriately protected OBM system on the vehicle test stand, directly at the final exhaust pipe of the car, makes it possible to collect the undiluted exhaust gas. It temporally supplies high-dynamic results by not using unnecessarily long hoses. The parallel use of OBM technology could form the first step for the introduction of large analyzers at the test stand, which are being tested for type approval.

Some results of city cycle measurements on the roller test stand with the OBM system and with tested and certified large analyzers in the context of the CVS technology are shown in **Figure 5**.

In the practice very large amounts of data must be processed. During field experiments the parameters are compressed. These compressed parameters have to be resolved in steps of 1, 10 or 100 s according to the requirements of the evaluation, **Figure 6**.

5 Summary

The introduction of micro-system sensors can effectively support measures for the reduction of emissions and fuel consumption. However they require long term stable use. At this point further development efforts are necessary. These investments in research and development are worthwhile, since the importance of OBM technology will increase with the introduction of the EU 5 and EU 6 pollution standards.

Presently we lack the process engineering solutions for the monitoring of whose permitted values have recently been strongly reduced NO_x and PM_{10}. To examine emissions of heavy commercial motor vehicles, robust measuring sensors are needed, which can detect the following:

- NO, NO_2 concentration while driving, under normal loads
- fine particle concentrations, not the turbidity, in the exhaust gas during driving.

Today both procedures represent the weak point of monitoring technology. Nitrogen oxide (NO) molecules do not develop when engines are idling. Thus the current measurements do not supply usable statements about the quality of the SCR technology. The turbidity measuring technology, with the long proven opacimeter, can not analyse the concentration of fine particles from the modern diesel engines which have the new common rail technology. Here intelligent OBM technology is urgently necessary both for monitoring and maintenance.

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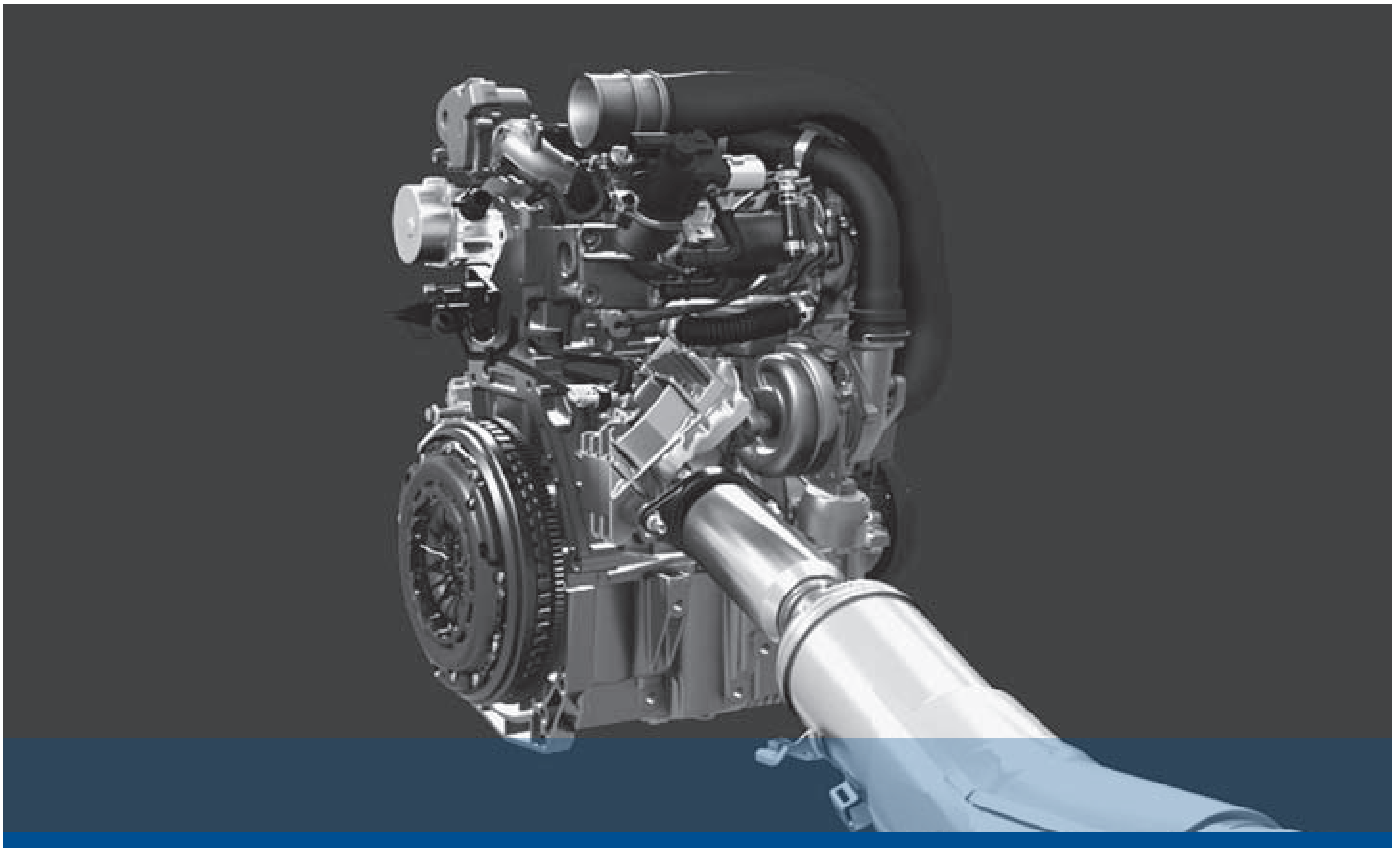

Exhaust Fuel Injection System for Efficient DPF Regenerations

Low speed driving profiles, like city ones, are very critical for passenger cars equipped with Diesel particulate filters (DPF). The main problem one must face is the significant oil dilution increase when using standard in-cylinder post-injection strategies. Following the experience gained in the field with the 1.5-dCi- Euro-4-engine with 78 kW equipped with DPF and exhaust fuel injection system (EFI), this device has been further optimized in the view of the new Euro 5 Renault's engine family. In comparison to the other solutions to reduce oil dilution (new combustion system design, innovative in-cylinder injection strategy) this system has demonstrated a high autonomy on urban cycles with a considerable improvement of the fuel consumption at low loads.

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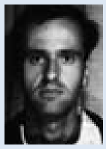

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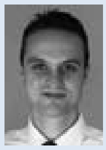

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1 Introduction

Low speed driving profiles, like city ones, are very critical for passenger cars equipped with Diesel particulate filters (DPF). According to the low exhaust gas temperatures in normal mode, the engine must generate a significant increase of the gas temperature upstream DPF via some specific air-fuel strategies (retarded injection pattern, reduced air mass flow, etc). One main issue is the use of post-injections that lead to fuel impingement on the cylinder liner walls and results in oil dilution that could damage the engine if the level exceeds a certain threshold (typically 8 %). However, it is not advised to avoid regeneration when the vehicle operates at low speeds as far as it will lead to DPF soot overload and substrate failures. In addition, the Euro5 regulation concerning the exhaust particulate emissions (PM < 5 mg/km) require the OEMs to avoid any crack of the DPF. Consequently, oil dilution must be controlled and reduced at a level that brings to the customer a high autonomy on this type of driving. For this purpose, Renault has carried out an extensive study of customer's driving profiles. For the oil dilution issue, the PAP cycle, **Figure 1** has been defined as the most severe one: the average speed is 14 km/h with a maximum of 50 km/h. The elementary cycle lasts 27 min and is 6 km long. One main feature of this driving profile is the very frequent and long idle phases that contribute to the oil dilution increase. It is representative of the so-called "delivery" or "postman" driving. Renault has characterized several Euro4 competitors vehicle on this cycle and compared it to the new Laguna III equipped with the 2.0-dCi-127-kW-Euro-4-engine, **Figure 2**. The cylinder displacement varies from 1.2 l (4 cyl.) to 3 l (6 cyl.) and the types of after-treatment systems are representative of the different philosophies already in mass production (close-coupled DOC + CSF, close-coupled

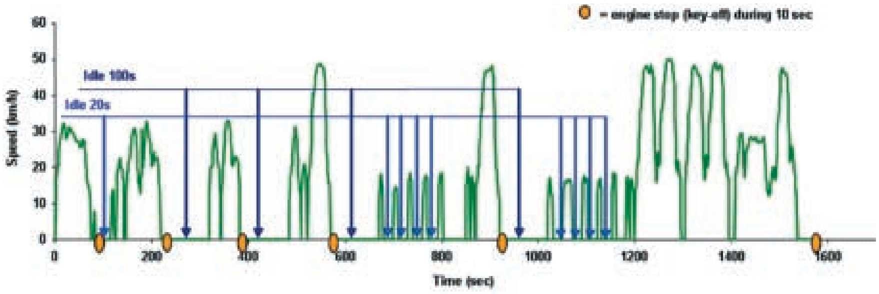

Figure 1: PAP driving cycle

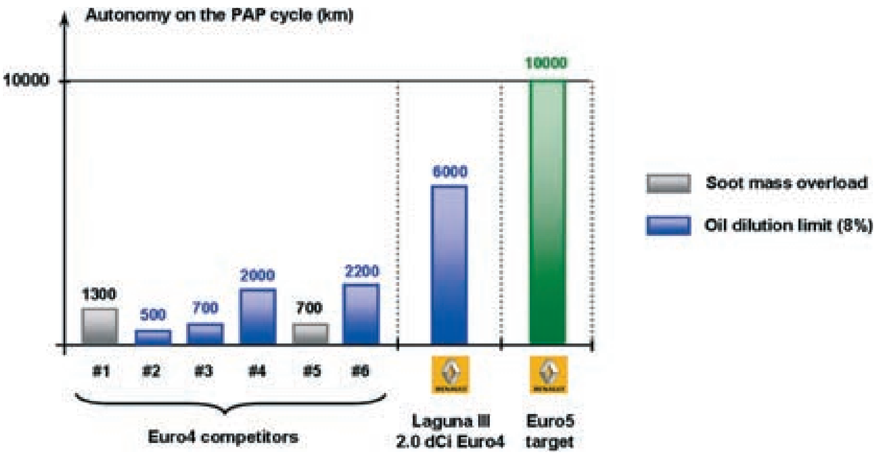

Figure 2: Comparison of the autonomy of Renault and competitors vehicles on the PAP cycle

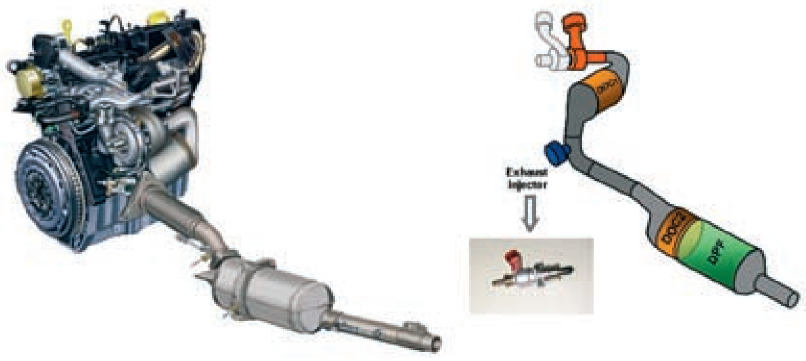

Figure 3: 1.5-dCi-Euro-4-DPF-engine (78 kW) equipped with the first generation of an exhaust fuel injection system

DOC + under-floor CSF, w/ and w/o additive). It has to be noticed that all these vehicles use post-injection strategies (up to two events per cycle) to provide active regeneration conditions at the DPF inlet. No specific devices to reduce oil dilution like exhaust fuel injection systems are considered. This comparison pointed out two philosophies of DPF strategies for this very low speed cycle:

- Some vehicles (like the Laguna) regenerate so as to avoid DPF soot overloading and thus DPF cracks. As a consequence, the autonomy is limited by the oil dilution criteria (a typical limit of 8 % has been used for this comparison). The optimization carried out by Renault on the new Laguna III leads to a rather high autonomy of 6000 km in comparison with the competitors (with a maximum of 2200 km).
- The others vehicles do no regenerate on the PAP cycle which leads to DPF soot overloading and DPF cracks. In this case, the autonomy is limited by the lightening of a MIL on the dashboard. A maximum autonomy of 1300 km has been measured which is not acceptable for the customer.

For the Euro 5 applications, Renault has settled an ambitious target of 10000 km for the autonomy on the PAP cycle. As discussed above, it was necessary to achieve such a level in order to provide to the customer a robust DPF system with a high oil drain interval and also to avoid any cracks of the substrate regarding the Euro5 PM regulation. To achieve this target, an oil dilution reduction of at least 40 % was necessary in comparison with the Laguna III 2.0 dCi Euro 4. Renault has already launched on the market an innovative DPF system with an exhaust fuel injection for small and medium size vehicles (Clio, Kangoo, Modus and Mégane). Since 2006, this DPF system is available as an option on the 1.5-dCi-78-kW-engine, **Figure 3**. The fuel injector is installed after the closed-coupled oxidation catalyst and introduces fuel that is burned within the small DOC which is placed upstream of the CSF. Thanks to this system it's possible to avoid (at least reduce) the fuel injected by the in-cylinder post-injections. As a result, a significant oil dilution reduction and increased oil drain interval for urban conditions have been reached. With the upcoming Euro 5 regulation and the DPF being mandatory, Renault has carried out an extensive study to compare the different solutions to reduce oil dilution beginning with the optimization of the exhaust fuel injection system already in series production. In chapter 2, these different solutions will be compared and in chapter 3 we will present in detail the newly designed DPF system with EFI (Exhaust Fuel Injection) that has been implemented on the Euro 5 engine family (from the 1.5 dCi to the 3.0 dCi).

2 Solutions to Reduce Oil Dilution

2.1 In-cylinder Injection Strategy

In terms of injection pattern, several approaches exist in series production but the most common strategy consists of retarded pilot and main injections in combination with two post-injections (one close and one late). Post-injection events are the major contributors to oil dilution increase during regeneration and especially the late one. Indeed this late injection event takes place at a point in the cycle when the gas density and temperature are low and the piston is far away from the cylinder head. As

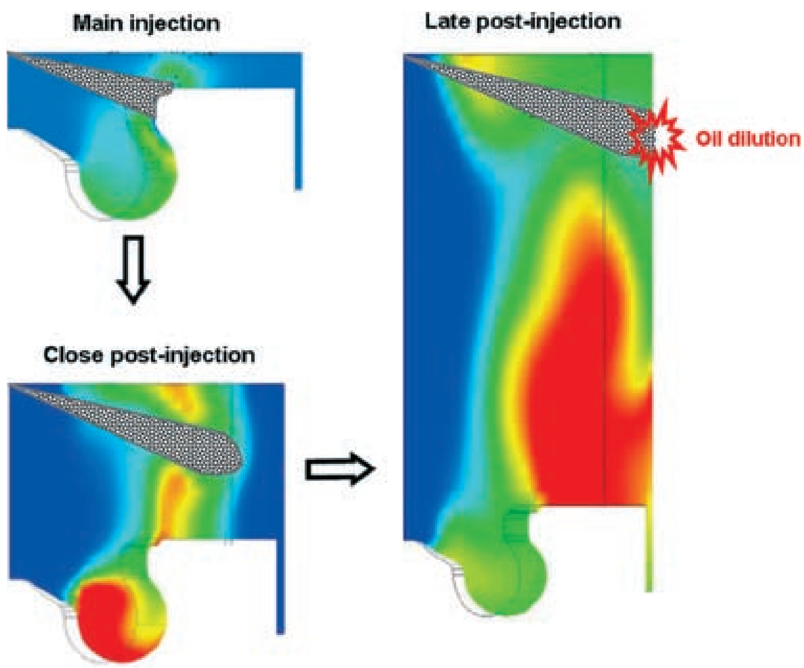

Figure 4: 3D CFD showing the temperature gradients and the fuel spray penetration within the combustion chamber at the main, close and late post-injections timings

illustrated on **Figure 4**, this leads to high fuel spray penetration and thus impingement of the fuel on the cylinder liner walls. One way to improve oil dilution is to reduce the spray momentum and this can be achieved by dividing in two parts the post injected quantity, **Figure 5**. This injection strategy (so called ”split of the late-post“) has been tested and optimized in terms of fuel quantity and separation between the two late post-injections in order to maximize the oil dilution reduction. In comparison to the unique late post-injection, the tests on the engine bench showed a 25 % improvement of the oil dilution level for speed lower than 2500 rpm and B_{MEP} below 7 bar. On the PAP cycle, Figure 11, it leads to a 20 % lower oil dilution level keeping the same regeneration efficiency. Those results can be explained by two main phenomena:

- The reduced injected quantities of the two injections lead to smaller injection duration and thus reduced spray penetration. Consequently, the fuel impingement on the cylinder liner is decreased which leads to lower oil dilution.
- The ”seat throttling effect“ that occurs two times with the splitted late post-injection instead of one. The seat throttling effect during the nozzle opening/closing leads to lower mean injection pressure and thus reduced spray momentum and penetration, **Figure 6**. The result is a decreased oil dilution level.

Nevertheless, this 20 % oil dilution reduction is not sufficient regarding the target of 40 % and this will be further discussed in chapter 2.4.

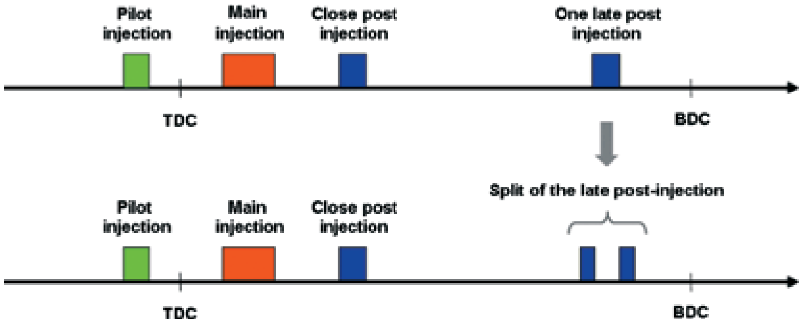

Figure 5: Principle of the split of the late post-injection strategy

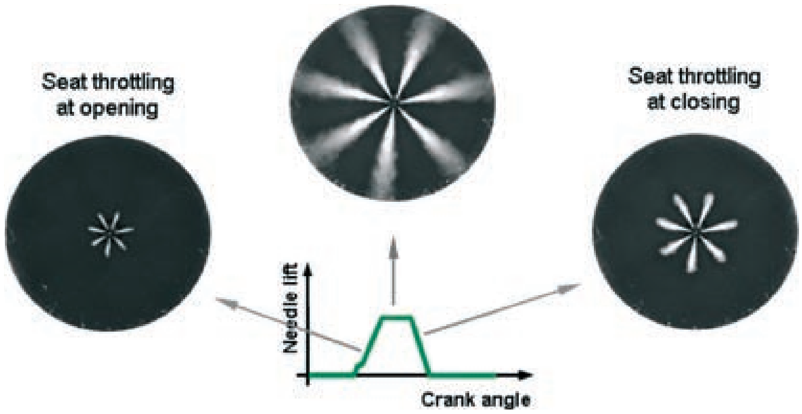

Figure 6: Visualization of the seat throttling effect on the fuel sprays during nozzle opening and closing

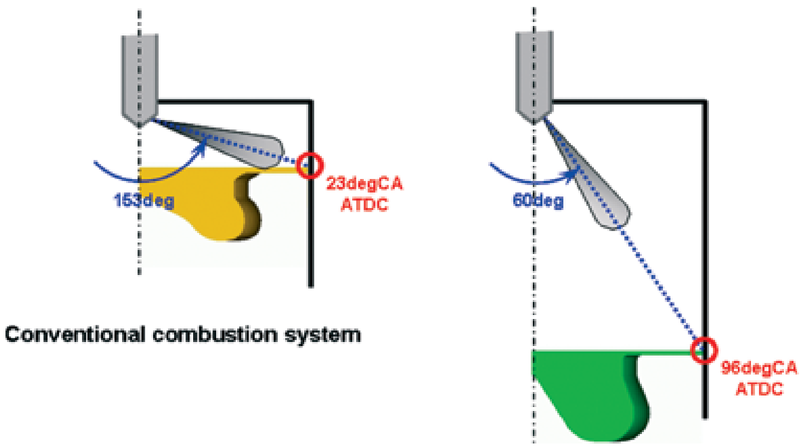

Figure 7: Spray design (nozzle tip protusion and cone spray angle) of the conventional and new combustion systems

2.2 Combustion System Design

Another approach to reduce oil dilution is to adapt the combustion system design and especially the spray pattern to avoid, or at least reduce the fuel impingement on the cylinder liner walls. To achieve a significant oil dilution reduction, it is necessary to increase drastically the crank angle limit beyond of which the fuel impinges the cylinder liner walls, **Figure 7**. Indeed, post-injection timings are late (> 40 deg CA ATDC) and it was mandatory to redesign the combustion system with a very low spray cone angle of 60 deg. Specific combustion bowl shape (deep and narrow) and swirl ratio (very low) were necessary to fit with this low spray cone angle. As shown on Figure 7, the crank angle limit before fuel impingement on the walls is increased by 73 deg in comparison with the conventional combustion system. The two systems have been compared in terms of oil dilution and regeneration efficiency at steady state on several operating points. With respect to the significant increase of the crank angle limit, the oil dilution level has been drastically reduced. For example at 1500 rpm / B_{MEP} 3 bar which is representative of a low speed condition, the oil dilution has been reduced by 70 %. This combustion system has a strong potential to reduce

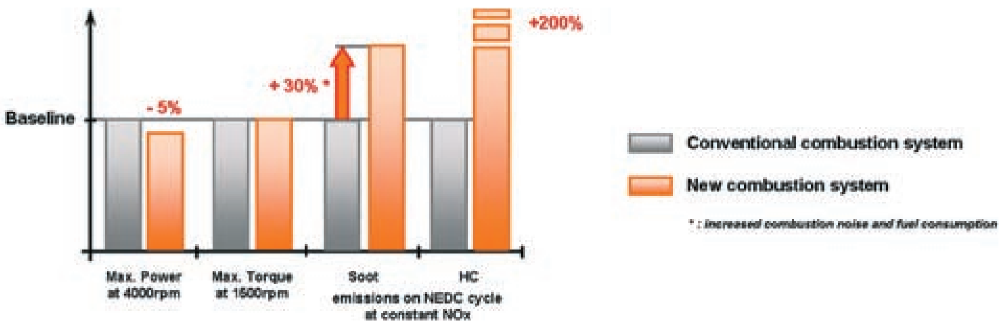

Figure 8: Comparison of the conventional and new combustion systems: power/torque at full load and emissions on the NEDC cycle

oil dilution and to fulfill the 40 % improvement. However, it induces some drawbacks on performances and emissions at part load in comparison with the conventional design. Low end torque is kept constant but maximum power is reduced by 5 %, **Figure 8**. Furthermore, the HC emissions are very critical and the NO_x/Soot trade-off is deteriorated: for a given NO_x level on the NEDC cycle, soot has been increased by 30 % in comparison with the conventional system and it was not possible to keep the same combustion noise and fuel consumption (both are deteriorated). This will result in more frequent regenerations and will reduce the overall improvement of the oil dilution level on the PAP cycle that is estimated to 50 %, Figure 11. Another main issue concerning this new combustion system is the strong modifications that must be applied to the engine design (bowl shape, swirl level of the cylinder head) that are not always feasible on existing engine families. This will be further discussed in chapter 2.4.

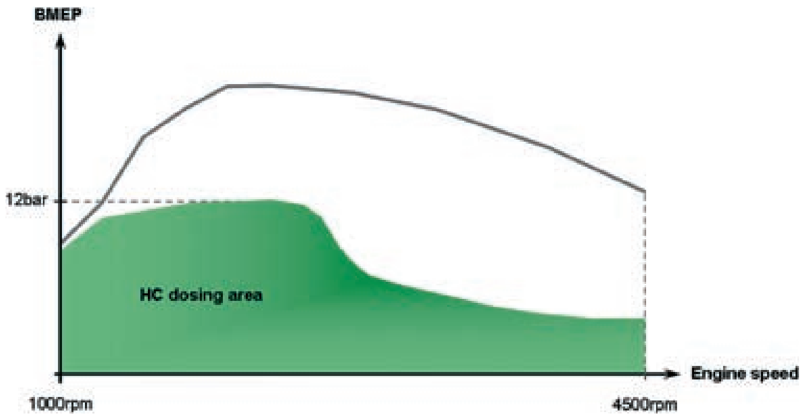

Figure 9: HC dosing area of the 1.5-dCi-Euro-4-engine

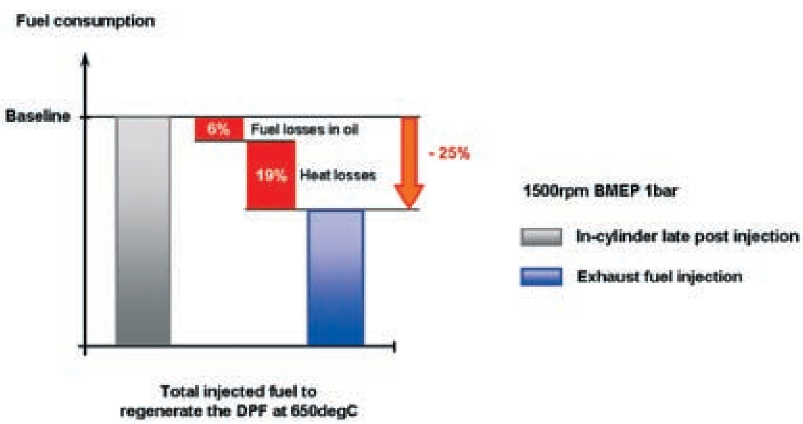

Figure 10: Comparison of the total fuel injected when using exhaust fuel injection versus late post-injection to regenerate the DPF at 1500 rpm / B_{MEP} 1bar

2.3 Exhaust Fuel Injection

An effective solution to reduce oil dilution is to inject less fuel (especially the late post quantity) within the combustion chamber and to introduce it directly at the exhaust side upstream the oxidation catalyst. As discussed above, since 2006 Renault is using an exhaust fuel injection (EFI) system on the 1.5 dCi 78 kW engine equipped with DPF, Figure 3. The main advantage of positioning the EFI on the exhaust line after the turbine is that existing engines can be easily fitted with such a system. It is not necessary to redesign engine parts like the cylinder head if we consider the system described in [1].

The main drawback of an EFI located downstream the turbine is that one can not benefit of the maximum gas temperature available on the exhaust side so as to vaporize the injected fuel. A minimum gas temperature of 350 deg C is necessary in order to evaporate 95 % of a conventional Diesel fuel. Taking into account the effect of drifts and scattering, a minimum exhaust gas temperature of 400 deg C before DOC2 has been settled so as to achieve a good evaporation of the fuel in all the operating conditions. When the engine operates at low loads, this threshold has been obtained by intake throttling and close post-injection strategies. Even if the use of this post-injection has a cost in terms of oil dilution, we will see in the following that this solution is sufficient to fulfill the target of a 40 % oil dilution reduction. The engine map area where the late post-injection can be replaced by exhaust fuel injection is illustrated on **Figure 9**. The low and me-

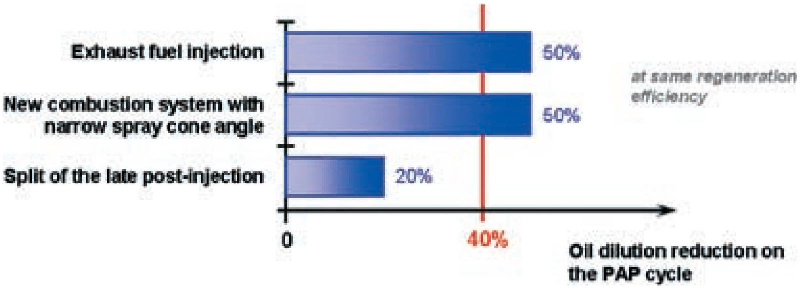

Figure 11: Comparison of the different solutions in terms of oil dilution reduction on the PAP cycle in comparison to the standard in-cylinder strategy (up to two late post-injections)

dium loads, which are the most critical for oil dilution, are well covered. For high speeds and loads, the HC dosing area is limited by the HC conversion efficiency of the catalyst. In this case, conventional late post-injection is used. As shown on Figure 11, this exhaust injection system brings a 50 % oil dilution reduction at nearly-constant regeneration efficiency on the PAP driving profile. In addition, the exhaust fuel injection is more efficient than the late post-injection strategy especially at low engine/load conditions which are representative of urban driving. For example at 1500 rpm / B_{MEP} 1 bar, the total injected fuel that is necessary to reach a gas temperature upstream DPF of 650 degC is 25 % lower with an exhaust injection in comparison with the late post-injection strategy, **Figure 10**. This difference is explained by two reasons:

- With the exhaust fuel injection, there is no fuel loss in the oil. For the operating point considered, 6 % of the injected fuel goes into the oil
- Even if the pressure and the temperature within the cylinder are rather low during the late post-injection event, a part of the fuel burns. The resulting gas temperature increase at the exhaust valve opening is not totally recovered at the DPF inlet due to the thermal losses. In the case of the EFI, the fuel is injected just before the DOC and the thermal losses are limited. This loss of energy corresponds to 19 % of fuel loss.

The exhaust fuel injection system is an efficient technology to reduce oil dilution and improve fuel consumption during regeneration. Besides, the oil dilution reduction (50 %) is higher than the target of 40 %.

2.4 Comparison of the Different Solutions

As depicted in **Figure 11**, only two solutions satisfy the target of 40 % oil dilution reduction on the PAP driving cycle: the exhaust fuel injection device and the new combustion system with a narrow spray cone angle. As described before, a combustion system modified to fit with a narrow spray cone angle has some drawbacks on the performance and emissions of the engine that are hard to compensate. Besides, the strong modifications on the cylinder head and piston bowl definition imply a great design effort and are not always feasible on existing engine platforms. An exhaust fuel injection system combines excellent performances in term of oil dilution and fuel efficiency with a limited impact on the engine design. Thus the integration costs are significantly lower. According to its first successful experience with the exhaust fuel injection of the 1.5-dCi-Euro-4-engine, Renault has decided to go on with this technology for the Euro 5 engine family. As described in chapter 3, this system has been further optimized so as to improve its cost to efficiency ratio.

3 Renault's Euro 5 Exhaust Fuel Injection System

3.1 Exhaust Line Design

The Euro 4 design of the exhaust line incorporates two oxidation catalysts, Figure 3: one located after the turbine outlet for HC & CO after-treatment in normal mode and a second one located upstream the DPF to burn the fuel injected by the EFI during regeneration mode. The Euro 5 design of the exhaust line has been simplified so as to reduce its cost and to improve its efficiency, **Figure 12**. The DPF has been located closer to the engine so as to reduce thermal losses and a single oxidation catalyst is used for HC & CO after-treatment in normal mode and HC burning in regeneration mode. The exhaust fuel injector is located directly at the turbine outlet. So as to reduce the cost of system, the precious metals inside the DOC have been optimized in terms of loading and formulation. It leads to the adoption of Pt/Pd coating with 20 % of Palladium. Besides its reduced cost, this coating has proven to be more efficient in HC conversion during regeneration. Thus, it has been possible to extend the HC dosing area and further reduce the oil dilution.

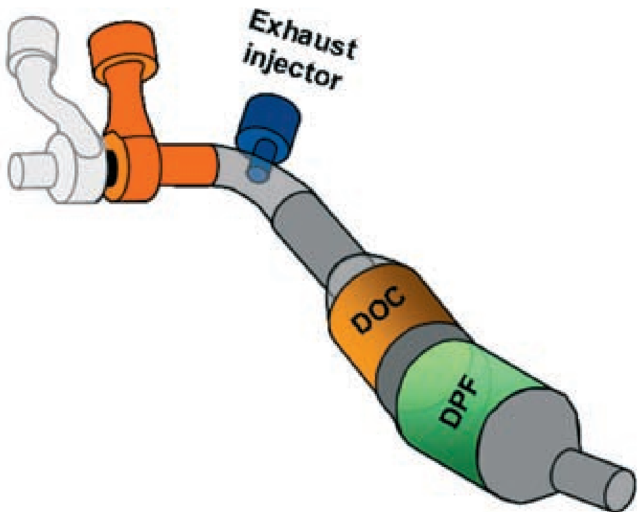

Figure 12: Euro 5 exhaust line layout

3.2 HC Dosing System

As described on **Figure 13**, the HC dosing system is composed of:

- one element for the fuel dosing and spraying (liquid injector)
- one system for the injector cooling and fixing
- one system for the fuel supply
- one system for the injection control.

The injector is derived from the gasoline technology. It runs at a maximum pressure of 6 bar and is drived by the engine ECU at a frequency that is independent of the engine speed. Compared to the Euro 4 definition, the injector has been optimized to fit to the HC dosing requirements with Diesel fuel: the nozzle has been redesigned regarding spray angles, droplets' size and flow rate.

The major guidelines of this redesign have been to maximize the sprayed surface in the exhaust line in compromise with a fine atomization of the fuel. These evolutions have been associated to a specific control strategy and have contributed to the global optimization of fuel evaporation and mixing for a better regeneration efficiency (see chapter 3.3).

Finally, in order to reduce the cost of the system the following optimizations have been carried out:

- The water cooler used on the Euro4 version has been replaced by an air cooling device
- On the Euro 4 version, the fuel was supplied to the injector by a specific electric pump. For Euro 5, the fuel comes directly from the common-rail fuel pump via a low pressure derivation.

3.3 Optimization of Fuel Evaporation and Mixing

For the Euro 5 version, emphasis has been put on the optimization of the fuel evaporation and mixing with the help of 3D CFD. The purpose was to reach a good HC uniformity upstream DOC with the new exhaust line design. As illustrated on **Figure 14**, the process of mixing is the following:

1. The fuel is injected and impinges the inner wall of the exhaust tube
2. Thanks to the hot gas and walls, the fuel evaporates gradually and is mixed with the gas before reaching the inlet of the oxidation catalyst.

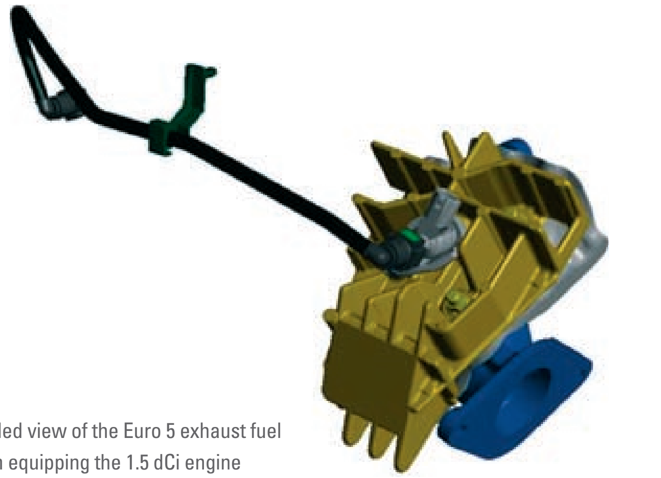

Figure 13: Detailed view of the Euro 5 exhaust fuel injection system equipping the 1.5 dCi engine

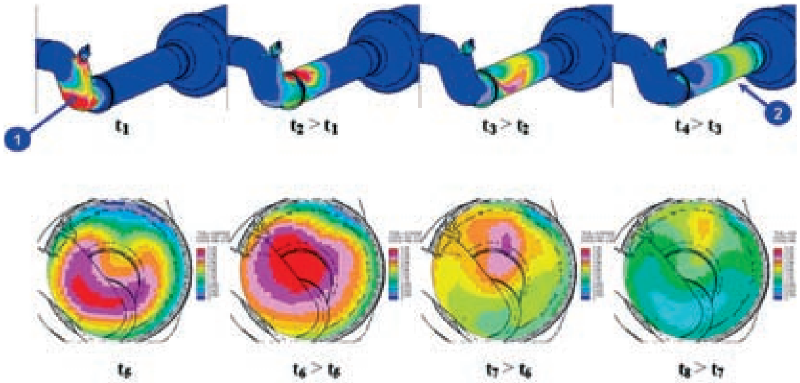

Figure 14: 3D CFD showing the evaporating and mixing processes as a function of time

Figure 15: Optimization of the exhaust line design

Compared to the Euro 4 design, the Euro 5 exhaust line has received several optimizations in terms of HC uniformity: first, the fuel injection system has been improved. As described above, the new one provides a finer atomization of the fuel, which leads to an easier evaporation. The spray's targeting has been also optimized in the view of benefiting from the fuel impingement on the exhaust line walls. This contact between liquid phase and hot walls provides a very efficient HC evaporation. A lot of work has been carried out on the interaction between HC spray and exhaust gas flow. Fuel-gas mixture is very dependent on aerodynamic within the exhaust line and a turbocharged engine produces complex aerodynamic motions downstream the turbine that have a major impact on fuel-gas mixture. Thanks to the optimization of the exhaust line design, it is a very effective way to improve the HC dosing system performance, **Figure 15**. In comparison to the Euro 4 engine, the HC dispersion upstream DOC of the Euro 5 engine has been reduced by more than 90 %. This improvement leads to a better regeneration efficiency of the soot inside the CSF, which decrease oil dilution.

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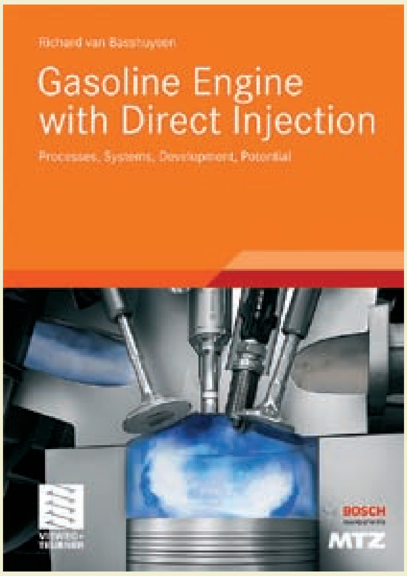
Gasoline Engines are the answer to the challenges of future



Richard van Basshuysen

Gasoline Engine with Direct Injection

Processes, Systems, Development, Potential

2009. xviii, 437 pp. With 399 Fig. Hardc. EUR 49,00
ISBN 978-3-8348-0670-3

Direct injection spark-ignition engines are becoming increasingly important, and their potential is still to be fully exploited. Increased power and torque coupled with further reductions in fuel consumption and emissions will be the clear trend for future developments. From today's perspective, the key technologies driving this development will be new fuel injection and combustion processes. The book presents the latest developments, illustrates and evaluates engine concepts such as downsizing and describes the requirements that have to be met by materials and operating fluids. The outlook at the end of the book discusses whether future spark-ignition engines will achieve the same level as diesel engines.

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450 cm^3 High Performance Engine for a BMW Sports Enduro

The BMW G450X is designed for uncompromising use in competitive Enduro sports. It is a completely new development and has no predecessors at BMW Motorrad. The key concerns when defining the specification sheet were saving weight and perfect tailoring to the needs of off-road riders. This article describes the new engine concept.

1 Introduction

Typically for motorbike, the engine and its peripherals are an integral part of the overall vehicle concept. That's why it was necessary to develop the entire driveline new tailored for integration in the new vehicle concept. The specification sheet for the power train prioritizes compact dimensions and light weight, while a balance shaft is required to reduce the stress on the rider. The power output of the engine must permit both trial-type riding with excellent torque and drivability, as well as offering high performance and a broad usable rev range for fast sections. The highest possible performance should be achieved, while at the same time fulfilling the homologation requirements.

2 Basic Engine Concept

The most important feature of the concept is the merging of the bearing axle of the rear swing arm with the rotating axle of the drive pinion. This makes it possible to lengthen the swing arm and to place the engine further back on the vehicle while retaining the same wheelbase. At the same time, the cylinder can be tilt forward, **Figure 1**. This means that the clutch needs to be moved from the transmission input shaft to the crankshaft in order to make space for the fork bearing. The forward twist of the cylinder provides for optimum performance in the intake port.

The engine consists of four large cast parts, the vertically divided crank-/transmission casing, the cylinder and the cylinder head. Three side panels and the cylinder head cover close off the engine to the outside, **Figure 2**. With a bore of 98 mm and a stroke of 59.6 mm (= 449.5 cm^3 displacement), the engine is designed with a short stroke compared with the competition. The main dimensions are listed in the **Table**.

2.1 Crank Casing, Cylinders, Crankshaft Drive, Clutch, Mass Compensation

The vertically divided, die-cast crank case, Figure 2, made from AlSi9Cu3Mg, contains all bearing points for the crankshaft, balance shaft, transmission shafts, shifting drum and shifting axes, as well as the oil pumps and the water pump. The two halves of the crank case are sealed by paper seal. In order to achieve the lightest weight possible, thin-walled castings were favored. The zones between the main bearing and the cylinder/ cylinder head, which are subject to particular stress, were optimized in terms of design.

Figure 1: Vehicle package

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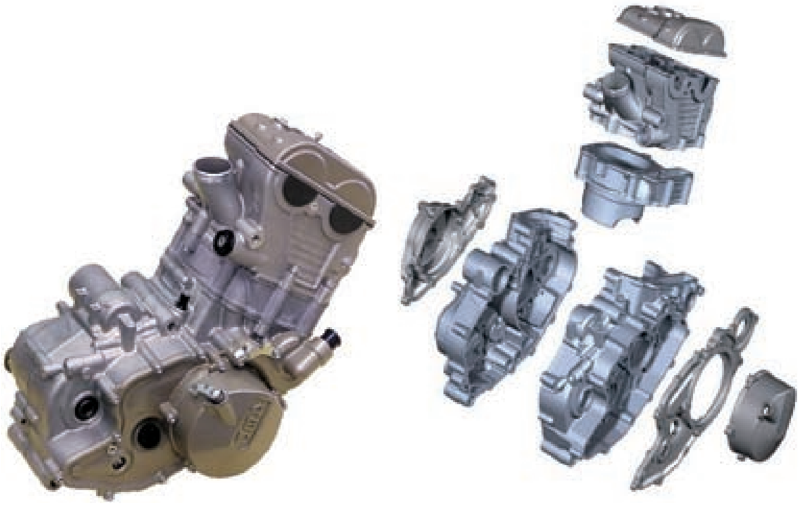


Figure 2: Core engine, engine main parts

Because of the placement of the clutch directly on the right crankshaft end, the dynamic behavior of the crankshaft requires a combined process involving both calculation and vibration analysis. One result of the dynamic optimization is the placement of a third roller bearing in the generator cover. This bearing increases the rigidity of the vibration system and, from an engine mechanics perspective, allows the crankshaft drive to run smoothly over 11,000 min^{-1}.

The single-component conrod is made from high-alloyed case-hardening steel. The nitrided piston bolt runs in a bearing bush.

The forged piston with a diameter of 98 mm is optimized in terms of weight and friction and has a coated, extremely short shroud. The piston has two rings, the first of which is a 1.2 mm high nitrided steel plain compression ring and the second a 2.3 mm high oil scraper ring in a single piece hose spring version. To increase the piston's service life, it is essential that the base of the piston should be intensively cooled by two oil spray nozzles.

The crankshaft is driven by means of a primary gear, which is part of the wet multiple-disk clutch which is borne directly on the crankshaft, Figure 4. Because the clutch runs with the speed of the engine, the torques to be transferred are must less than with a standard layout. The consequence is a correspondingly small diameter of the 6 disk pairs. The contact force is applied by a cup spring. The clutch is lubricated by means of a stream of oil passed through the crank-

Table: Main technical data

		G 450 X
Design	4 Stroke	1-cylinder
Displacement	ccm	449,5
Bore / Stroke	mm	98 / 59,6
Number of Valves	–	4
Valve angle IV / EV	°	10 / 12
Diameter IV	mm	40
Valve Lift IV	mm	10,220
Diameter EV	mm	33
Valve Lift EV	mm	9,380
Main Bearing	Roller Bearing	NJ206
Conrod big end	Roller Bearing mm	35 / 43
Conrod small end	mm	20
Conrod length	mm	107
max. Power ECE / Sport	kW / 1 / min	31 / 7000 38 / 8500
max. Torque ECE / Sport	Nm / 1 / min	43 / 6500 44 / 7700
max. engine speed	1 / min	10800
Average piston speed	m / s	21,4
Compression Ratio	ε	12.0

The short cylinder is produced as a closed deck using a die casting process. A friction-optimized nikasil coating is used as the cylinder's liner surface. The cylinder is fixed directly in the crank case by means of the threaded joint of the cylinder head.

The crankshaft, Figure 3, consists of two parts, which are joined together by press fit. One half of the crankshaft has the lifting journal attached in a forged bond. The crankshaft is made from a high-alloyed, heat-treated case-hardening steel.

The main bearings used are cylinder roller bearings size NJ206 with end-profiled rollers. The cage-controlled 19-roller conrod bearing runs directly on the lifting journal, while the large conrod eye acts directly as an outer running partner. A coordinated oil supply is essential for the correct function of the roller bearing and the axial dynamic friction point between the conrod and the crank web. The pressure oil is supplied centrally to the crankshaft end on the generator side.

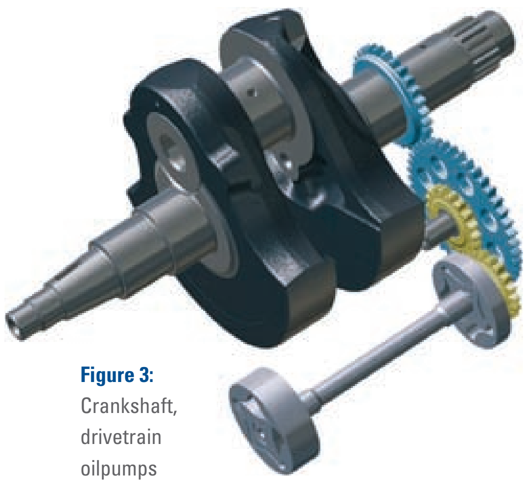


Figure 3: Crankshaft, drivetrain oilpumps

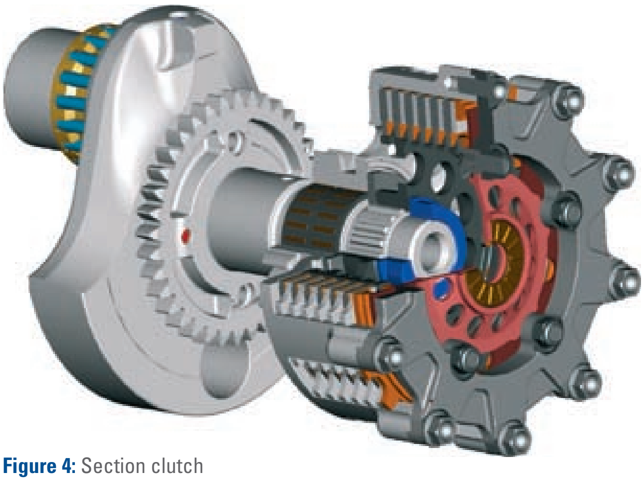


Figure 4: Section clutch

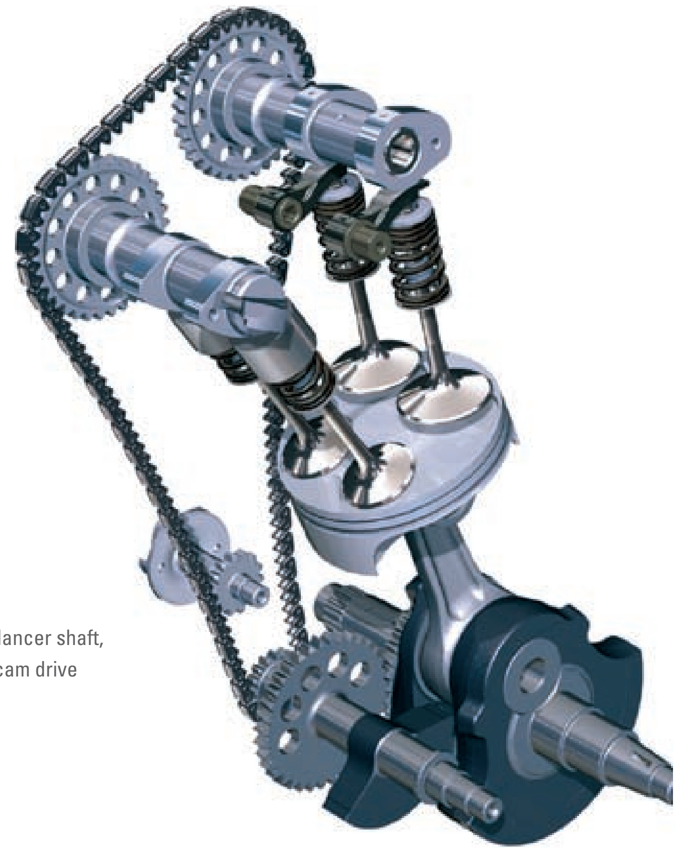


Figure 5: Balancer shaft, valve train, cam drive

shaft, Figure 8. The clutch is activated by means of a central, mechanically activated release lever.

A balancer shaft, **Figure 5**, is used to reduce vibrations. This is driven by spur gears, runs in the opposite direction at the same speed as the crankshaft and eliminates the first order of inertia forces. Like the crankshaft, it is carried on roller bearings. The single-component conrod makes it possible to get very close to the crankshaft, thereby keeping the first order mass moment at a low level. The balancer shaft is also used to drive the timing chain and as a bearing in the starter drive.

2.2 Cylinder Head, Valve Drive, Timing Drive

The cylinder head, **Figure 6**, which is made from AlSi10, is produced using a low-pressure die-casting process and is heat treated. The camshafts are fixed with a bearing cover on the cylinder head. The camshafts are carried by a roller bearing on the chain side. The major part of the valve actuation forces is transferred by a plain bearing located between the cams. The cylinder head is bolted to the crankcase together with the cylinder by means of four tie rods. There are two more threaded joints in the crankcase on the outside of the chain case. Directly beneath the intake and output ducts, two short threaded joints from the cylinder head and cylinder ensure an even contact with the multiple-layer steel cylinder head seal.

The timing drive, Figure 5, is applied directly to the two camshafts by the balancer shaft by means of a silent chain. The silent chain is guided by plastic tracks and the tension is controlled by a hydraulic chain tensioner for optimized friction power.

The high engine speeds and the required valve speeds make it necessary to use four titanium valves, as well as weight-optimized aluminium spring re-

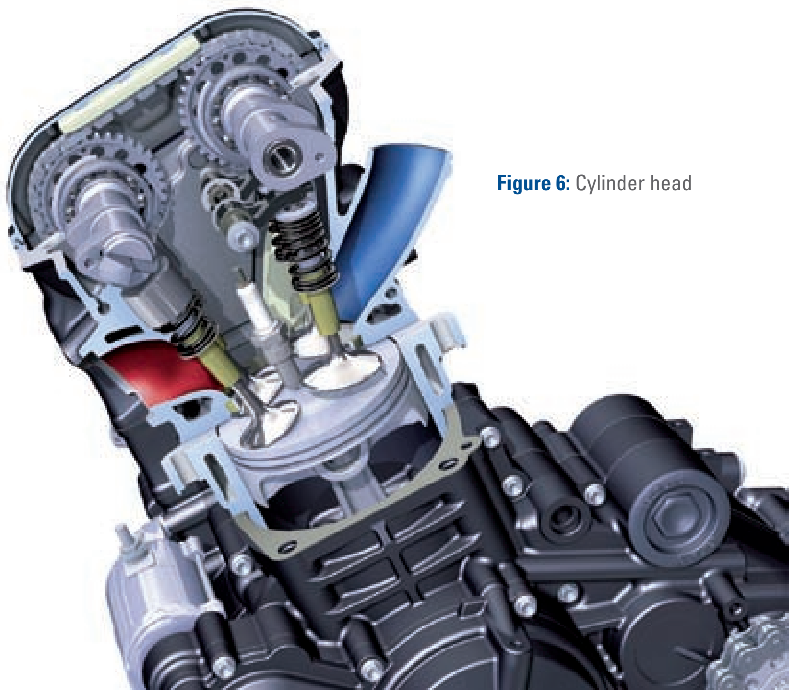


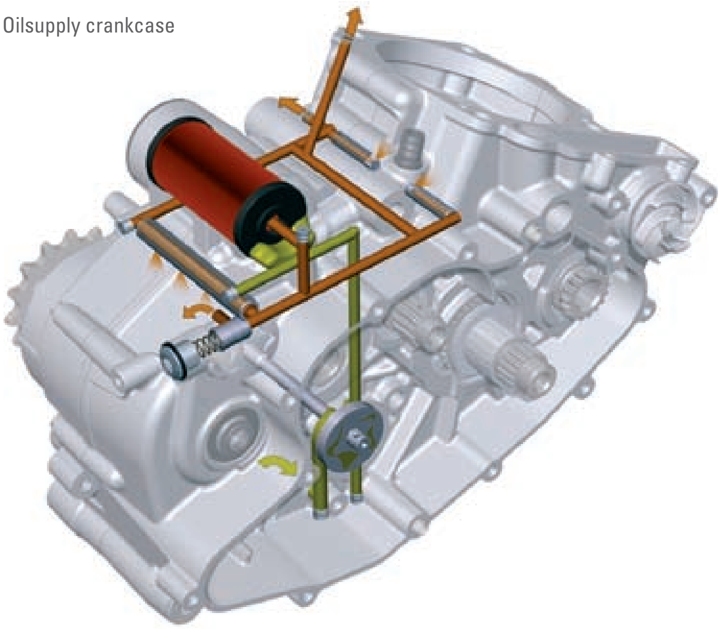
Figure 6: Cylinder head

tainers. The 40 mm diameter heavier weight intake valves are activated by DLC-coated cam followers, which, as carry-over parts, could be reused from existing BMW Motorrad engines [1, 2]. The layout of the cam followers also favors the passage of the steeply rising intake port. The lighter-weight outlet valves, diameter 33 mm, are activated by barrel tappets with a diameter of 28 mm with similarly reduced mass. This allows to lower the cylinder head height in the front area and also allows to use an automatic decompression device, Figure 5.

2.3 Oil Circuit, Crankcase Ventilation

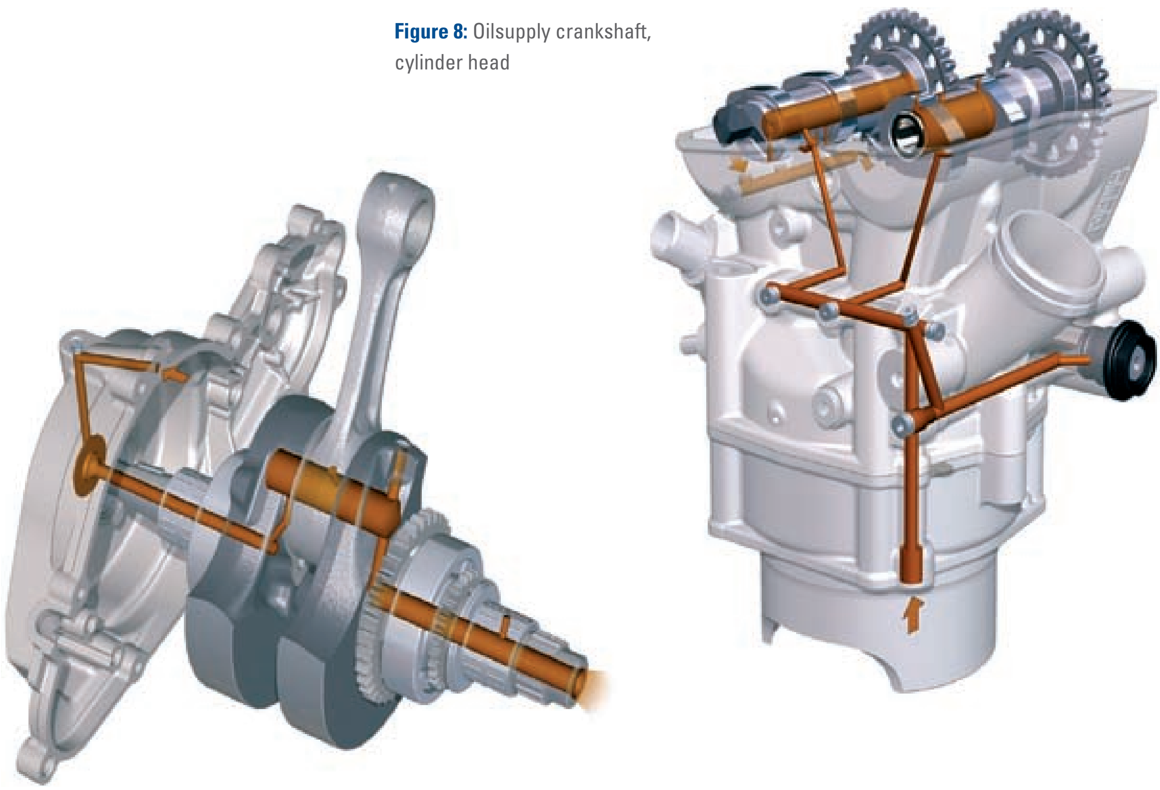
The engine is lubricated by a combined wet/dry sump lubrication system and the entire oil circuit only contains 1.1 l of oil. The two trochoid oil pumps, Figure 3, sit on a shared shaft, which is driven by the crankshaft by means of an intermediate transition. The larger of the two pump units generates a partial vacuum in the crank chamber, draws off the oil and transfers it to the transmission chamber, which serves as a semi-dry sump. The underpressure generated in the crank chamber is positive for the effective friction power behavior of the basic engine.

Figure 7: Oilsupply crankcase



In order to supply pressure oil to the engine, the smaller of the two pumps draws the engine oil through a screen from the oil sump on the transmission side, **Figure 7**. Next the pressure oil is passed through the filter element and then applied to the main oil duct with the spring-loaded oil pressure regulating valve. The oil filter and pressure regulating valve are easily accessible from the outside. The main oil line is used to supply the oil spray nozzles for cooling the piston, the axial oil supply in the crankshaft and the cylinder head lubrication points, **Figure 8**.

Figure 8: Oilsupply crankshaft, cylinder head



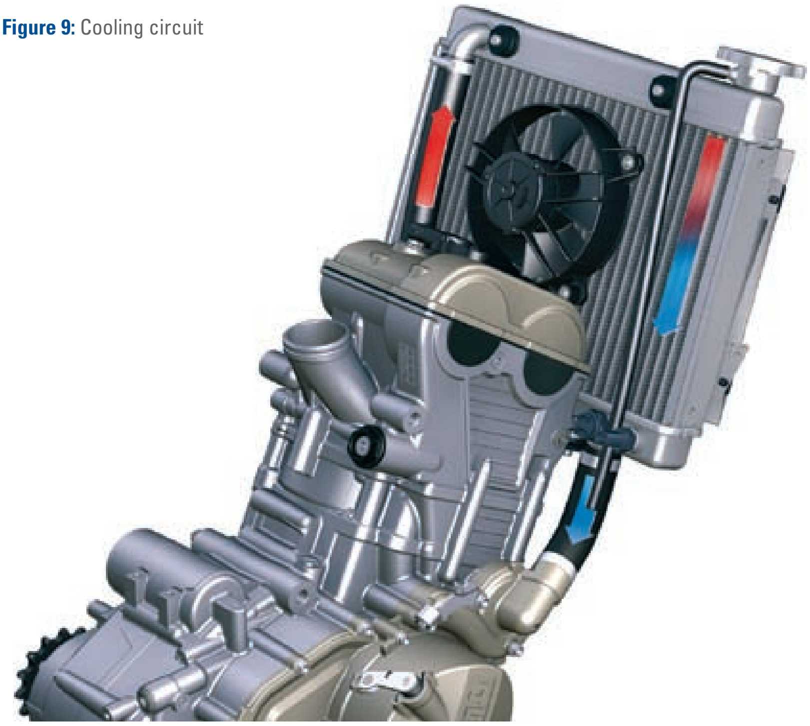
A riser line in the cylinder and cylinder head supplies the chain tensioner and camshaft bearing with oil, Figure 8. From the camshaft plain bearing points, the floating pairs of cam followers and barrel tappets are specifically lubricated by means of an intermittent oil supply.

The crankcase is ventilated by means of a labyrinth system in the rotating interim shaft of the primary drive, Figure 10. The interim shaft acts as a centrifugal oil separator.

2.4 Water Cooling Circuit, Positioning of the Coolant Pump

The compact design of the unit is also made clear by the layout of the coolant pump, **Figure 9**. It is driven by the timing chain, Figure 5. A gear wheel engages in the toothed chain in the area of the chain guide track and the pump wheel sits on the outer edge of the same shaft. This means that the pump is operated without additional chain or wheel drive and is also placed at a very low position on the engine.

Figure 9: Cooling circuit



The placement of the engine further back in the vehicle according to the concept allows a single block radiator to be used. The cooling circuit is not regulated either by a thermostat or by flow. When temperatures above 90 °C are reached, the fan is started by means of a temperature switch integrated in the radiator. An overpressure valve in the radiator cover limits the system pressure to 2.2 bar.

2.5 Intermediate Shaft and Transmission

The placement of the clutch directly on the crankshaft is an unusual solution. The wet clutch transfers the torque of the crankshaft to the clutch basket and its gear wheel, **Figure 10**. The power flux to the transmission input shaft takes place by means of a pair of intermediate gears with a ratio of 1:2.618 in the direction of deceleration. There is a slipping clutch integrated in the gears of this intermediate shaft. The set transmission torque reduces load peaks, which have a retroactive effect via the chain and transmission as far as the clutch and crankshaft in the event of skips during off-road use. This intermediate shaft produces a very favorable shaft layout in the engine; one of the special features is that it leads to a reversal of direction, so that the crankshaft rotates backwards in the vehicle. Force is applied to the input shaft of the 5-speed gearbox by means of a gear wheel that also has spur teeth.

The dog-type constant-mesh countershaft transmission typical for motorcycles, Figure 10, is integrated in the vertically divided crank case and consists of the roller-mounted gear input and pick-up shafts and five loose and fixed gears with spur teeth which are always meshed. Switching is achieved by moving the fixed wheels on the axle, while the torque is applied by means of claws that engage in the corresponding pockets of the relevant loose gear. The fixed wheels are moved by three steel shift forks that engage in the curved paths of the cam. They are guided by two steel axles on the axial plane. A ratchet mechanism on the switching shaft sequentially turns the cam by 60°. A spring-loaded roller lever that engages on an indexing gear fixes the shifting drum and therefore locks the gear.

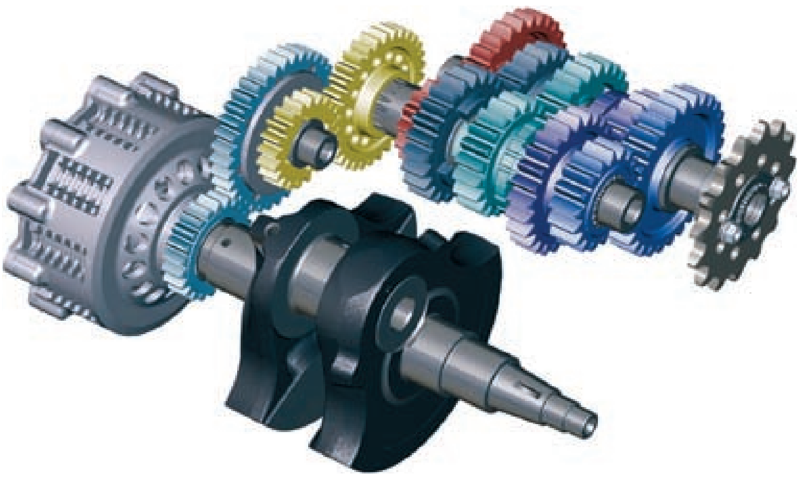


Figure 10: Clutch, intermediate shaft, gearbox

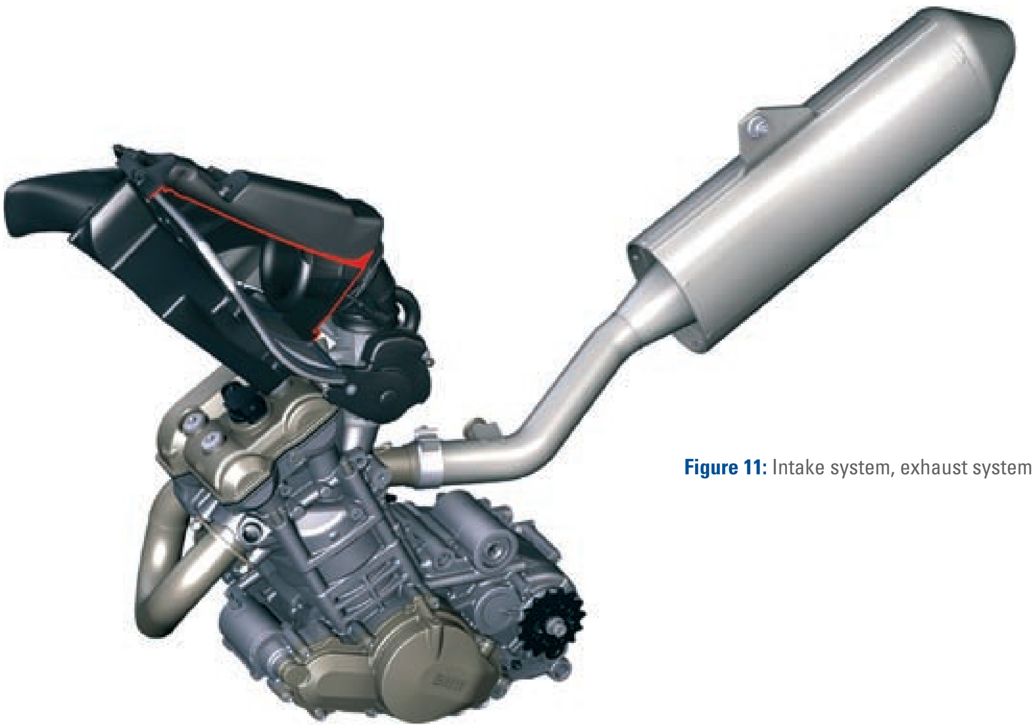


Figure 11: Intake system, exhaust system

The torque generated is transferred from the pinion on the final shaft to the rear wheel by means of a chain.

2.6 Starter Drive and Decompression Device

The extremely compact high-speed starter is mounted on the front of the engine case. A reduction gear is used to lower the speed of the starter to the starting level of the crankshaft. In the first transmission level is a slipping clutch which prevents the overloading of the starter gears and starter freewheel. The subsequent reduction level transfers the start torque to the gearwheel of the one way clutch. This is linked to the magnetic body of the generator. The link between the generator body and crankshaft is friction locked by means of a cone, while a disk spring positions the generator body in relation to the crankshaft. When the engine is running, the starter drive is uncoupled by the one way clutch.

3 Engine Peripherals

3.1 Intake Manifold, Fuel System, Throttle Flap Assembly

Air is taken into the vehicle at a cool area at the top front of the vehicle that is kept free of dirt and splashes of water. The incoming air reaches the lower part of the airbox, **Figure 11**, by means of an inlet air intake snorkel and flows into the clean air side of the intake system by means of a oiled filter fleece. The intake air reaches the double throttle flap system by means of an intake snorkel at an angle of 90°. In this two-flap system the flap closer to the cylinder head is activated by the accelerator cable and the upper flap is controlled by means of a stepper motor, depending on the mapping.

The fuel is supplied by means of an electronically regulated intake manifold injection system. The injection valve is located in the throttle body. The fuel pump housed in the fuel tank starts when the crankshaft speed signal is detected and delivers a constant 3 bar fuel pressure under the control of a mechanical pressure regulator.

3.2 Exhaust System

Appropriately for the segment, there are several exhaust systems variants available for the different uses of the vehicle. In the homologized version, Figure 11, there is an oxygen sensor in the stainless steel manifold. The rear muffler contains the catalytic converter and operates according to the reflection and absorption principle. Its outer skin consists of an extruded aluminium pipe in which the fixing holder is also integrated. The damping elements are located inside the pipe; which closed off at the top end by the catalyst track and at the other end by an end cap. When the motorcycle is used for racing, there is a slip-on titanium muffler for the standard manifold and a modified titanium manifold that also permits higher gas flow. Overall, the fully titanium exhaust system is about 3 kg lighter than the standard system.

3.3 Engine Control, Sensors, Actuators

The engine is managed by an ECU (engine control unit) that is tailored to the needs of very lightweight single cylinder engines. The load sensing is registered by the position of the two throttle valves, the intake pipe pressure and the intake pipe temperature. The position of the crankshaft and the speed of the engine are registered by an inductive sensor and teeth on the outer rotor of the generator. The ignition is achieved by means of an ignition coil and a spark plug with M10x1 thread diameter. The engine temperature is registered by a coolant sensor. A discrete-level sensor is used to regulate the oxygen.

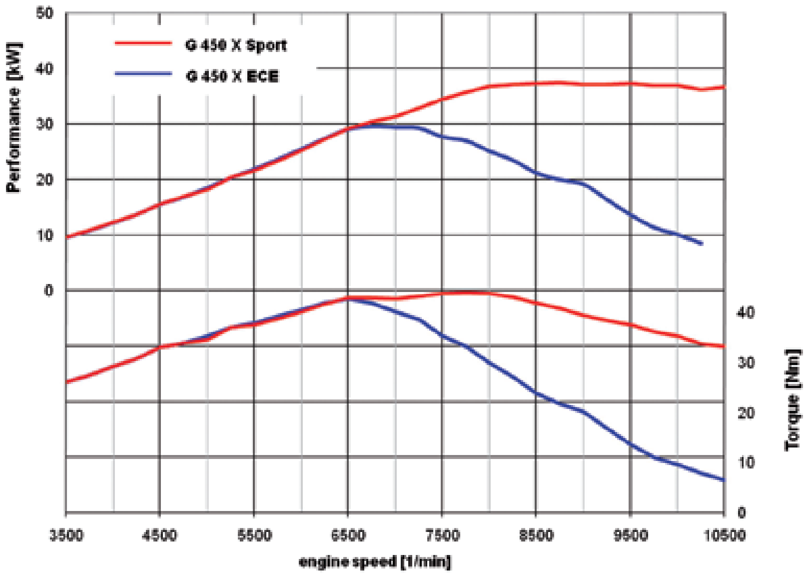


Figure 12: Comparison of full load curves

4 Functional Features, Safety

Top-class off-road competition, which was the primary purpose the developers had in mind when creating the vehicle, makes huge demands on the functional features of the engine. On the one hand, trial-type sections that require a slow pace demand a controlled response and a small cutoff angle, while, on the other hand, high-level performance and spontaneity are essential in faster sections. In order to avoid the nuisance of frequent gear changes on difficult terrain, the broadest possible rev range is required.

With 31 kW at 7000 min^{-1} and 43 Nm at 6500 min^{-1}, the G 450 X is ahead of the competition, while still complying with emission and noise level legislation. When the motorcycle is to be used solely for competition purposes, there is an optional race mapping feature in the engine controller that ensures an output of 38 kW and a torque of 44 Nm in conjunction with the sports muffler, **Figure 12**. In addition, the response was also adapted to the needs of the best riders.

The different performance modes are mainly influenced by the reduced exhaust gas back pressure in conjunction with an optimized mixture control and the double throttle body system. The option of closing the upper throttle flap by means of a stepper motor or not to open it at the same speed as the lower valve mechanically actuated by the rider, offers a wide application range. Influencing of the response is just as possible as the rating at full load at the filling limit of the engine. In addition, the increase in speed at cold start is achieved by means of the second throttle valve.

The development of the engine, and of the vehicle as a whole, was influenced by very early racing trials with high-profile competition riders. The load collectives registered during the process formed the basis for intensive test bench trials and for balancing the design load in the simulation.

5 Summary

The new G 450 X from BMW Motorrad is the company's first motorcycle suitable for use in Enduro competition that also allows new customer groups to be addressed. Typically for BMW Motorrad, the entire concept includes a whole range of innovations. The completely newly developed drive train features excellent functional features such as the power characteristic and the optimized engine and peripherals package, as well as lightweight design. The output of 31 kW at 7000 min^{-1} is extremely highly homologized in comparison with the competition and allows a very engaged riding style, by simultaneous complying with all legal requirements.

The drive train was developed in close cooperation with the BMW racing team. This meant that the input of the professional racers was incorporated directly in development right from the first engine assembly. The special requirements of Enduro sport thus directly formed the basis for designed loads and general conditions in the simulation and for designing the test programs on engine and assembly test benches.

BMW Motorrad continues to be represented in several Enduro race series with a works team riding this motorcycle.

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Friction Optimisation and its Effect on Specific Fuel Consumption

As part of a complete-engine development project by Mahle, various optimisation measures were investigated on a modern baseline engine and analysed for their ability to improve the friction behavior of the engine. In addition to evaluation criteria such as feasibility and cost, the main emphasis was placed on increasing the effective efficiency in the partial load range. Thus, one of the main criteria during the final analysis of each optimisation measure was a reduction in specific fuel consumption and CO_2 emissions both in stationary operation and during the driving cycle.

1 Introduction

For combustion engines, the reduction of friction losses is a central development objective for future engine concepts. This is evidenced by the fact that in new engine generations, friction losses within the engine are generally lower compared with previous generations, despite increases in specific output and torque. Further reduction of friction losses represents a major challenge in the optimisation and ongoing development of components. Efficient implementation of additional potential for improvement requires an understanding of the tribological principles of the optimisation measures and the ability to apply them in a targeted manner.

One of the primary development objectives for future products of the Mahle Group is the ongoing development of the existing product portfolio also with focus on friction behavior. In addition to new technologies such as assembled camshafts with roller bearings (LFC – Low Friction Camshaft) and detail optimisations on existing products such as pistons and piston rings play an important role during the ongoing development process. In particular, these focus on the interaction of the friction partners (piston rings and cylinder liner) and the influence of design elements such as piston skirt profile and cylinder liner honing. However, for pistons and piston rings especially, contradictory boundary conditions often arise in the application of optimisation approaches due to a plethora of requirements for the complete system. The objective of the investigations presented here was to compile a package of individual measures with optimal complementarity that could be used to achieve measurable friction reductions-despite high demands on functionality and durabilty, and despite a baseline engine that already demonstrated good frictional characteristics at the outset. Finally, these frictional improvements were to be evaluated in terms of their influence on the specific fuel consumption and also in view of feasibility and costs. The investigations were based on a sixteen valve four cylinder gasoline engine with single-stage turbocharging and homogeneous direct injection.

2 Valve Train Optimisation

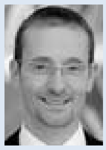
Modern valve train systems already exhibit very low friction levels. Currently, the valve train accounts up to 20 % of the total mechanical friction in gasoline engines. Especially at low engine speeds and loads relevant for the NEDC (New European Driving Cycle), a high percentage of the friction occurs in the valve trains. With increasing engine speeds and loads this relative proportion decreases considerably.

The optimisation measures described here were implemented on a DOHC system with two overhead camshafts and roller cam followers. This valve train design currently represents the most effective system in terms of low friction values. The friction distribution of this type of valve train system is such that friction losses are the greatest at the main camshaft bearing. As a result of the kinematics and the off-center load transmission during the opening and closing phase of the valves, strong shear forces that can lead to mixed friction conditions are produced in the oil film of the hydrodynamic bearings. Due to the roller actuation used, the contact friction that arises during the transfer from cam lobe to cam follower does not have a significant influence on the total friction of the valve train.

The specific optimisation of a DOHC system therefore should be focused on reducing friction in the main bearings of the camshafts. This can be achieved with two different approaches. One involves reducing the bearing loads by lightening the component weights and lowering the valve spring forces. Another approach is to reduce the bearing friction directly through the application of a coating at the plain bearing surface or the use of a roller bearing instead of the hydrodynamic plain bearings. The results of different optimisation measures based on the same test engine are shown in **Figure 1**.

The Mahle lightweight valves can reduce the weight of the intake and exhaust valves by up to 50 %. As a consequence of the lighter component weight, the valve spring rates can be reduced significantly depending on the engine application without negatively impacting the dynamic properties of the valve train.

The Authors



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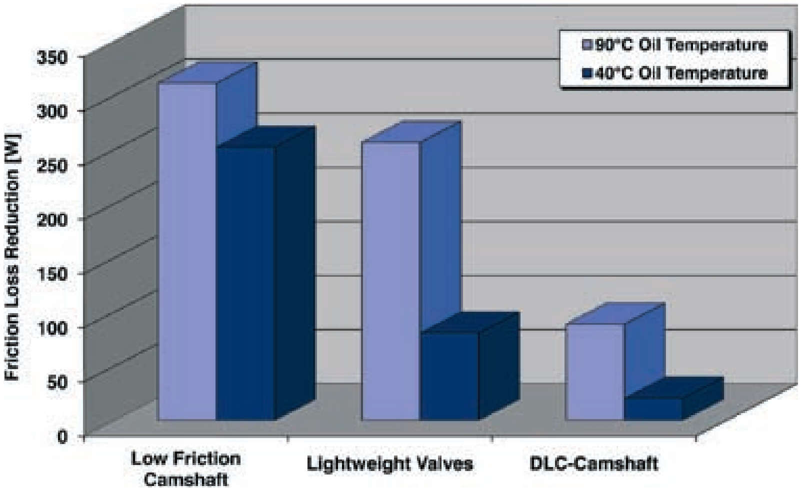

Figure 1: Average reduction in frictional loss resulting from optimisation measures in the valve train of a 16 V four cylinder engine

The lightweight valve itself consists of individual sheet metal components joined together using laser-welding. During the subsequent grinding process lightweight valves could be handled in the same way as conventional valves. The valve design features hollow spaces that can be filled with sodium, for example, to improve cooling on the exhaust side. A significant advantage of lightweight valves is that changes to the cylinder head design are not required to reduce engine friction. By using lightweight valves, friction can be reduced on average by up to 60 W per cylinder (four valves) at an oil temperature of 90 °C. This results in an average friction reduction of 240 W in a four-valve four-cylinder engine.

Another way to optimise DOHC valve train systems is by directly reducing the main bearing friction itself. This can be accomplished in two ways. One involves improving the friction coefficient of the bearing surface by applying a coating. In the second approach, the hydrodynamic bearings are replaced by roller bearings.

In the test engine, a DLC (diamond-like carbon) coating was applied to the camshafts at the main bearing area. This reduced friction in the valve train by 80 W at an oil temperature of 90 °C. Compared to the friction reduction achieved through the use of lightweight valves, only a minor advantage was achieved with this approach. A coating on the bearing surface is only effective in the case of direct contact between the friction partners, for example, in the case of static friction or mixed friction. This condition occurs rarely in hydrodynamic bearings and therefore offers low potential for improvement.

The Low Friction Camshaft is an assembled camshaft with roller bearings [1], **Figure 2**. Mahle has integrated this technology into the standard production process for assembled camshafts, thus enabling efficient and cost-effective production. The roller bearing used is a special needle bearing, which minimises space requirements in the cylinder head. For the needle bearing, no inner ring was used so the needles run directly on the locally hardened cam tube. Also a special bearing cage is used to position the bearing during the assembly process in the cylinder head and has additional grooves for supplying oil to the needle bearings by means of the splash oil in the cylinder head. During the joining process, only fully machined cam lobes are used, thus eliminating subsequent grinding and ensuring the cleanliness of the bearings. The LFC reduces the frictional loss in the valve train by 300 W on average. Comparison of the improvement potential of the LFC with that of the lightweight valves and the DLC-coated camshafts reveals a significant advantage of the LFC over the other technologies, in particular at low oil temperatures, such as occur during the cold start.

In addition to reduced mechanical losses, another advantage of the roller bearing camshaft is that less oil is required in the cylinder head. Because the hydrodynamic bearings are dispensed, less oil is required, and the delivery rate of the oil pump can be reduced. The potential savings that can be achieved by reducing the required oil pump power is equivalent to reducing mechanical losses by using a low friction camshaft.

To enable a detailed examination of the reduced oil consumption when using the LFC, a direct comparison was made on a cylinder head between a camshaft supported by roller bearings and a camshaft supported by plain bearings with the same bearing diameter. The comparison demonstrated that the oil

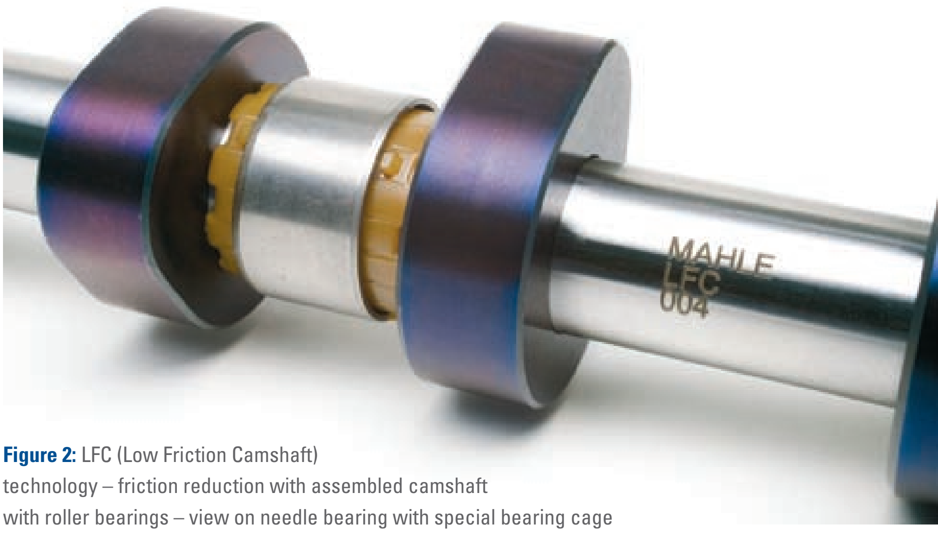

Figure 2: LFC (Low Friction Camshaft) technology – friction reduction with assembled camshaft with roller bearings – view on needle bearing with special bearing cage

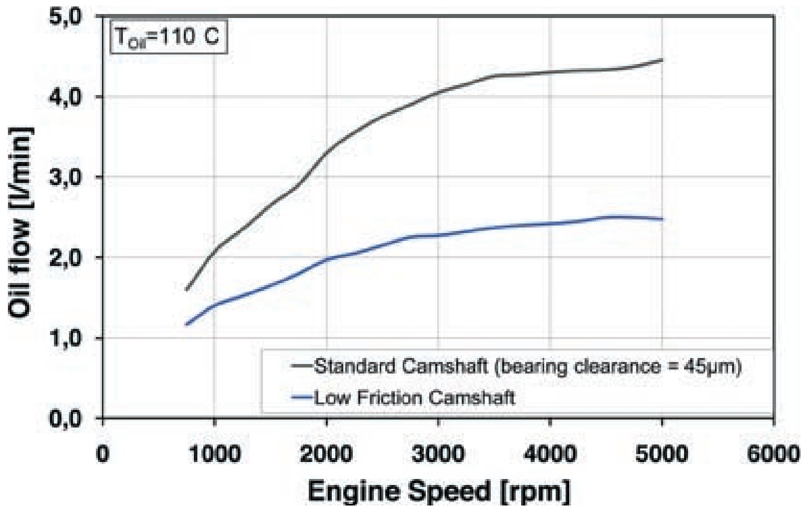

Figure 3: Reduced oil flow in the cylinder head through the use of an assembled camshaft with roller bearings

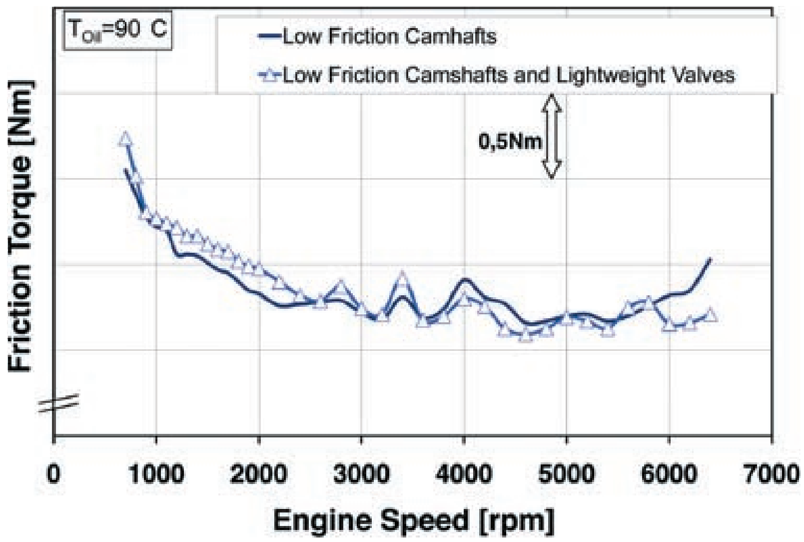

Figure 4: Comparison of Low Friction Camshafts with standard valves and lightweight valves with reduced valve spring forces

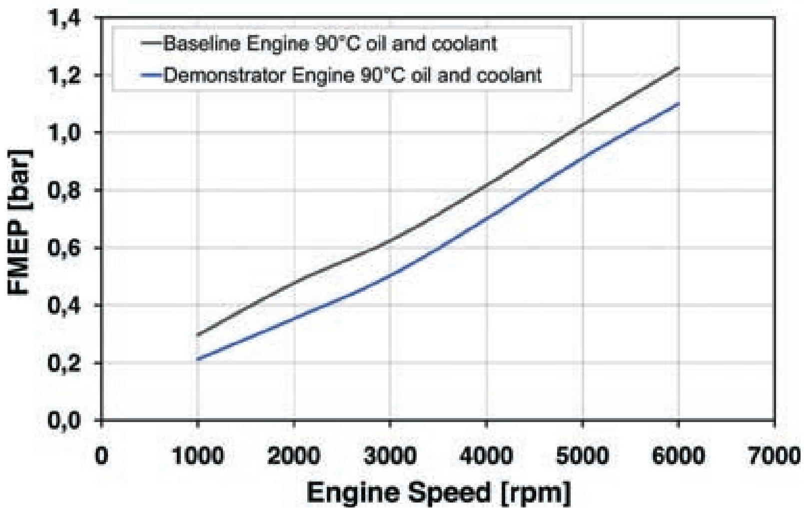

Figure 5: Comparison engine block friction of baseline engine and demonstrator engine

delivery rate in the cylinder head could be reduced by 40 % on average, **Figure 3**.

The use of low friction camshafts and the additional reduction in the oil delivery rate enabled fuel savings of 1 % in the NEDC for a four cylinder engine.

An additional investigation was performed to determine whether it was possible to combine the reduction potential of the LFCs and the lightweight valves. The results, **Figure 4** indicate that frictional losses in the valve train are not further reduced by combining the two technologies. The bearing friction is already reduced to a minimum by the roller bearings, and thus an additional decrease in the bearing load provides at the most only minimal advantages for the total friction in the valve train.

To verify the neutral acoustic behavior of an assembled camshaft supported by roller bearings, structure-borne and airborne sound measurements were performed on an acoustic test bench with a motored cylinder head. In addition to a just slightly increased overall noise level, the measurements revealed a shift to lower frequencies, which are subjectively perceived as more pleasant than high frequencies. The analyses did not identify any critical changes in the noise behavior of the engine due to the use of the LFC.

3 Engine Block with PCU (Power Cell Unit) and Crankshaft

The complete stripped engine including pistons, piston pins and rings, connecting rods and crankshaft accounts for the largest proportion (over 40 %) of friction losses in the engine.

Within the context of PCU optimisation, different piston and ring variants were examined. The optimal configuration was determined to be a combination of Monotherm steel pistons with a Grafal-coated piston skirt, DLC-coated piston pins and an optimised ring set.

The piston ring set comprised new designed rings developed under the boundary conditions of engine operation with the same oil consumption and blow-by values. A ring height reduction of approximately 30 % was achieved, along with a 25 % reduction of the tangential forces. In addition, the steel nitrided top ring

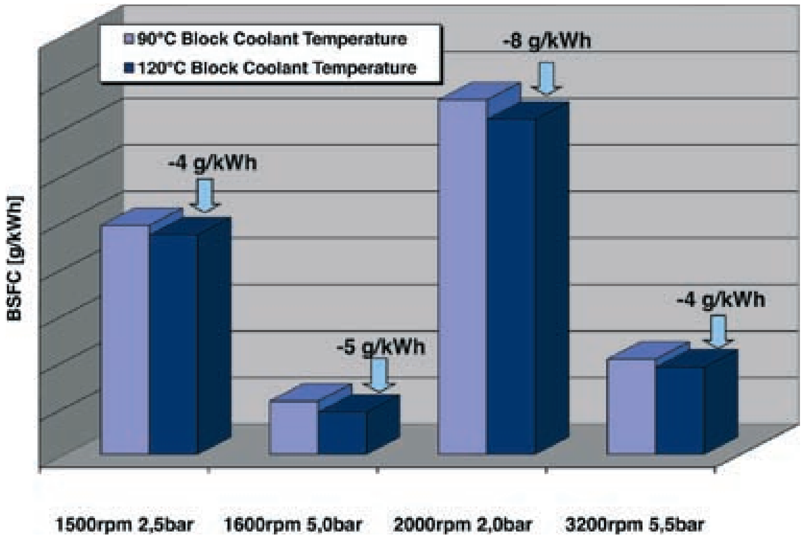

Figure 6: Fuel consumption results with split cooling system

was coated with a PVD (Physical Vapor Deposition) coating on the tread area.

The entire bearing unit of the stripped engine was also reworked. This involved designing new hydrodynamic bearings for the connecting rods and main crankshaft bearings. The objective was to reduce the bearing diameters and widths by 5 % and adapt the bearing design to the new dimensions without imposing limits on the functionality or durability.

By applying the measures described, it was possible to reduce friction related to the stripped engine by 10 to 30 %, which corresponds up to 15 % of the total mechanical friction in the engine at low engine speeds, **Figure 5**.

4 Split Cooling System

The temperature of engine coolant and oil has a direct influence on the piston friction. A higher coolant temperature causes the cylinder to expand further, thus increasing the clearance between the piston and cylinder liner and reducing the overlap. In addition, the hot cylinder wall increases the oil temperature locally and reduces the viscosity. Both these changes have a positive influence on piston friction. However, the coolant temperature cannot be increased without consequence, as it is essential to prevent thermal stress of the cylinder head, especially around the exhaust valves. For this reason, a split cooling system with separate cooling circuits for the block and the cylinder head was developed for the test engine. This split cooling system makes it possible to increase the block coolant temperature without affecting the coolant temperature of the head. To design the split cooling system, CFD simulations were carried out to provide a detailed preliminary analysis of the cooling effect and cavitation behavior. The results were then validated on a test engine fitted with thermocouples. To enable a clearer demonstration of the impact of this individual measure, the final fuel consumption tests with the split cooling system were conducted on the baseline engine. As the results show, fuel consumption savings of up to 8 g/kWh BSFC can be achieved with this system by increasing the cooling temperature by 20 to 30 K at the four part load operating points analysed, **Figure 6**.

5 Demonstrator Engine

A major objective of the development project presented here was to determine the ideal combination of individual optimised components in a complete engine. Two engine configurations were selected to analyse the stationary fuel consumption and the cycle fuel consumption in the New European Driving Cycle (NEDC). The basic setup included the configuration described under Engine block with PCU (Power Cell Unit) and Crankshaft with a valve train containing the low friction camshafts and the lightweight valves with reduced valve spring forces. This combination was selected primarily due to the effective cooling properties of the lightweight valves. Because Monotherm steel pistons were used in the demonstrator engine, the lightweight valves were intended to counteract elevated combustion chamber temperatures.

Figure 7 shows the mechanical friction losses of the demonstrator engine in comparison with the base engine. The mechanical friction losses can be reduced by up to 22 % in low engine speeds through the full use of the optimisation measures.

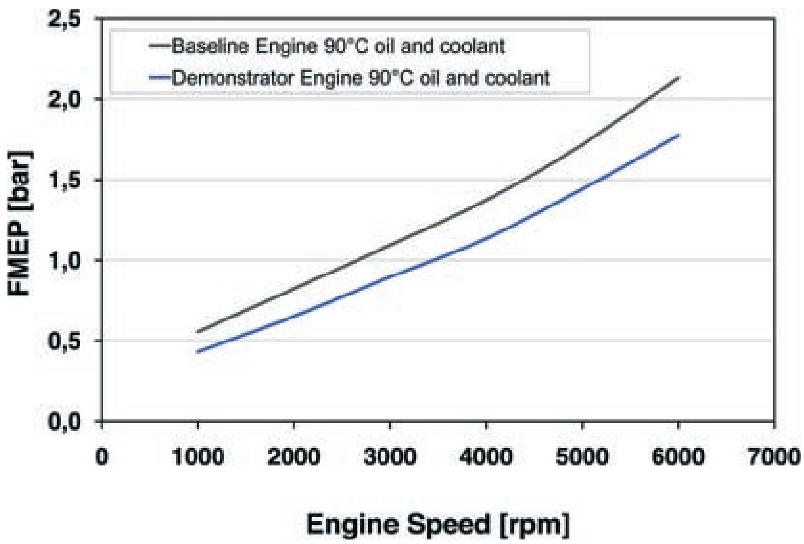

Figure 7: Engine friction – comparison between baseline engine and demonstrator engine

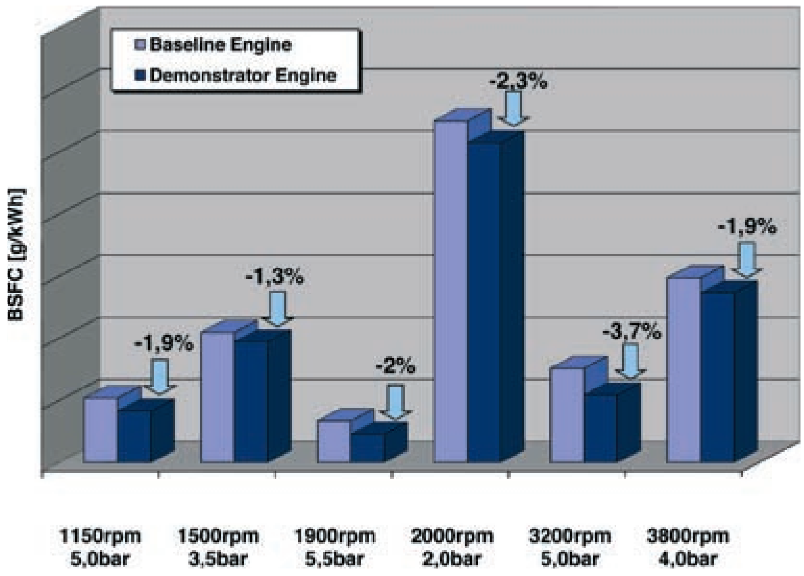

Figure 8: Fuel consumption results of stationary measurements

For the consumption measurements on a stationary test bench, the split cooling system was implemented on the demonstrator engine as well. No additional modifications (Split Cooling System, adapted Oil Pump Design) were made for the cycle consumption analysis, as the main focus here was on measures to achieve mechanical optimisation.

6 Stationary Consumption

The stationary fuel consumption measurements were conducted on a test bench with high-precision conditioning of engine coolant and oil. The operating points to be analysed for the stationary consumption measurements were selected based exclusively on the NEDC-relevant operating range. At these operating points, savings potentials of up to 12 g/KWh were realised for the specific fuel consumption, **Figure 8**. However, in the gasoline engine used, it is not possible to convert these friction improvements completely to fuel efficiency. This is due to the conventional load control by means of a throttle flap, which is the major factor preventing the complete conversion of achieved friction improvements.

The analysis of the fuel consumption is carried out in reference to a specific load point, which corresponds to a specific brake mean effective pressure (BMEP). Because the brake mean effective pressure corresponds to the indicated mean effective pressure minus the frictional mean effective pressure (BMEP = IMEP − FMEP), the indicated mean effective pressure must also be reduced if the frictional mean effective pressure is reduced. To achieve a load reduction with conventional load control using a throttle flap, the engine must be chocked. Consequently, the throttle losses increase and the internal efficiency worsens. As a result, the friction improvements cannot be fully converted in the throttled gasoline engine. In this respect, diesel engines and modern gasoline engines with throttle-free load control offer greater conversion potential.

7 Fuel Consumption Measurements on the Vehicle

Stationary fuel consumption measurements provide initial insight into fuel consumption advantages based on the optimisation measures. Such measurements enable specific observation of the impact of load and engine speed. However, the decisive factor when evaluating optimisation measures is the consumption advantage in the cycle, i.e., in the hands of the customer. For this reason, the demonstrator engine was installed in a midsize vehicle and measured on a certified chassis dynamometer in direct comparison with a base engine.

The New European Driving Cycle (NEDC) was used as a comparison cycle. In this comparison, three complete cycle measurements were performed for each engine variant under standard conditions so as to improve the statistical significance. Despite the reduced scope of optimisation measures, it was still possible to measure a 2.3 % improvement in fuel consumption in the cycle. With an additional reduction in the oil delivery rate and the implementation of the split cooling system on the demonstrator engine, increases in fuel efficiency of 3 to 4 % are realistic.

8 Conclusion

In the development program presented here, we have been able to demonstrate potential for friction reduction and thus optimisation of fuel consumption in the combustion engine through the application of new technologies. We have shown fuel consumption advantages in the NEDC of over 2 % through the use of optimised pistons and piston ring sets in combination with an assembled camshaft with roller bearings or alternative lightweight valves with reduced valve spring forces. An additional investigation of the effect of an oil pump optimised in terms of delivery rate and the split cooling system, combined with the aforementioned measures, demonstrates an overall consumption reduction potential of up to 4 % through the specified friction optimisation measures.

Reference

[1] Lavieuville, M.; Artur, C.; Fix, V.; Schneider, F.; Flender, T.; Kreisig, M.: Low Friction Camshaft – A Solution to Reduce Mechanical Losses in Combustion Engines. Diesel Engine Conference 2008

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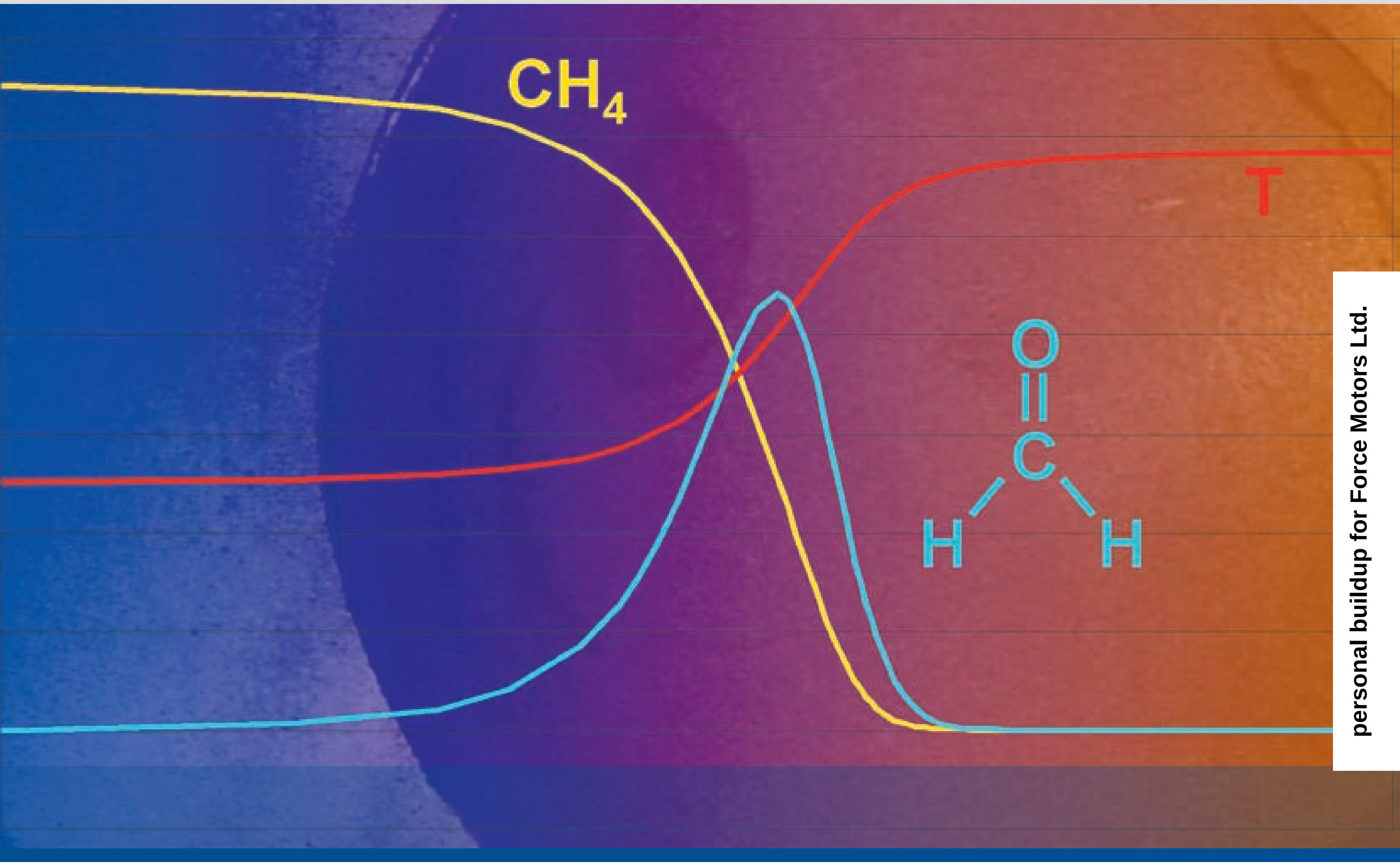

Formation of Formaldehyde in Lean-burn Gas Engines

In recent times stationary gas engines, especially those fuelled with poor gases such as biogas or landfill gas, have shown amounts of formaldehyde (HCHO) emissions exceeding the given limits of the German emission regulation TA Luft. In order to achieve compliance with the emission regulations, research was conducted at the Lehrstuhl für Verbrennungskraftmaschinen of the Technische Universität München to discover the source of formaldehyde emissions as well as determining the influence of internal engine modifications on emissions. This research project No. 918 was initiated by the Forschungsvereinigung Verbrennungskraftmaschinen e. V. (FVV, Research Association for Combustion Engines). For investigations a 4-l and a 1.7-l single-cylinder SI engine were used.

1 Introduction

As a result of the Renewable Energy Act [1], a large amount of biogas plants have been built in the past few years. In these facilities, biomass is fermented into biogas, a mixture of methane, carbon dioxide, nitrogen, oxygen and a number of trace gases. The biogas is used to fuel internal combustion engines in order to generate electric power. Being stationary engines, they are subject to the TA Luft emission standards which restrict the emission of harmful substances [2].

For internal combustion engines the restricted substances comprise of particular matter, carbon monoxide, nitrogen oxides, sulphur oxides and formaldehyde. In the past, measurement of formaldehyde emissions were found to exceed the limit of 60 mg/Nm³ (Nm³ = dry standard cubic meter) in various biogas plants. This led to the cooperative FVV research project "Formaldehyde formation – interactions".

2 Formaldehyde in Combustion

The combustion of hydrocarbons is not a one-step reaction; in fact there are many intermediate reactions. Even methane (CH_4), the simplest form of all hydrocarbons, burns with a number of stable and unstable intermediates, **Figure 1**. Formaldehyde (HCHO) is a stable intermediate that forms in cold regions of the flame at temperatures from 400 to 800 K. Below 1000 K, it is relatively stable but combusts rapidly at temperatures above 1200 K [3].

According to chemical reaction kinetics calculations, the equilibrium mole fraction of formaldehyde in the exhaust gas of stoichiometric and lean methane/air flames is less than 0.1 ppb [5]. This is equivalent to 10^{-4} mg/Nm³ and thus five orders of magnitude smaller than the TA Luft emission standards sets. Therefore the source of the formaldehyde emissions is subject to incomplete combustion of gas/air mixtures.

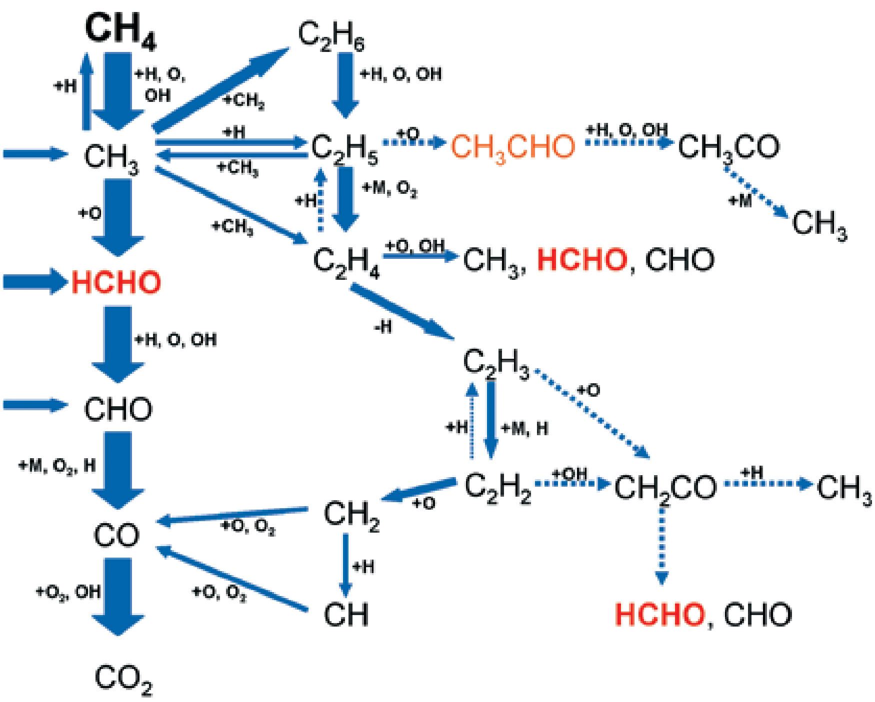

Figure 1: Reaction scheme for the combustion of methane [4]

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Received .... January 20, 2009
Reviewed .... February 4, 2009
Accepted .... February 16, 2009

Table 1: Technical data of the research engines

Gas engine	MTU396	AVL520LVK
Stroke [mm]	185	140
Bore [mm]	165	125
Displacement [cm³]	3956	1718
Number of cylinders	1	1
Compression ratio [1]	12.05	13
Number of valves	4	2
Shape of combustion chamber	omega-shaped piston recess	frustum-shaped piston recess
Swirl number according to Tippelmann	0.57	0.32

3 Test Engines and Test Bench

The engine investigations for the FVV research project "Formation of Formaldehyde" were carried out on two single cylinder research engines at the Lehrstuhl für Verbrennungskraftmaschinen (LVK): an AVL research engine, specifically converted for gas application, and an already gas engine converted MTU396 engine, which has been used in a number of other FVV research projects before. A summary of the engine specifications can be found in **Table 1**. Due to the AVL520's cylinder head and the modifications carried out, the AVL test engine shall further be referred to as AVL520LVK. Its shrouded intake valve enables the variation of the swirl number within a range of $-0.36 \leq D \leq 1.28$ (swirl number according to Tippelmann [6]).

Both engines can be operated with either natural gas or gas mixtures of up to six different gases. The natural gas is taken from the supply net of Stadtwerke München (SWM); the other gas types were used from bottles. The fuel gas is introduced into the charge air pipe via clocked injection valves and a homogenizer (a gas mixture chamber). A vessel (volume: ten times the engine displacement), situated between the homogenizer and the inlet manifold, dampens the pulsations of the intake airflow. The exhaust system comprises a vessel with a volume five times the engine displacement and an exhaust throttle valve in order to adjust turbocharger efficiency by means of backpressure. Ignition is accomplished by a capacitor discharge ignition system and multi-electrode spark plugs. A schematic of the test assembly's set-up is shown in **Figure 2**.

To enable a thermodynamic analysis, the engines are equipped with high and low pressure indication. Cylinder liner and cylinder head temperatures are measured near the combustion chamber wall in several positions. Coriolis mass flow meters are used to measure mass flow of the fuel gases. Air mass flow is measured with a rotary gas meter. For emission analysis a FT-IR spectroscope (Fourier transformed infrared spectroscopy) of the type "AVL Sesam IV" is used in addition to the conventional motor exhaust gas analysers. Both engines are equipped with wideband lambda sensors (Etas LA-3/4).

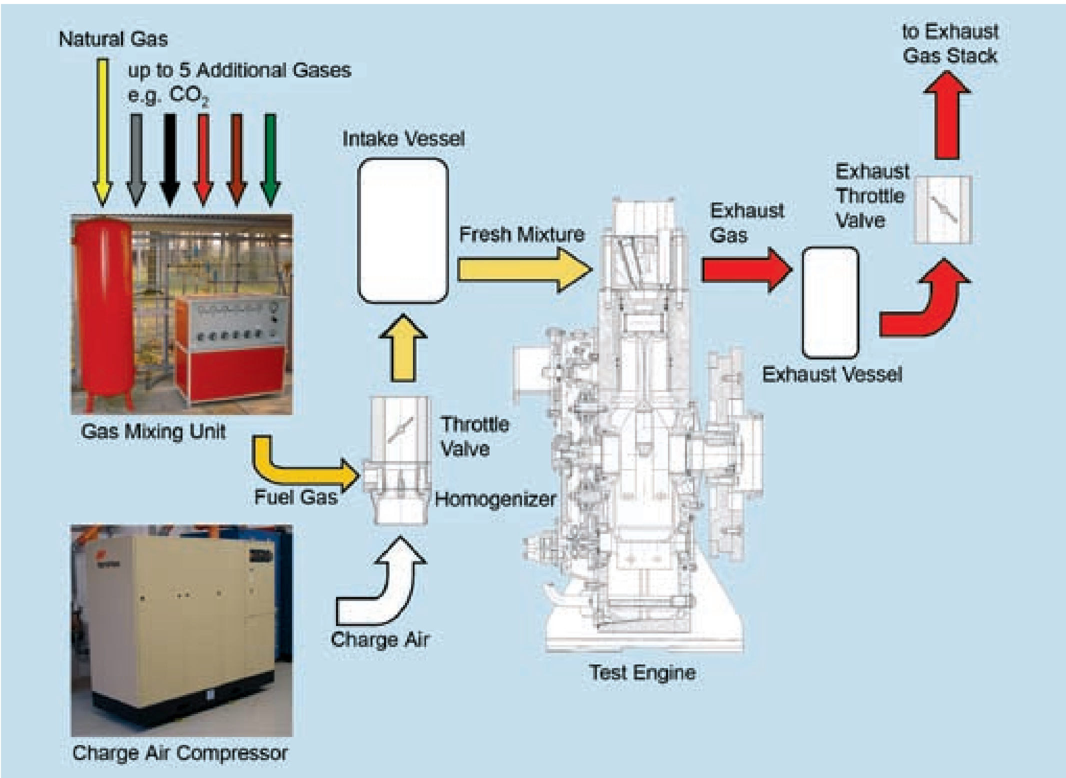


Figure 2: Schematic of the test assembly's set-up

4 Engine Tests

Various operating parameters and engine modifications have been analysed regarding their influence on formaldehyde emissions. In basic experiments the effects of engine load, excess air ratio and ignition point (IP) on formaldehyde emissions were investigated. Based on the experimental findings, further research was performed on the effect of operating and design parameters influencing the amounts and characteristics of formaldehyde emissions.

4.1 Methodology of Testing

The test engines run at a constant 1500 rpm, in steady-state conditions. For each investigated parameter the IP was varied. Within the basic experiments charge air pressure, excess air ratio and exhaust back-pressure were kept constant. This approach leads to a change in turbocharger efficiency, engine load and the emission of nitrogen oxides (NO_X) within a variation of ignition timings.

In order to show the full potential for the reduction of HCHO emissions at constant NO_X emissions a reverse approach was carried out. This meant keeping turbocharger efficiency, engine load and NO_X emissions constant for each IP, while other influential parameters were varied.

4.2 Basic Measurements

For these investigations, both engines were operated at low and medium charge pressures. In **Table 2** charge pressures (absolute pressure in air intake, p_{ai}) and excess air ratios (measured with lambda sensor, λ) are shown.

The leaner the air-fuel mixture at a constant charge pressure is, the more formaldehyde both engines emit. This effect is more pronounced for higher charge-air pressures than for lower ones, **Figure 3**. But there is no explicit correlation between charge air pressure and the formaldehyde emissions. Adversely, the emission of total hydrocarbons (THC) rises with increasing charge air pressure.

Rich mixtures show a decrease in formaldehyde emissions for late ignition timings. For increasingly lean mixtures, their characteristics remain approximately constant during early ignition timing, but increase to some extent significantly the later the timing (leanest mixtures with the MTU396). Due to the lean air-fuel ratio and the combustion lasting long into the expansion stroke, the flame quenches, which shows through the sudden rise of THC emissions.

The AVL520LVK engine emits larger amounts of formaldehyde than the MTU396. Amongst others, this is caused by the missing exhaust vessel on the AVL-520LVK during the basic experiments. This causes a larger amount of in-cylinder residual gas so the conditions of flame propagation decline, which leads to higher emissions of formaldehyde.

4.3 Variation of Fuel Gas

The influence of the fuel gas quality on formaldehyde emissions was researched on the basis of two gas mixtures consisting of natural gas and carbon dioxide. These two mixtures are equivalent to biogas with a high ("Biogas1") and biogas with a low ("Biogas2") net calorific value (NCV), **Table 3**. Because the natural gas supplied by the SWM consists to 98 % of methane, the synthetic biogas could be mixed from carbon dioxide and natural gas.

The variations of fuel gases on the MTU396 were carried out analogue to the basic experiments with matching mix-

Table 2: Charge air pressure and excess air ratio for basic experiments

Test engine	MTU396			AVL520LVK		
	p_{ai} [bar abs.]			p_{ai} [bar abs.]		
λ = 1.40	XXX	XXX	XXX	1.14	1.61	1.93
λ = 1.50	1.23	1.63	2.03	1.14	1.61	1.93
λ = 1.60	1.23	1.63	2.03	XXX	1.61	1.92
λ = 1.65	1.23	1.63	2.03	XXX	XXX	XXX

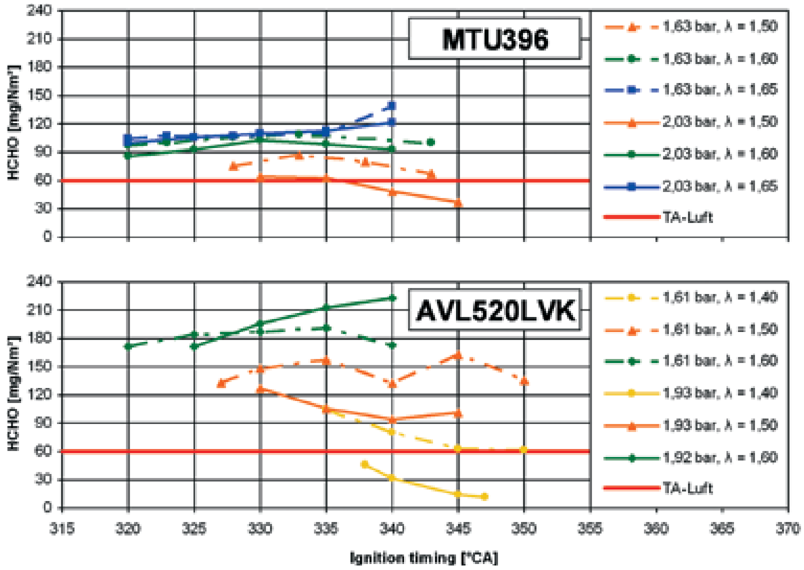


Figure 3: Emissions of formaldehyde in basic experiments for both research engines

Table 3: Mixtures of fuel gases; components' fractions

Fraction	Natural gas [% by vol.]	CO_2 [% by vol.]	NCV [MJ/kg]
Natural gas	100	0	49.1
"Biogas1"	70	30	23.0
"Biogas2"	50	50	13.4

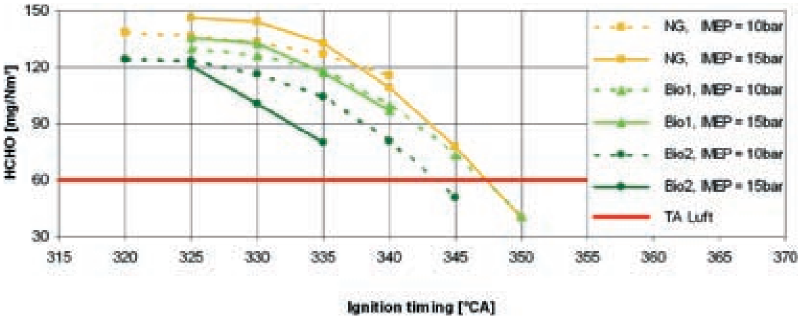


Figure 4: Emissions of formaldehyde for variation of fuel gases at AVL520LVK

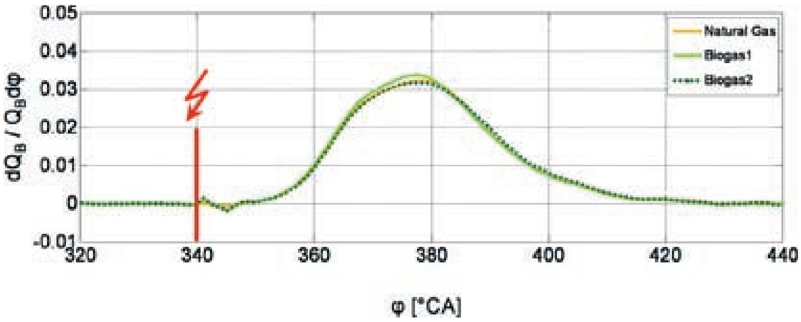


Figure 5: ROHR of the three fuel gases at IMEP = 10 bar and IP = 340° CA

ture calorific values. This was achieved by adjusting the excess air ratio. Due to the limited gas mass flow through the homogenizer, “Biogas1” was the only fuel gas mixture researched.

Fuelled with “Biogas1”, the MTU396 shows similar characteristics as with natural gas, and produces approximately the same amount of formaldehyde emissions. Even though the characteristics are similar, they shifted towards an advanced IP by several degrees of crank angle (°CA). Furthermore, the duration of combustion of “Biogas1” and the ignition delay are longer and the centre 50 % MFB (mass fraction burned) is later than fuelled with natural gas. The effects on the rate of heat release (ROHR) are caused by the inert gas fraction in the fuel gas and lead to the offset in the characteristics of formaldehyde emissions.

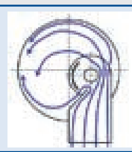
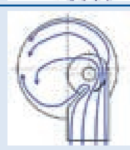
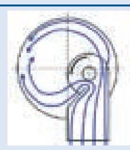
With the AVL520LVK, the influence of fuel gases was researched after mounting the exhaust gas vessel which had not been done for the basic experiments. Ignition timing was varied for indicated mean effective pressures of IMEP = 10 and 15 bar at constant NO_X emissions of 500 mg/Nm³ and turbocharger efficiency of η_{ETC} = 70 %.

Fuelled with the biogas mixtures, all investigated engine loads and ignition timings emit less formaldehyde than when fuelled with natural gas, **Figure 4**. This is mainly because the biogas mixtures allow the usage of significantly richer mixtures at constant NO_X emissions. According to TA Luft standards, all emission values have to be referenced to 5 % oxygen content in the dry exhaust gas, thereby favouring rich mixtures. This mathematical conversion reduces the formaldehyde emission values of biogas given in mg/Nm³, though the ppm values in humid exhaust gas are partially higher than those of natural gas.

All fuel gases show an almost constant trend of formaldehyde emissions over advanced ignition points, but decrease towards late IPs. Constant NO_X emissions allow richer mixtures at late IPs inhibiting the flame to quench early so that the emissions of formaldehyde do not rise. Apart from “Biogas2” at IMEP = 15 bar, there is no shift of the characteristics of formaldehyde emissions with the biogas mixtures. With the MTU396 the conditions of flame propagation decline for “Biogas1”, though the mixture calorific value is held constant. In comparison to natural gas, the NO_X emissions are lower. These emission values are held constant on the AVL520LVK allowing the investigation of rich mixtures beyond constant mixture calorific value. The improved conditions of flame propagation manifest in the almost perfectly matching ROHR of all three fuel gases, **Figure 5**.

Increasing the engine load only has a positive effect on the emissions of formaldehyde at late IPs. In the case of advanced ignition, a negative effect is noted. The reason is found in the excess air ratio: higher engine loads result in higher NO_X emissions, which can be compensated by a leaner mixture. This causes the degradation of the flame propagation conditions, thus increasing formaldehyde emissions.

Table 4: Investigated swirl numbers and positions of shrouded intake valve (schematically)

Name	Swirl number according to Tippelmann (Paddle wheel)	Position of shrouded intake valve (schematically)
S_{ref} Reference swirl, initial value	0.32 (1.49)	
S_{equ} Equivalent swirl	0.31 (1.37)	
S_{low} Low swirl	0.10 (0.43)	

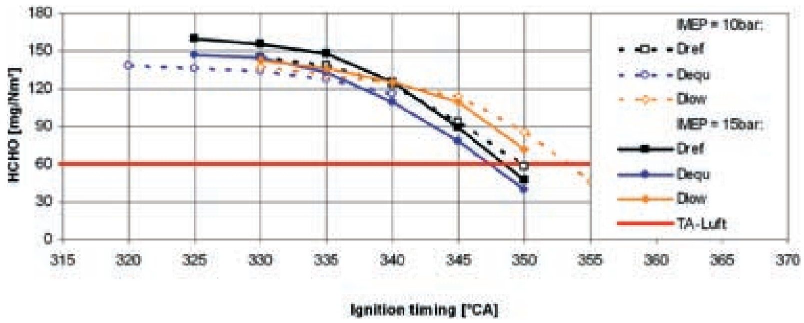


Figure 6: Influence of swirl variation on the characteristics of formaldehyde emissions

4.4 Variation of Swirl

In addition to the original shroud intake valve position on the AVL520LVK engine, the influence of swirl was investigated on two other positions. Both positions were defined on short tests' data obtained during the basic experiments. **Table 4** shows the examined shroud positions and the corresponding swirl numbers. Each swirl number features familiar formaldehyde emission characteristics, as known from the variation of fuel gases. Relatively constant emissions at early IPs are followed by a decrease at late IPs, **Figure 6**. The swirl numbers S_{ref} and S_{equ} show the same characteristics while S_{equ} has slightly lower emissions. The S_{low} characteristic shows a 5° CA shift towards later IPs and a quicker combustion (not shown graphically).

Higher loads lead to an increase of formaldehyde emissions during early ignition times, which decreases towards later IPs, as seen before in the variation of fuel gases. Within the investigated range of ignition timing the excess air ratio is greater at high than at low loads.

4.5 Variation of Compression Ratio

In comparison to the original compression ratio of ε = 13, two further ratios ε = 12 and 15 were investigated. This was achieved by changing the diameter and depth of the piston recess, thereby keeping the manipulation of in-cylinder flow conditions to a minimum. All investigated compression ratios show the well-known characteristics of formaldehyde emissions, **Figure 7**. The compression ratio of ε = 12 leads to lower emissions of formaldehyde than ε = 13, because lower peak in-cylinder pressures and temperatures allow an enriched air-fuel mixture, still keeping within NO_x-emission limits.

The opposite is the case with the higher compression ratio of ε = 15, as higher peak in-cylinder pressures and temperatures cause much leaner mixtures with the effect of significantly higher formaldehyde emissions. This lean mixture is also the reason for the remarkable fact that indicated efficiency is not improved by increasing the compression ratio.

Due to the very lean mixture leading to declining conditions of flame propagation the combustion at ε = 15 is largely incomplete, also resulting in high THC emissions. The reduced fuel utilisation levels the potential gain of efficiency induced by the increased compression ratio. At late IPs a richer mixture leads to significantly improved conditions of flame propagation causing the HCHO emissions to decline almost to the levels of ε = 12 and 13 and, at the same time, the indicated efficiency to exceed those of the both lower compression ratios.

4.6 Effect of Top Land Crevice

The top land crevice is a major contributor of spark ignition (SI) engines' unburned hydrocarbons emissions. During compression and combustion, fresh air-gas mixture, whose temperatures can reach activation energy levels enabling the formation of formaldehyde, is trapped in top land crevices. On the piston's down-stroke the trapped gases flow back into the combustion chamber. If these gases contain formaldehyde and the further oxidation process stalls, it will be part of the exhaust gas. Moreover, the formation of formalde-

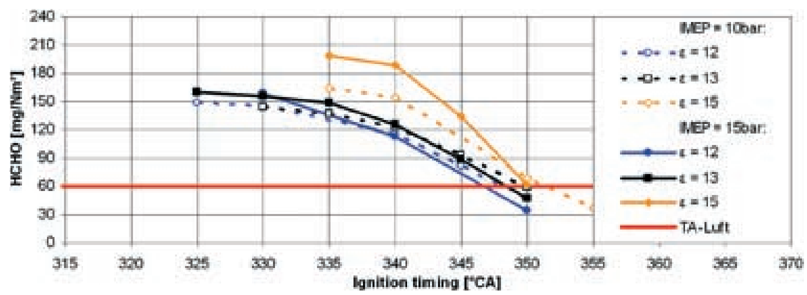


Figure 7: Emissions of formaldehyde at variation of compression ratio

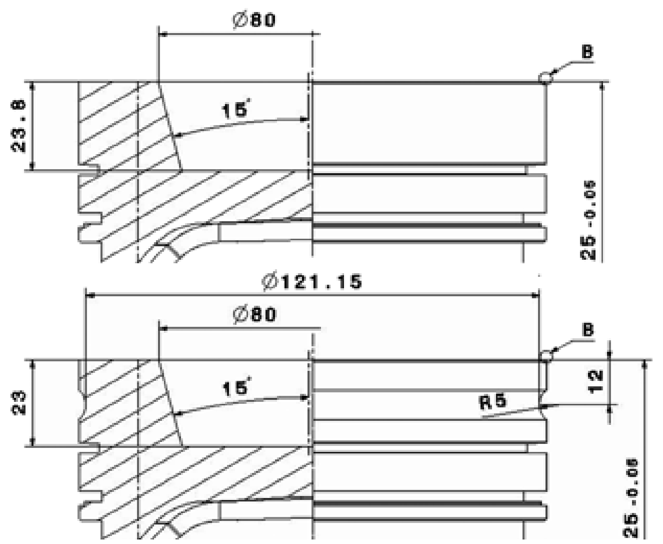


Figure 8: Geometry of the original piston (above) and the piston with increased volume of top land

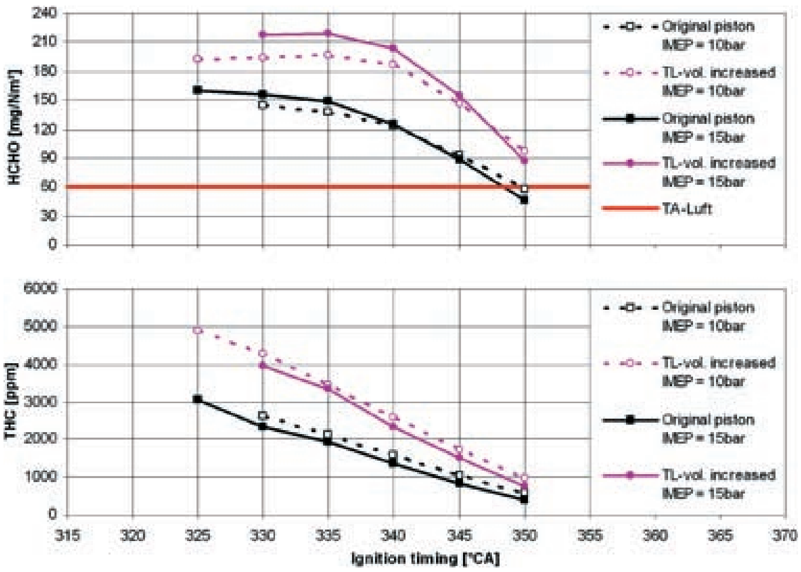


Figure 9: Formaldehyde (top) and hydrocarbon emissions (bottom) with enlarged top land volume

hyde can also occur by partial oxidation of Methane flowing back out of the top land crevice into the combustion chamber.

To investigate what effect the top land crevice has on formaldehyde emissions, an arc-shaped groove was cut into the piston, doubling the top land crevice's volume. As a result the compression ratio declined, which was compensated by reducing the depth of the piston recess, **Figure 8**.

Two indicated mean effective pressures where investigated, IMEP = 10 and 15 bar. The familiar characteristics of the formaldehyde emissions do not change with the doubled volume of the top land crevice, but the emitted formaldehyde amounts are significantly higher. The same can be said about the hydrocarbon emissions, which increase with advanced IP, **Figure 9**.

Assuming a direct proportional relation between the volume of the top land crevice and the total rise of the thereof originating emissions, one can estimate the fraction of HCHO and THC emissions that derive from the top land crevice. The formaldehyde emissions from the top land first rise with advancing ignition and then lessen after reaching a maximum. In contrast, the hydrocarbon and formaldehyde emissions from other sources steadily increase with advancing ignition timings, **Figure 10**. So, formaldehyde and hydrocarbon emissions from other sources show the familiar characteristics, whereas emissions originating from the top land show significant characteristic differences. This leads to the conclusion of a complex relation between the amount of formaldehyde emissions out of the top land and the in-cylinder conditions during compression, combustion and expansion. In order to provide more detailed information about these mechanisms, additional research needs to be pursued. At late IPs the top land is the dominant source for the formation of formaldehyde, up to 85 % of the total amount. Advancing the ignition timing reduces the contributed amount to appreciable 34 %. However, in terms of hydrocarbon emissions the top land is still the major source over the scope of the investigated ignition timing.

4.7 Variation of the Combustion Chamber's Geometry

A spherical shaped piston recess as shown in **Figure 11** was investigated. The surface/volume ratio was held constant and the quench area was reduced by half. The ROHR increases significantly with this piston, leading to an earlier 50 % MFB position. For this reason the formaldehyde emissions characteristic shift towards later ignition timing. The amount does not change significantly, so we can forego showing the results in a diagram.

4.8 Further Parameters

Research was done on the influence of exhaust gas pressure, charge air temperature and coolant temperature on formaldehyde emissions. A distinguishable trend to reduce formaldehyde is achieved by increasing charge air and coolant temperatures as well as reducing exhaust gas pressure, at the same time the NO_x emissions increase. Leaning fuel mixtures to reduce NO_x emissions to the original values raises the emissions of formaldehyde to their original values and partly beyond.

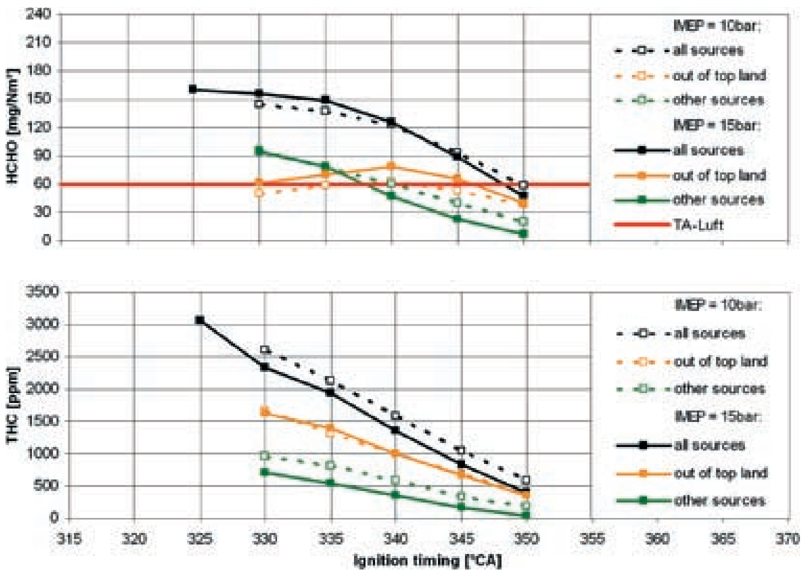


Figure 10: Distribution of formaldehyde (top) and hydrocarbon emissions (bottom) to top land and other sources

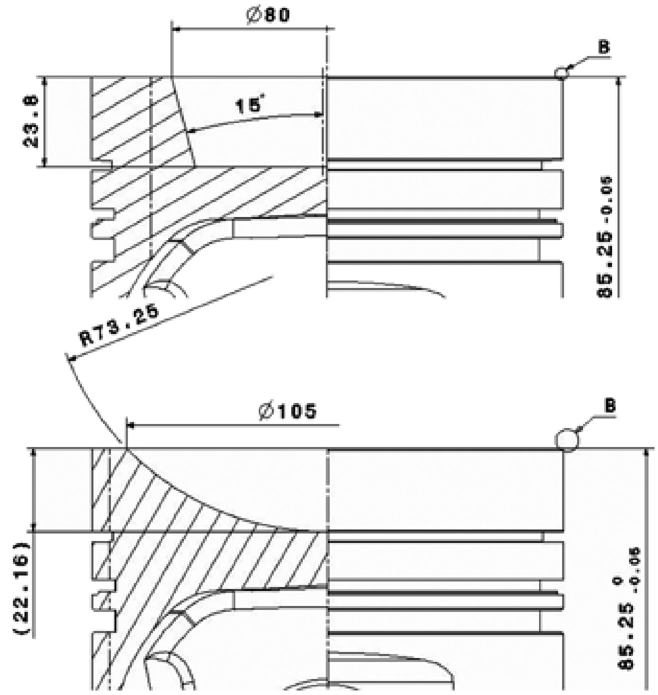


Figure 11: Geometry of the original piston (top) and the piston with spherical shaped piston recess (bottom)

For some engine operating points the exhaust-gas composition was also investigated after flowing through a catalyst from Süd Chemie AG. As a result of this catalytic after-treatment the emissions of formaldehyde were reduced considerably.

5 Summary

In the scope of the FVV research project "Formaldehyde formation – interactions" (No. 918), numerous operating parameters and engine modifications were investigated in regard to their influence on lean-burn gas engines' formaldehyde emissions. The basic experiments were carried out on two single-cylinder engines, and showed fundamental relations between formaldehyde emissions, excess air ratio and ignition timing. Lowest formaldehyde emissions were reached with rich mixtures and a high charge air pressure at late ignition timing.

Various fuel gases with different percentages of carbon dioxide had the same characteristics in formaldehyde emissions when emissions of nitrogen oxides were held constant. In this case, formaldehyde emissions remained constant in the early range of ignition points, and declined towards later ignition points. A proportional increase of carbon dioxide in the fuel gas resulted in a drop of formaldehyde emissions due to an enriched air-fuel mixture.

Reducing swirl accelerated the combustion. This did not influence the amount of formaldehyde emitted, but led to an offset of the formaldehyde emissions' characteristics to later ignition timing. Changing the geometry of the combustion chamber had the same effect, shifting the characteristics but not the amount of formaldehyde emissions.

A reduced compression ratio only showed a minor effect on the formaldehyde emissions, whereas an increased compression ratio resulted in significantly higher emissions of formaldehyde. This was caused by leaning the mixture in order to reduce emissions of nitrogen oxides. The conditions for flame propagation deteriorate and, besides the higher emissions of formaldehyde, lead to a reduction of thermal efficiency and an increase of hydrocarbon emissions due to incomplete combustion.

Experiments with an increased top land volume on a piston identified the top land to be a significant, towards late ignition timings the dominant source of formaldehyde emissions. In contrast to the emission of the other sources, no coherence between formaldehyde and the THC emissions concerning the top land could be established. Varying charge air temperature, coolant temperature or exhaust gas pressure had no mentionable influence on the formaldehyde emissions, as long as emissions of nitrogen oxides were held at a constant level.

Exhaust-gas measurements behind a catalyst supplied by Süd Chemie AG showed a considerable reduction of formaldehyde emissions due to catalytic after-treatment.

Apart from reducing the volume of the top land crevice, which is structurally limited, no parameters were found capable of reducing formaldehyde emissions without taking the disadvantages of other parameters into account (nitrogen oxides, thermal efficiency and exhaust gas temperature). It is therefore necessary to continue practical and theoretical research, to comprehend the involved mechanisms that define the characteristics of formaldehyde emissions deriving from top land crevices, as was discovered in this research project.

Acknowledgements

This investigation was performed in the context of a research project (No. 609180) by the Forschungsvereinigung Verbrennungskraftmaschinen e. V. (FVV) which was accompanied and supported by a working group populated by representatives of the industry. The authors would like to express their thanks to the working group and its leader, Dipl.-Ing. Heinrich Baas of MWM GmbH, for substantial support. Special thanks are also extended to GE Jenbacher for providing materials.

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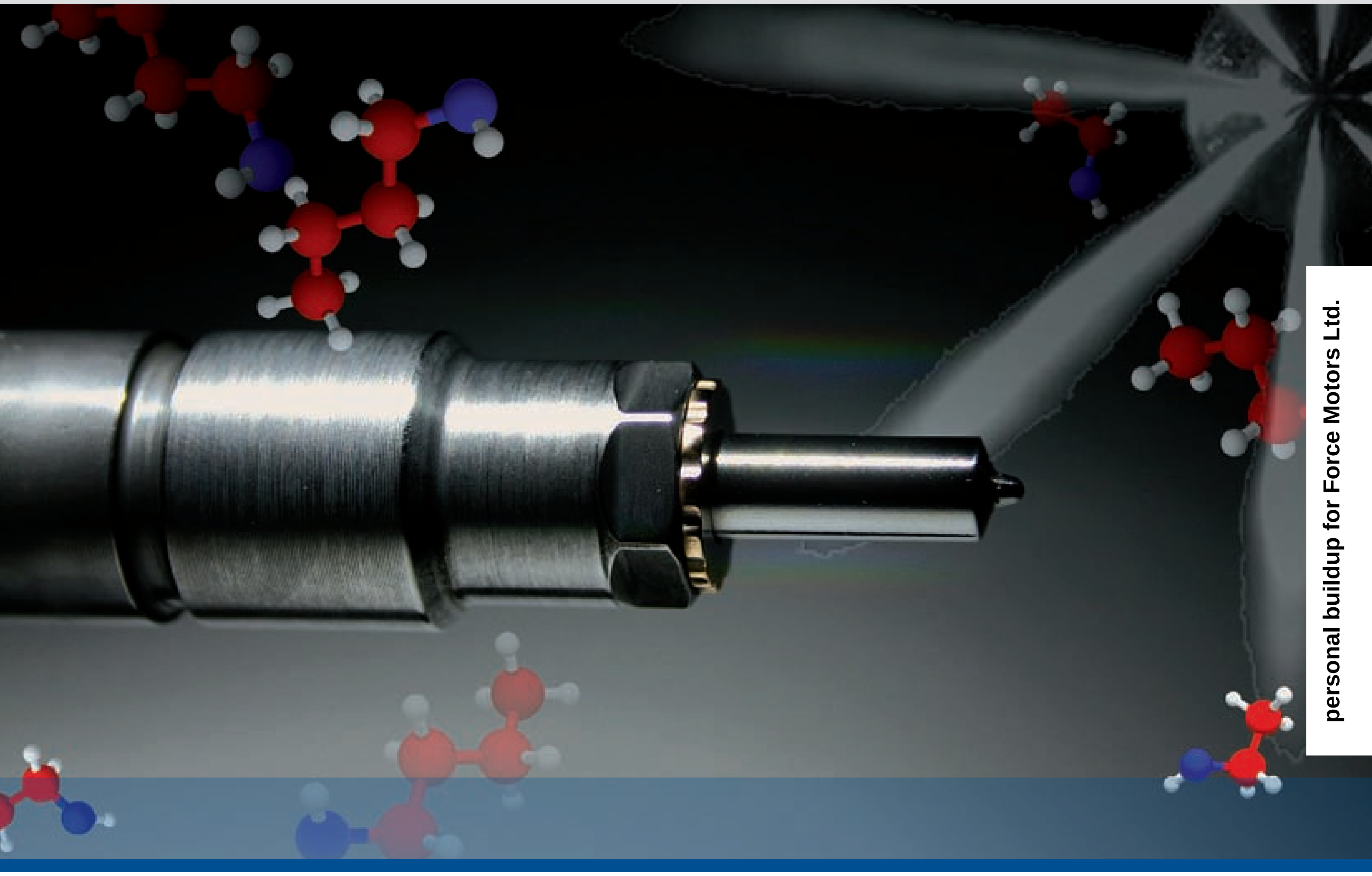

Alcoholic Biofuels as an Admixture Component for Conventional and Alternative Diesel Combustion Processes

Biogenic admixture increases the possible savings in carbon dioxide emissions and reduces the dependency on fossil fuels. In this context especially alcohols not only provide the opportunity to optimize the trade-off between nitrogen oxides and particular matter (NO_x/PM trade-off) with conventional combustion processes but they can also support alternative combustion processes by their advantageous properties concerning reaction kinetics. At the Institute of Internal Combustion Engines of the TU Braunschweig investigations on a single-cylinder heavy duty research engine regarding FAME/ethanol and butanol admixtures employing conventional and alternative combustion processes were carried out.

1 Introduction

Besides the reduction of legally limited exhaust gas components, decreasing carbon dioxide emissions continuously gains relevance due to the debate about climate protection. In addition to the reduction of fuel consumption, the decrease of carbon dioxide emissions of mobile applications in Europe by means of biogenic admixtures is advanced by the legislator. Currently the German bioquotas-law (BioKraftQuG) prescribes a biofuel share of 4.4 % in terms of energy for diesel fuel and 2.8 % for gasoline, correlating with 5 respectively 4.3 % volumetrically. At present, actually biofuels as rapeseed methyl ester (Rapsölmethylester, RME) fatty acid methyl ester (FAME) in case of diesel fuel and ethanol in case of gasoline are used. Further abbreviations and symbols are shown in **Table 1**.

The bioquotas-law provides for an increase in the biogenic share of gasoline to 3.6 % in 2010. Initially a quota of 6.25 % in 2009 was provided by law, intended to be increased step by step up to 8 % in 2015 [1]. Due to problems concerning engine and exhaust gas aftertreatment compatibility the total quota was reduced to 5 % in 2009 at short notice. A current draft law provides for a fixation of the total quota at 6.25 % and of the ethanol quota at 2.8 % from 2010 until 2014 [2]. Current announcements of the EU even provide for a total quota of 10 % in 2020 [3]. However, the vehicle manufacturers specify the maximum compatibility to 7 % vol. for diesel and 5 % vol. for spark-ignition engines, correlating with a total quota of 5 % in terms of energy.

Besides technical limits, the availability of biofuels appears to be a further obstacle on the way to fulfillment of the prescriptions by law, **Figure 1**. The comparatively low yield in diesel fuel equivalent per acreage inevitably leads to a sharpening in the competition between biodiesel and food production. At present, diesel fuel and gasoline are produced simultaneously in a joint production process causing a gasoline surplus with rising diesel fuel demand. Currently this oversupply can still be sold overseas, but this sales potential is expected to shrink as the local ethanol production increases.

Thus the concept of an investigation at TU Braunschweig regarding the suitability of biogenic gasoline substitutes in shape of alcohol/diesel blends for CI engine applications emerged. A single cylinder diesel engine was used as the research engine. The reduced ignitability and increased volatility caused by admixture of alcohol implied a combination with alternative combustion processes whose fuel requirements are actually quite similar to those of the alcohol/diesel-blends. As appropriate alcohols, ethanol and butanol were chosen, whereas on the part of the industry biogenic butanol is said to be producible in commercial scale by the beginning of 2010 [4].

2 Research Engine

The investigations were carried out on a single cylinder research engine with 2.059 l displacement. Engine components concerning the combustion process like piston and cylinder head originate from the MAN D28-series. In case of the test series presented in this article, the compression ratio was reduced from $\varepsilon = 16.5$ to $\varepsilon = 14$ in order to provide for both the conventional and (partially) homogeneous combustion process without the need of a change in the engine setup. This was achieved by means of a piston featuring an enlarged combustion bowl, taken from the race truck engine also based on the MAN D28. The main engine specifications are shown in **Table 2**.

The injection system consisted of D28 series injectors and a Bosch CP3.3 high pressure pump. With all measurements, standard measurement equipment for exhaust gas emissions and pressure indication was employed. In order to evaluate the emissions of particulate matter, the Filter Smoke Number (FSN) was determined. **Figure 2** shows the test bench setup.

3 Combustion Process

The conventional combustion process employing diffusion combustion is well described in [6] for example. After discharge of the fuel, which is comparatively cold compared to the combustion chamber temperatures, from the nozzle

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MTZ Peer Review
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Reviewed by experts from research and industry.

Received .... April 03, 2009
Reviewed .... April 14, 2009
Accepted .... May 08, 2009

Table 1: Abbreviations and symbols

A/F	Air/Fuel	**PM**	Particulate Matter
ATDC	After Top Dead Center	q_m	fuel mass injected per stroke / mg/stroke
B	n-Butanol	$q_{m,hom}$	homogeneously burnt fuel mass / mg
CA	Crank Angle	$q_{m,het}$	heterogeneously burnt fuel mass / mg
DF	Fossil Diesel Fuel	**RME**	Rapeseed Methyl Esther
E	Ethanol	**ROHR**	Rate Of Heat Release / J/° CA
EGR	Exhaust Gas Recirculation	**SC**	Split Combustion
FSN	Filter Smoke Number / -	**SOI**	Start of Injection / ° CA ATDC
HDC	Homogeneous Diesel Combustion	**SOA**	Start of Actuation / ° CA ATDC
HR_{50}	center of 50 % energy-conversion mass / ° CA	**TDC**	Top Dead Center
ISFC	Indicated Specific Fuel Consumption / g/kWh	T_m	mass average temperature / K
IMEP	Indicate Mean Effective Pressure / kPa	ε	Compression ratio / -
$IMEP_{HP}$	Indicate Mean Effective Pressure in HP loop / kPa	λ	Relative A/F ratio / -
n_M	Engine speed / rpm	λ_S	Relative A/F ratio at starting point / -
NCV	Net Calorific Value / MJ/l	φ	Crank position / ° CA ATDC
$p_{injection}$	injection pressure / MPa	$\varphi_{SOA,MI}$	Crank position of start of actuation of main injection / ° CA ATDC
p_{intake}	intake pressure / kPa		

air is absorbed into the fuel jet ("air-entrainment"). A cloud of very rich air-fuel mixture forms around the liquid core of the fuel jet which ignites after chemical pre-reactions. This leads to the characteristic products of a rich combustion, including carbon monoxide, unburned and partially burnt hydrocarbons as well as primary particles. Continuous growth of particles follows downstream. Subsequently these products are pushed ahead to the border of jet, which is surrounded by a diffusion flame. Here the final conversion proceeds, in which CO, HC and particles are ideally oxidized to CO_2 and water. Very high temperatures occur in this process, enhancing the formation of nitric oxides in an environment with high oxygen content. The particles produced cannot be oxidized entirely due to the short dwell time in areas of sufficient temperature.

To avoid the problems of conventional combustion the intensive search for alternatives to a conventional combustion process with pilot injection started. The homogeneous diesel combustion (HDC) represents one possible solution in order to minimize NO_x and particle emissions. The results with HDC displayed in this article are based on the investigation published in [7]. Besides the limited uti-

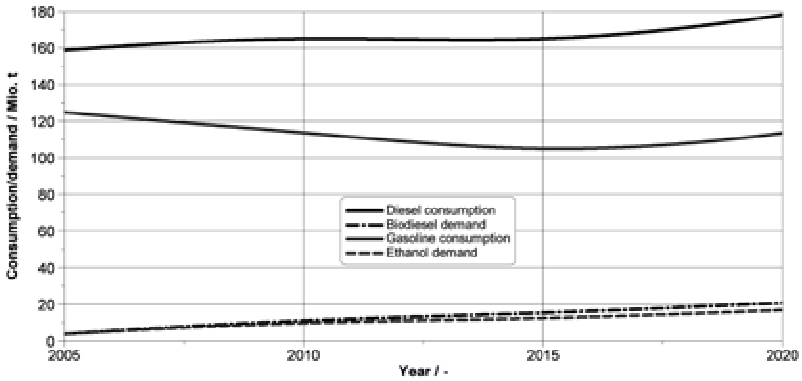

Figure 1: Fuel consumption and demand in the European Union [5]

Table 2: Special specifications of the single cylinder research engine used

Project name	IVB 1
Basis	MAN D28
Bore in mm	128
Stroke in mm	160
Connecting rod length in mm	251
Ignition pressure limit in MPa	22
Compression ratio	14,0
Number of valves	4
Injector	Bosch CRIN 2
Nozzle	8-hole, 155°-cone, 800 cm³/ 10 MPa / 30 s
Rail pressure in MPa	200

lizable range of load up to approximately IMEP = 800 kPa, especially the very complex control of the combustion process by means of high EGR rates was identified as the major challenge.

Also in [7], a partially homogeneous combustion process called split combustion (SC) was introduced. Without the application of EGR the exhaust gas emissions could be reduced significantly employing this combustion process, by means of moderate EGR-rates even below Euro V limits. As well as for the homogeneous processes advantages for the partially homogeneous combustion by fuels containing alcohol were anticipated. The various injection patterns are depicted in **Figure 3**.

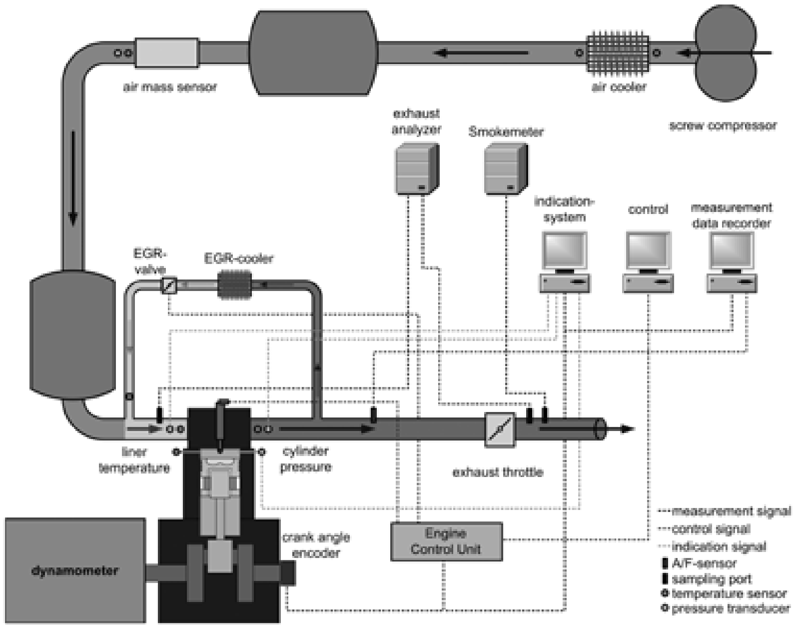

Figure 2: Schematic test bench setup

4 Fuels and Blends

Due to its polarity ethanol is only miscible with fossil diesel fuel to a very limited extent, resulting in the separation of a second phase in the mixture without the use of an additive. This behavior is depicted in **Figure 4** and illustrated by markers. Besides specific additives, however, RME can be used as a solubilizer to produce stable micro-emulsions. Examinations have shown that depending on the ambient temperature at least as much RME as ethanol has to be blended [8]. Regarding the upper limit of 7 % biodiesel share declared by the vehicle manufacturers, this was rounded up to a combined biogenic share of 15 % volumetrically.

In contrast to ethanol, butanol shows a nearly unlimited miscibility with fossil diesel fuel without any separation of a second phase at ambient temperature. Consequently diesel-based blends of 15, 30, 50 and 70 % volumetric share of butanol could be produced and tested in the research engine. Because the fuel properties relevant for engine operation are partially significantly different from those of the CEC reference diesel fuel, the properties of the blends had to be calculated respectively estimated as far as possible to provide for proper engine operation and interpretation of the experimental results. The net calorific values and the cetane numbers of the blends were determined by weighting of the individual properties according to the volumetric composition of the mixture. Calculation of the stoichiometric air/fuel-ratio was done analogically according to the gravimetric composition. The most essential properties of the fuels and blends used are listed in **Table 3**.

Another sensitive parameter with the use of alcohol as a fuel for CI-engine applications is lubricity. Due to the rather limited share and the good lubricity of RME compared with diesel fuel, a negligible deterioration with ethanol respectively the ethanol-blend could be assumed. With the butanol-blends, the additive Lubrizol 539M was employed to improve lubricity with a share of 50 % butanol and above.

5 Emission Potentials

In the following, the emission potentials of the different fuel/combustion process

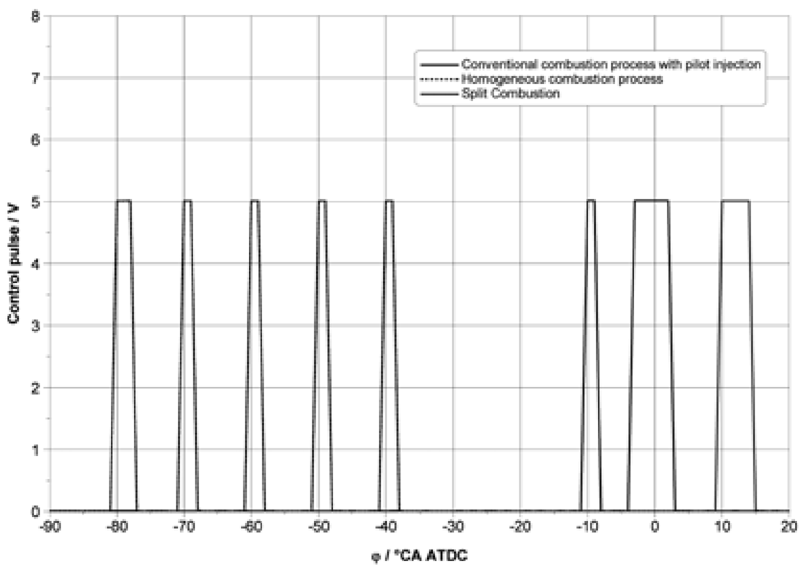

Figure 3: Injection patterns for the three combustion processes

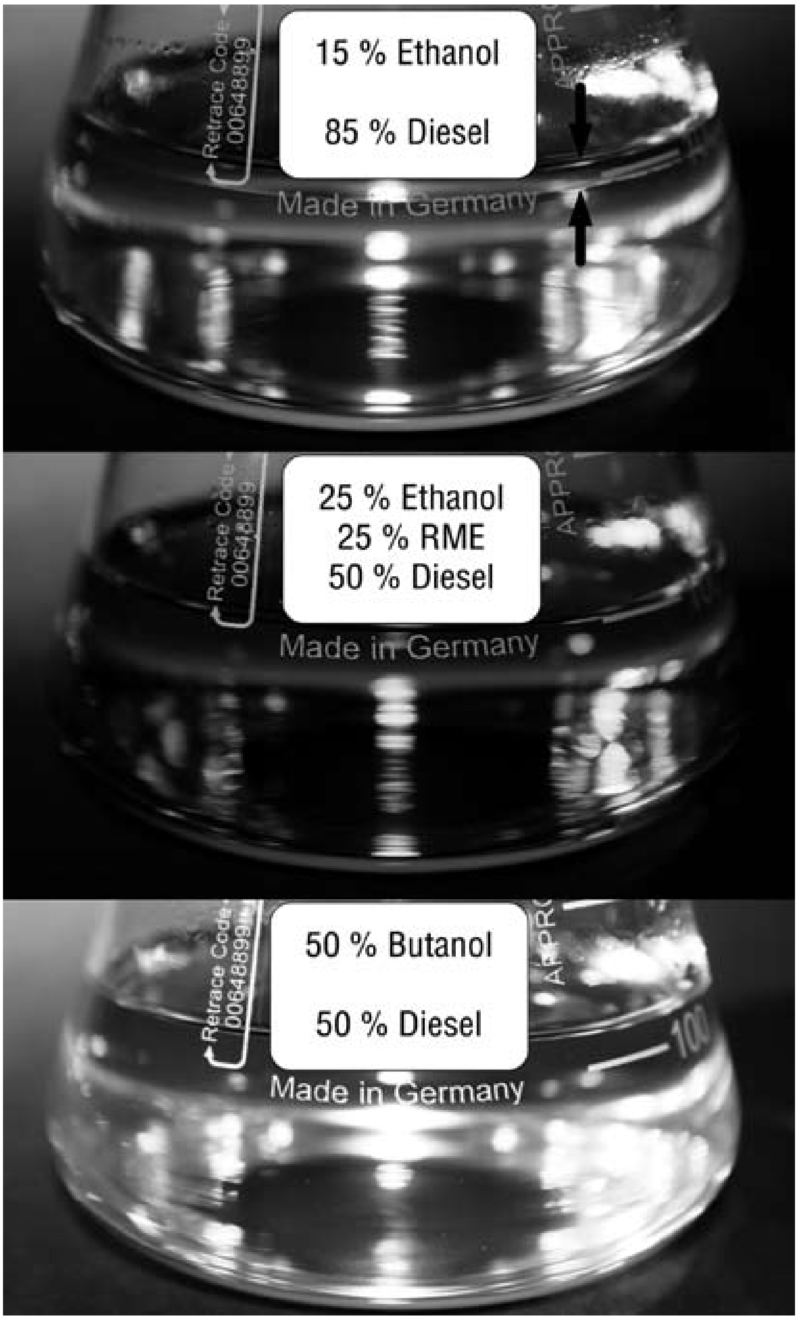

Figure 4: Study of miscibility of diesel with ethanol, RME and butanol

combinations will be investigated regarding the legally limited exhaust gas components. Beginning with the HDC at IMEP = 600 kPa, the results of the SC and the conventional combustion process, each at IMEP = 1200 kPa, are discussed hereupon.

5.1 Homogeneous Combustion at IMEP = 600 kPa

For the investigations on HDC, according to the results presented in [11], the starting A/F ratio has been adjusted to λ_S = 3 and the transformation was controlled by the EGR rate. The injection pattern from Figure 3 was not changed during the tests, except the single injection durations. The total fuel mass per cycle was split into five equal and equidistant injections. To avoid heavy knocking operation a minimum EGR-rate of about 50 % was applied. IMEP, engine speed and injection pressure were kept constant.

The emissions and the fuel consumption in **Figure 5** display the typical behavior of homogeneous combustion systems. With rising EGR-rates NO_X and PM emissions are decreasing. Simultaneously, the CO emissions rise. As expected, the blends produce lower PM emissions than neat diesel fuel. This can be attributed to the oxygen content and the molecular structure of the alcohols. In addition, the fuel consumption of the blends is lower compared to diesel fuel.

The lower NO_X emissions correlate to the higher CO emissions. This leads to the conclusion that on the one hand the lower local temperatures cause lower post-flame NO_X production but also more quenching effects resulting in higher CO production, as the water gas shift reactions freeze earlier. The unburned HC probably arise mainly from the fuel in the squish area because the level does not change with EGR rate.

The cylinder pressure analysis proves this explanation, **Figure 6**. Starting with the DF/RME/$E_{85/7.5/7.5}$ blend compared to diesel fuel there is nearly no change in the pressure curve and hence only a small deviation in the HR_{50} point. Therefore the fuel consumption and the emissions are almost equal. With the DF/$B_{85/15}$ blend the shift is more significant and the period with high cylinder temperature is smaller as well. This results in lower NO_X emissions and higher CO emissions. Thus the lower ignitability mainly improves homogenization. The DF/$B_{70/30}$ blend results in a further delay of ignition. Thus the HR_{50} point is moved in the range around 10° CA after TDC which is optimal for fuel consumption.

5.2 Split Combustion and Conventional Diesel Combustion

With split combustion at partly-homogenous process, emissions and fuel consumption show significant benefits for the butanol-blends, **Figure 7**. With the DF/RME/$E_{85/7.5/7.5}$ blend only small advantages are measurable. Only the fuel consumption is improved by nearly 3 % due to the delay of the homogeneous combustion, **Figure 8**. With DF/$B_{85/15}$ similar results in ISFC were obtained, PM and NO_X emissions are slightly lower. The greatest improvements are possible with the DF/$B_{70/30}$ blend.

Table 3: Fuel properties [9, 10]

	CEC legislative fuel	Ethanol	n-Butanol	RME EN 14214	DF/RME/E-Blend (85/7,5/7,5)	DF/B-Blend (85/15)	DF/B-Blend (70/30)	DF/B-Blend (50/50)	DF/B-Blend (30/70)	DF/B-Blend (10/90)
Density at 15 °C in kg/m³	833 - 837	790	810	860 - 890	836	832	828	822	817	812
Net calorific value in MJ/l	35.3	21.1	26.8	32.3	34.0	34.0	32.8	31.4	29.6	27.7
Stoichiometric air/fuel ratio	14.6	9.0	11.2	12.5	14.0	14.0	13.5	12.9	12.2	11.5
Oxygen content in wt.-%	< 0.03	35.0	21.0	10.9	3.4	3.1	6.4	10.7	15.0	19.4
H:C ratio (molar)	0.157	0.250	0.208	0.157	0.162	0.165	0.170	0.179	0.189	0.201
Cetane number	52 - 54	< 10	< 18	52 - 56	≈ 50	≈ 48	≈ 43	≈ 35	≈ 28	≈ 21

For comparability with the conventional combustion process even higher butanol-shares have been investigated. It was found that with higher shares the trade-off between nitrogen oxides and particles (NO_X/PM trade-off) and the net calorific value corrected consumption were affected positively both with SC as well as with the conventional process. The comparison of the combustion processes is shown in **Figure 9**. Optimized operation points are chosen and compared.

The results clearly show that even with the conventional combustion process very low emissions can be obtained. With the engine settings chosen in this investigation the SC has some advantages regarding NO_X emissions, but disadvantages concerning fuel consumption and the other emissions.

Results with the conventional combustion process are shown in **Figure 10**. The use of EGR enables a distinct reduction in NO_X emissions. With a butanol-share of 70 % the simultaneous rise in PM emissions is absent. At the same time the consumption is lower due to a faster combustion. A significant increase in cylinder peak pressure has not been recorded; only CO emissions are on a higher level.

6 Summary and Outlook

Based on former investigations [12, 13] with butanol-shares up to 30 % vol. the butanol-share has been increased up to 70 % vol. in this research project of TU Braunschweig. The effects on operation of a single cylinder research engine with different fuel blends were

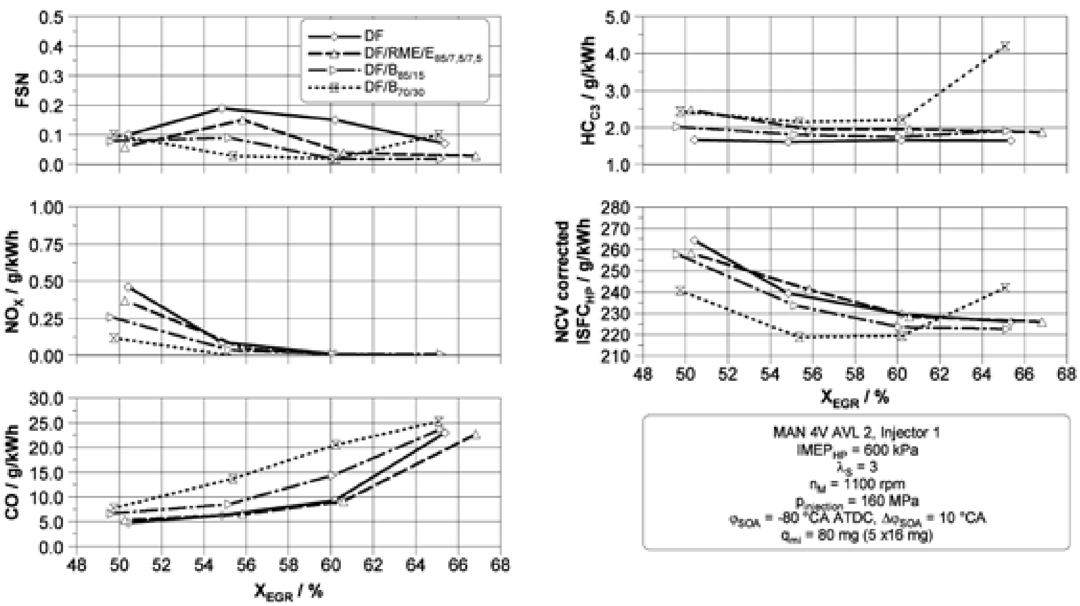

Figure 5: Emissions and specific fuel consumption, variation of EGR rate for different fuels at HDC, IMEP = 600 kPa

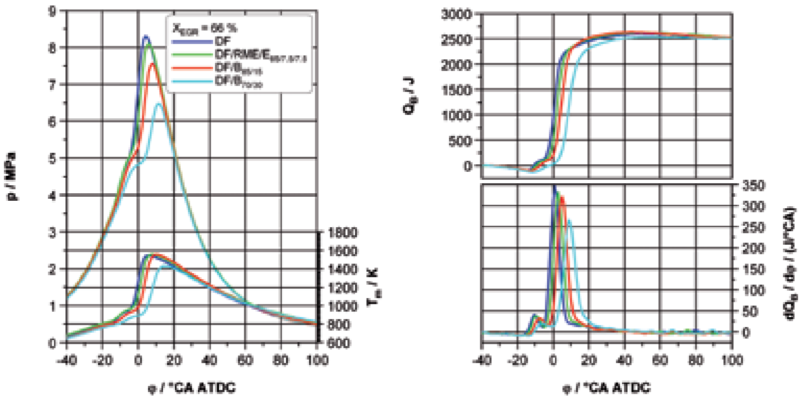

Figure 6: Combustion pressure p and heat release data QB, EGR rate of 66 % for different fuels at HDC, IMEP = 600 kPa

determined. The tests with different combustion processes showed the following trends:

- In operation with the homogeneous diesel combustion blends with up to 30 % alcohol share show some advantages regarding fuel efficiency compared to pure diesel fuel but no improvements in controllability.
- In the partially homogeneous combustion process (SC) with rising butanol-shares both NO_X/PM trade-off and indicated specific fuel consumption show advantages. Regarding CO and HC emissions there were disadvantages.
- In the conventional combustion mode with a pilot injection in order to reduce pressure gradients, especially at low A/F ratios (λ = 1.5) or high EGR rates distinct advantages regarding PM and CO emissions at equal NO_X and HC emissions were detected.

The investigations have shown that fuel blends with butanol shares higher than 30 % have a significant engine potential. With rising butanol shares their characteristics are changing towards a fuel blend showing good evaporation and bad ignition properties. Such characteristics imply a suitability for homogeneous diesel combustion systems which was proven by the tests.

Additionally the conventional combustion system with pilot injection and cooled EGR showed high potential as well in comparison. A share of 50 % butanol was sufficient to meet Euro-V-emission limits at stationary operation with the engine described above.

The implementation of mixture preparation and ignition conditions to provide for proper diesel engine operation throughout the whole load range with the application of blends with high butanol-shares have to be more investigated. The application of a variable valve train to vary the effective compression ratio could be a possible solution. Due to their characteristics, which make them well suited for homogeneous diesel combustion, and the sustainability based on the capability of butanol to be produced from biomass or crops, butanol/diesel blends are a very interesting field for further investigations.

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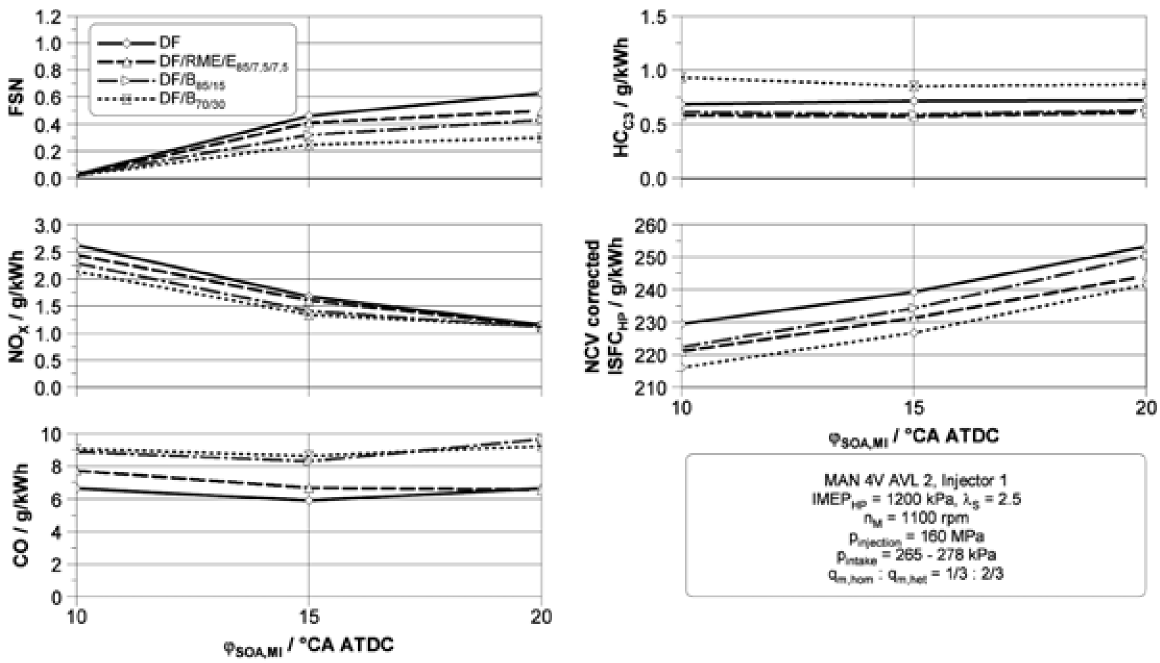

Figure 7: Emissions and consumption at split combustion, IMEP = 1200 kPa

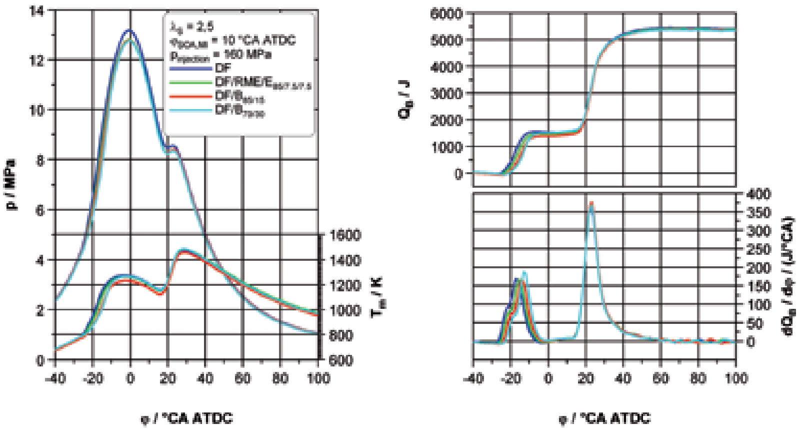

Figure 8: Heat release at split combustion with $\varphi_{SOA,MI}$ = 10° CA after TDC for different fuels, IMEP = 1200 kPa

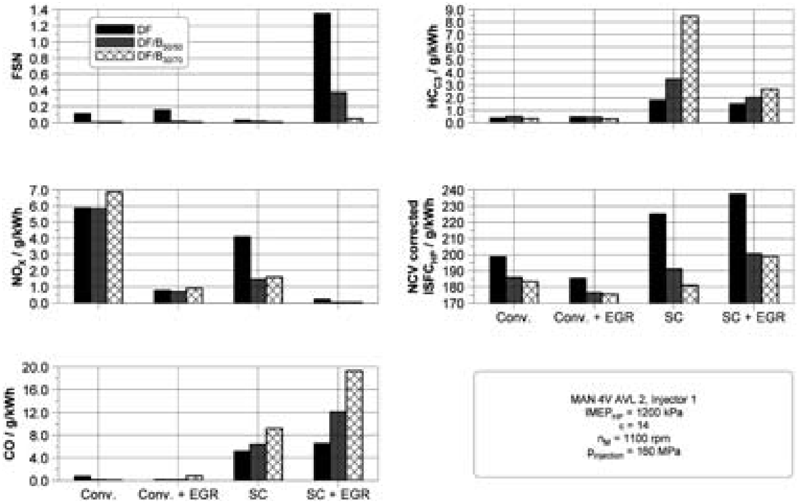

Figure 9: Emissions and consumption for different blends and combustion processes

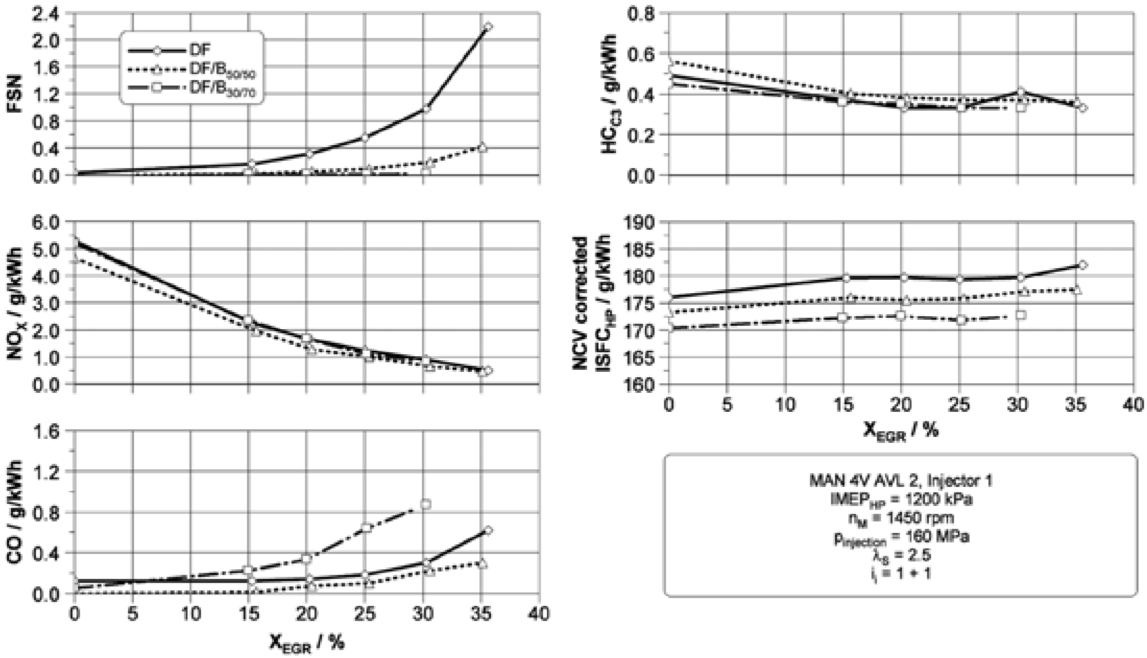

Figure 10: Emissions and consumption at conventional combustion process

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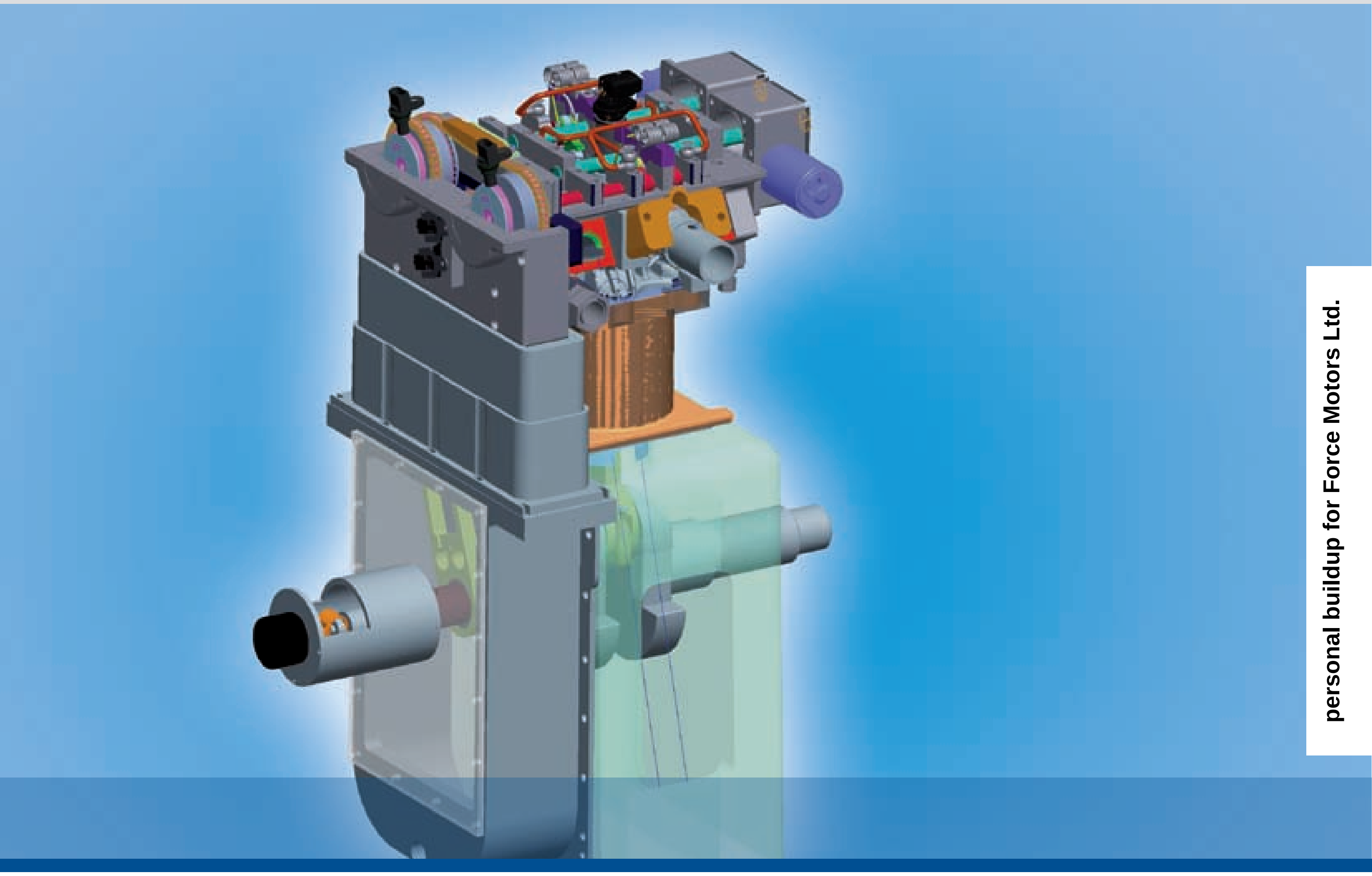

Fully Variable Valve Train and Variable Compression Ratio

First Measured Results

The Institute for Internal Combustion Engines at the Technical University of Kaiserslautern has analyzed whether variable compression ratio can be combined with throttle-free load control over mechanically fully variable valve trains and direct injection, or rather just over direct injection to optimize the efficiency of gasoline engines. Furthermore, the question must be considered as to which efficiency advantages will possibly get lost due to an increase of friction; from additional losses of the forces, which adjust and fix valve lifts and get lost ultimately due to the influence of combustion efficiency. Although the effective compression ratio can be increased, depending on load, via "loading" the cylinder with residual gas – for example, with variable valve timing on the exhaust valve – the combustion also becomes influenced. That is why, in the end, the effect that a fully variable compression ratio has on the fuel consumption can only be determined through tests.

1 Introduction

It was the turbo-charged diesel engine with common-rail direct injection that helped the European automobile industry to achieve its CO_2 targets. However, particle filters and aftertreatment of NO_x emissions can hinder any further and obvious reductions of CO_2 emissions.

Regarding gasoline engines, new technologies were introduced in large series production in order to improve the engines' efficiency. These technologies are, for example, variable valve timing, variable valve trains with two step valve systems and mechanically fully variable valve trains with and without direct injection, which can be used homogenous and stratified. On the other hand fuel consumption, and therefore CO_2 emissions, was also decreased by reducing the cylinder capacity and downsizing the gasoline engine at the same time through optimized gears. Improving the efficiency and downsizing are the two ways which were and are still combined to attain the new CO_2-target of 120 g/km.

According to several engineers the next consequent step will be to decrease fuel consumption of charged gasoline engines at part- and full load, by using variable and fully variable compression ratio.

The thermal efficiency η_{th} of gasoline and diesel engines depends on the compression ratio ε and the thermodynamic properties of the gas being compressed, **Figure 1**.

The effective compression ratio ε_{eff}, which results from the load control of gasoline engines at part load, is considerably smaller than the geometrical compression ratio ε_{geom} which can be achieved at full load.

If one takes into consideration the technical developments of gasoline engines, the question arises as to whether variable compression ratio can be combined with throttle-free load control over mechanically fully variable valve trains and direct injection, or rather just over direct injection to optimize the efficiency of the engine. Both options are possible.

Furthermore, the question must be considered as to which efficiency advantages will possibly get lost due to an increase of friction; from additional losses of the forces which adjust and fix valve lifts and get lost ultimately due to the influence of combustion efficiency. As the effective compression ratio which depends on load can also be increased through "loading" the cylinder with residual gas – for example with variable valve timing on the exhaust valve – only tests can determine in the end what effect fully variable compression ratio has on the fuel consumption.

2 Technical Design

After GT-Power simulations, the first evaluation showed that the geometric compression ratio should cover a range from 9 to 19. At a geometric compression ratio of 19, the effective compression ratio in a typical four-cylinder gasoline engine at an effective middle pressure p_{me} = 2 bar lies around 11. Due to the fact, that a compression ratio of 19 with valve pockets in the piston cannot be reached, the valve lift on the exhaust side should also be adjusted in a variable way.

That is why the Technical University of Kaiserslautern, with the support of

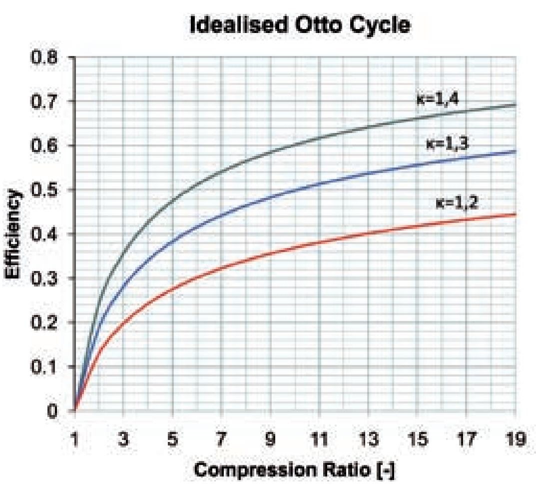

Figure 1: Thermal efficiency of idealized otto cycle with different compression ratios

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MTZ Peer Review

The Seal of Approval for scientific articles in MTZ.

Reviewed by experts from research and industry.

Received .... January 19, 2009

Reviewed .... January 27, 2009

Accepted .... February 05, 2009

the EnTec company, designed a singe cylinder experimental engine with a large series cylinder head design and the fully variable valve train ”UniValve“ on inlet- and exhaust side, **Figure 2**.

Literature research was completed before designing the variable compression ratio, as was an analysis of the risks and functions of the different principles.

The advantage of the selected variable compression-ratio system, **Figure 3**, with an adjustable distance of the cylinder-head-liner-unit to the middle of the crankshaft was above all that adjustments of the compression ratio were neither made at the turning crankshaft nor at the oscillating piston or conrod. Moreover the adjusted compression ratio can be precisely and easily measured in a ”static“ system.

When changing the compression ratio, the distance between camshaft and crankshaft alters. That is why a chain drive was designed that adjusts itself automatically to distance changes, **Figure 4**.

The chain drive consists of three part-chain-drives which are connected to each other through one common shank. The intermediate chain sprocket is lead over a turnable lever arm which is mounted on the crankshaft. The intermediate chain sprocket is moved on a circle curve over that turnable lever, when the distance of the cylinder head and the crankshaft is adjusted. The intermediate chain sprocket is driven by a chain drive 1 from a sprocket on the crankshaft and drives the exhaust camshaft over chain drive 2 on the other side.

The drive of the intermediate shaft moves the exhaust camshaft. If the cylinder head is lifted up, the rocker arm will swing to the inside and adjust itself to the change in distance. This chain drive can be integrated in a serial engine package without limiting the speed of the engine.

The adjustment of the compression ratio is achieved by a worm gear which moves through a screw gear liner and cylinder head. At a bore diameter of 83 mm, the length of the distance required to change the compression ratio from 9 to 19 is 10 mm. The worm gear is powered by an electronic engine and can be adjusted even when in motion. **Figure 5** shows the integration of optical measurement technology and direct injection.

Engine timing is adjusted by a freely programmable ECU, designed by the company Motec, which also powers the two phase adjusters. The engine is currently run by manifold fuel injection (MPI). The direct injection of liquid fuel and the methane direct injection are prepared, so that an asynchronous, balanced mode of methane and liquid fuel can also be outlined.

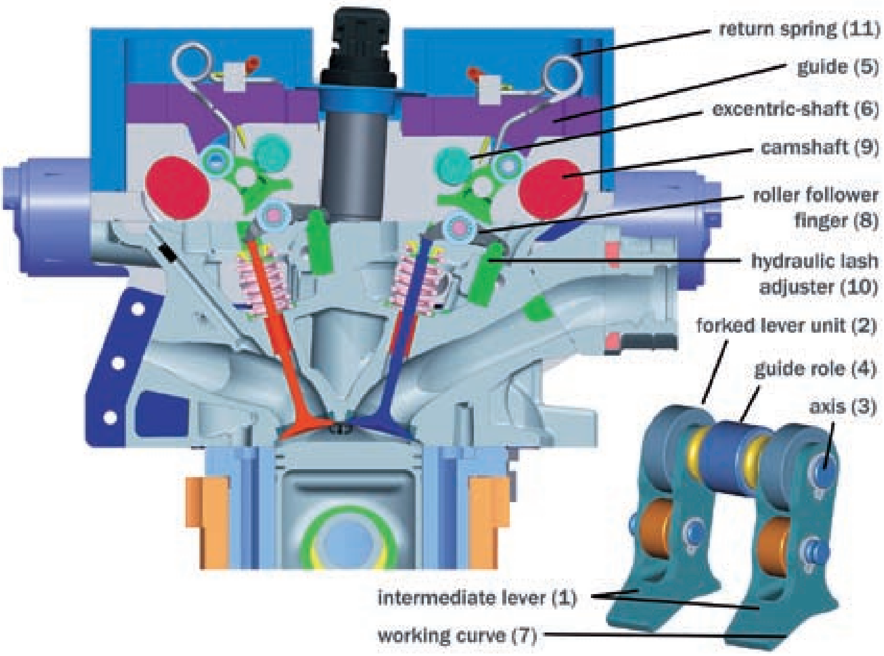

Figure 2: Modified cylinder head with “UniValve“

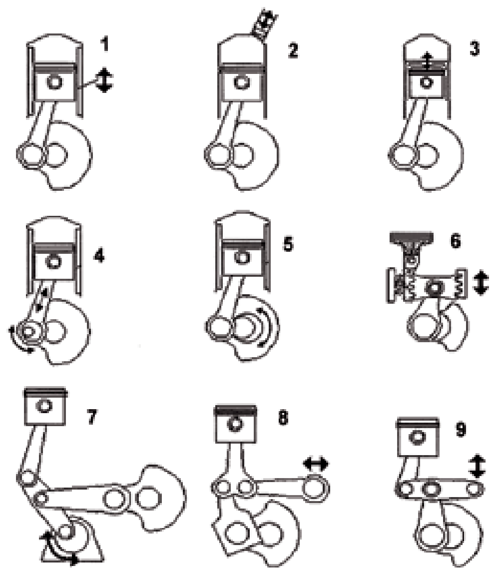

Figure 3: Possible designs of variable compression ratio [4]

3 First Measured Results

Apart from the degree of freedom concerning compression ratio variability, the single-cylinder experimental engine also has an additional fully variable valve train on the inlet-and exhaust side. The fully variable valve train on both sides is, on the one hand, used to realize a collision free use of the engine at a high compression ratio, and on the other hand to be able to optimize the charge-cycle-work, consumption and emissions to a minimum level.

The load of the experimental engine can be controlled either by a throttle valve or driven throttle-free over an inlet valve lift. It is the aim of future research works to not only consider the influence of variable compression ratio on the fuel consumption with different mixture preparations but also especially to investigate new charge-cycle-work and residual gas strategies with the variable exhaust valve train.

In the following paragraphs a short demonstration of a few measurement results of the single-cylinder experimental engine will be presented. The map point of all measurements lies at n=800/min and p_{me}=3 bar.

3.1 Throttle-free Load Control with Variable Compression Ratio

The influence of the inlet spread on the break specific fuel consumption (BSFC) is shown in **Figure 6**.

The fuel consumption decreases in an almost linear way with the reduction of the inlet spread, as already seen in four-cylinder engines. At an inlet spread in the range of 60° CA the fuel consumption increases again. Yet it was not possible for the different compression ratios and their concomitant inlet spreads to be matched in an exact way in order to find the lowest level of consumption. The lowest fuel consumption at this map point/speed was found to be at a geometrical compression ratio of ε=13. At a geometrical compression ratio of ε=14 and an inlet spread of IS=60° CA knocking combustion occurs.

In **Figure 7** the lowest fuel consumption of each inlet spread variation, Figure 6, is illustrated over the geometrical compression ratio.

While the indicated fuel consumption improves by nearly 9 %, **Figure 8**, the variable compression ratio only decreases the BSFC at this map point by about 2.5 % at an exhaust spread of OS=110° CA. In doing so, a relatively big leap of the compression ratio increase from 10 to 11 and from 13 to 14 can be witnessed.

The charge-circle-work that fundamentally influences the fuel consumption at a constant exhaust spread decreases also in an almost linear way with the inlet spread, as already seen in the measurements of the four-cylinder engine.

To keep the map-point constantly at p_{me}=3, even at an increasing compression ratio and thereby a higher efficiency, the aspirated amount of fresh load and accordingly the valve lifts must also be reduced, in line with an increasing ε. As a result of the reduced valve gap, the throttle losses increase. Furthermore the low pressure at the button dead center which occurs due to a low cylinder load and change in compression ratio has a negative influence on the charge-cycle-work (according to the UT-UT-definition from [1]).

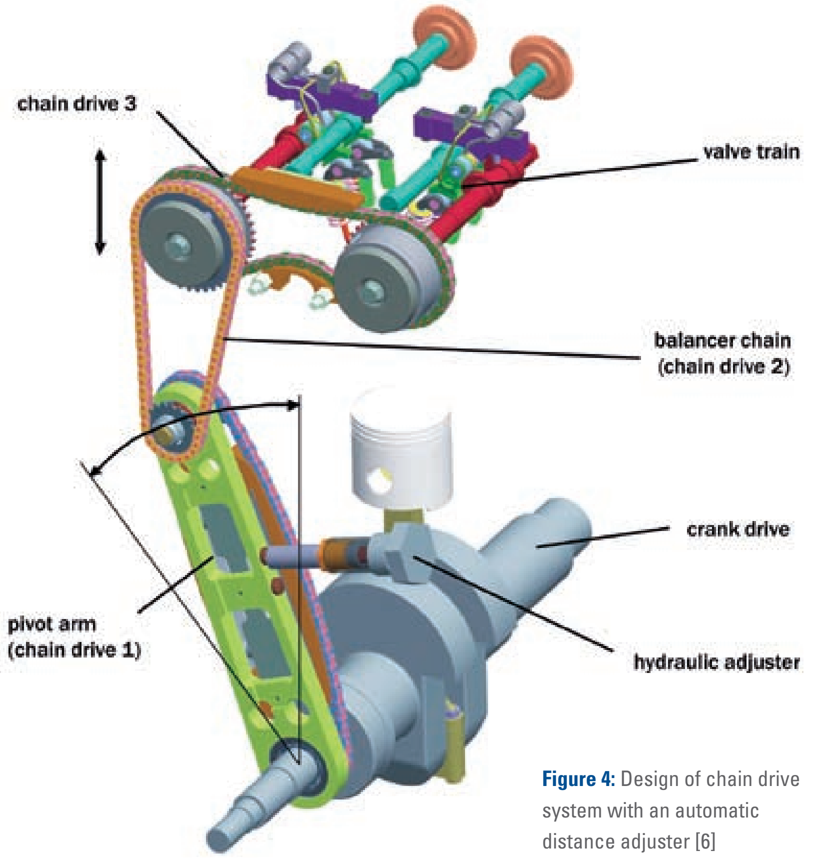

Figure 4: Design of chain drive system with an automatic distance adjuster [6]

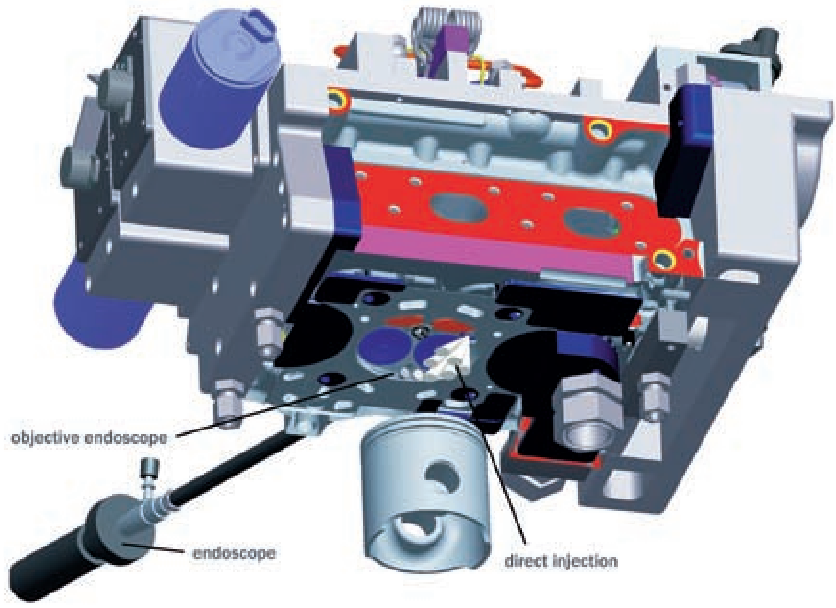

Figure 5: Integration of optical measurement technology and direct injection

However, a further reduction of the BSFC cannot be achieved by an increase of the geometrical compression ratio from ε_{geom}=13 to εg_{eom}=14, as the friction losses increase in line with a rising compression ratio. At a geometrical compres-

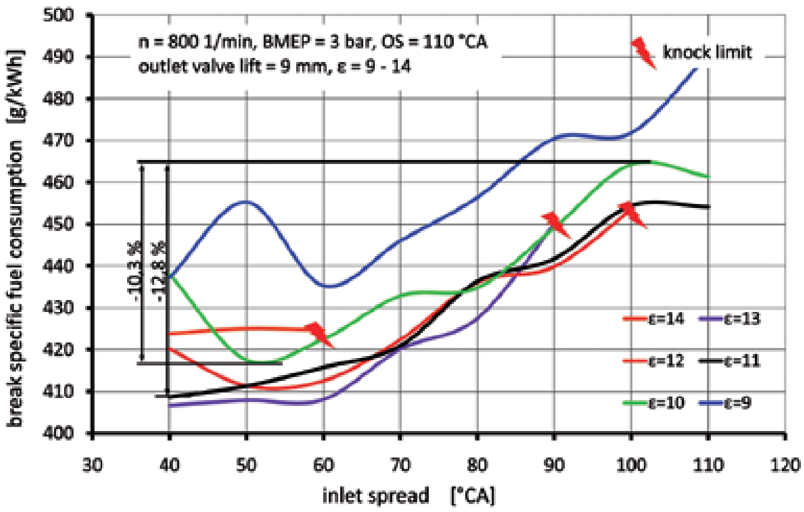

Figure 6: Break specific fuel consumption (BSFC) over IS at different geometric compression ratios

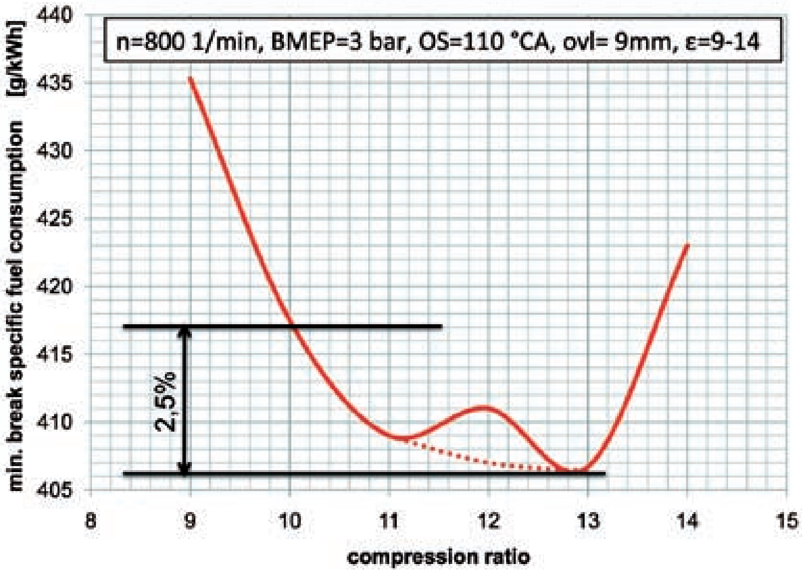

Figure 7: Break specific fuel consumption over geometrical compression ratio (BSFC)

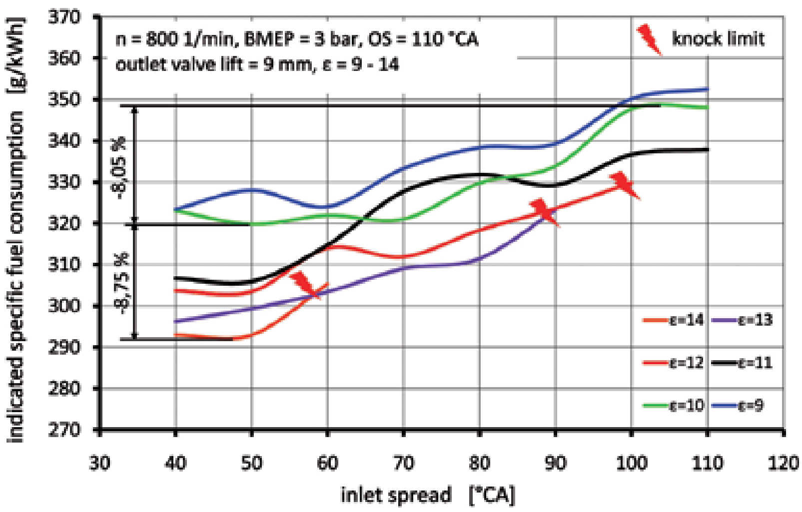

Figure 8: Indicated break specific fuel consumption over IS with variable compression ratio

sion ratio of ε_{geom}=14 the risen friction mean effective pressure compensates the advantage of increased efficiency.

With a rising geometrical compression ratio, the surface to volume ratio of the combustion chamber increases. As the combustion chamber becomes more disc-shaped, the boundary conditions for a clean and complete combustion deteriorate even more. Additionally heightened efficiency leads to low exhaust gas temperatures and consequently to worsened boundary conditions for the after oxidation of unburned hydrocarbons in the exhaust gas. That is to say, when increasing the combustion ratio one can determine a rise of HC-emissions, **Figure 9**.

To sum up, with variable combustion ratio as an addition to the potential of fully variable valve trains an improvement of efficiency can be attained. Yet, increasing the combustion ratio not only leads to a rise in friction in the system, but also to a bigger share of unburnt hydrocarbons. The reduction of the BSFC is therefore only possible to the point when a level of ”optimal“ geometrical compression ratio is reached. When the compression ratio is increased further, an over proportional rise of losses occur, and this leads to a rise of the BSFC.

3.2 Throttle-free Load Control and Fully Variable Exhaust Side

Running this single-cylinder engine throttle-free, as well as the inlet valve lift and inlet spread, the exhaust valve lift, exhaust timing and exhaust spread can be adjusted fully variable too.

The exhaust spread sweep at throttle-free load control, at a fixed inlet spread of IS=40° CA and a full valve lift on the exhaust side – as for an outlet timing of 290° CA – shows a minimum of fuel consumption at an exhaust spread of OS=70° CA, **Figure 10** (blue line). This offers the best compromise of expansion losses, exhaust part of gas exchange losses and valve overlap (due to influencing the residual gas amounts, but scavenging losses of fresh load). If the exhaust timing is further reduced, the fuel consumption will also decrease further. The ”optimal“ exhaust spread at this point is shifted in direction to OS=100° CA.

Minimal fuel consumption at an ”optimal“ exhaust spread cannot be accomplished anymore at a spread around OS=110° CA

However, if the drivable spread range is extended to an exhaust spread of OS=150° CA, minimal fuel consumption will be measured at an exhaust spread of 130° CA, an exhaust valve lift of 4,5 mm and valve timing of 220° CA. The decrease in BSFC occurs as a result of reduced expansion losses and higher effective compression ratio due to a higher amount of residual gas, **Figure 11**.

One should note nevertheless, that during this complete series of tests the inlet spread was fixed at IS=40° CA. The early opening of the inlet valve due to the fixed inlet spread can be seen at short exhaust valve timing and high exhaust spread as a disadvantage, because the pressure in the combustion chamber rises as a result of the re-compression of the residual gases enclosed in the combustion chamber. As soon as the inlet valves opens, the pressure is released into the induction pipe. Consequently low-pressure expansion losses are revealed which could be avoided through a later opening of the inlet valves.

A variation of inlet as well as exhaust spreads will be examined in future test series.

4 Conclusion

A single-cylinder engine shows at throttle-free load control over variable inlet valve trains the same consumption behavior as the measurements known from a four-cylinder engine.

An additional increase of the geometrical compression ratio from ε_{geom}=10 to ε_{geom}=13 allows in the tests run so far a further reduction of the BSFC of 2.5 %. It becomes obvious in the tests, that the increase of efficiency and compression ratio is limited by an increase in friction and a consequent rise in the level of losses.

Another degree of freedom available is the adjustment of exhaust spread and exhaust valve, which allows a further decrease of fuel consumption as well as a variable adjustment of the internal backflow of the amount of residual gas shares. As a result of a reduced exhaust valve, high amounts of residual gas shares can be attained by the retention of exhaust fumes.

Which potentials an increase of the geometrical compression ratio through

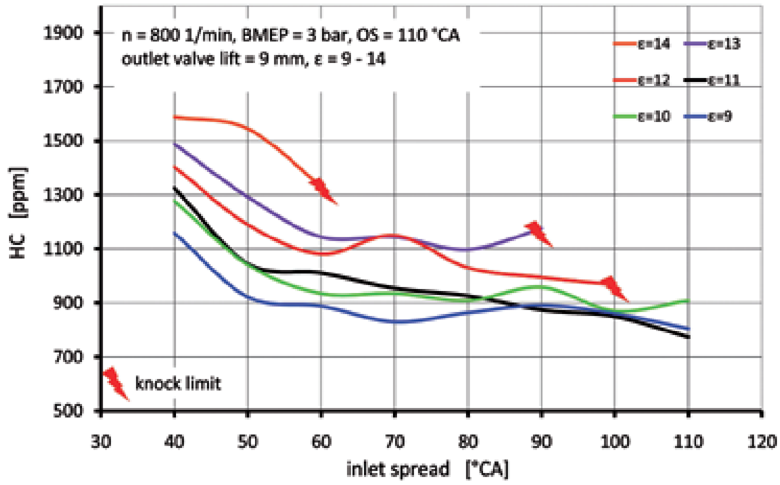

Figure 9: HC-emissions over IS with variable combustion ratio

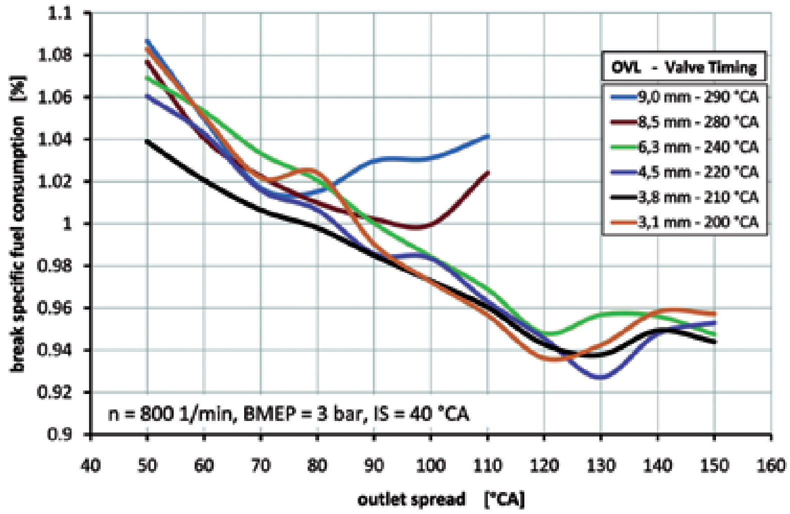

Figure 10: BSFC over exhaust spread with throttle-free load control

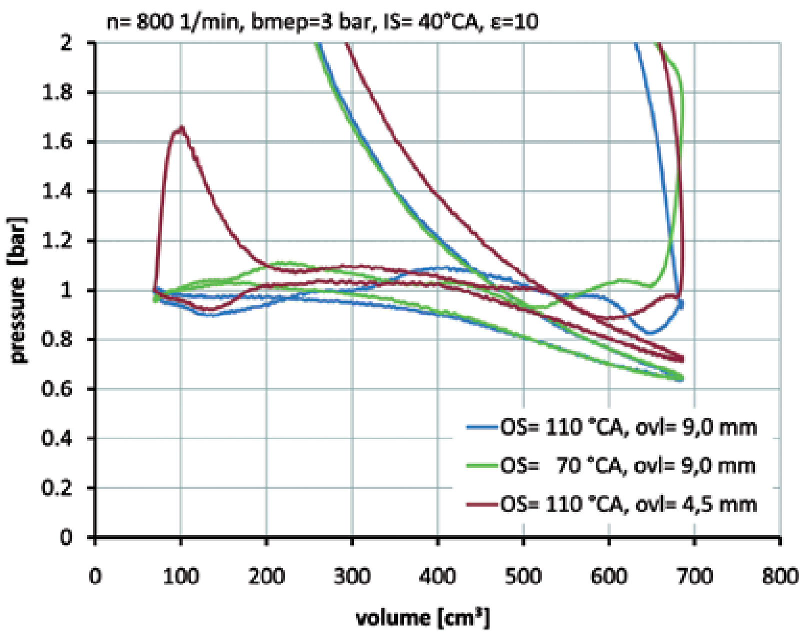

Figure 11: p-V diagram with variable exhaust valve

targeted valve timing of residual gases can cover will be examined in the following series of tests. The first measurements already show that a simple addition of selected consumption potentials is not possible, as they influence each other.

While testing the single-cylinder experimental engine all degrees of freedom could be varied. That is why it was not only possible to compare the measurements but also to trace them clearly back to their respective origins and consequently being able to find obvious reasons.

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Abbreviations

Abbreviation	Term	Unit
° CA	degree crank angle	° CA
ε_{eff}	effektive compression ratio	
ε_{geom}	geometric compression ratio	
OS	Outlet Spread	° CA
IS	Inlet Spread	° CA
TDC	Top Dead Center	
BMEP	Break Mean Effective Pressure	bar
PMEP	Pumping Mean Effective Pressure	bar
BDC	Bottom Dead Center	

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07-08|2009 · July-August 2009 · Volume 70
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30th International Vienna Motor Symposium

As every year, leading automotive engineers and scientists from all over the world met at the 30th Vienna Motor Symposium which was held on May 7th and 8th, 2009. They presented the latest findings in engine developments and gave an outlook on future trends in the automotive industry. This report contains a summary of the lectures presented in the individual sessions.

1 Introduction

After a welcome fanfare, **Figure 1**, which was performed by members of the orchestra of the Vienna University of Technology, **Prof. Lenz**, **Figure 2**, welcomed the participants to the 30th International Vienna Motor Symposium.

Prof. Lenz pointed out that despite the all-pervasive sentiment of crisis, the symposium was fully booked. However, individual companies had substantially reduced the number of participants for whom they had originally reserved, which ran counter to the claim made at the same time that "they would now concentrate, in particular, on intensive development, research, and further education efforts".

With regard to the economic crisis, which currently dominated the general discourse, Prof. Lenz stressed that the automotive industry had not caused the crisis. In the automotive industry, developments followed an optimum trend, progress was continually made in all areas, and the same applied to production which was characterised by high efficiency and excellent quality.

The argument that the wrong models had been developed was simply not true, Prof. Lenz stressed. The automotive industry, he went on to say, always produced the models which buyers wanted and did not act as a teacher who would tell consumers what cars they should buy.

The problem resulted much rather from the hesitations of customers to buy cars, and this disinclination to buy had been triggered not by the automotive industry but by unscrupulous "financial jugglers".

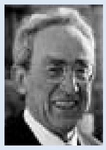
Although the objective of the Symposium is and has always been to disseminate the most recent findings of automotive engineering and to cast a glance into the future, at this anniversary conference, a summary of all previous symposia was also presented.

Professor Gruden, **Figure 3**, who had formerly worked for Porsche and had attended all symposia without exception right from the start, described in his presentation entitled "30 years of the International Vienna Motor Symposium" how demands relating to environmental standards, oil reserves, alternative fuels, and powertrains had evolved over time. Prof. Gruden concluded his speech by saying that 30 years of International Vienna Motor Symposia meant 30 years of progress and success. If we applied the yardstick of our human lifespan, he pointed out, the International Vienna Motor Symposium was now in its prime.

Prof. Mikulic offered Prof. Lenz the illustrated book „30 Jahre Wiener Motorensymposium" (30 Years of the Vienna International Motor Symposium) and a bouquet of flowers to Mrs. Lenz, **Figure 4**.

After the joint plenary opening session, the participants split up into two parallel groups (**Figure 5** and **Figure 6**), in which technical papers were presented under the chairmanship of Professors **H. Eichlseder**, **B. Geringer** and **G. Jürgens**.

The Author



Univ.-Prof. Dr. techn. Hans Peter Lenz is President of the Austrian Society of Automotive Engineers (ÖVK) in Vienna, Austria.

Figure 1: Welcome fanfare performed by members of the orchestra of the Vienna University of Technology

Figure 2: Univ.-Prof. Dr. techn. Hans Peter Lenz

Figure 3: Prof. Dr. techn. D. Gruden

A comprehensive and impressive exhibition of new engines, components and vehicles complemented the technical presentations, **Figure 7** and **Figure 8**.

Accompanying persons were offered an ambitious cultural and social programme which comprised a one-day excursion entitled "Tracing the Footsteps of Joseph Haydn", a one-day tour "From Belvedere Palace to the Inner City – Baroque Buildings in Vienna" as well as the half-day tours "Ernst Fuchs, Artist and Designer – A Representative of Fantastic Realism" and "Downtown Vienna off the Tourist Track".

Upon invitation of the Mayor of Vienna, the participants and accompanying persons spent an evening at the wine tavern Fuhrgassl-Huber in Neustift.

2 Plenary Opening Session

Dr. H. Demel, **Figure 9**, COO Magna Vehicle and Powertrain Group, Oberwaltersdorf, delivered the first lecture in the plenary entitled "Well-to-Wheel Energy Efficiency of Different Vehicle Concepts":

The total energy requirement of a vehicle, starting with the production of fuels and all the different materials for components, the manufacturing and utilisation of a vehicle through to its recycling had come to play an ever greater role, Dr. Demel stressed. He went on to say that for the calculation of the total energy requirement, a well-to- wheel analysis did not suffice, as the overall life cycle had to be taken into account. Therefore, the following conclusions can be drawn:

- Lightweight construction permits substantial energy savings, but results in higher costs on account of the materials used.
- By using recycled materials or by processing and recycling used materials, considerable cost and energy savings can be achieved in vehicle production.
- If it is assumed that in the future mainly electric motor powered vehicles will predominate in the market, CO_2 emissions and other impacts on the environment depend, to a great extent, on the percentage of electrical energy that comes from fossil fuels and from renewable sources. Hence an electric motor powered vehicle with a consumption of 13.2 kWh per 100 km emits 58 g of CO_2/km in Austria, on an average 86 g CO_2/km in the EU, 110 g CO_2/km in the US and 191 g CO_2/km in China.

Prof. Dr. H. List, **Figure 10**, Chairman of the Board, AVL List GmbH, Graz, presented a report on the topic: "Future Powertrain Systems in a Rapidly Changing Global Environment":

In Prof. List's view, the rapidly changing environment for powertrain systems will not fundamentally affect the criteria that consumers apply when buying cars, such as purchasing price, fuel consumption, operating costs, driving dynamics and comfort as well as technology orientation, but the importance attached to these individual criteria in relationship to one another is certainly evolving rapidly. In order to be able to respond quickly to these changes in customer expectations, manufacturers will have to use modular powertrain systems which will permit them to conceive flexible, specific solutions for the individual market segments.

The electrification of powertrains will lead to a widening range in the technologies applied and thus increase the complexity of modular systems. At the same time, in addition to the further development of individual components, a shift in paradigm will call for a far-reaching optimisation of the overall system. The weighting of the thermodynamic and electrodynamic components in the powertrains of the future will be very significantly determined by the evolution of battery technology and further progress in the design of internal combustion engines.

The shift of paradigm in the design of powertrains, Prof. List pointed out, would also call for a paradigm shift in the development process itself as, for example, the consistent use of simulation models throughout the entire process would constitute an important prerequisite. In his opinion, only in this way would it be possible to design components rapidly and efficiently with a view to optimising overall solutions for the different market segments in an integrated, simultaneous process development environment.

Dr. K.-T. Neumann, **Figure 11**, Chairman of the Board, Continental AG, Hanover, was the last speaker at the opening plenary session. He gave a report on the topic "The Electrification of the Powertrain – Opportunities and Challenges for the Automotive Industry":

Dr. Neumann considered the electrification of the powertrain as the key to future mobility.

For lithium-ion batteries, assuming 100 % cell costs in the reference year 2010, he expects a cost reduction of 35 % by 2020, thanks to improvements in production, with 15 % resulting from the use of new materials, 10 % from standardisation, and 5 % from lower material costs.

At the same time, he believes power density and energy density will be strongly increased.

In addition, he emphasised, electric motor-powered vehicles should lead to entirely new business models.

3 New Otto Engines

Dipl.-Ing. P. Lückert, **Dipl.-Ing. F. Kreitmann** (lecturer), **Dr.-Ing. N. Merdes**, **Dr.-Ing. R. Weller**, **Dipl.-Ing. A. Rehberger**, **Dr.-Ing. K. Bruchner**, **Dipl.-Ing. K. Schwedler**, **Dipl.-Ing. H. Ottenbacher**, **Dipl.-Ing. T. Keller**, Daimler AG Stuttgart: "The New 1,8l Four-Cylinder Turbocharged Direct Injection Gasoline Engine by Mercedes-Benz for All Passenger Cars with Standard Drivetrains":

Over the past few years, the four-cylinder engine produced by Mercedes-Benz and designated internally as M 271, has proved to be an excellent model especially in the C, E and SLK vehicle categories.

Figure 4: From left to right: Prof. Dr. L. Mikulic, Mrs. M. Lenz and Prof. Dr. H. P. Lenz

The four-cylinder engine M 271 evo described by the lecturer represents the outcome of a consistent further step in the engine downsizing strategy which Mercedes-Benz has been applying for many years in its production vehicles.

Special attention was given to design measures that resulted in lower fuel consumption. At the same time, the performance and torque characteristics of the new engine were improved.

Essential changes in the design of the new engine comprise a homogeneous direct injection system instead of port fuel injection, and the substitution of a compressor by a single-stage waste-gate exhaust gas turbocharger. The configuration of the base engine with a displacement of 1.8 l was retained. As compared to the predecessor model, on balance, the new engine displays significantly better fuel economy thanks to improvements to the combustion process and a reduction of friction losses combined with more sophisticated control and application strategies.

Dipl.-Ing. T. Wasserbäch (lecturer), **Dipl.-Ing. M. Kerkau**, **Dipl.-Ing. F. Maier**, **Dipl.-Ing. J. Hawener**, **Dr.-Ing. H.-J. Neußer**, Dr.-Ing. h.c. F. Porsche AG, Weissach: "High Performance Engines Offering Maximum Efficiency – The New Family of Flat 6 Engines from Porsche"

The second generation of the Porsche Carrera 997, Boxster 987 and Cayman sees the introduction of a new family of water-cooled flat-six engines at Porsche. These engines represent a completely new design, which is characterised by a modular concept, several displacement variants and higher degree of integration with enhanced power unit rigidity. The engine family is structured to reflect the typical Porsche differentiation between the individual vehicle types in economic terms. The power output of the new flat-six engines ranges from 188 kW to 283 kW. The characteristics which are key for a high performance engine, such as a wide engine speed range, spontaneous responsiveness, low weight with a low centre of gravity and high lateral acceleration, have all been significantly improved in comparison with the predecessor engines. Performance and fuel consumption levels unmatched by competitors have been achieved thanks to the first-ever use of direct fuel injection in a flat-six engine, coupled with consistently enhanced cylinder charging and a reduction of internal engine friction.

Figure 5: Festival hall

Dr.-Ing. J. Böhme (lecturer), **Dipl.-Ing. H. Müller**, Audi AG, Ingolstadt; **Dipl.-Ing. M. Ganz** (lecturer), **Dipl.-Ing. M. Marques**, quattro GmbH, Neckarsulm: "The New 2.5-Litre-TFSI-Five-Cylinder Engine for the Audi TT RS":

Audi looks back on a long tradition of designing five-cylinder engines with turbocharging. The combination of direct injection with turbocharging is a consistent further development . With a displacement of 2.5 l, the engine has an output of 250 kW in the range of 5,400 to 6,700/min and 450 Nm at 1,600/min. In combination with an optimally adapted six-speed manual transmission, these engine data result in outstanding acceleration and elasticity characteristics comparable to the performance of sports cars, with reasonable fuel consumption.

4 Hybrid 1

Dipl.-Ing. M. Weiss (lecturer), **Dr. N. Armstrong**, **Dipl.-Ing. J. Schenk**, **Dipl.-Ing. F. Nietfeld**, **Dr. R. Inderka**, Daimler AG, Stuttgart: "Hybrid Propulsion with Highest Electric Power Density for the ML 450 BlueHYBRID":

The ML 450 BlueHYBRID is based on the current M-Class and is equipped with the innovative AHS-C two-mode hybrid system with two high-performance electric motors. The complete drive system, the battery and power electronics as well as the operating strategy which have been especially devised for this vehicle result in a significant reduction of fuel consumption and emissions.

Alongside the two high-performance electric motors, the power-split AHS-C two-mode transmission comprises four clutches and three sets of planetary gears. This transmission was designed jointly with Daimler's development partners at the production site in Troy in Michigan, USA.

The ML 450 BlueHYBRID utilises a high-voltage battery on a nickel-metal hydride basis. Liquid cooling is used for the first time, which offers stable operation in all situations. Together with the AHS-C two-mode hybrid system, the battery permits all hybrid-specific operating modes, such as electric-only driving, engine start-stop, regeneration and boosting.

Series production of the ML 450 BlueHYBRID will take place at the Mercedes-Benz production plant in Tuscaloosa, USA.

M. Cisternino (lecturer), GM Powertrain Europe, Torino; **J. Hendrickson**, **A. Holmes**, General Motors Corporation, Pontiac, USA: "General Motors' Front-Wheel Drive Two Mode Hybrid Transmission":

General Motors is now expanding the application of two-mode hybrid technology to front-wheel drive vehicles with the development of a hybrid electric transmission packaged into essentially the same space as a conventional automatic transmission for front-wheel drive. This was accomplished by means of a space-efficient arrangement based on two planetary gear sets and electric-motor generators with large internal diameters. A combination of damper and hydraulically controlled clutch permit comfortable shutdown and restarting of large-displacement engines in front-wheel drive vehicles. The hybrid system, which assures electric low-speed urban driving, features two continuously variable ranges of transmission speed ratios, four fixed transmission speed ratios, electric acceleration boosting, and regenerative braking. In its first vehicle application, the two-mode hybrid contributes to a reduction of fuel consumption by approximately one third.

R. Shimizu (lecturer), **K. Nakata**, **M. Kanda**, Toyota Motor Corporation, Shizuoka, Japan: "Analysis of a Lean Burn Combustion Concept for Hybrid Vehicles":

Lean-burn combustion is one of the most efficient technologies for improving an engine's thermal efficiency.

Toyota regards hybrid technology as the key to building vehicles with lowest fuel consumption levels. When the new lean-burn engine is combined with a hy-

Figure 6: Ceremonial hall

brid powertrain, the thermal efficiency under high load must be improved in order to lower CO_2 emission further. This means that under high load, NO_x emissions must also be reduced.

The lecturer explained that various combustion concepts had been analysed and homogenous direct injection in combination with lean-burn combustion had been selected as the new strategy for lowering CO_2 and NO_x emissions. With a view to achieving ultra-lean combustion with a homogenous air/fuel ratio, turbulence intensification measures had to be applied and the ignition system had to be adapted.

In combination with downsized boosting, it could be expected, he stressed, that when this new combustion concept would be applied in hybrid vehicles, massive reductions of CO_2 and NO_x could be achieved.

Figure 7: Exhibition: race cars in the design competition Formula Student

5 New Diesel Engines and Concepts

Dipl.-Ing. F. Rudolph (lecturer), **Dr.-Ing. J. Hadler**, **Dipl.-Ing. H.-J. Engler**, **Dipl.-Ing. A. Krause**, **Dipl.-Ing. M. Stamm**, Volkswagen AG, Wolfsburg: "Volkswagen's New 1.6l TDI Engine":

With its new 1.6 l 4V TDI, Volkswagen presents an entirely newly conceived unit which will form the basis for all of Volkswagen's future four-cylinder diesel engines.

On the basis of the 2.0 l 4V TDI which has a power output ranging from 81 kW to 125 kW, a wide variety of modifications were made to the new 1.6 l 4V TDI which has a maximum output of 77 kW. These modifications include the changeover of the injection system to the PCR 2-System supplied by Continental and operating at an injection pressure of 1600 bar and a two-piston high-pressure pump with an integrated mechanical pre-injection pump. In each cycle, up to six injections are possible. This unit also incorporates a piezo actuator with a seven-hole injector.

In accordance with the significant reduction of engine capacity, almost every single engine component had to be redesigned and friction losses were significantly brought down.

S. Kimura , **E. Matsumoto**, **M. Yamane**, Nissan Motor Co., Ltd., Kanagawa, Japan; A. M. de Hoyos (lecturer), Nissan Technical Centre Europe, Barcelona: "Nissan's New Clean Diesel Technology for Japanese Real New Long-Term Regulation":

A new 2.0 l L4 in-line diesel engine with direct injection was designed which optimised acoustic behaviour, vibrations and thermal emissions. Nissan's technologies, which are based on the EURO IV-technologies, were integrated into the new engine: a lean-burn NO_x trap (LNT) was combined with Nissan's new control technology.

This engine, fitted with piezo-electric injectors, delivers an output of 63.5 kW/l and its torque achieves 180 Nm/l at a low compression ratio. Instead of an oxidation catalyst, a three-way catalyst is used as an exhaust gas aftertreatment unit which will allow this engine to comply with the new long-term legal provisions. In order to achieve accurate rich-spike and desulphurisation control rapidly, Nissan devised a model-based control strategy which combines VGT, an exhaust gas return valve, an electronically activated throttle valve and an injector unit.

Dr.-Ing. W. Held, **Dipl.-Ing. G. Raab** (lecturer), **Prof. Dr.-Ing. K.-V. Schaller**, **Dipl.-Ing. W. Gotre**, **Dipl.-Ing. H. Lehmann**, **Dipl.-Ing. H. Möller**, **Dipl.-Ing. W. Schröppel**, MAN Nutzfahrzeuge AG, Munich, Nuremberg, Steyr: "Innovative MAN Euro V Engines without Exhaust-Gas Aftertreatment":

MAN seeks to offer its customers products in the different markets that are of economic interest. Therefore, it devised an AdBlue-free technology, which combined an externally cooled EGR system and a PM-Cat filter, for all MAN series even before the EURO IV legislation came into force. This technology was very well received by customers as it not only has well-known advantages as compared to an SCR technology but also does not result in higher operating costs.

As a result of the early introduction of EURO V in some European countries which generated benefits with regard to road toll fees, MAN Nutzfahrzeuge AG made use of its many years of experience with SCR technology, since the EURO V EGR solution described in detail by the lecturer was not yet available.

The motivation for the development of an AdBlue-free concept resulted from the positive response of customers to the MAN EURO IV EGR/PM cat technology.

Thanks to the new EURO V EGR solution, other EURO IV solutions can be devised to meet customer demands in the emerging markets, which means that MAN can offer a technology worldwide that requires no exhaust-gas aftertreatment.

This technology constitutes the basis for a platform concept for EURO IV / V and EURO VI, in which EURO IV can be realised

Figure 8: Exhibition

without aftertreatment, EURO V in conjunction with an oxidation catalytic converter, and EURO VI with an SCRT system.

6 Powertrain

Dipl.-Ing. E. Schneider (lecturer), **Dipl.-Ing. J. Müller**, **Dipl.-Ing. M. Leesch**, **Dipl.-Ing. R. Resch**, IAV GmbH, Chemnitz: "Optimised Transmissions as a Result of Integrated Powertrain Design":

On the basis of the power and energy required, the lecturer examined the potential for reducing CO_2 emissions through vehicle and transmission design measures. In keeping with efficiency improvements and the ensuing trend towards reducing swept volume and narrowing speed spreads, transmission systems as well as the influence of the total speed range and the number of speeds play a vital role. The integration of electric motors constitutes a further potential for lowering CO_2 emissions. In addition, the lecturer illustrated computer-aided synthesis programmes which systematically generate new transmission systems with optional hybrid functions with a view to tapping existing potential. As a result of this targeted research, a new eight-speed dual-clutch transmission system and an eight-speed hybrid automatic transmission system were developed. These structurally optimised transmission systems reconcile the conflicting goals of efficient map conversion and compact, cost-effective design.

M. Uchida, **S. Ishii**, Nissan Motor Co., Ltd., Kanagawa Japan; **K. Usuki**, Jatco Ltd., Kanagawa Japan; **M. Rowland** (lecturer), Nissan Motor Manufacturing (UK) Ltd., Louvain-la-Neuve, **K. Takemoto** (lecturer), Jatco France SAS, Lardy: "Brand-new NISSAN/JATCO Seven-Speed Automatic Transmission and Its HEV Derivative": Nissan and Jatco have developed a new seven-speed AT, which can reduce CO_2 emissions by up to 7 % as compared to the existing five-speed AT.

The lecturer addressed several new features used in the AT and its derivative which will be launched in the near future. As to the hardware, friction was minimised by means of an efficient design of each component combined with a newly developed transmission fluid.

As to the software, a new gear shift control logic and electric acceleration pedal "ECO-Pedal" which is controlled harmonically with the gear shift schedule optimises drivability and fuel economy. Finally, the lecturer described the derivative for an HEV version. He then also illustrated a derivative for electro-hybrid vehicles.

"ECO-Pedal"is a world first system, which assists eco driving by controlling the reaction force of the acceleration pedal harmonically with the gear shift schedule. The system detects the pressure applied to open the accelerator; when excess pressure is applied, a counter push-back tells the driver to ease up in order to save fuel. With this ECO-Pedal excessive acceleration can be avoided and fuel economy in the real world improved by 5 to 10 %.

The drive for electric-hybrid vehicles consists of an integrated motor and two clutches instead of a torque converter.

Dr.-Ing. J. Greiner (lecturer), ZF Getriebe GmbH, Kressbronn; **Dr.-Ing. B. Vahlensieck**, **Dr.-Ing. M. Mohr**, **Dipl.-Ing. P. Casals**, ZF Friedrichshafen AG, Friedrichshafen: "Fuel-Efficient Driveline Systems":

In the USA and Japan more than 90% of all customers have traditionally preferred automatic shifting, while, not long ago, the use of automatic transmissions on the European market was almost exclusively limited to premium cars with 6, 8 and 12 cylinder engines. One reason was high additional costs for the automatic transmission as an "extra", another reason was the image of this type of transmission which had the reputation of being very convenient but, above all, gas-guzzling and the opposite of sporty. This changed dramatically with the introduction of automatic transmissions featuring up to 8 gears and a high ratio spread as well as due clutch transmissions, which are regarded as being very sporty. By using optimised starting devices, intelligently arranging the gear sets and applying efficient electrohydraulic control systems, engineers have managed to design units which differ only marginally from manual transmissions both in terms of cycle and actual consumption.

Against the background of enormously increasing fuel costs it is to be expected that, based on such automatic transmissions, a significantly higher volume of micro, mild, and full hybrid versions will find their way into drivelines as so-called parallel hybrid systems. The range of functions includes the start-stop system with the micro hybrid, recuperation and boost mode in the case of the mild hybrid, and electric driving with the full hybrid.

The lecturer demonstrated the influence of the individual systems on fuel consumption by presenting the transmission and hybrid portfolios of ZF with due regard for the internal efficiency and load point shifting as a function of the number of speeds and the gear ratio. He pointed out that the different systems were compared and analysed and their interactions with future engine and vehicle technologies were assessed by means of simulations and efficiency measurements.

7 Future Powertrains

Dipl.-Ing. P. Langen (lecturer), **Dipl.-Ing. W. Nehse**, BMW Group, Munich: "BMW Efficient Dynamics – A Look into the Future":

The lecturer drew attention to the fact that no other car manufacturer had more strongly reduced CO_2 emissions across its whole model range than the BMW Group with its Efficient Dynamics strategy.

In the meantime, 21 BMW models and six MINI models emit less than 140 g of CO_2 per kilometer.

Important elements of this development strategy include brake energy regeneration, the gear shift indicator, the engine start-stop function, intelligent energy and drive train.

The focus continues to be directed at the primary use of energy in the internal combustion engine. This is where a combination of different components from BMW's modular technology kit (Valvetronic, Bi-VANOS, High Precision Injection, Turbocharging) could contribute towards reducing CO_2 emissions to an ever greater extent in the future.

Especially through the use of exhaust gas energy for turbocharging in combination with fewer cylinders, an analysis of the entire drivetrain topography and the vehicle as a whole opens up new opportunities.

A comparison of various customer-oriented profiles reveals considerable potential for more extensive use of brake energy, especially in urban areas.

If an electric motor is used to convert this energy, we are then faced with the questions of how and when this energy is to be fed back into the vehicle. If the electric motor is sufficiently powerful, it can also perform drive functions and it has to be considered how the two drive units should be dimensioned relative to one another. In the context of a marginal analysis, the electric motor is capable of replacing the internal combustion engine completely.

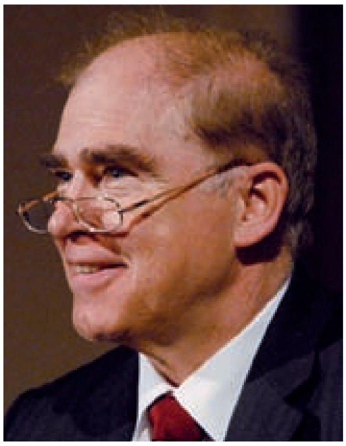


Figure 10: Prof. Dr. H. List, AVL List GmbH

The MINI E, the first electric vehicle launched in the market by the BMW Group, constitutes a field trial on the road as regards the alternative drive units of tomorrow.

H. S. Lee Ph.D., (lecturer), Vice Chairman R&D Division, Hyundai-Kia Motor Company, Gyeonggi-Do, Korea: "Hyundai-Kia's Powertrain Strategy for Green and Sustainable Mobility":

With "Blue Drive" products and technologies, Hyundai-Kia Motor hoped to be able to achieve a fleet average of 35 miles per gallon by 2015, the lecturer explained. This was five years ahead of the US government's deadline for fuel reduction. In the meantime, Mr. Lee pointed out, Hyundai-Kia Motor Company would also meet the European emission standards of 130 g CO_2/km by 2012. In his presentation, the lecturer explained his company's powertrain strategy in three parts:

- fuel efficiency improvements in the conventional internal combustion engine
- the development of engines using alternative energies
- the design of environmentally friendly vehicles.

He illustrated various new engines and transmission systems which were to improve fleet fuel economy while, at the same time, complying with more stringent emission standards. Bioethanol, biodiesel and CNG vehicles are being developed by Hyundai-Kia Motor to reduce its dependence on petroleum. The ultimate goal of developing environmentally friendly vehicles is to develop a pollution-free vehicle that has zero gas emission. The Hybrid Electric Vehicle (HEV) is considered as a possible alternative while the Fuel Cell Electric Vehicle (FCEV) is viewed as an ultimate solution. Hyundai-Kia Motor aims to rank among the world's top five automakers by 2010 in terms of sales, corporate social responsibility, and environmental performance.

Prof. Dr. L. Mikulic (lecturer), **Prof. Dr. H. Kohler**, Daimler AG, Stuttgart: "Powertrain Technologies for Sustainable Mobility at Mercedes-Benz":

Prof. Mikulic stressed that Daimler AG would meet future challenges with their roadmap for sustainable mobility. In addition to searching for better energy sources, this roadmap consisted of three steps, he explained. The first step began with the optimisation of internal combustion engines. The goal is to make the petrol engine as economical as the diesel engine and the diesel engine as clean as the petrol engine. Milestones on the way to the convergence of these two combustion principles are for example the petrol engine with second generation direct fuel injection and the BlueTEC-technology for diesel engines. These two combustion principles were reconciled in the DIESOTTO engine concept that combined the advantages of both.

Figure 9: Dr. H. Demel, Magna International

For further efficiency enhancements, the combustion engine will be assisted by electric motors in modular hybrid concepts as a the next step. Depending on the purpose for which it is used and on the demand of customers, the engine will be equipped with a start-stop function, boosting, recuperation as well as pure electric driving will be the options.

As a third step, these technologies will lead to emission-free driving with battery-electric and fuel cell vehicles. The future has just begun: Daimler AG is presently running the biggest fleet of test vehicles with battery-powered electric and fuel cell drives worldwide.

8 Emission Reduction

Dipl.-Ing. W. Müller (lecturer), **Dr.-Ing. I. Lappas**, **Dr. rer. nat. A. Geißelmann**, Umicore AG & Co. KG, Hanau: "After-Treatment Systems for Heavy Duty On- and Off-Road Applications":

Figure 11: Dr. K.-T. Neumann, Continental

In view of the future exhaust gas emission legislation it can be expected that all options for internal engine modifications destined to bring down emission levels must be utilised. In addition, a combined exhaust-gas after-treatment of particulate and nitrogen emissions will be required. The SCR technology which uses urea as a reducing agent is increasingly emerging as the method of choice for NO_x after-treatment in heavy-duty utility vehicles for on- and off-road applications.

The lecturer illustrated designs and technological solutions for the individual system components of catalytic converters (DOC, particulate trap, SCR) that will permit heavy duty vehicles to comply with the new standards.

He also described the activity features of a number of state-of-the-art Fe-zeolith-, cuzeolith- and vanadium-based SCR technologies. New trends in the development of SCR catalytic converters demonstrate a significant improvement of durability at high temperatures and intensified SCR activity at low temperatures even when exhaust-gas emissions have a low NO_x content.

Dr.-Ing. B. Mahr (lecturer), Mahle GmbH, Stuttgart; **Dr. sc. techn. M. Warth**, **Dipl.-Ing. J. Rückauf**, **Dr.-Ing. A. Elsäßer**, Mahle International GmbH, Stuttgart: "Innovative Exhaust-Gas Recirculation System for Cost and Fuel Efficient Compliance with Emission Standards"

The lecturer elaborated on an innovative exhaust-gas recirculation system which, by using a rotating flap, generates temporary negative pressure impulses in order to produce the desirable exhaust-gas recirculation rate under different engine operating conditions. Through an integrated control of both charge air and exhaust-gas mass flow, the EGR rate can be increased while the pumping work can be reduced at the same time by means of this system. Thanks to the highly efficient, flexible electronic activation flap on the intake side, high exhaust-gas recirculation rates can be attained without the need for adjusting exhaust-gas counter-pressure. With this arrangement, the EGR system is not burdened by high exhaust-gas temperatures and contamination. The highly dynamic response of the system and flexible electronic activation allow a nearly instantaneous control of EGR rates as required in different operating states across the entire map and during engine warm-up, which will become a much more significant feature in view of future test cycles. The lecturer presented the experimental results of an application of this system in a utility vehicle engine. He demonstrated that over wide map ranges, EGR flow rates could be increased by more than 50 %, with a marked reduction of NO_x emissions and specific fuel consumption.

Dr.-Ing. M. Flik (lecturer), **Dr.- Ing. S. Edwards**, **Dr. E. Pantow**, Behr GmbH & Co. KG, Stuttgart: "Emission Reduction in Commercial Vehicles through Thermal Management":

Through the combination of exhaust-gas recirculation and exhaust-gas after-treatment, significant progress was made over the past few years so that compliance with future emission standards is assured. At the same time, it was possible to reduce fuel consumption through these measures.

Despite the progress made over the past few years, roughly 50% of the energy contained in the fuel still goes unused as it is released into the atmosphere in the form of hot exhaust gases. Against the background of climate change and the significant role freight traffic plays for national economies, greater attention must be given to energy recovery in vehicles.

The design of the vehicle cooling system and the specific components used are decisive factors for harnessing the energy contained in hot exhaust gases. The lecturer illustrated methods for energy recovery in vehicles and the impact of these on the vehicle cooling system and analysed these methods for their economic feasibility.

Under real-life driving conditions, the lecturer stressed, the potential for reducing fuel consumption through thermal management, including the recovery of heat contained in exhaust gases, amounted to approximately 10 %.

9 Hybrid 2

Dr. R. Fischer, AVL List GmbH, Graz: "The Electrification of the Powertrain – From Turbohybrid to Range Extender":

AVL expects a distribution of electrification by the year 2025 which presents the following picture: a focus on both mild hybrid systems, which will be applied on a broad basis, and electric-motor powered vehicles with integrated range extenders which will primarily be used in intra-urban traffic.

In order to obtain the optimum cost/benefit ratio, new engine concepts are being developed, with the key always being system optimisation, i.e. the engine, the transmission, the electric motor, the battery and control strategies must be optimally adjusted to one another. The lecturer quoted as examples two concepts on which developments are mainly focused:

- The turbohybrid: using additional functions of the combustion engine permits a simplification of the electric system: Through supercharging and electric motor traction support during starting, the potential for higher fuel economy through load point shifting can be tapped to a great extent by means of a low-cost, mechanical transmission with optimised efficiency. The overboosting capacity of a turbocharged engine permits permanent recharging of the battery which results in a marked reduction of the required battery capacity and thus also in lower system costs. Such a combination, as implemented in the turbo-hybrid, constitutes an attractive universal strategy assuring excellent fuel economy and great driving pleasure.

– Range extenders: two different approaches are possible: Solutions with direct drive or range extenders without direct drive. The demands made upon pure range extenders (PRE) are completely different from those made upon today's combustion engines. The basis is an electric vehicle, and potential customers are buyers of electric vehicles, which means that these are primarily designed for urban driving. For cost and weight reasons, however, the battery should be as small as possible, i.e. for distances that are rarely driven this battery would not be suitable. The operating range of the combustion engine is limited to single-point operation for recharging the battery. This gives rise to entirely new, low cost, weight-optimised concepts for combustion engines. The acoustic behaviour of and packaging space for such PREs are a particularly significant parameter.

Dipl.-Ing. P. Langen, **Dr. M. Klüting**, **Dr. M. Wier**, **Dipl.-Ing. F. Kessler**, **Dr. B. Curtius**, **Dipl.-Ing. H.-S. Braun** (lecturer), **Dipl.-Ing.G. Thiel**, BMW Group Munich: "The Full Hybrid Powertrain for BMW X6 (BMW ActiveHybrid)":

In 2008, the BMW Group launched the new BMW X6 XDrive and the new BMW 7-series.

In line with its EfficientDynamics-Strategy, the BMW Group developed a full hybrid powertrain for the X6 and a mild hybrid powertrain for the BMW 7-series to complement the highly efficient gasoline and diesel powertrains.

As a result of the close interaction between the combustion engine, the electric motor and the transmission, the features of this product are strongly determined by software functions. Efforts to achieve this interaction predominated in the development of the hybrid powertrains.

Despite the additional weight of the hybrid components, typical BMW dynamics have been maintained and even improved. CO_2 emissions were significantly reduced at the same time, and fuel consumption rates are extremely low.

Even though the transmission hardware and basic software were conceived in co-operation with GM and Daimler, BMW succeeded in retaining the typical characteristics of its brand in the BMW X6 Active Hybrid.

In combination with the V8 twin power-turbocharged engine and an electric engine mounted on the crankshaft and the new eight-speed transmission, BMW was able with its BMW 7-series active hybrid to lower fuel consumption drastically while improving driving behaviour at the same time.

Dipl.-Ing. O. Vollrath (lecturer), **Dr. N. Armstrong**, **Dipl.-Ing. J. Schenk**, **Dipl.-Ing. O. Bitsche**, **Dr.-Ing. A. Lamm**, Daimler AG, Stuttgart: "S 400 BlueHYBRID – the First Hybrid Vehicle with Li-Ion-Technology":

Mercedes-Benz is forging ahead with the electrification of the powertrain for vehicles in all performance classes and across all model series from the start-stop system to the full hybrid vehicle.

The S 400 BlueHYBRID thus represents the first Mercedes-Benz hybrid. Equipped with the features of a start-stop system as well as regenerative braking and electrical drive support, it achieves a reduction in fuel consumption of approximately 20 %.

The goal in packaging was to prevent any impairment of customer benefit.

The design of the components and the selection of standard installation spaces allowed all components specific to the hybrid system to be accommodated in the front end. In this respect, the battery technology played a special role, because it was possible to design a hybrid battery that was no larger than a conventional starter battery, and also fitted into the installation space. This lithium-ion battery was used in a passenger car for the first time.

10 Supercharging and Gas Exchange

Dipl. Ing. N. Klauer, **Dr. M. Klüting**, **Ing. F. Steinparzer** (lecturer), **Dr. H. Unger**, BMW Group, Munich: "Turbocharging and Variable Valvetrains - Fuel Economy Technologies for Worldwide Use":

With its holistic BMW EfficientDynamics approach, the BMW Group started already many years ago to respond adequately to changed requirements so as to safeguard sustainable mobility. Electrification, energy recovery, engine start-stop units and efficient powertrain technologies constitute the vital elements of this strategy.

In the foreseeable future, combustion engines will continue to play the main part as drive systems and therefore further efforts to reduce fuel consumption will remain focused on combustion engines.

With the integration of the Valvetronic 2001 into BMW's gasoline engines, a significant reduction of fuel consumption was achieved for the first time in mass produced engines. With direct injection and turbocharging, further milestones were reached on the path towards higher fuel economy through new engine technology.

Consistent further improvements in design and an intelligent combination of these engine and powertrain concepts could play a central role for innovative drive systems for worldwide use.

T. Tomoda (lecturer), **T. Ogawa**, **H. Ohki**, **T. Kogo**, **Dr. K. Nakatani**, **E. Hashimoto**, Toyota Motor Corporation, Shizuoka, Japan: "Improvement of Diesel Engine Performance through Variable Valve Timing":

Toyota studied the effects of variable valve timing and lift in order to improve the thermal efficiency of diesel engines,

Figure 12: Koei Saga, Toyota

while reducing engine-out emissions at the same time.

Under high load, the early closing of one or the early opening of both intake valves intensifies swirl motion without leading to higher pumping losses. Late closing of the intake valves results in a reduction of the effective compression ratio, so that it is possible to increase the EGR ratio and advance the start of fuel injection. As a result, low NO_x formation and improved thermal efficiency can be achieved simultaneously.

Under low load conditions, heat release is small and fuel is dispersed finely through the swirl because of the small fuel injection quantity. Therefore increasing the effective compression ratio through advanced intake valve closing is effective for reducing HC-formation.

Variable valve timing can also be applied to a wide operating range, thus bringing down NO_x raw emissions by as much as 40 % and improving fuel efficiency by 4 % in the NEDC.

Furthermore, low-end torque was raised by 40 % through exhaust pressure pulsation. In order to achieve these results, a new piston chamber design with deep valve pockets was also required.

Dr.-Ing. B. Huurdeman (lecturer), **Dipl.-Ing. (FH) J. Kosicki**, **Dipl.-Ing. (BA) H. Schick**, **Dipl.-Ing. (BA) D. Talmon-Gros**, MANN+HUMMEL GmbH, Ludwigsburg: "Development of Turbo Charger Parts Using High Performance Plastics":

As a downsizing measure for gasoline engines, the turbocharger has reached the volume market of base engines which has increased the pressure on cost and weight. For Diesel engines, additional variability in the flow path is being developed. The switching operation requires flaps, valves or variable guide vanes in the area of the turbine or the compressor. These switching devices make high demands on temperature and corrosion resistance and have to be integrated into the charging components because of packaging space and mass restrictions.

In order to develop cost-effective, lightweight and integrated solutions for ducts, housing and switching devices on the compressor side of the turbocharger, it makes sense to use plastic parts. But at high temperatures, only high performance plastic materials can fulfil the requirements of durability and dimensional stability.

Over the past few years, MANN+HUMMEL which is a supplier for air intake systems has carried out many investigations and designed innovative parts.

The lecturer described two switching devices in the flow path close to the compressor and also a compressor housing made of the high performance thermoplastic resin PPS, and illustrated the options and restrictions that must be realised when using plastic materials. The results of tests on component and engine test benches indicated, he went on to say, a promising future for plastic materials on the basis of the current state of the art.

11 Future Energy Supply

Dr. J. Adolf, Shell Deutschland Oil; **Dipl.-Ing. R. Huibers**, **Dr. W. Warnecke** (lecturer), Shell Global Solutions Deutschland GmbH, Hamburg: "Shell Passenger Car Scenarios to 2030 – Facts, Trends and Options for Sustainable Auto-Mobility in Germany":

The 25th edition of the Shell Passenger Car Scenarios is characterised by transition. Following a uniform socio-economic lead scenario, the present Passenger Car Study 2009 starts by analysing the possible consequences of demographic change on future auto-mobility in Germany. It tracks continuation of today's car ownership and mileage patterns up to 2030, differentiating by age and sex. The state of auto-mobility in Germany will in future be characterised more strongly by women and by older people. The motorisation of the German population will continue to increase. Passenger car mileage will continue to rise till 2020, dropping back to today`s level towards 2030 .

The study uses two mobility scenarios to examine the sustainability of auto-mobility in the coming years, in terms of energy consumption and CO_2 emissions. One of these scenarios, called "Automobile Adaptation", assumes a continuation of today's trends and behaviour patterns in the future, and already shows a substantial reduction in energy consumption and CO_2 emissions of the passenger car fleet. The alternative scenario "Auto-Mobility in Transition" is characterised by rapid technological change and greater diversification of propulsion and fuel technologies. But conventional propulsion systems and fuels will still be playing a key role by 2030.

Figure 13: D. M. Hancock, GM Powertrain

Dr. A. M. Lippert, **Dr. G. J. Smyth** (lecturer), General Motors R&D and Strategic Planning, Warren, USA: "Global Energy Systems in Transition: Next Steps in Energy Diversity for Transportation":

The challenges and opportunities for transforming the world's transportation energy systems for long-term sustainability of mobility have never been greater. Volatility in petroleum markets has contributed to economic disruption. Environmental and energy supply concerns continue to intensify. These externalities must be addressed with a robust portfolio of solutions that promise to transform the energy system for personal mobility.

In the past two years alone, these solutions have made significant advances:

- improved energy efficiency of internal combustion engines and transmissions, and increasing capability to utilise existing biofuels
- electrification of the vehicle propulsion system through hybrids (including plug-in), extended-range electric vehicles, and fuel-cell electric vehicles
- a shift in the use of energy resources towards low-carbon fuels, such as biofuels and renewable electricity, which can be used in both conventional and electrified propulsion systems.

In view of today´s large vehicle population, the introduction of new technolo-

Figure 14: Prof. Dr. M. Winterkorn, Volkswagen AG

gies initially has only a small overall impact, but an accelerating displacement of the existing fleet can be expected as time progresses. It is imperative to maintain constancy of purpose with regard to the technological transformation of vehicle propulsion and energy supply systems, as well as policies, infrastructure, and social objectives. Doing this will ensure that market tipping points can be achieved by 2025 across this portfolio of solutions, and the challenge of sustainable personal mobility will be solved by a commercial roll-out of new technologies in large numbers.

Univ.-Prof. Dr. G. Brauner, University of Technology, Vienna: "Electrical Energy Supply and Mobility in Europe":

Two parallel paths are followed in order to assure a sustainable supply of energy for mobility. On the one hand, combustion engines and their exhaust gas after-treatment systems are further improved with a view to lowering emissions significantly, thus reducing specific energy demand. It can be assumed that combustion engines will be used in the long run in vehicles offering high driving convenience. Second and third generation biogas and biofuels can contribute to their sustainability.

On the other hand, electric-motor driven vehicles will gain ground especially in short-distance traffic in the suburban areas of metropolises. In this sector, small electric cars which can run on renewable energies offer major advantages, as they operate without emissions and produce little noise. On account of the high efficiency of the electric drive and braking energy recuperation they also have a lower specific energy demand. Eletric vehicles are best suited for suburban office hour traffic in the megacities of the future, which will be characterised by stop-and-go traffic. The electric batteries of these vehicles can also be used as a storage medium for renewable energy in energy-active settlements in the grid-to-car or car-to-grid mode. Thus day and night balancing of photovoltaic energy supply or compensation for natural fluctuations in the supply of energy from predominantly regenerative sources is possible.

12 Combustion and Downsizing Concepts

D ipl.-Ing. J. Willand, **Dr. J. Jakobs** (lecturer), **Dr. E. Montefrancesco**, **Dr.-Ing. M. Daniel**, **Dipl.-Ing. V. Vortkamp**, **MBE**, **Dr.-Ing. B. Läer**, Volkswagen AG, Wolfsburg: "The Volkswagen GCI Combustion System for Gasoline Engines - Potential of and Limits to CO_2 Emission Reduction":

The Golf with the 1.6 l GCI-engine which was launched in late 2006 was the first passenger car worldwide presented to the public and ready for driving that is equipped with a self-ignition gasoline engine. The Volkswagen GCI combustion system integrated into the engine uses a part load combustion process based on homogenous self-ignition of the fuel which combines improved fuel economy as compared to conventional spark plug-induced stochiometric combustion with minimal nitrogen raw emissions.

The lecturer described the method applied in designing the GCI combustion process which involved the intensive use of various calculation methods, three-dimensional simulation of mixture formation and coupling CFD flow simulation with chemical reaction kinetics for shaping the self-ignition process.

Subsequently he explained the steps that had to be taken in engine management when transferring the GCI combustion process from the engine test bench to the vehicle.

A fundamental analysis of options and limits of part-load combustion over the entire system for reducing fuel consumption in real life driving showed that while fuel economy can be improved through these combustion processes, the combustion rates attained through downsizing with TSI technology cannot be reached under the boundary conditions relevant for Volkswagen. This can mainly be ascribed to the short time spans during which the engine is operated in the fuel-efficient, limited part load range. When an engine having the same capacity is operated outside this part load range, the relevant specific fuel consumption rates are clearly higher than those of a lower-capacity downsized engine.

Dipl. Ing. R. Weinowski (lecturer), **Dipl. Ing. A. Sehr**, **Dipl. Ing. S. Wedowski**, **Dr. Ing. S. Heuer**, **Dipl. Ing. T. Hamm**, **Dipl. Ing. C. Tiemann**, FEV Motorentechnik GmbH, Aachen: "Future Downsizing of S.I. Engines – Potential and Limits of Two- and Three Cylinder Concepts":

Today, downsizing, in combination with intake air pressure charging, is one of the most promising opportunities for the reduction of CO_2-emissions from internal combustion engines. In the small size vehicle class with relatively low capacity engines, three cylinder, instead of conventional four cylinder engines are increasingly being applied. Any further decrease of engine swept volume gives rise to the question as to whether a further reduction of cylinder displacement

Figure 15: Contented faces at the end of the successful symposium: from left to right: D. M. Hancock, GM; K. Saga, Toyota; Prof. Dr. M. Winterkorn, VW; Prof. Dr. H. P. Lenz

or the introduction of two cylinder engines would be preferable. Furthermore, the high cost sensitivity in this vehicle class and the required technological input have to be considered.

Based on the current state-of-the-art of small gasoline engines, the potential for lowering CO_2-emissions through downsizing while maintaining similar driving performance was analysed and the limits to reducing single-cylinder displacement were described.

The lecturer pointed out that besides the combustion chamber configuration determined by design and thermodynamic considerations, questions relating to the response behaviour of potential charging units, maximum engine performance, emission levels and friction losses played a vital role.

At the end of his presentation, the lecturer briefly evaluated the NVH behaviour of two- and three-cylinder engines.

Dr.-Ing. P. Kreuter, **Dipl.-Ing. U. Peter**, **Dipl.-Ing. M. Kier**, **Dipl.-Ing. S. Wegner**, **Dipl.-Ing. M. Müller**, **Dipl.-Ing. R. Bey** (lecturer), Meta Motoren- und Energie-Technik GmbH, Herzogenrath: "Meta Downsizing Concept: Reduction of CO_2-Emissions to 75 g/km":

In the light of today's state-of-the-art, downsizing strategies for base engines used in smaller vehicles, i.e. small displacement engines, can be successfully applied only to a limited extent. For the subcompact vehicle category, the boundary conditions for effective downsizing in terms of thermodynamics can be markedly improved through the reduction in the number of cylinders. However, a smaller number of cylinders in combination with high supercharging rates also lead to high rotational irregularities at the crankshaft, thus resulting in lower driving comfort.

The Meta Downsizing strategy applied to a two-cylinder engine comprises the following technology approaches and measures:

- downsizing and simultaneously reducing the number of cylinders
- supercharging by means of a spontaneous charging device
- balancing torque fluctuations on the crankshaft
- Fuel: CNG.

As compared to the base engine this strategy results in improved performance and a reduction of CO_2-emissions by 75 g/km in the NEDC (in the compact car category) if vehicle design modifications are also made.

13 Downsizing – Otto Engines

Dr.-Ing. J. Hadler, **Dr.-Ing. R. Szengel**, **Dr.-Ing. H. Middendorf** (lecturer), **Dipl.-Ing. A. Kuphal**, **Dipl.-Ing. W. Siebert**, **Dipl.-Ing. Lars Hentschel**, Volkswagen AG, Wolfsburg: "Minimum Consumption – Maximum Power: TSI-Technology in Volkswagen's New 1.2l Engine":

After Volkswagen, three years ago, laid the foundations for the worldwide success of small gasoline engines with outstanding low-end torque and very low fuel consumption with its 1.4 l 125 kW TSI by combining double supercharging and direct injection, the 1.4 l 90 kW TSI engine which was launched last year represented a breakthrough for TSI technology also in the mid-size engine category.

Therefore, Volkswagen extended its TSI strategy by launching the new 1.2 l TSI engine which, having an output of 77 kW, is destined for the Polo and Golf class. The further evolution of engine technology for the small but powerful EA111 engine series included a consistent optimisation of friction losses and lightweight design. The engine, which boasts a new weight-optimised aluminium crankcase and an entirely novel combustion process, combines high performance, low fuel consumption and affordable costs for mass production.

Dipl.-Ing. D. Borrmann (lecturer), **Dipl.-Ing. B. Pingen**, **Dipl.-Ing. B. Müller**, **Prof. Dr. P. Kelly**, **Dipl.-Ing. K. Küpper**, **Dr.-Ing. M. Wirth**, Ford Werke GmbH, Cologne: "The Powertrain with a Small Downsized Engine: Design Strategies and System Components":

The technology for downsizing gasoline engines is currently gaining ground in the compact cars of the C- and also the B-class. The additional combination with downspeeding results in a further reduction of CO_2-emissions.

This, however, calls for engine sizes and swept volumes which may have considerable disadvantages in the practical handling of vehicles.

Special attention was given to starting behaviour, switching requirements and transient acceleration capability.

Therefore, the entire drivetrain must be regarded for the successful implementation of a downsizing and downspeeding strategy, in order to find solutions that eliminate potential weaknesses on the one hand, but can also be applied to cost-sensitive market segments on the other.

Alongside calibration strategies for optimum starting behaviour, the stepping of gears was modified, which facilitates starting in the first gear but at the same time permits downspeeding over a broad range. If the available torque is properly conceived, an increase in the number of gears is not necessary. Automatic transmission guarantees customer acceptance. In order to reach similar driving behaviour with manual transmission, intelligent coupling systems can be used which decouple the clutch engagement from typical pedal motion characteristics of the driver.

Y. Boccadoro, **O. Tranchant** (lecturer), **R. Pionnier**, **H. Engelhardt**, Renault s.a.s., Rueil-Mailmaison: "The New TCe 130 1.4 l Turbocharged Gasoline Engine of Renault":

Renault is extending its TCe engine family which began with the TCe 100 with a swept volume of 1.2 l launched two years ago.

The lecturer described the supercharged 1.4 litre gasoline engine TCe 130. The goal was to design an engine that could replace a 2.0-litre naturally aspirated engine in the 100 kW-class and reduce CO_2-emissions.

The new engine has a maximum torque of 190Nm above the speed of 2250/min and a maximum performance of 96 kW at 5500/min. Special attention was given to acceleration in the low speed range.

As compared to the 2.0 l naturally aspirated engine, the TCe 130, which is used in the new Mégane III, shows 19% lower fuel consumption in the European Driving Cycle.

The engine is based on the Nissan 1.6 l naturally aspirated engine. The main design characteristics of this engine, which is entirely made of aluminium with iron-cast liners, are a forged crankshaft with eight counterweights, and a variable valve timing system on the inlet camshaft.

The TCe 130 was developed jointly by Reanult and Nissan .

14 Mixture Formation

Dr.-Ing. M. Dürnholz, (lecturer), **Dr.-Ing. R. Busch**, **R. Baskaran**, **B.Sc., B.Tech.**, **MBA**, **S. L. Kulkarni**, **BE (Mech.)**, **G. Anthony**, **BE (Mech.)**, Bosch Ltd., Bangalore, India: "Bosch Common Rail System for Small Diesel Engines in Emerging Markets":

India in particular is showing signs of becoming a potential global hub for the small car market in the future. The per capita income in India is about 1000 US$, whereas in Germany the per capita income is about 38.000 US$. This translates into a very small population of Indian consumers who can afford products currently marketed in Western Europe and the USA. The challenge both for OEMs and suppliers is to offer unique products at dramatically lower costs to meet the special needs and match the lower purchasing power of buyers in most emerging markets.

One outcome of such an innovative model is the Tata Nano which was unveiled at the Auto Expo India 2008. Developing and creating low-cost products is not tantamount to low-tech products. It is about having a holistic approach consisting of local development and local manufacturing in line with clearly defined price-performance targets.

These ultra low cost automobiles target the huge consumer base in the emerging markets who drive two-wheelers today (7 million new motor cycles are bought every year in India alone). By 2010, automobiles in major Indian cities will have to comply with BS4 emission standards (equivalent to Euro IV). Under these boundary conditions, Bosch conceptualised a robust, highly competitive common-rail system for small diesel engines which meets the demands of car-makers for a state-of-the-art technology that will comply with future emission standards while offering a realistic price-performance ratio. The lecturer then explained the need for adopting highly advanced technology to the segment of low-cost vehicles and meeting the requirements made upon an injection system under difficult boundary conditions. At the end of his presentation, the lecturer gave an outlook for the future potential of this system.

Dr.-Ing. D. Schöppe (lecturer), **Dipl.-Ing. S. Zülch**, **Dipl.-Ing. D. Geurts**, **C. Gris**, **Dr.-Ing. R. W. Jorach**, Delphi Diesel Systems, Europe: "Delphi's New Direct Acting Common Rail Injection System":

In the Direct Acting Common Rail System, the injector needle is set in motion by a piezo ceramic actuator instead of an indirect control incorporated in an electro-hydraulic servo mechanism used by conventional fuel injection technologies. Thus the nozzle needle can be opened and closed very rapidly, independently of rail pressure. With the aid of a two-stage needle motion amplifier, the injection process can be accurately controlled at any time. The integrated additional fuel accumulator operates as a "rail in the injector" and improves injection quality, especially for multiple injection. The injector operates without any leakage.

The use of a piezo actuator as the driver for the direct acting injector has given rise to a number of new demands upon electronics. Therefore, a new electronic control system was devised in order to assure optimum actuation of the direct acting injector.

The lecturer elaborated on the design and operating principles of the direct acting CR system and its performance characteristics, which constitute the basis for a premium diesel engine.

Dipl.-Ing. M. Miyaki, **Dipl.-Ing. K. Takeuchi**, **Dipl.-Ing.K. Ishizuka**, **Dipl.-Ing. S. Sasaki**, Denso Corporation, Aichi-Ken, Japan; **M. Nakagawa** (lecturer), Denso Automotive Deutschland GmbH, Wegberg: "Breakthrough in Common Rail Systems: Controlled Injection by means of an Injector with Built-in Pressure Sensor":

Denso has developed the world's first common rail system injector with built-in pressure sensor. This technology permits closed loop injection control in the cylinders. With the recently developed combustion technology, the desired reduction of NO_x-emission is achieved primarily through high EGR rates and process strategies close to conventional limits (and the range for attaining these is extremely narrow).

With this newly developed technology, the fuel injection rate, which determines the combustion process in the engine cylinders, is directly detected and controlled in a closed loop to allow precise compensation of the injection system for its entire service life. Highly advanced injection control allows extremely close multiple injections, and variable injection rate control can be assured in mass production for the first time. In addition, the use of this technology offers three major advantages for the overall engine system. Alongside the expansion of the possible calibration range, which improves fuel economy and reduces emissions, it heightens the overall robustness and reliability of the engine. With closed-loop control, the number of man-hours required for calibration can be significantly reduced.

15 Plenary Closing Session: A View into the Future

K. Saga, Managing Officer, Toyota Motor Corporation, Aichi-ken, Japan, **Figure 12**: "Does Hybrid Technology Mean the End of Conventional Combustion Engines?":

Fossil fuel depletion, global warming due to increased atmospheric CO_2, and urban air pollution caused by exhaust emis-

Figure 16: From right to left: Prof. Dr. M. Winterkorn, VW; Prof. Dr. F. Piëch, VW; Mrs. U. Piëch, Mrs. M. Lenz

sions are some of the issues surrounding the future of the automobile. One high profile solution to these problems is the hybrid vehicle. Currently, coming into wider use throughout the world, HVs use energy regeneration and other functions to boost fuel economy and reduce CO_2. Based on the history of Toyota's HV development, the lecturer explained Toyota's view that hybrid technology will not mean the end of conventional combustion engines, but, on the contrary, will extend fossil fuel availability as a result of lower consumption rates and will thus prolong the life of the conventional combustion engine.

Combustion engines would no longer be able to survive once all conventional fuel reserves were depleted, the lecturer stressed.

D. M. Hancock, Vice President, General Motors Corporation, Pontiac, USA, **Figure 13**: "GM´s Voltec Propulsion System: A Further Step in the Electrification of the Vehicle":

In late 2010, General Motors will begin volume production of the Chevrolet Volt, followed approximately one year later by the Opel/Vauxhall Ampera. These automobiles are extended-range electric vehicles (E-REVs) based on GM's Voltec electric propulsion system. They use a rechargeable onboard energy storage system – specifically an advanced lithium-ion-battery – and an electric traction motor to provide full-performance electric propulsion exclusively for up to 64 km driving range (based on Euro MVEG cycle, 40 mile range based on the EPA city and highway cycle). The battery can be recharged by plugging into a standard European 220 Volt (U.S. 120 V or 240 V) outlet or other level 1 and level 2 charging standards. The Voltec propulsion system also includes an onboard electric generator powered by a small internal combustion engine. The generator is engaged automatically to extend the vehicle's total operating range.

The car's 64-km range on initial charge is well suited to most European and North American drivers, who typically travel less than that distance each day. In those cases, the Volt and Ampera may never engage their onboard generators and thus would produce little or no carbon dioxide or other tailpipe emissions over much of the vehicle's life cycle.

Prof. Dr. M. Winterkorn, Chairman of the Board, Volkswagen AG, Wolfsburg, **Figure 14**: "The Multi-Brand Philosophy of the Volkswagen Group":

In his impressive closing lecture, Prof. Winterkorn pointed out that the automotive industry was facing enormous challenges in many respects. Two factors would be crucial to future success: the first was strong, clearly positioned brands and attractive vehicles, and the second was economic and technological capability.

This is where the multi-brand philosophy of the Volkswagen Group came into its own. Nine successful automobile brands – from Audi to Scania, from Volkswagen to Lamborghini – met every customer wish all over the world. And the Group alliance not only gave the brands the required critical mass, but also offered a technological, ecological and economic potential which no other automaker could match. The Volkswagen Group's cross-brand Modular Matrices were the technological backbone for leveraging this potential to the full, the lecturer concluded. On this basis, Europe's largest automaker has set its sights on an ambitious goal: to lead the global automotive industry by 2018.

With this closing lecture, which was followed by a discussion to which leading figures from the automotive industry contributed, the 30th International Vienna Motor Symposium was concluded, **Figure 15** and **Figure 16**.

Conference Documentation

The lectures presented at the 30th International Vienna Motor Symposium are published in their in extenso versions, in the VDI-Fortschritt-Berichte, series 12, no. 697, volumes 1 und 2 (including a CD), and additional brochures.

Illustrated Book "30 Jahre Wiener Motorensymposium" (30 Years of the Vienna International Motor Symposium).

This book contains photographs from 30 years of the Vienna Motor Symposium.

All documents can be obtained from the Austrian Association of Automotive Engineers (Österreichischer Verein für Kraftfahrzeugtechnik, ÖVK).

Invitation

The 31st International Vienna Motor Symposium will take place on April 29th and 30th, 2010 in the Congress Centre Hofburg Vienna.

We should like to extend a cordial invitation to you already at this point in time. After the announcement of the programme on the internet, which will probably be made around mid-December 2009, we urgently recommend that you apply in good time.

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