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NEURAL Mathematical Model for the Determination of the Combustion Pressure Profile

NEW Approach to Turbochargers for Four-cylinder Gasoline Engines

BIOFUELS
AN ALTERNATIVE SOLUTION?

CONTENTS

COVER STORY

BIOFUELS
AN ALTERNATIVE SOLUTION?

4 I Due to their energy density, there will be no substitute for liquid fuels in the foreseeable future. What potential do biofuels have as a replacement for fossil fuels? The Karlsruhe Institute of Technology (KIT) reports in detail on the Bioliq process. The concept, technology and state of development of the decentralised solution for producing biofuels are presented.

COVER FIGURE & FIGURE ABOVE iStock

FAREWELL

Dear Reader,

Most of us find it difficult to say goodbye. Certainly when changing jobs or even when leaving our favourite holiday resort. But saying "farewell" to a person we like and respect is a much harder task.

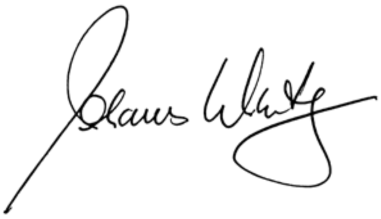
At the end of the year, our publisher Dr. Richard van Basshuysen, who has accompanied the development of ATZ and MTZ for two decades, will be leaving. During this time, his achievements in advancing technical progress in our field have been extraordinary.

It is, of course, quite impossible in these few lines to give a comprehensive appreciation of the life work of this exceptional engineer, journalist and author. Therefore, I would simply like to recall two points from his time as Head of Development at Audi that made a lasting impression on the German automotive industry. As a pioneer of TDI technology, it was he who first launched the direct-injection turbocharged diesel engine into series production. As a result, it was Dr. van Basshuysen who laid the foundation stone for the overwhelming success of the diesel engine in passenger cars, even in the luxury class. What is more, as Head of Development for premium class vehicles, he went on to conquer this class with the predecessor to the A8, the V8, which was an important milestone on Audi's long journey into the premium segment.

Dr. van Basshuysen brought his expertise and contacts from his time in industry with him when he joined ATZ and MTZ. Benefiting from his untiring support, my predecessor Wolfgang Siebenpfeiffer succeeded in turning these magazines into a modern technical media family, including what has since become an extensive range of books published by us.

As a young man, having just joined ATZ and MTZ in the early 1990s, Richard van Basshuysen impressed me above all by the way in which he was not only open to new ideas – and surprised us with many of his own – but also insisted on absolute perfection when it came to putting these ideas into practice. His boundless energy motivated us, and perhaps even overstretched us occasionally. In short: He was a good mentor to us.

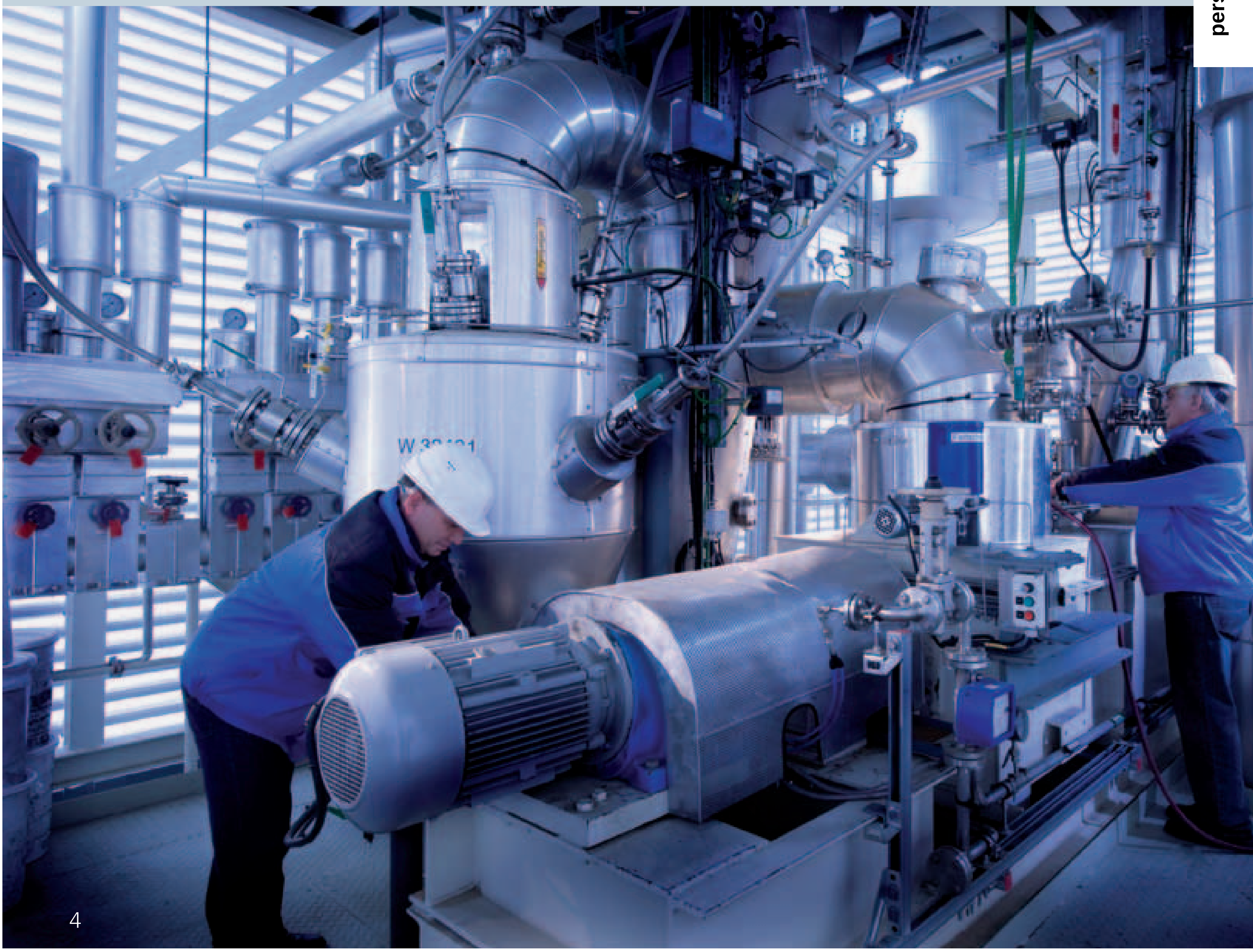
Now, as our paths go separate ways, we not only give him our heartfelt thanks but also bid him "farewell", in the truest sense of the word.



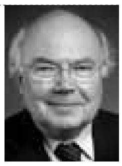
JOHANNES WINTERHAGEN, Editor-in-Chief
Stuttgart, 19 October 2010

THE BIOLIQ PROCESS CONCEPT, TECHNOLOGY AND STATE OF DEVELOPMENT

Synthetic fuels from residual biomass may contribute considerably to the global motor fuels demand. Gasoline and diesel of high quality can be produced by that way. For large scale use of biomass logistical and technological obstacles are to be considered, resulting in bioliq process currently developed at the Karlsruher Institute for Technology (KIT). A de-central pre-treatment of biomass for energy densification by fast pyrolysis allows for the economic supply of centralized industrial facilities in large scale, in which synthesis gas is produced and further converted to the desired synthetic fuels.



AUTHORS



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BACKGROUND

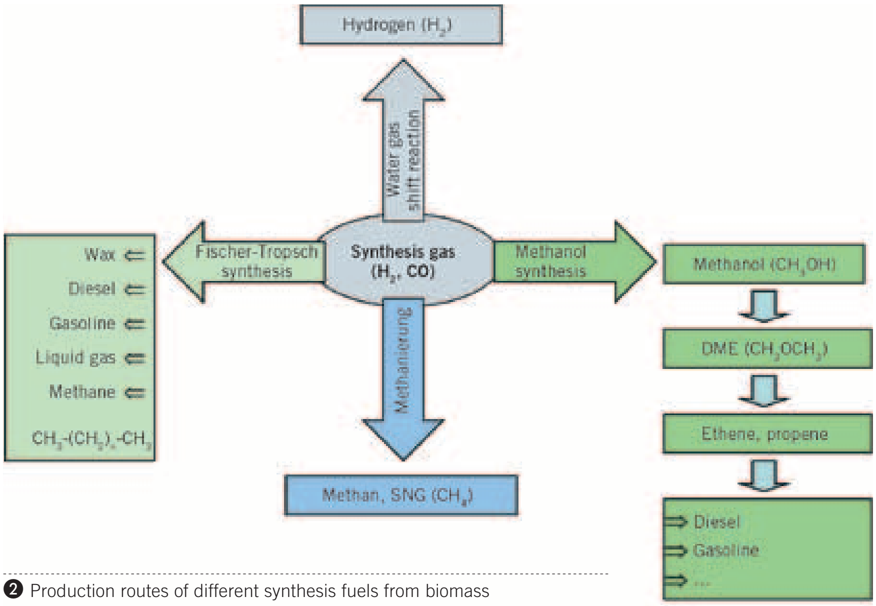
Fossil energy sources, in particular crude oil, are the basis of today's supply of fuels. Even if the forecasts about their date of depletion differ greatly, there is no doubt about their long-term shortage. The widely varying and increasing energy demand of emerging states such as India and China accelerates this effect. In addition to that, apart from the finite nature of fossil resources, problems with securing energy supplies, costs of exploitation and transport and the demand for environmentally compatible handling of resources play a major role. As shown by the recent developments of the world market prices for crude oil, even small disruptions already suffice for causing massive variations in price on a global scale with serious consequences for the world economy.

The consistent use of renewable energy sources is a way of reducing the dependence on fossil raw materials (crude oil, natural gas and coal) in particular in the industrialized countries. Moreover, the use of renewable energy sources can make a considerable contribution to reducing CO_2 emissions and thus diminishing the anthropogenic greenhouse effect. Among the renewable energies, biomass is the only renewable carbon source and should therefore be used in the long run as raw material for producing carbon-containing products and energy sources. Even if the proportion of biomass in the primary energy consumption has exceeded the 10 % mark in Germany in 2009, presently it is used predominantly for the generation of heat and power (combined 80 %). Biofuels, in particular biodiesel, make up 20 % of the energetic use of biomass. The contribution of biomass to mobility is being discussed controversially in the public and in professional circles. However, without any doubt, liquid fuels with their so far unattained high energy densities will continue to make a significant contribution to passenger and cargo transport for a long time to come. Up to now, the focus has been on bioethanol and biodiesel from sugar-containing crops and oil seeds as first generation biofuels, along with critical questions about their effects on the food and feed market and worldwide applicable and sustainable plant growing standards.

Second generation biofuels rely on residues and side products from agriculture and forestry, which are available in large quanti-

<table>
<tr><th>HYDROCARBONS</th><th>MTG</th><th>FT (CO-CAT.)</th><th>FT (FE-CAT.)</th></tr>
<tr><td>LIGHT GAS</td><td>1.4</td><td>5</td><td>8</td></tr>
<tr><td>ETHANE, ETHENE</td><td>5.5</td><td>0</td><td>4</td></tr>
<tr><td>PROPANE, PROPENE</td><td>0.2</td><td>3</td><td>13</td></tr>
<tr><td>ISOBUTANE</td><td>8.6</td><td rowspan="2">1</td><td rowspan="2">1</td></tr>
<tr><td>N-BUTANE</td><td>3.3</td></tr>
<tr><td>BUTENES</td><td>1.1</td><td>2</td><td>9</td></tr>
<tr><td>GASOLINE FRACTION (C_{5+})</td><td>79.9</td><td>19</td><td>36</td></tr>
<tr><td>GASOIL, MEDIUM DISTILLATE</td><td>–</td><td>22</td><td>16</td></tr>
<tr><td>HEAVY OIL, WAXES</td><td>–</td><td>46</td><td>5</td></tr>
<tr><td>OXYGENATES</td><td>No data available</td><td>1</td><td>5</td></tr>
</table>

❶ Hydrocarbon compositions of the MTG process and of products from Fischer-Tropsch synthesis obtained from cobalt- or iron-catalysts



❷ Production routes of different synthesis fuels from biomass

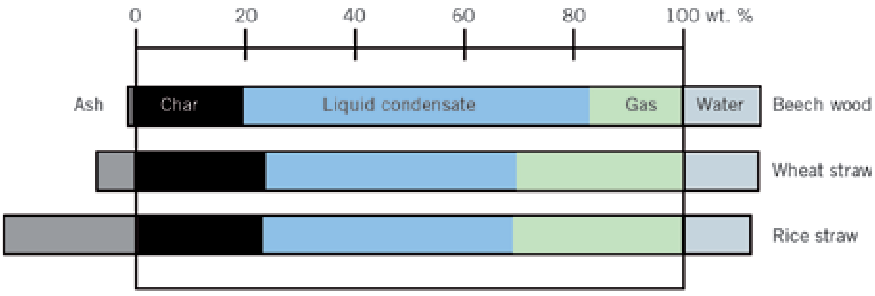
ties. These synthetic fuels include hydrogen, methane, (SNG – Substitute Natural Gas), ethanol from lignocellulose and the wide range of BTL fuels (Biomass to Liquid) generated from synthesis gas. The latter comprise methanol, ethanol, dimethyl ether and BTL diesel and gasoline. The route whereby such fuels are generated from fossil raw materials, predominantly from coal and natural gas, has been basically known for a long time and already introduced as Gas to Liquid (GTL) or Coal to Liquid (CTL) processes proceeding via synthesis gas as intermediate. Coal or natural gas is first reacted with steam and oxygen to give a crude synthesis gas, a mixture of hydrogen and carbon monoxide. The crude synthesis gas is freed from particles, CO_2, HCl and trace elements, which would interfere in the subsequent synthesis, in which the syngas is reacted catalytically at high pressures and temperatures to the desired product. XTL fuels can be very similar to conventional fuels or even better in terms of their combustion and emission behavior. They can be used directly using the presently available distribution infrastructure, require no new drive technology and allow a similar radius of action as petroleum-based fuels.

Different catalytic process lead from the synthesis gas either directly to hydrocarbons. In the Fischer-Tropsch process hydrocarbons of different chain length are produced, which have to be separated, depending on the desired product range. Another route proceeds via methanol to give dimethyl ether or olefins (Methanol to Olefins – MTO), and finally to gasoline (Methanol to Gasoline – MTG) or diesel fuels (Methanol to Synfuel – MTS), ❶ and ❷. This is how Sasol produces approximately 6 million t/a in the worldwide largest Fischer-Tropsch plants, which corresponds to about one third of the fuel consumption in South Africa. Natural gas is processed on a large industrial scale in Mossel Bay, South Africa, and in Bintulu, Malaysia, at an annual capacity of approximately 2 (South Africa) and 1 (Malaysia) million t, respectively, to give synthetic fuels. In addition to fuels, the production and further processing of methanol also allow a large number of important key chemicals to be prepared. Currently, a whole series of methanol plants based on natural gas and coal are built in particular in the Middle East and in China, which already today is the main consumer and producer of this substance. If the world's annual production in 2008 was 45 million t, the production capacity in 2010 is estimated at 85 million t.

THE BIOLIQ PROCESS

The use of biogenic starting materials is a particular challenge for the production of synthesis gas products, for which an adequate technology still has to be developed. The collective term biomass comprises a wide range of different materials, most of which have low volumetric energy density and are distributed over a wide space. On the other hand, in order to enable economical operation, the complex technology of producing synthetic fuels requires large-scale production plants. The Karlsruhe Bioliq process allows decentralized pre-treatment of biomass in regionally distributed plants. The high-energy intermediate biosyncrude can be transported economically even over large distances and subjected to further processing in the required large-scale plants. The process comprises several process steps, which are currently set up at the Karlsruhe Institute for Technology (KIT) in the form of a pilot plant:

The pre-treatment of biomass is performed by a so-called fast pyrolysis. The finely comminuted feedstock is heated in the absence of air together with hot sand as heat transfer agent in a twin-screw reactor to 500 °C within seconds. A large



❸ Product yields of fast pyrolysis using different feedstocks portion of the vapors can be condensed to give brownish pyrolysis oil that has a strong smell of smoke aromas. The rest is made up of a flammable gas, which can be used for reheating the circulated sand. Their proportions in the product differ, depending on the biomass used, ❸. In addition to approximately 20 % of coke, 50 to 60 % of pyrolysis oil and 20 to 30 % of gas on a water- and ash-free basis are formed, which contain about 10 % of the heating value. Coke and pyrolysis oil are admixed to the biosyncrude, which contains about 85 % of the energy originally contained in the biomass, but has only less than a tenth of its original volume and an energy density that is comparable to that of lignite. This intermediate is stable on storage and transport and constitutes a fuel that is highly suitable for the next process step: In entrained flow gasification, the biosyncrude is reacted with oxygen at more than 1200 °C to give a tar-free synthesis gas low in methane, as required for the subsequent chemical syntheses. As these processes take place under high pressures of between 30 and 80 bar, the Bioliq process also performes gasification under high pressure in order to avoid expensive compression of the synthesis gas. The Bioliq entrained flow gasifier is designed for an operating pressure of 80 bar at a thermal fuel capacity of 5 MW. To account for the high ash contents of the biomasses to be used, the pilot gasifier is equipped with a cooling screen with a refractory material. By maintaining the gasifier at a temperature optimized to the slag melt properties of the biomass, a firmly adhering slag coat is applied to the refractory cladding, protecting the material from abrasion and the reactor from corrosion. The draining slag melt is discharged, after passing through a water quench, via a slag sluice as a solid material. Apart from being highly compatible with fuels of high ash content, the cooling screen also results in a high reactor life and allows quick start-up and discharge, thus ensuring safe operation. The crude synthesis gas is purified and conditioned. For this, the KIT pilot plant uses an alternative hot gas purification method, which, compared with the conventional method used on a large industrial scale, has better energy efficiency and lower investment costs, in particular with smaller plant sizes. This is based on

❹ Process building of the pyrolysis pilot plant at the KIT

the fact that the order of magnitude of, for example, a mineral oil refinery having a production capacity of 10 million t/a cannot be achieved using biomass as raw material, but is lower by a factor of about 10 and thus would be a priori less economical. To compensate this scale effect, an efficient technology would be, among other things, is requied.

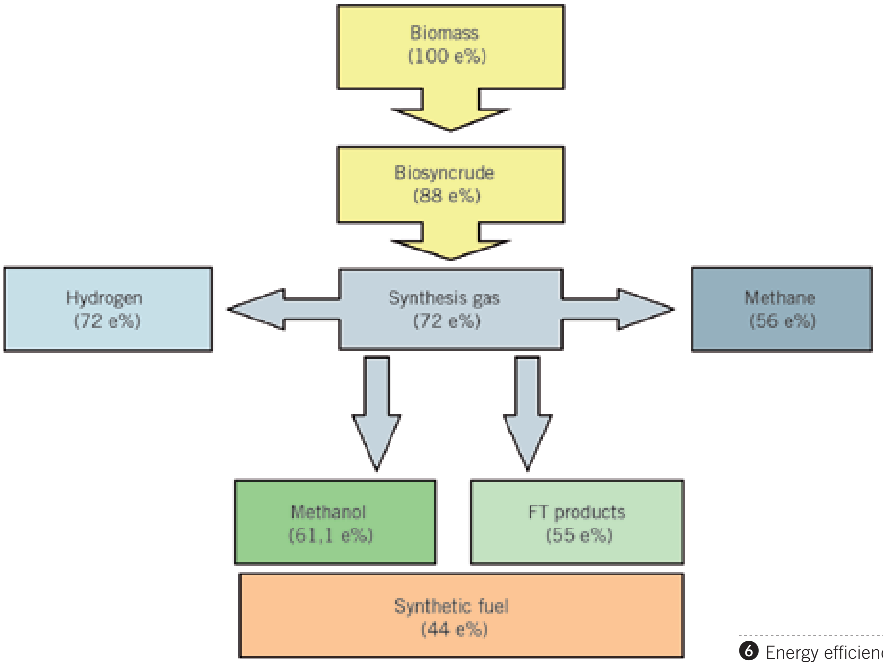
The gas purification unit designed for 80 bar removes a partial flow of 700 Nm³/h (2 MW_{th}) and precipitates particles by means of ceramic filter candles, removes acid gas components and alkalis by means of inorganic adsorbents and catalytically converts NH_3, HCN and organic components in a final catalyst stage at a continuous temperature level of initially 500 °C, which can be varied later on, in order to optimize the heat balance of the process. In the synthesis stage, the pre-purified synthesis gas is freed of CO_2 in a conventional solvent wash and then converted directly to DME in a single-step synthesis. In preliminary work on the process, it has been shown that no adjustment of the hydrogen/carbon monoxide ratio (for example one with biomass as feedstock, while for methanol synthesis, it would have to be set to 2) via a separate water gas shift reaction ($H_2O + CO => H_2 + CO_2$) is required. The next step is a zeolite-catalyzed dehydration of the DME with oligomerization and isomerization of the hydrocarbons used. Whereas the fuel synthesis proceeds with practically quantitative conversion, only about half of the synthesis gas is converted in the DME stage, the other half being fed back to the DME synthesis.

NEW PLANTS

In accordance with the process stages, the KIT pilot plant is being set up in several phases. Starting in 2005, first the pilot plant for fast pyrolysis was set up and put into operation in 2008, ❹. In the same year, the gasification stage was started, which is still in the setup stage. Both plants were set up and operated in cooperation with Lurgi, Frankfurt. 2009 saw the kick-off of the planning of the gas purification (MUT Advanced Heating, Jena) and synthesis stages (Chemieanlagen Chemnitz CAC, Chemnitz). Plant construction takes place in parallel to the gasification stage, so that the simultaneous completion of all plants in construction is expected for the end of 2011. The project for building the pilot plant is funded by BMELV and the agency of renewable resources FNR, Gülzow.

	STAGE 1	STAGE 2	STAGE 3	STAGE 4	STAGE 5
PROCESS	Fast pyrolysis	Entrained flow gasification	Gas purification	DME synthesis	Gasoline synthesis
PRESSURE [BAR]	–	80	80	55	50
TEMPERATURE [°C]	500	> 1200	500	250	300
THROUGHPUT	500 kg/h of biomass (2 MW_{th})	1000 kg of biosyncrude (5 MW_{th})	700 Nm^3 (2 MW_{th})	50 kg/h	30 kg/h
PRODUCT	Biosyncrude	Crude synthesis gas	Pure synthesis gas	DME	Gasoline

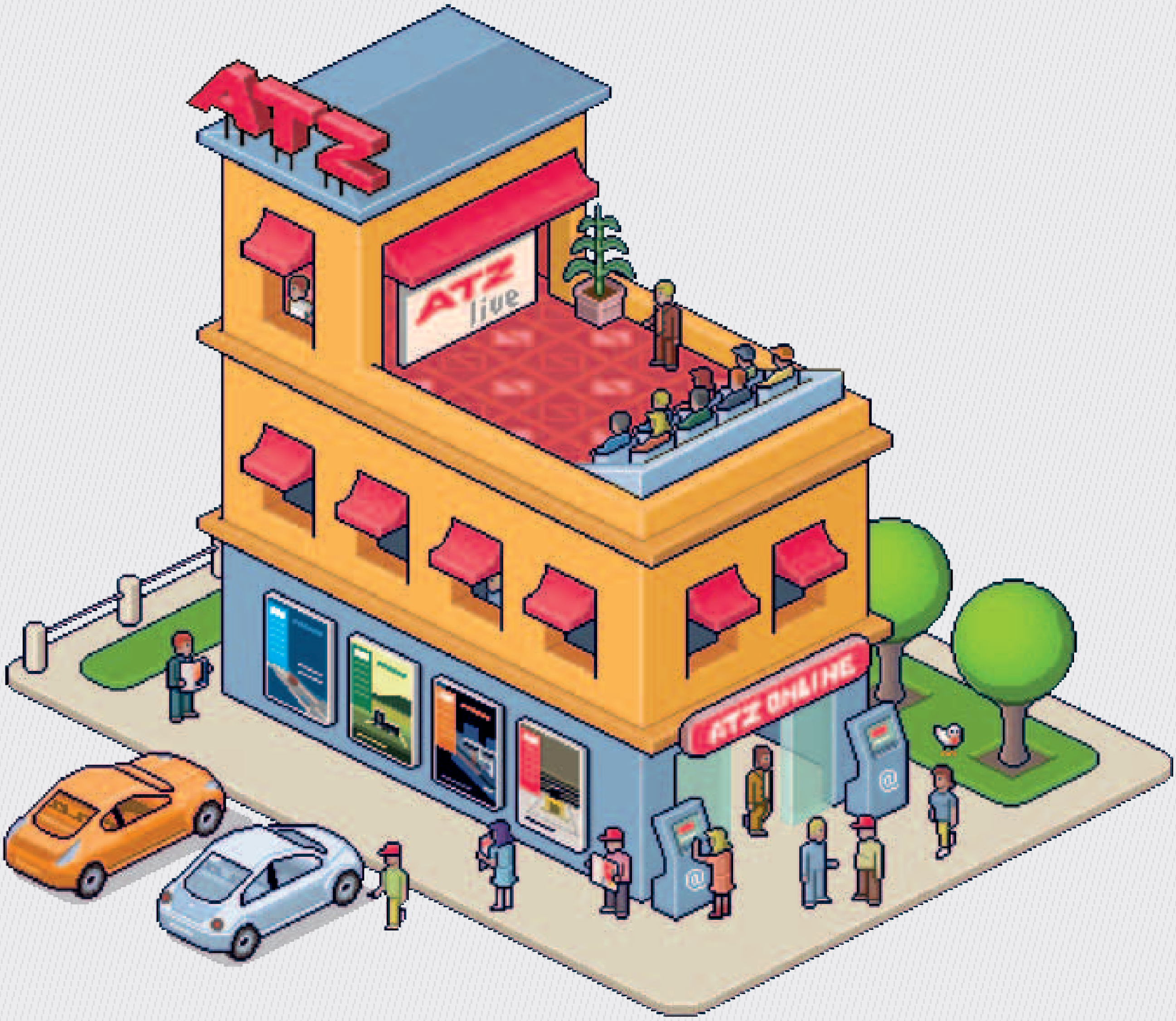
❺ Characteristics of the Bioliq pilot plant



❻ Energy efficiencies of different synthetic products from biomass

A few data on the KIT pilot plant are summarized in ❺. Approximately 40 % of the energy originally contained in the biomass are expected to show up again in the fuel ❻. Depending on the process, additional products such as liquid gas or chemicals will be formed. An important aspect is that heat and power are formed as byproducts, which are used to cover a large portion of the energy demand of the process. This results in a high CO_2 reduction potential of BTL fuels. However, ⑥ also shows that the generation of pure hydrocarbons from biomass gives the lowest energy efficiencies (relative to the energy content of the products, compared with the starting material). Hydrogen preserves the highest energy content, whereas further processing of the synthesis gas by exothermic reactions leads to a lower energy content. Moreover, biomass with its averaged empirical formula of $C_5H_9O_4$ has a high oxygen content, compared with fossil raw materials. When pure hydrocarbons are prepared, the oxygen is cleaved off in the form of water and carbon dioxide. This leads to a hydrogen deficiency within the process and a lowering of the carbon efficiency, which, however, is undesirable in terms of using the carbon contained in the biomass. A more advantageous method for preserving energy and carbon efficiencies is to prepare oxygen-containing products that allow a large portion of the oxygen present in biomass to be preserved in the product. In that respect, biomass is suitable in the short term for providing synthetic oxygenates (anti-knock agents), which find a useful application in blends with mineral fuels. Oxygen-containing pure fuels are also a longer-term option without restrictions or drawbacks whatsoever in driving applications and with respect to their combustion and emission behavior. On the other hand, their lower energy density compared with pure hydrocarbons leads to a shorter range at the same the tank volume. However, pure synthetic hydrocarbons are also useful as high-quality fuels if grades can be produced that meet the requirements of the engine combustion concepts. This is the focus of current research and development activities at the KIT in chemical catalysis, whose successful development concepts will then also be tested and put into practice in the Bioliq pilot plant.

ATZ. A VARIETY IN AUTOMOTIVE KNOWLEDGE.



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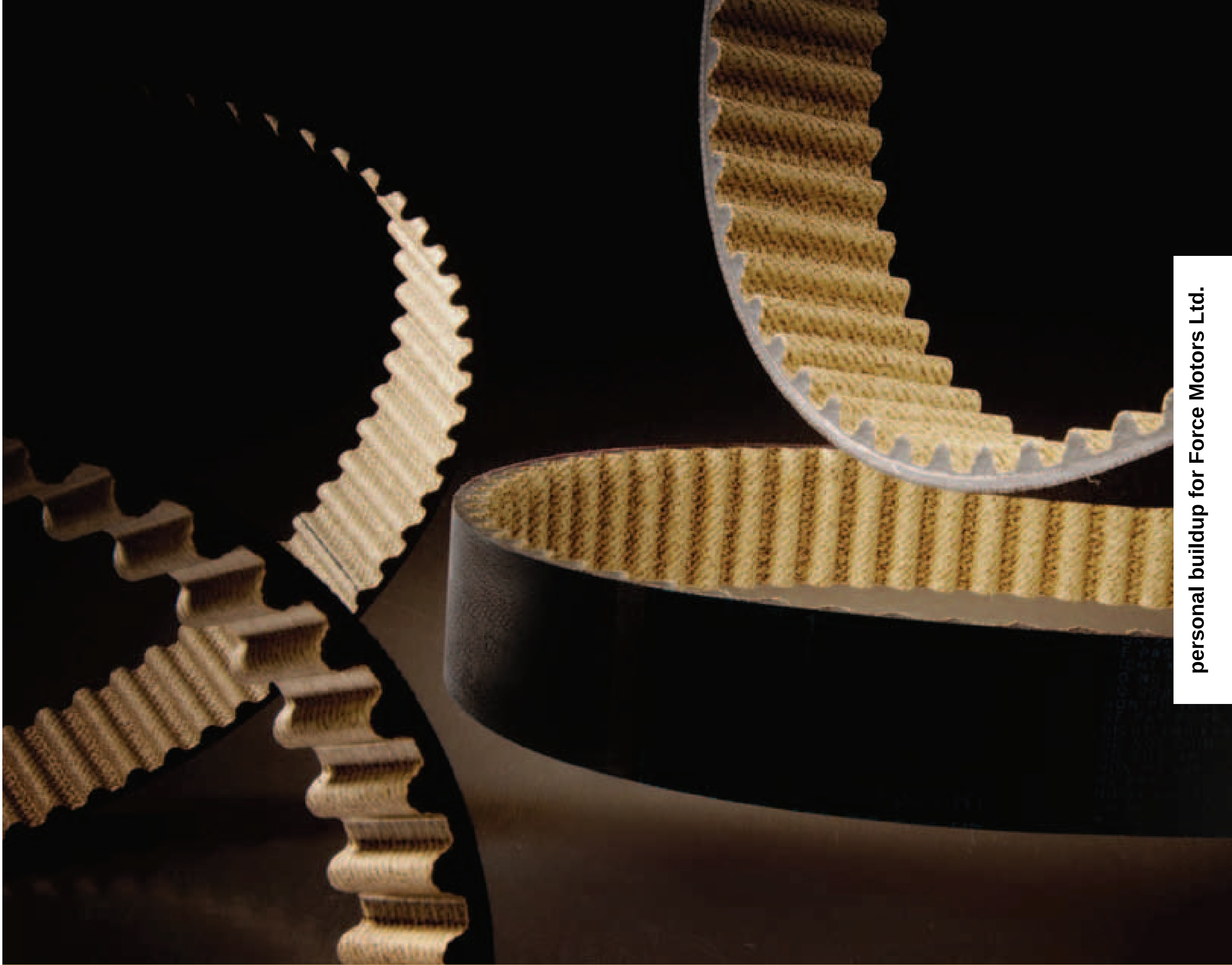

TIMING BELT IN OIL BATH
FOR OIL PUMP DRIVE

Volkswagen and Dayco have worked in partnership to develop a belt drive operating in oil inside a diesel engine to replace current chain technology for the oil pump drive. It is engineered as a life time belt system. This new Dayco technology offers benefits of lower friction, improved NVH and reduced weight.

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GROWING DEMANDS

Developments on engines with the targets of engine downsizing, increase of specific power, improving engine efficiency, reduce friction losses, optimise vehicle mass, and cost-saving strategies are key factors for all carmakers. In the field of combustion engines, Dayco Power Transmission is an active participant in these efforts. One innovation is the development of a timing belt that is working without tensioner directly in engine oil, driving the oil pump. The power saving in comparison with the original chain drive showed approximately 50 W [2].

The following article will show the principal developments on the 1.6 litre TDI common rail engine. This engine was presented at the 30th International Vienna Motor Symposium, as the basis for all future four-cylinder diesel engines for the Volkswagen Group [2]. Dayco also delivers the camshaft belt of this engine.

OIL PUMP DRIVE – SYSTEM DESCRIPTION

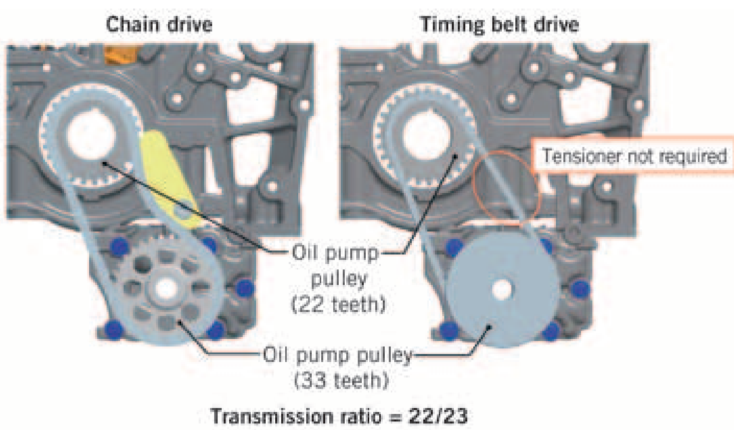
Oil pumps in combustion engines are currently mainly driven by chains, using layouts with fixed centre distance. A well known assembly is reported in ❶ (left), where the typical components (chain sprockets, chain and its tensioner) are detailed. ① (right) shows the alternative timing belt drive, fitted in the same fixed centre distance, providing the same function, tensioner-less system.

The object to be “alternative to chain” had to face the following requirements/constraints:

: no need of drive redesign
: no change in assembly line
: improve/guarantee the system performance
: meet lifetime target
: achieve a cost advantage.

Starting from the original chain drive and from the surrounding geometrical constraints, a similar design was followed for the belt system. The driver pulley is a sintered part, directly press-fitted on the crank-shaft and without flanges.

The driven pulley, ❷, is also a sintered part, directly press-fitted on the pump shaft and with two flanges. The pictures beside show also that this pulley is partially submerged in the oil. A Tensioner device is no longer needed.



❶ Comparison between chain system and timing belt

DRIVE LAYOUT – GEOMETRICAL FEASIBILITY ANALYSIS

The oil pump is held to the lower part of the engine block via fixing screws. Pump positioning is guaranteed via reference bushings. First the axial tolerances on the cam shaft and on the oil pump assembly were analysed. In this case the belt width was selected to be 9.4 mm. Samples of parts were manufactured in order to check (by fitting and by testing) the extreme ranges of the dimensions defined on the drawing.

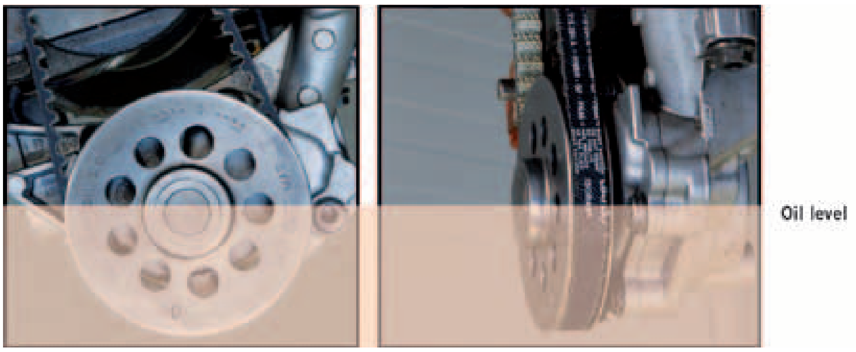
Then the radial tolerance were analysed and the following parameters were considered:

: fixed centre distance of the pump
: pulley diameters and their tolerances
: belt length and elasticity
: drive thermal expansion from -40 up to 150 °C
: the setting of belt/drive after initial running of the engine
: belt dimensions at the end of the life.

By running tolerance analysis, a nominal belt-length was found. This procedure had been previously adopted during a co-development with Audi to drive the water-pump on the rear side of the engine, ❸. Also in this case (lifetime installation) there is no tensioner device.

③ shows how the basic parameters, the delta between drive and belt length are used to take under control the static tension. Finally the belt is designed as a self-tensioning-system, working in cold and/or warm conditions with a fixed drive centre distance. Robustness of this solution is guaranteed by the following facts:

: the tightening of belt tolerances achieved by optimized production process
: belt elongation at end of life that ranges today close to 0.03 %
: the belt can slide axially, recovering crankshaft axial oscillation and thermal expansion
: resistance to contaminants: motor-oils and fuels, with various oil/fuel dilutions and various acidification levels, were used to validate this Dayco structure.

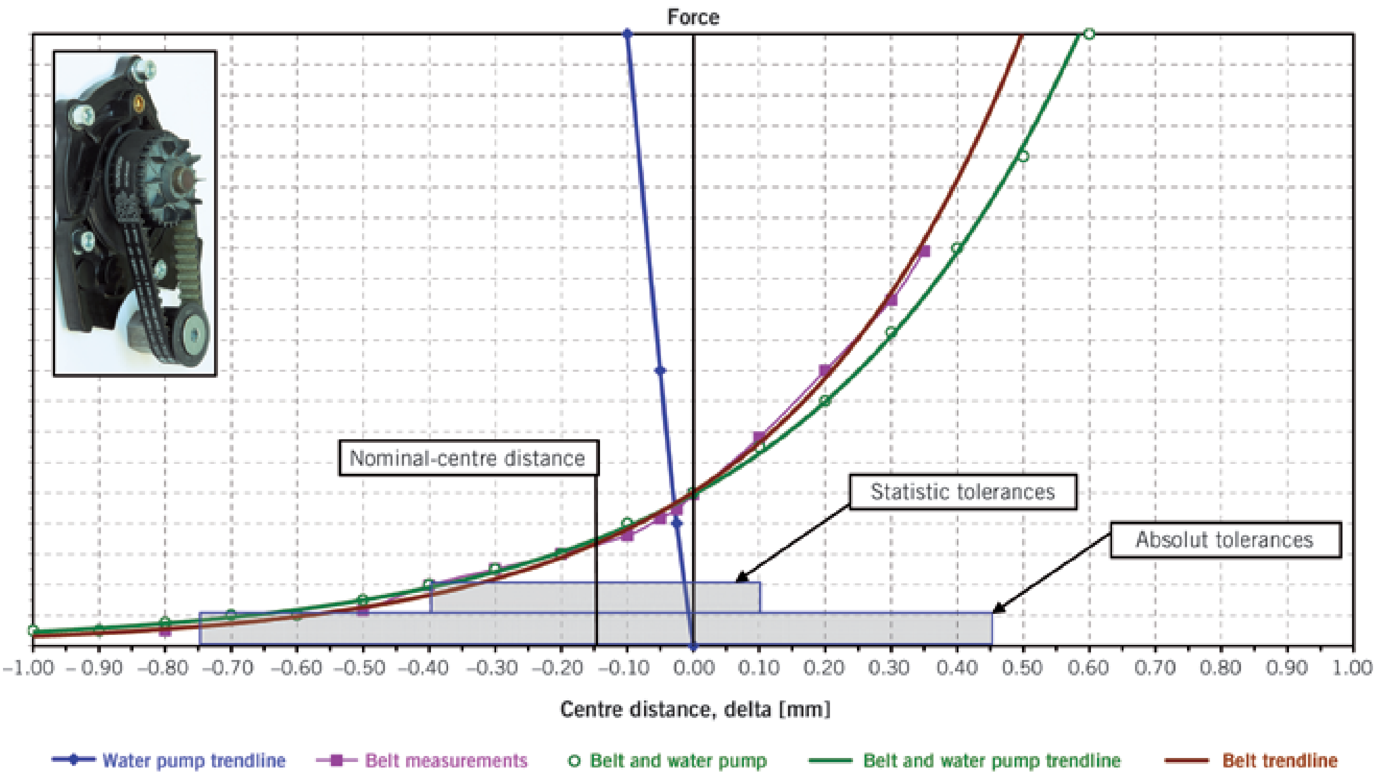


❷ Timing belt system for oil pump; front view and lateral view

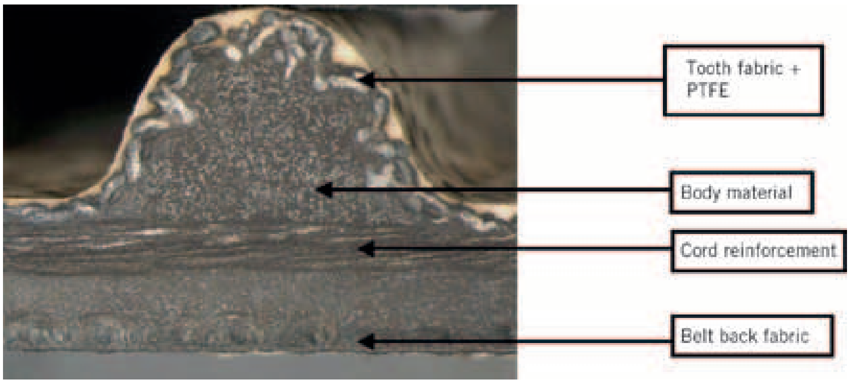
TIMING BELT STRUCTURE

A toothed belt comprises elements as described in the following, ❹:

: the tooth fabric is a polyamide with PFTE skin; the white treatment is the basis of the belt technology from Dayco,



❸ Influence of centre distance (tolerance analyses on water pump drive)



❹ Belt structure description

rated by many car-manufacturers as a benchmark for heat and wear resistance
- body material is an special grade of HNBR, fibre-reinforced
- cord reinforcement by a high modulus glass cord
- belt back fabric made out of polyamide, to protect against edge wear and to enhance the ageing resistance of the structure.

FEASIBILITY ANALYSIS

A timing belt was historically designed to work in dry-air ambient. In case of ingress of liquid or contaminants, a given requirement of function and life must be guaranteed, beside the standard approval process. With the "belt in oil project", Dayco targets were extended: the belt had always to run in a chemically aggressive environment "lifetime and worldwide". The experience from the sealing technology could not be simply carried over, while timing belts are submitted to dynamic stresses.

The feasibility phase was extremely complex for Dayco engineering. A large variety of inputs and boundary conditions were considered, like more than 100 different motor oils, diluted with fuels, water, acids, cleaning liquids at different temperature level, under different status of ageing and so on.

To define the most realistic working conditions was a key point: thanks to cooperation with car manufacturers and motor oil suppliers, it was possible to come, step by step, to a structured matrix of tests, able to cover even some border-line cases (like acidified oil able to attack oil-pan wall).

Finally, to validate the performance in wet ambient, a focused testing procedure was necessary: bench and laboratory tests had to reproduce the real contamination and to focus on the essential belt parameters.

EXAMPLES OF MATERIAL VALIDATION

By storing samples in oil/fuel it's possible to map the damage for that material. By combining time and temperature of exposure, the material gets a "damage factor" for the parameter under analysis, ❺. Controlled parameters are:
- elongation/tensile strength/modulus
- dimensional variation/delta weight
- adhesion/cord pull-out/hardness
- tooth stiffness stability, dynamic and static.

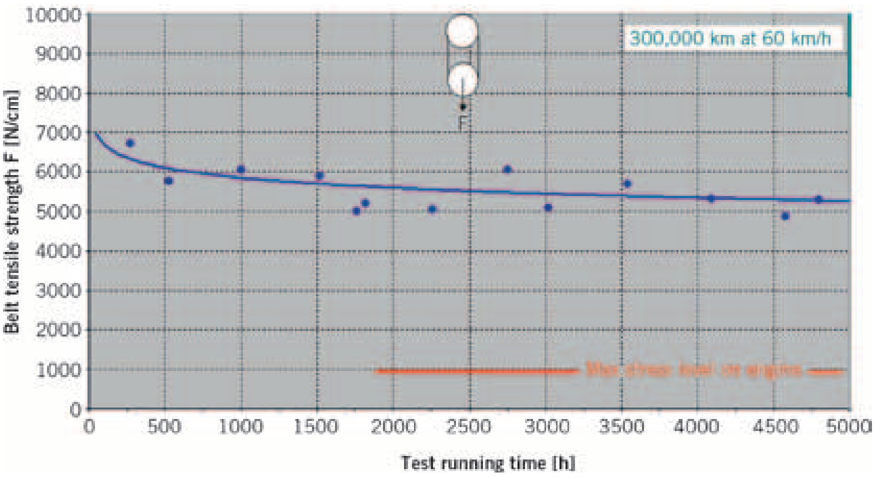
The engine producer has the information about different thermal and duty cycles, for different drivers. By studying this data set, it is possible to calculate the number of hours corresponding to a lifetime target. Finally, an equivalent duty cycle can be defined, shortening test time by increasing oil temperature. The test engineers used this data, to calculate cycles for test benches, to be able to reproduce field cases. This example Dayco specific bench-test was able to simulate with 800 h at 140 °C a worst-case of belt ageing.

TEST PROGRAM FOR OIL PUMP DRIVE

The customer standard validation program was completed successfully. Specific tests were designed, to add to the mechanical stress also the "fatigue" from engine oil. Particular attention was paid to the stability of the belt parameters under exposure

	DAMAGE MAP				
HOURS/TEMPERATURE	90 °C	100 °C	110 °C	120 °C	130 °C
0	0.00	0.00	0.00	0.00	0.00
1000	0.20	0.22	0.26	0.33	0.43
1500	0.27	0.35	0.51	0.67	1.00
2000	0.40	0.52	0.75	1.00	
3000	0.53	0.69	1.00		
4000	0.76	1.00			
5000	1.00				

❺ Damage map example for material parameter



❻ Residual tensile strength extracted from test-bench and test-car (60 km/h speed)

PARAMETER	CHAIN DRIVE	BELT DRIVE
Friction loss		up to 50 W saving
Number of components and assembly	five (two sprockets + one chain + one tensioner + one screw) one screwed hole in the engine wall, one screwing phase	three (two pulleys + one belt)
Acoustic		better
Weight [%]	100	80 to 90
Temperature range		from -40 up to 150 °C

❼ Comparison of oil pump drive by chain and belt

to oil and contaminants. An example of these specific tests is here given:

- firing engine cycles, able to self-generate the highest fuel's content
- firing engine cycles, able to self-generate the highest temperature in the oil
- rig tests, with aged oils artificially acidified, running up to 150 °C
- rig/engine tests, to simulate other fluids in the oil, for example cooling or cleaning liquids.

Under these special conditions, the monitoring of the main belt parameters was performed. This results finally in building trend-lines, as reported in ❻, referring to belt tensile strength. Also the safety limits for this oil pump-drive are shown.

COMPARISON CHAIN VERSUS BELT

As described at the Vienna Motor Symposium, on the 1.6 litre common rail engine the amount of power required to drive the oil pump is reduced by about 50 W when a belt-drive is adopted [2]. Experiments show in general that absolute power loss is dependent on surrounding conditions: mainly at low engine-oil temperature a belt drive has a higher benefit. Measurements and modelling show that the power loss is mainly coming from reduced tensioning and sliding blade friction. On this aspect, the belt can allow longer free-span without any guiding contact, and this contributes to lower power dissipation. Overall, experience suggests that the primary reason to move to a timing belt for an oil pump drive is mainly linked to the comparison in ❼.

CONCLUSIONS

The technology of timing belt working in oil ambient allows Dayco to work on different applications on combustion engines and to be an economic alternative solution to the chain drive systems.

Dayco is currently the only supplier of this product for combustion engines. Stability of materials against oil and other contaminants was achieved by a tailored selection of belt components and their treatments and checked on a testing plan focusing on lifetime targets.

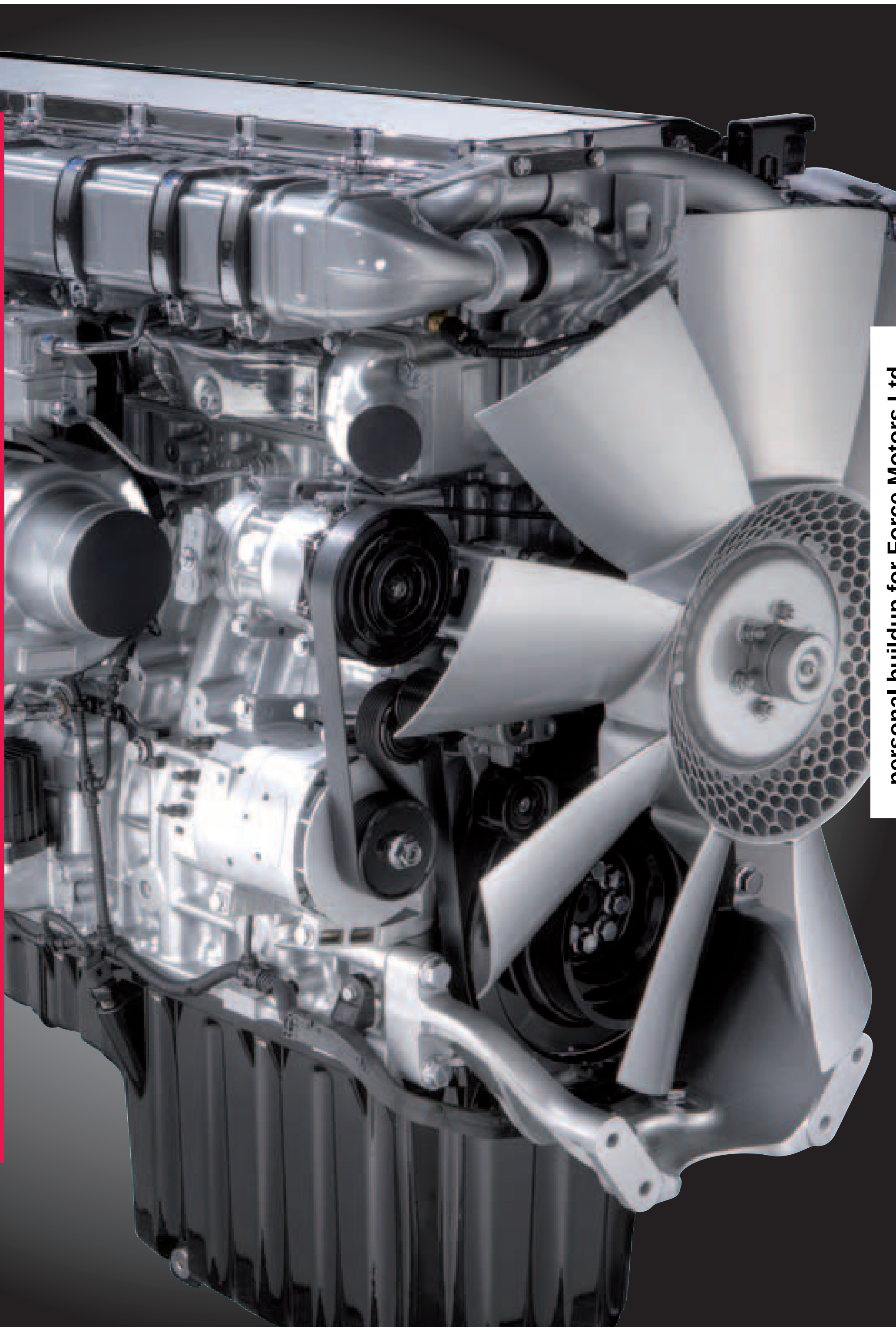
Work in co-development with car makers and oil suppliers, allowed the definition of realistic working conditions when chemicals in the surrounding have to be added to the traditional mechanical and thermal stress. Furthermore Dayco showed how to apply this belt to an oil-pump drive, allowing to the designer to avoid tensioning devices in layouts with fixed center-distance. Advantages of lower friction, better acoustic and lighter drives are offered by this technology. The life time target and the worldwide reliability are given with the new materials. Further applications of timing belts running in oil, are soon expected to come in series production.

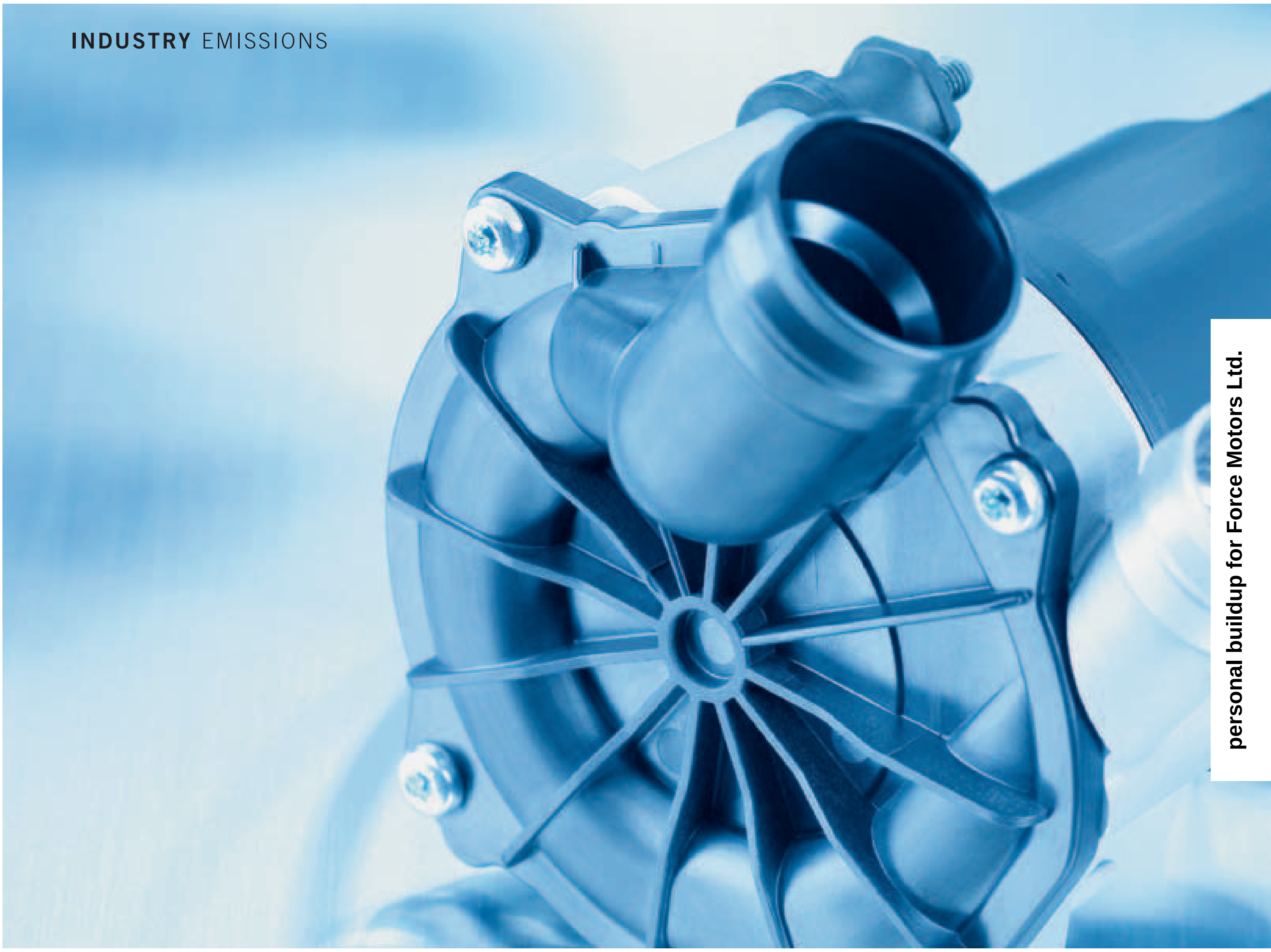
REFERENCES

[1] Hahne, B.; Neuendorf, S.; Paher, G.; Vollmers, E.: Neue Dieselmotoren für Volkswagen-Nutzfahrzeug-Anwendungen. In: MTZ 71 (2010), Nr. 1, S. 26-32

[2] Rudolph, F.; Hadler, J.; Engler, J.; Krause, A.; Stamm, M.: The new 1.6 l TDI engine from Volkswagen. 30th International Vienna Motor Symposium, 2009

[3] Tiemann, C.; Dohmen, J.; Steffens, C.; Wedowski, S.; Walter, R.; Schulte, H.; Di Giacomo, T.: Belt vs. Chain – Study on the Potential for CO_2 Saving by the Timing Drive. Aachener Kolloquium Fahrzeug und Motorentechnik, 2008





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GROWING SIGNIFICANCE

Secondary air injection is a key component in addressing the extensive requirements of low-consumption gasoline engine strategies with regard to low cold-start emissions, system robustness and durability. Even now it is a known variable and firm content of many applications. In order to achieve the mentioned requirements, however, it will in future grow in significance. The importance of secondary air injection is also emphasized by the SULEV/PZEV component of a 15-year endurance limit requirement for emission systems. This gives rise to further tough challenges with regard to component longevity and diagnosability. Market drivers and legislative requirements result, moreover, in a growing demand for a reduction in CO_2 emissions for internal combustion engines. In the wake of these developments, consumption-reduction strategies, such as downsizing, are of growing market significance and at the same time suggest the suitability of secondary air injection.

MODE OF OPERATION

Secondary air injection is primarily an instrument for reducing emissions following cold starts and for accelerating the heat-up of the catalytic converter on gasoline engines. To this end, during rich engine operation air is injected for a brief phase (around 20 to 60 s) directly into the exhaust gas system. As a consequence a lean mixture develops in the exhaust gas stream. The key concept behind secondary air injection is to ignite the unburnt fuel contained in the exhaust (exhaust gas ignition or EGI).

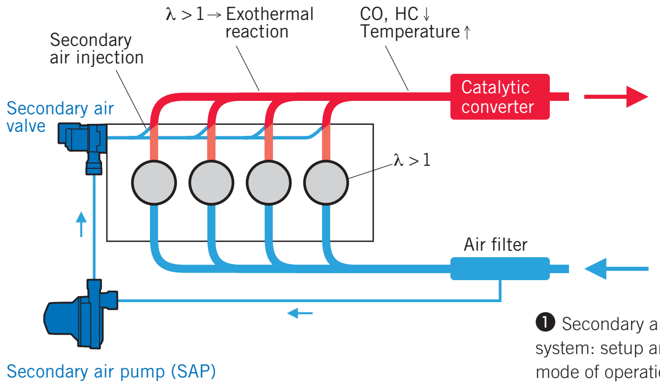
SECONDARY AIR INJECTION
A COMPONENT OF LOW-CONSUMPTION, LOW-EMISSION STRATEGIES

The new emission stages, Euro 6 in Europe and SULEV/PZEV in the USA, pose severe challenges on the quality of a car engine's combustion and exhaust aftertreatment systems. Since most of today's systems achieve very good emission control results under hot operating conditions, the tougher challenges are presented by the reduction of cold-start emissions and the rapid heating of the exhaust aftertreatment system. The following article by Pierburg and Audi shows how secondary air assists in achieving these goals besides examining other aspects of the secondary air process.

During cold start, the engine runs on a rich mixture resulting in smooth, stable and zero-misfire operation at relatively low idling speed. An accompanying phenomenon of this rich mixture is the emission of incompletely combusted carbon monoxides (CO) and hydrocarbons (HC). But without secondary air injection, these would be discharged into the environment. With the aid of secondary air injection, most of the constituents are oxidized in the exhaust gas manifold upstream of the catalytic converter in an exothermal reaction; hence most of the CO and HC emissions are significantly reduced and, on the other hand, the inflow temperature in the catalytic converter rises allowing it to more quickly achieve the operating temperature, ❶.

Despite the higher concentration of CO and HC in the combustion chamber under rich operating conditions, the emissions, as a consequence of aftertreatment with secondary air in the exhaust gas manifold, are lower than with leaner combustion without secondary air injection [3]. [2] shows a comparison of two versions of an engine indicating that secondary air injection allows a catalytic converter heating strategy which achieves a 80 °C higher converter inflow temperature 20 s after test start. The cumulative HC emissions at 45 s following test start on the version with secondary air is 60 % lower than on the version without secondary air injection, ❷. The rapid heating of the catalytic converter allows an early transition to controlled catalytic converter operation. The outcome is a significant reduction in NO_x emissions and this is not directly attributable to the exothermal reaction of secondary air injection but a consequence of the rapid heating of the catalytic converter. Most of the NO_x emission reduction occurs at a time in which the secondary air injection process has already been deactivated, ②.

Typical engine air/fuel ratios during the secondary air phase are lambda 0.7 to 0.9. Commonly in most applications, secondary air is continuously injected. A pulse injection rate synchronized with the engine charge exchange does not yield any advantages [3]. The air flow rate of the secondary air/fuel pump is not controlled as a factor of the engine operating point. The air/fuel ratio in the exhaust is the result of uncontrolled air injection and the air ratio within the engine. The exhaust air ratio is not constant regarding time and place. The mean is lambda 1.2 to 1.4. In most applications, air is injected in all exhaust ports. The reactivity of the exhaust gas is higher



❶ Secondary air system: setup and mode of operation

the closer secondary air injection is located to the exhaust valve.

Alongside the already-mentioned primary advantages of secondary air injection, in the past the sole focal point, nowadays new aspects have taken front seat, ❸. Secondary air injection is increasingly being taken into consideration in the realization of low-consumption engine strategies. Also, since the emission advantages obtained through secondary air are not subject to any obsolescence process and given the excellent ruggedness of the system as such, this technology is proving attractive for further applications. The CO_2 advantage obtained through secondary air does not result from the operating phases of the secondary air system itself. The advantage is indirect given that the use of a secondary air system allows the implementation of engine concepts and catalytic converter layouts which, during the entire service life of the engine, deliver fuel savings.

One example is the location of the catalytic converter. In order to achieve a rapid rate of heating, this is frequently positioned close to the engine. This gives rise to the need for converter cooling through mixture enrichment which, in turn, spells disadvantages in terms of fuel consumption. Particularly on the low-volume engines increasingly entering the market and frequently operating at high loads, this can impact on fuel consumption. The potential of additional exothermal effects through secondary air injection can be exploited in order to heat rapidly enough a catalytic converter mounted at some distance from the engine. Such an arrangement reduces with the need for catalytic converter protection through mixture enrichment which, in turn, gives rise to higher fuel consumption.

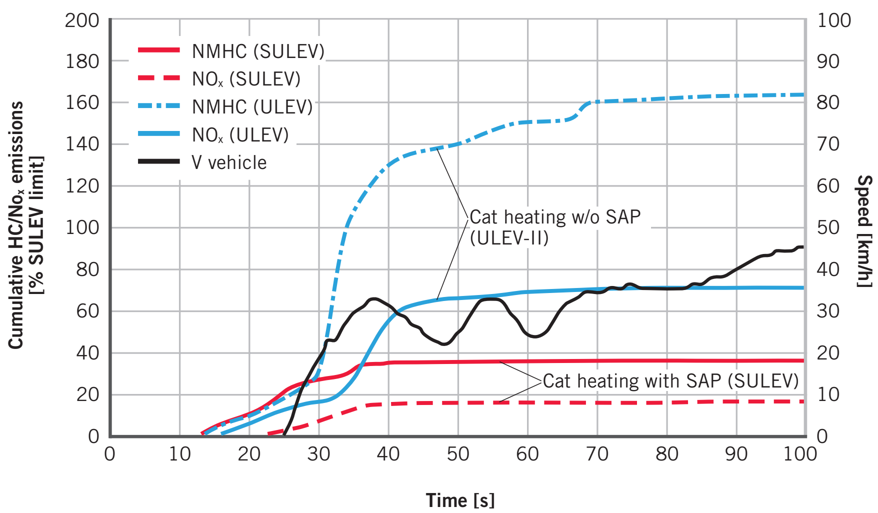
Another example is the compensation of the heat drop caused by the turbocharger. Turbocharged engines must compensate for the loss of heat in the turbine in order to heat as quickly as possible the catalytic converter downstream of the turbine. The exothermal process of secondary air can support this and is therefore an opportunity of implementing low-consumption concepts such as direct-injection turbocharged gasoline engines.

The ruggedness of systems based on secondary air injection is indicated, among other factors, by the relatively wide lambda window in which the engine can be operated following cold start. Strategies without secondary air and a lean engine start-up have a narrower window and suffer more from any changes resulting from factors such as ambient conditions, fuel quality or possible production deviations during series manufacture. Moreover, emission advantages from secondary air are free from any negative effects. In contrast to catalytic converters which, for example, age thermally or chemically, secondary air throughout the service life of the engine, is a contribution toward emission reduction.

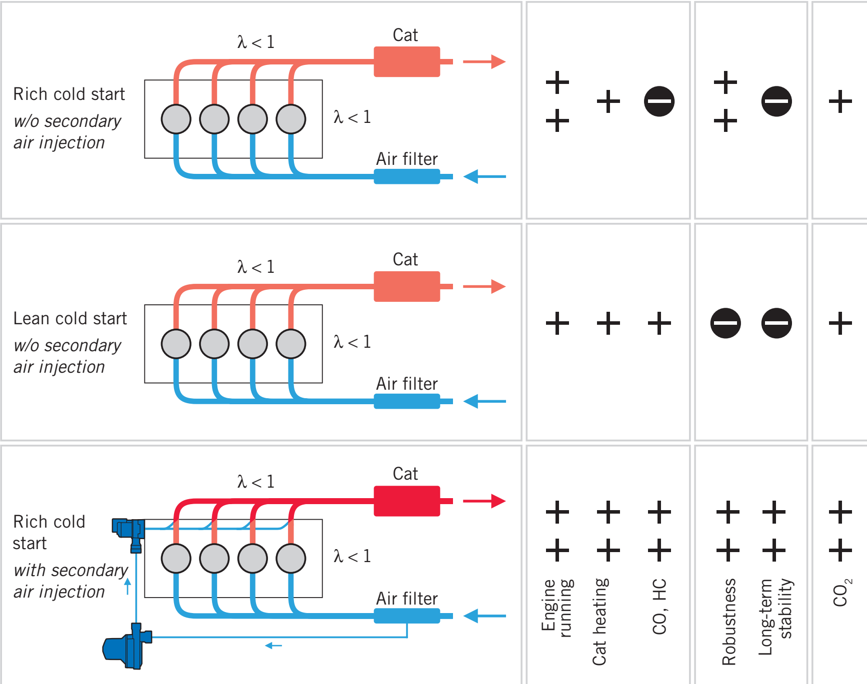
SYSTEM STRUCTURE AND COMPONENT REQUIREMENTS

The key constituents of a modern secondary air system are the secondary air pump (SAP) and the secondary air valve, ①. The SAP normally extracts the air from the engine air filter via a line or, alternatively, its own secondary air filter. The air is pumped to the secondary air valve. This valve opens either the air path for the secondary air or blocks this in order to prevent the reflow of exhaust gas into the secondary air system. Just as the secondary air pump which operates unregulated at constant voltage, the valve is not controlled in its stroke in order to prevent any effects on the volume of secondary air. Downstream of the valve, the secondary air is routed into the exhaust gas system. Preferably this is through an air distribution bore and branch passages in the cylinder head. Certain systems are supplemented with sensors (pressure or mass air flow sensors) if only for diagnostic purposes.

Secondary air systems have been used in this way since the end of the 1980s. Apart from the sensors, there has not been any change to the fundamental setup. However, over the years the require-



❷ Cumulative emissions in the FTP-75 exhaust test [2]



❸ Comparison of various cold-start and catalytic converter heating strategies

ments to be met by the system and its components have been steadily adapted to the existing conditions. Whereas in the past, the operating times ranged between 60 and 90 s, for future applications these will be less than 20 s. The typical exhaust gas backpressures for the secondary air systems had been some 100 mbar. As a consequence of turbochargers and exhaust gas aftertreatment components, the operating pressure is rising in the direction of 140 mbar and even higher. The materials making up the components have to be adapted to the increased use of biogenetic fuels. Not least of all due to the requirements of on-board diagnosis (OBD), systems are changing in the selection of components, operating modes and as a result of the addition of sensors. Among the requirements to be met by the secondary air pump are a shorter maximum capacity run-up time (the time required to reach 90 % the rated capacity), a compact footprint, a high rate of air flow at high operating pressure and low air and structure-borne noise emissions. Whereas in the past the period up to the rated pumping capacity had not been a decisive criterion, this has now changed with the ever shorter operating time. The shorter the operating time, the more important it is to achieve the rated pumping capacity.

COMPARISON WITH ELECTRICALLY HEATED CATALYTIC CONVERTER

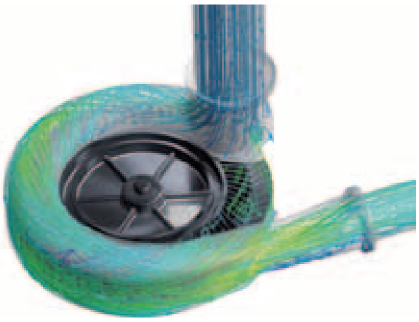
Theoretically, rapid converter heating to operating temperature is also possible with an electrically heated converter. The advantage of this arrangement is the possibility of very quick heating. To fully exploit this potential, it is necessary to ensure very high electric power density. Since this is not possible with a conventional 12-V vehicle electrical system, the electrically heated catalytic converter must be operated via a separate partial vehicle electrical system with a higher voltage. Basically, this comprises the following components: an electronic power switch, additional battery, DC/DC converter, power control unit and high-voltage wiring. A further advantage of the electrically heated catalytic converter is the option of starting the heating of the converter even before the engine is switched on, for example via an ignition signal (terminal 15). In contrast, the best possible engagement time for the SAP is the moment of starter actuation with the exothermal reaction by nature being slightly delayed, beginning after the initial ignition.

All the above-mentioned components of the electrical catalytic converter system including the electrical catalytic converter itself must be diagnosable (mandatory OBD) and provide constant performance throughout service-life. Given the high complexity of the electrically heated converter, the manufacturing input is much higher compared with a secondary air system. It will not be possible to achieve the robustness of a secondary air system. A further disadvantage of the electrically heated catalytic converter occurs during repeated short-distance operation. This is where excessive discharge will lead to a gradual draining of the battery each time the engine is restarted. If this occurs, it will be necessary to deactivate the converter heating. This leads to OBD problems and must be indicated by a warning light. With its relatively low power consumption the secondary air pump, in contrast, does not overdrain the battery since its energy requirement can be completely met by the generator. Finally, the costs of an electrically heated converter system, if only because of the expense involved by the separate voltage supply, is a multiple of the secondary air system. As a consequence, the advantages are evident. From today's vantage point, an electrically heated catalytic converter is not a viable option.

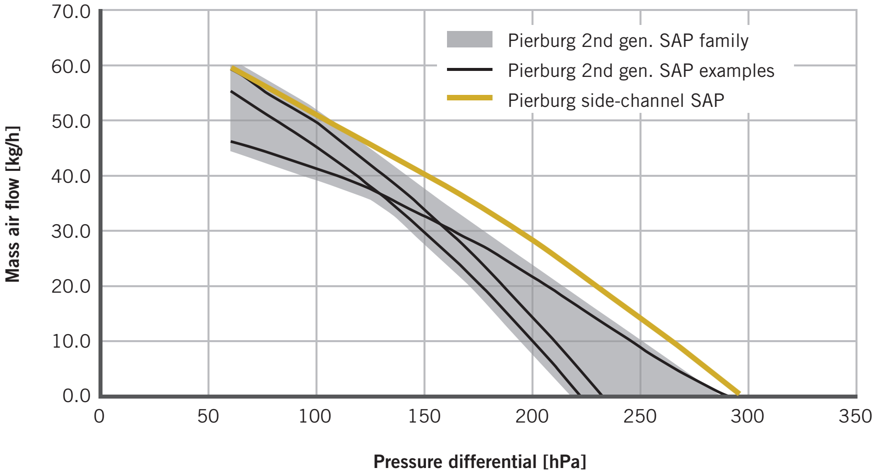
SIDE-CHANNEL SECONDARY AIR PUMP

Pierburg GmbH has developed a new secondary air pump that addresses the increasingly sophisticated requirements, ❹. In contrast to the well-known second-generation Pierburg pumps with two-stage radial blower [1], the new pump delivers the air with the aid of a single-stage side-channel blower parallel and with one channel on

❹ The new side-channel secondary air pump from Pierburg



❺ Air flow on the new side-channel secondary air pump



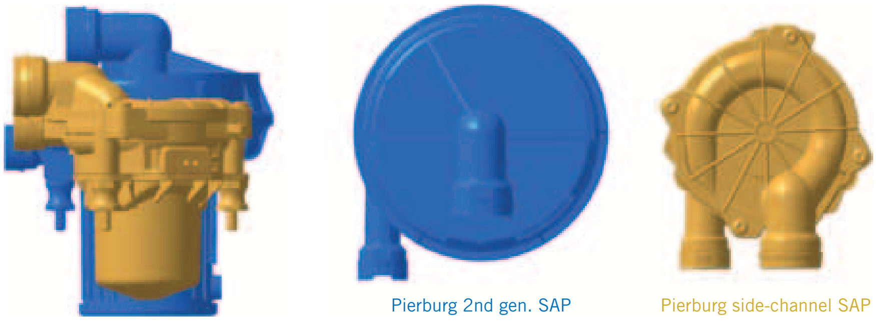
❻ Air throughput characteristics on a selected secondary air pump from Pierburg

either side of the impeller, ④ and ❺. The impeller is driven by a DC motor bolted to the pump housing. Housing and cover options along with a variety of motor designs allow the assembly to be flexibly and quickly adapted to customer requirements. The new pump combines the advantages of very high capacity with, at the same time, a small footprint. At a low operating pressure (around 100 mbar) the first versions of this pump are already at the top range of what had been achievable with the second-generation pumps, ❻. With higher operating pressures (around 140 mbar) the much smaller side-channel pump is superior to the larger pumps of the second-generation Pierburg model ❼ and competition. By optimizing the moment of inertia and motor characteristics, it has been possible to reduce by up to 75 % the time taken to achieve maximum capacity and this is now only 300 ms.

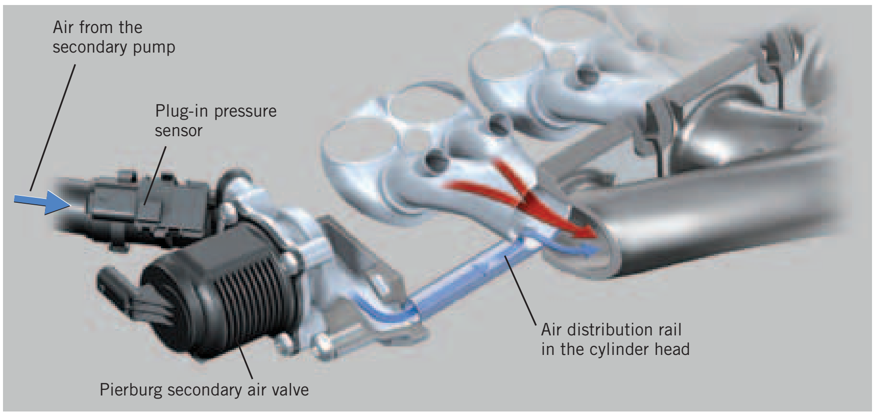
A major challenge was to retain the excellent acoustic properties of the second-generation pump with centrifugal blower and even improve these. With minimum imbalance of motor and impeller, the rotor of the secondary-channel SAP from Pierburg unit is ideal for reducing structure-borne noise. By designing the pump with its center of gravity in the mounting level of the vibration damper and carefully selected vibration damper geometry, the structure-borne noise transmitted to the engine or vehicle body is further reduced. This pump is therefore suitable for engine or vehicle body mounting. Airborne noise, frequently an unwanted property of side-channel blowers, has been further decreased by adapting the channel contours and the impeller; as a consequence, the aggregate noise level of this pump is superior to that of competitive products and even slightly better than that of the second-generation Pierburg SAP. Furthermore, the frequently irritating noise emitted by the first set of blades on side-channel blowers has been eliminated.

SECONDARY AIR VALVES

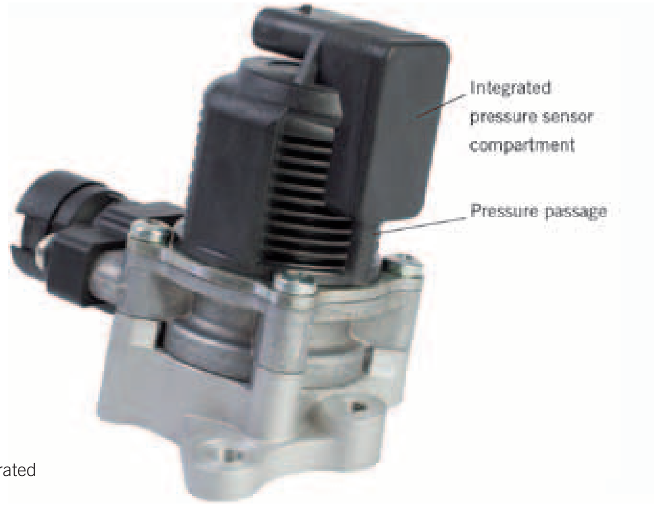
Pierburg's proven secondary air valves, model ARV (vacuum actuated, independently of the pump) and SLV (actuated by



❼ Size comparison between the second-generation Pierburg SAP and the new side-channel SAP



❽ Example of installation of a Pierburg ESV with plug-in pressure sensor [2]



❾ Pierburg ESV with integrated pressure sensor

pump pressure, no separate actuation) continue to meet the requirements of many applications. Increasingly, however, new applications have been provided with a solenoid operated (model ESV) Pierburg valve first introduced into series production in 2007, ❽. The main advantages of this valve are its rapid opening and closing independently of any vacuum, pump pressure or ambient pressure, the high opening force and the small footprint. The ESV shares with the two other models, ARV and SLV, the valve closing unit design with integrated automatic valve destined to prevent unwanted return flows of exhaust into the secondary air circuit. The outstanding feature of Pierburg's ESV is a very high opening force of around 80 N despite the slight space taken up by the magnet's coil. This is achieved with the aid of a split armature. With closed valve, this allows high magnetic flux line focus and a reduced effective pole clearance.

All Pierburg secondary air systems and their components are OBD compatible with mass air flow or pressure sensors. For those arrangements working on the basis of pressure measurements in the secondary air system, Pierburg has two options. The first is a version of the plug-in pressure sensor, ⑧, which allows ample flexibility in the selection and positioning of the sensor. Secondly, Pierburg offers the option of a pressure sensor, ❾, integrated with the actuator. This reduces the footprint and eliminates the necessity of a separate electric contact. The integrated sensor is an absolute pressure sensor with a ratiometric, analog output signal. The characteristic curve is programmable. On both versions, with the internal sensor and with the plug-in version, the actuator and hence the electric plug can be rotated in 90° steps with relation to the housing. On the integrated sensor version, a circular channel permits this by transmitting the pressure signal from the housing to the actuator in each of the four settings. The low footprint, the modular structure and the replaceable contacts allow space-saving attachment directly at the cylinder head with reduced pressure losses.

SUMMARY

Secondary air injection has proven to be a robust, highly durable and, overall in the vehicle systems, cost-effective approach for emission reduction. Its potential of assisting in low-consumption strategies is increasingly gaining attention. The further development of these products must keep pace with changing requirements. Pierburg's new SAP and the further development of the valve are destined for secondary air injection on a variety of applications. Due to the growing significance of factors in which secondary air injection demonstrates its advantages, it will find increasing use. This is particularly the case on SULEV applications. Alternative strategies with similar objectives, e.g. HC traps or electrically heated catalytic converters, individually address these aspects do not, however, achieve such an overall positive impact as secondary air injection.

OUTLOOK

Secondary air systems are suitable not only for emission reduction on gasoline engines. On diesel engines, additional fresh air in the manifold could altogether improve the oxidation of hydrocarbons and carbon monoxide and hence reduce emissions altogether. Analyses carried out by Pierburg have indicated that secondary air can also be applied for significantly raising diesel exhaust temperature. Another study indicates that an increase in mean exhaust temperature of 20 to 30 °C would raise overall cycle CO conversion rates by up to 70 % [4].

REFERENCES

[1] Kochs, M. W.; Kloda, M.; van de Venne, G.; Golden, J.: Innovative Secondary Air Injection Systems. SAE Paper 2001-01-0658, 2001
[2] Eiglmeier, C.; Pfalzgraf, B.; Helbig, J.; Adam, S.; Grigo, M.; Dornhöfer, R.: Der neue R4 – 2.0 l TFSI SULEV/PZEV-Motor von Audi. 16. Aachener Kolloquium Fahrzeug- und Motorentechnik, 2007
[3] Hallgren, B. E.; Heywood, J. B.: Effects of Substantial Spark Retard on SI Engine Combustion and Hydrocarbon Emissions. SAE-Paper 2003-01-3237, 2003
[4] Neumann, D.; Schrade, F.; Basse, N.; Schäffner, J.; Tschiggfrei, W.; Krämer, L.: Kohlenmonoxid – Die neue Emissionsherausforderung für Diesel-Pkw. In MTZ 70 (2009) Nr. 7/8

CRANKSHAFT BEARINGS FOR ENGINES WITH START-STOP SYSTEMS

The challenges placed on engine bearings are growing along with the improving efficiency of modern combustion engines. Depending on the driving situation, fuel-efficient start-stop systems for instance increase the number of cycles that take the crankshaft and the bearing half shells through a phase of mixed film lubrication. A new overlay from Federal-Mogul based on reinforced polyamide-imide prevents wear which metallic sliding surfaces tend to show during such punishing use.

Crankshaft bearing shells with Irox overlay

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INCREASED WEAR CAUSED BY START-STOP SYSTEMS

Plain bearings used in the engine have to be designed for ever increasing specific load collectives. This includes rising firing pressures, higher temperatures, smaller bearing dimensions and a rising level of crankshaft deflection which is caused by lightweight designs. Oil viscosity that is getting lower as a tendency and the higher level of oil dilution caused by fuels such as E85 are only a few more of the many issues on an increasingly long list of tough operating conditions. Nevertheless, plain bearings shall have the lowest possible coefficient of friction to limit the engine's internal losses.

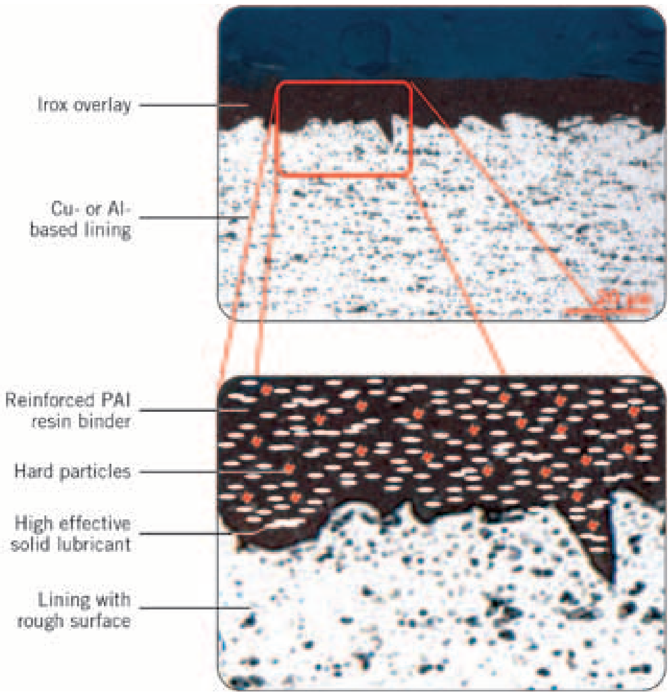
Start-stop systems, which can be found in more and more vehicles today, pose a particular challenge. They increase fuel efficiency by 5 % and more through switching off the engine during standstill as often as possible. To avoid any impact on drivability, the engine is cranked very quickly as soon as the driver engages the clutch.

For the crankshaft bearing half shells and the big end bearings this can translate into frequent high-speed rotary movement before a hydrodynamic film is established. During this phase of boundary lubrication metal-to-metal contact can occur between the crankshaft surface and the bearing's sliding surface. This was not an issue while the number of engine re-starts totaled at what was generally understood to be a normal magnitude. However, in a vehicle with start-stop system this effect can necessitate new technological solutions to avoid premature bearing wear, depending on the driving cycle. Consequently future engines for start-stop applications need to be designed for 250,000 to 300,000 starts. Traditional bearing shells with aluminum or copper lining show severe wear after only 100,000 cycles.

A newly developed coating called "Irox", for which patent is pending, is a counter measure based on polyamide-imide with additives. This overlay prevents bearing wear even during frequent mixed lubrication phases and is thus optimally designed for start-stop operation – this applies even to applications with cast iron crankshafts. On top of that the new overlay makes half bearings with aluminum (Al) based lining so much more robust that they can be used in applications previously considered to be the realm of copper (Cu) based linings.

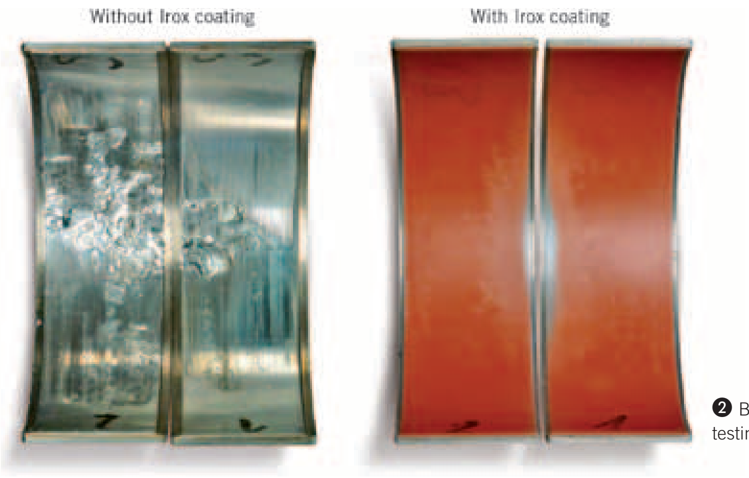
MATERIAL STRUCTURE

The new sliding bearing matrix consists of three elements: The steel back ensures dimensional and mechanical strength of the half bearing. On top of this a lining substrate is applied that provides the appropriate tribological material properties for the individual application. Throughout modern mass production vehicles these linings are made from lead-free substrates, most often either based on aluminium (Al) or copper (Cu) alloys. Typically Al based crankshaft bearings are used for low to medium specific loads (≤70 MPa), while Cu based (CuNi2Si) bearings are used for medium to high specific loads. To a large extent bearing durability depends on the top layer. With Irox bearings this overlay consists of polyamide-imide (PAI) with additives, ❶.



❶ Schematic representation of the matrix system's design

The hard-particle reinforced PAI overlay is permanently bonded to the lining and forms a durable layer. Within the resin matrix several additives are evenly dispersed, some of which have a very fine grain. The bearing's typical reddish brown color is owed to micro reinforcement by finely distributed oxide particles. Another



❷ Bearing shells lined with the A-650 Al alloy substrate after endurance testing on an Underwood rig with 80 MPa for 250 h

type of not quite so finely grained hard particles also serves to increase the sliding layer's wear-resistance. Additional solid lubricant particles embedded in the matrix ensure good sliding properties during local metal-to-metal contact.

PAI is an amorphous highly temperature resistant polymer that well resists chemical attacks and wear. The material also lends itself ideally to be processed for the current application. Basically the overlay made from PAI and oxides can be applied and bonded equally well to the typical bearing linings made from aluminum or copper or bronze materials.

BEHAVIOUR DURING OPERATION

The Irox overlay's wear resistance is owed to a combination of several specific material properties. Its wettability causes the hydrodynamic film, essential for the correct function of a plain engine bearing, to build up faster. Also the micro reinforced PAI overlay is an elastic layer. Due to its lower modulus of elasticity it is better suited to compensate asperities in the topography of its running counterpart and thus shows excellent wear resistance.

Prior to applying the hard particle reinforced PAI the substrate is roughened-up to a specific level, ①. During bearing operation this generates two benefits: Firstly, the reinforced PAI overlay can cope very well with axial thrust load from whichever direction as the mechanical interlocking with the substrate on a micro level ideally supports loads. Secondly, the micro-mechanical interlocking with the substrate caused more beneficial wear characteristics. If the overlay is worn away locally, the many substrate asperities prevent the abrasion to advance because the metal-to-metal contact is limited to the microscopic peaks. Even in the case of edge loading the overlay will not show fissures but limited local wear only. This appears to be quite relevant as bearing damage and seizure of Al based substrates typically begin with crack initiation, subsequent crack advance and finally escalation.

Another material property is relevant in this context: The hard particle reinforced PAI overlay's elasticity has a positive effect on the plain bearing's damping properties. At the same time the PAI matrix maintains a certain capacity for embedding micro particles.

As PAI is a poor heat conductor the overlay limits the heat input to the substrate below. This turns out to be particularly beneficial with Al based substrates. As far as the roughness requirements to the sliding counterpart are concerned, the Irox matrix is no different from other crankshaft bearings. Instead, the new overlay matrix shows a superior behavior when run against a cast iron crankshaft.

SUITABILITY FOR CAST IRON CRANKSHAFTS

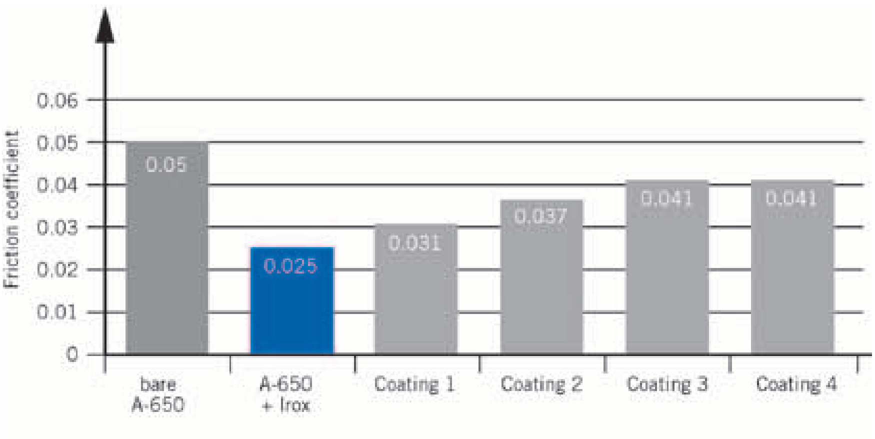
Using crankshafts made from grey cast with fine spherical nodules of graphite (Nodular Cast Iron, NCI) poses special requirements to plain bearings. A characteristic of NCI's morphological structure are hard ferrite envelopes around the graphite nodules. Despite even the most careful surface machining via polishing these circular structures covered by caps will not fully disappear. Those caps may break off during running condition and damage the bearing surface. If the bearing has no suitable mechanism to compensate this effect, cast iron crankshafts can quickly cause bearing seizure. With the Irox overlay, a combination of the embedded hard particles' polishing effect and the polymer's ductility prevent this type of problem from occurring.

TEST RESULTS

To verify the new layer matrix's suitability for the severe requirements of start-stop systems, the plain bearings had to pass multiple harsh tests. ❷ shows the surface conditions of two crankshaft bearing half shells lined with the same A-650 substrate, an Al alloy (AlSn6Si4CuMnCr) with improved micro structure. One of the bearings has an additional Irox overlay while the Al substrate itself serves as sliding surface in the case of the reference bearing.

Both material systems were tested at 80 MPa load on an Underwood rig under identical boundary conditions. Even though A-650 is a new high strength Al alloy with particularly fine silicon particles, its sliding surface shows considerable damage, while the Irox overlay shows only very limited local wear along the edges. This test result highlights very clearly just how much the overlay properties dictate whether a bearing substrate is suitable for a specific application or not.

Bearing half shells with Al substrate and the Irox overlay had a substantially improved fatigue resistance. Thus the new overlay matrix can toughen up Al based crankshaft bearing half shells for specifics loads that are up to 25 % higher – a magnitude that was hitherto reserved for Cu based systems. When applied to Cu based substrates, the Irox overlay successfully



❸ Coefficients of mixed friction of the tested sliding surfaces in comparison

passed engine test runs with up to 105 MPa specific loading.

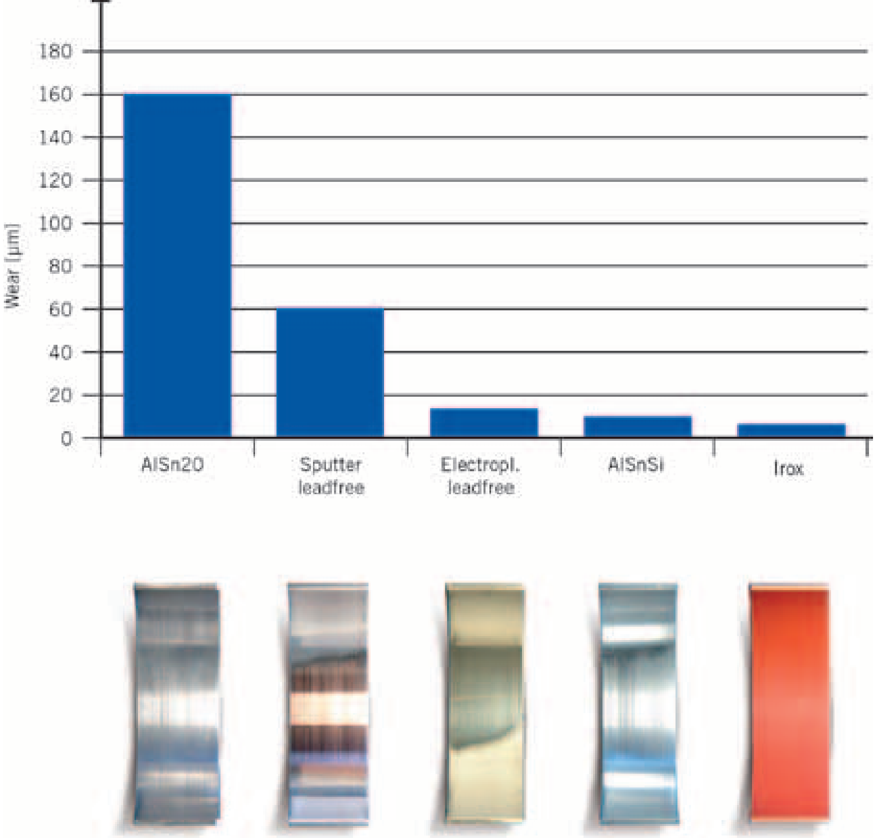
During further test runs on a High Frequency Reciprocating Rig (HFRR) the Irox overlay reduced the coefficient of friction by half when compared to a bare A-650 Al substrate sliding surface. Even in comparison to other polymer overlays, used to coat piston skirts for instance, the Irox overlay's coefficient of friction was lower by 20 % to 40 %, ❸.

In ❹, results of various bearing types in one of many tests are shown, here in a so called "Sapphire test bench". In many start-stop cycling tests, the wear of the Irox bearings amount to a significant lower level than that of sputter bearings, known for their perfect wear resistance. The result has been consistent and reproducible through all of the tests.

The Irox bearings were tested in engine tests with standard gasoline as well as with E-85. Despite the ethanol share of the E85 and the subsequent oil dilution caused by the alcohol, the Irox matrix's wear characteristics did not change, while a two-layer Al bearing lined with the common entry-level A-590 substrate (AlSn6Si-4CuMnCr with coarser silicon particles) failed during the E85 test run.

It can be deducted from these findings that the new layer matrix offers a potential for saving fuel by using engine oil with lower viscosity. It could also be proven during tests that the Irox overlay is less sensitive to small bearing clearances.

❹ Wall thickness under start-stop-conditions after 15,000 cycles

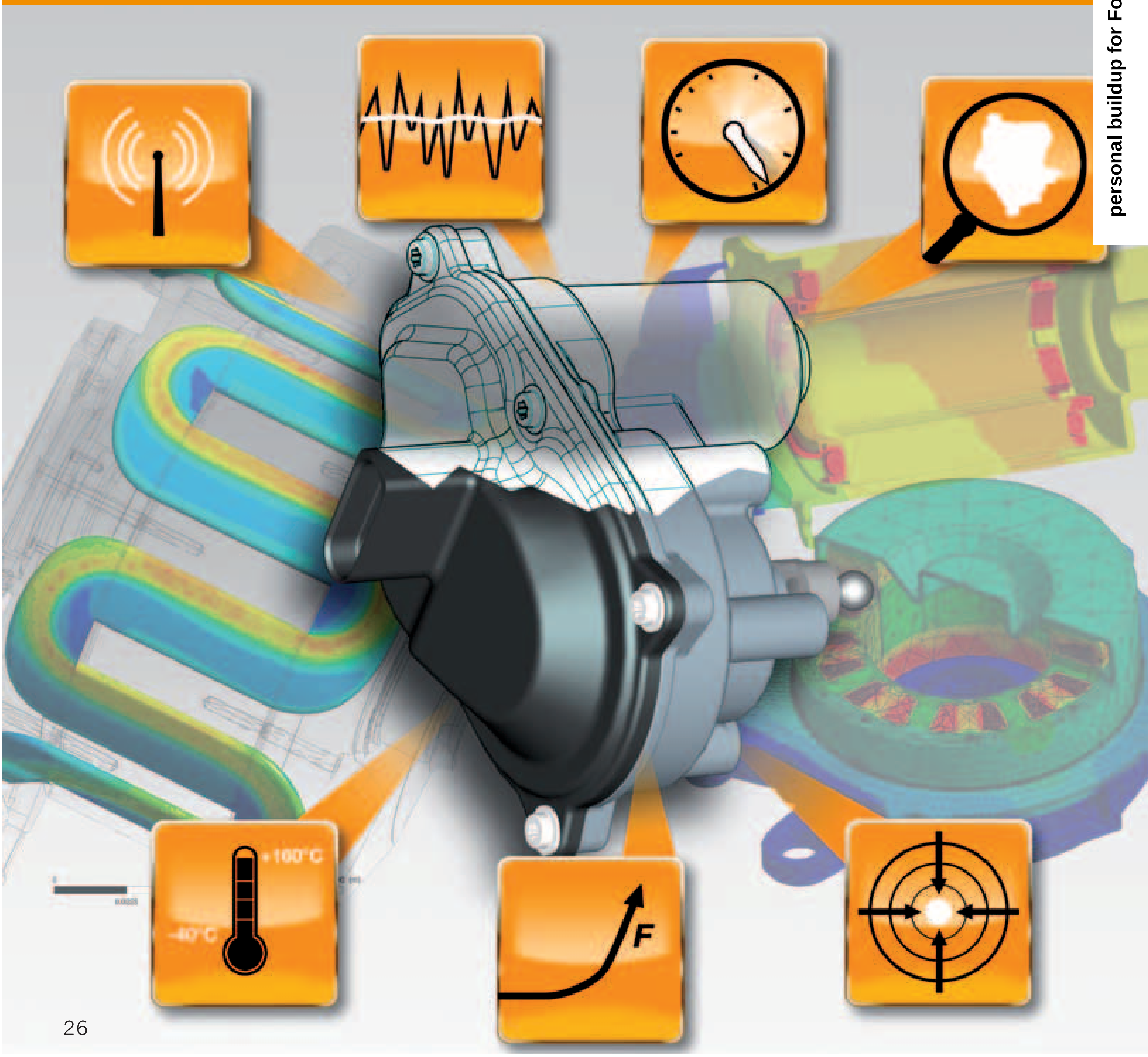


SUMMARY AND OUTLOOK

The new Irox overlay matrix is a solution tailored for combustion engines with a high share of mixed lubrication. The durable overlay made from micro reinforced PAI prevents premature abrasive wear, which metallic sliding surfaces can show under such operating conditions. Due to its high temperature resistance, its good damping properties and its robustness against oil dilution, e.g. caused when E85 fuel is injected, the new overlay matrix is suitable for use in modern passenger car and truck engines with high thermo mechanical efficiency and correspondingly tough requirements to engine plain bearings. The micro reinforced PAI overlay can also contribute to the use of engine oils with low viscosity (High-Temperature High-Shear Rate, HTHS 2.6 to 2.1). Large scale production for the first series application is scheduled to commence in April 2011. Further development work will be dedicated to exploring the Irox overlay's potential for use in highly loaded ancillary components, on thrust washers and for use in automatic transmissions.

ELECTRIC ACTUATORS FOR MODERN INTERNAL COMBUSTION ENGINES

Actuators convert the control variables of modern engine management systems into mechanical actions. Often, through continuous improvements in positioning accuracy and speed, it is these actuators that result in innovations which make engines even cleaner and more fuel efficient. Pneumatically operated actuators, however, come up against their limits in many cases. In the following report, Continental gives an insight into the selection and development of electric motor-driven actuators and their functional characteristics.



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ACTUATORS REALIZE FUNCTIONS

Vehicle drive systems need to address the demand for lower fuel consumption and reduced emissions. In addition to enhancing efficiency, this can also be achieved by taking the improved, cleaner combustion route. There are also, of course, alternative drive concepts which compete with and supplement optimized combustion engines.

Essentially, all drive architectures need to avoid wastage when converting resources into mobility. The excellent computing power of engine management systems can help us here by creating the best possible static and dynamic processes in the powertrain. At the same time, it is becoming increasingly important to provide reliable applications for the energy balance. In many instances, functions are implemented by actuators. For the application to run reliably, actuators are needed which function reproducibly as regards dynamics and position under all conditions and which remain unaffected by disturbance influences.

ELECTRICAL ACTUATORS WILL INCREASINGLY BE INCORPORATED

Function and application are at the heart of the modern drive system. Tools which can simplify and assist with this task represent a significant advantage for the application design engineer. Behavior which is both reproducible and constant whatever the external circumstances forms the foundation of a fully functioning end product. A world without throttles, valves and other components, not just for regulating fuel and air intake, the gas cycle, combustion and exhaust gas aftertreatment, but also for acoustic or comfort requirements is now unimaginable. All these actions require actuators, from simple versions such as thermostatic expansion elements, in coolant circuits for example, to complex sub-systems with integral intelligence. The requirements for the system as a whole are adjustment accuracy and speed, and freedom from interference.

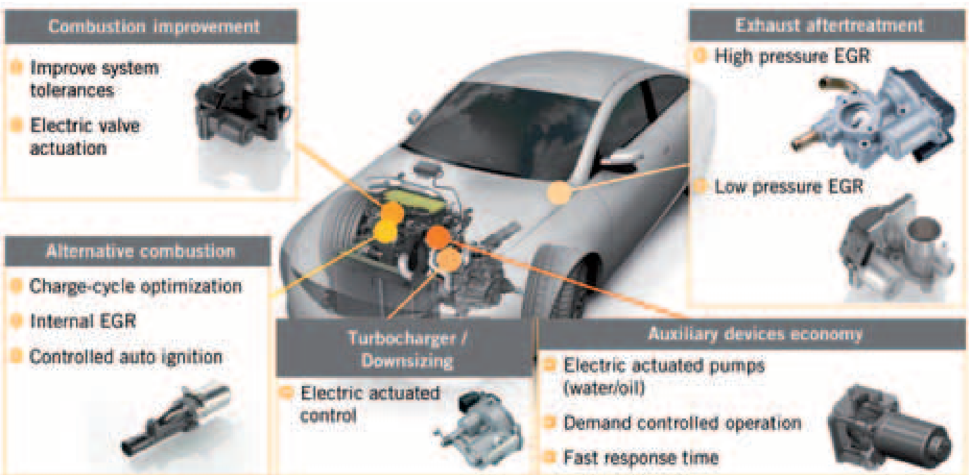
The trend is essentially towards closed-loop systems. High quality as regards optimum combustion management has already been achieved with steady-state processes. However, for highly dynamic processes, such as acceleration and gearshifting, pre-controls are also employed. The more precisely these can be tailored to the dynamic process and the smaller the positional tolerances, the better the consumption and emission characteristics. Consequently, pre-controls require actuators to have reproducible functions, in particular precise intermediate positions. This also applies when subject to disturbance factors such as temperature, vehicle system voltage or load peaks. Examples of this are throttling in the intake section in order to generate a specific vacuum for exhaust gas recirculation or the precise positioning of small wastegate apertures for regulating charge pressure.

Pneumatically driven actuators have, in many cases, reached their limits (vacuum variance/positional regulation accuracy). External load pulses are transmitted to the position virtually without any damping. Their only advantage is that they can easily be employed in so-called on/off systems. In future, electrical actuators will increasingly be incorporated into drive environments. They allow known functions to be carried out reproducibly and with great accuracy; they can upgrade these functions within the system (e.g. through their ability to adopt intermediate positions rather than just on/off) and they permit the development of essentially new functions within the network which only become possible if energy is provided in this way. Electrical actuators, properly designed to meet the requirements and logically integrated into the control strategy, are the preferred partners within the systems network if the application is to function correctly. This is equally true of applications for cleaner combustion engines, downsizing concepts and alternative drive concepts. ❶ shows some typical current applications for electrical actuators in combustion engines.

REQUIREMENT PROFILES

Actuators intervene in drive functions and usually have implications for emissions. Monitoring these functions and the ability to carry out onboard diagnosis are important criteria. The second essential requirement is excellent positional accuracy and reproducibility. In highly dynamic systems, in particular, positional accuracy also requires reliable adjustment dynamics, even under variable conditions (load, temperature fluctuations, service life factors).

Precise positioning must also be possible even when disturbance factors are present.



❶ Actuator applications in engine management systems

Ideally, actuators should offer an emergency operating function, allowing the function developer to provide the vehicle user the best possible degree of vehicle availability in the event of a malfunction. But desirable does not always mean practicable. A distinction has to be made between the essential and the desirable, with a view to conserving resources. There are also other significant constraints such as mechanical strength, EMC and environmental circumstances, ❷.

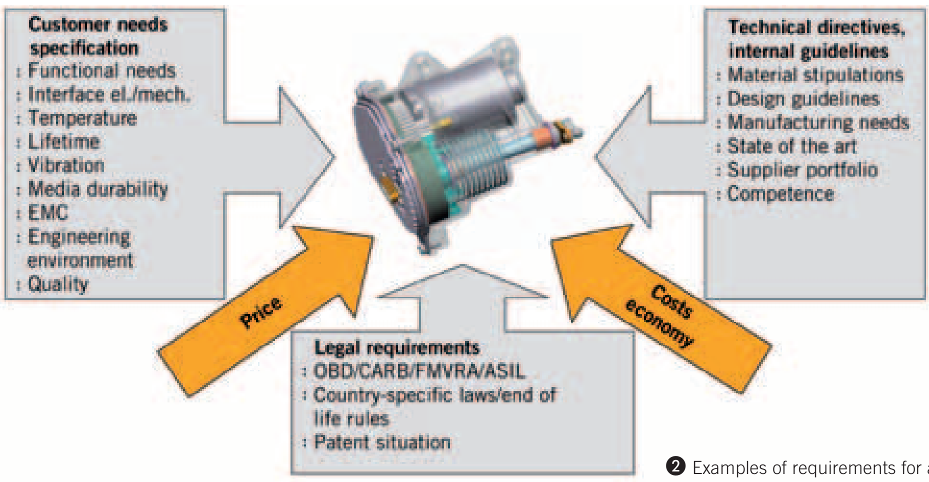
TYPES OF ELECTRIC ACTUATORS

Essentially, electrical actuators consist of a drive system, position sensors and an adjusting component (throttles, valves, levers, etc.). Contactless systems have gained acceptance for use as position sensors. It is important that sensors possess high resolution and a stable, reproducible characteristic under the constraints of temperature and interference. There is a variety of types available for use as drive systems. The aim is always to find the drive system whose energy is ideally matched to the load. Their design is influenced by the application profile, in particular by continuous load under temperature. The desired emergency operation behavior also exercises a major influence on design, cost and energy requirement.

Often, it is only precise simulation of the drive system under assumed design loads, taking account of both dynamic requirements and disturbance factors, that will produce a result. In very many instances, a gear drive from other drive systems (solenoid actuators, torque motors, stepper motors) will prove to be superior for engine compartment applications. Their advantages include interference resistance and excess torque.

DC GEAR DRIVE

In a DC gear drive, the motor providing the power output is operated in the speed range between best efficiency and maximum power, ❸. The torque is reinforced against the load by the gearing ratio. In general, this leads to minor holding currents and slight power loss. Its dynamic effect is to cause slight rotor inertia, resulting in rapid acceleration and braking of the movement. Since this sort of drive is also thermally stable in the best efficiency range, a maximum torque greater by several factors is briefly available. The drives can be designed with or without self-locking so as to satisfy many requirements. Account needs to be taken of operating time limits (gear wear or, in the case of DC motors, brush and bearing wear). From a control engineering point of view, they are easy to manage and facilitate rapid, reproducible and accurate positioning.



❷ Examples of requirements for actuators

SOLENOID ACTUATOR

Used as an on/off adjusting mechanism, easy to manage from a mechanical and control engineering point of view. In conjunction with energy accumulators (springs), higher adjustable loads can be realized from the respective holding positions on the yoke. The magnetic elements are the limiting factor; from a central resting position they must facilitate assured assignment to one side of the yoke. In this application, the drive also represents the system endstops.

If a solenoid actuator is used in a position-controlled system with intermediate positions, a balance is always needed between magnetic force and external force. This sort of balanced system usually exhibits no great excess forces. The dynamics are generally great. The drawback is the susceptibility to interference from external forces. Because there is little inherent damping, it is an oscillatory system and remanent holding forces on the yoke can cause ‘end position sticking’ (increased effort and cost for position control, limited adjustment accuracy). The theoretical adjustment mechanism dynamics can generally not be fully exploited. Advantages lie in wear behavior, compact design and the small number of components.

❸ Actuator with DC gear drive

TORQUE MOTOR

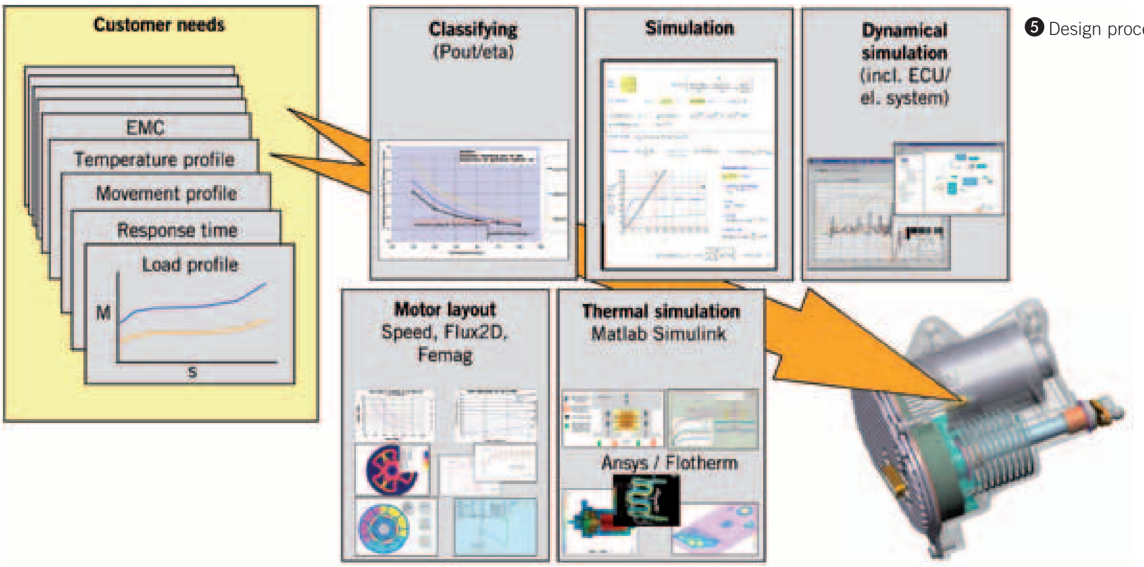
This is similar to the solenoid actuator but with proportional flow generation through the use of permanent magnets. A rotary movement is realized by the design of the yoke and the rotor.

STEPPER MOTOR

The stepper motor is well-known from its use in machine tools or plant and equipment drive mechanisms. The increment produces a digital resolution of the movement. Consequently, in some applications, there is no need for a position sensor. However, used as an actuator in a combustion engine environment, an absolute

❹ Summary of different types of drives related to typical applications of actuators in powertrain environment (green = good ability, yellow = medium ability, orange = bad ability)

		DC-geardrive	Solenoid	Torque motor	Stepper motor	Piezo motor
Performance	Range of performance					
	Dynamic behaviour					
	Shorttime excess force					
	Actuation range					
Complexity	Control engineering					
	Disturbance influence					
	Control effort					
	Parts requirement					
Case of application	Ideal design of output	rotary – linear	linear	rotary	rotary – linear	linear – rotary
	Ideal application	Drives with limp-home position	on/off-position small forces, end stop durability	on/off-position small forces, end stop durability	Application w/o required absolute position sensor	self-looking drives small stroke, high force, stable microstep
	Potential	BLDC-variants with high dynamic and high duration time			Combination with geartrain	Evolution of voltage amplifier and reduction of piezo voltage



❺ Design process

position sensor is needed because of the essential diagnostic function. A drive system based on a stepper motor requires in each case a significantly greater cogging/holding torque against the external load since loss of the position due to exceeding the breakdown torque can result in the drive 'spinning away'. The adjustable load reduces as the step frequency increases. Adjustability from stationary is similarly restricted since the position of the rotor relative to the stator exercises a significant influence. Even with an external absolute sensor, the design should be chosen so as to prevent step loss.

Stepper motors can be coupled with gears. The stepper motor's power requirement is greater than that of the position-controlled DC motor drive since it is vital that the holding position is achieved by current being applied without step loss. This also limits resistance to external dynamic interference factors. A disadvantage is the effort required at the output stages for bipolar activation of two or more phases.

PIEZO MOTOR

These motors (both linear and rotary) are self-locking. They feature high actuating forces and possible micro displacements. If actuating distances are lengthier, the actuating speed is slow compared to the drives described above. A further disadvantage is the need to generate piezo voltage in the 60 V to 150 V range. A piezo motor with typical actuating forces of 200 N and actuating speeds of less than 150 ms for a displacement of 20 mm cannot at present be realized within the given installation space limits. They already have an automotive application in injection valves.

RIGHT TYPES OF DRIVES RELATED TO TYPICAL APPLICATIONS

❹ summarizes the types of drive showing very different profiles. The aim is to find a solution which matches the requirements as closely as possible and an energy design solution which fulfils the functions while remaining affordable, ❺. One possibility is to employ scalable modular systems in which high-value core components are produced in large numbers, supplemented by elements which match specific applications. The DC gear drive is often the ideal basis. The DC motor provides the basic capacity in accordance with the requirement profile, with fine adjustment being generated via the gearing.

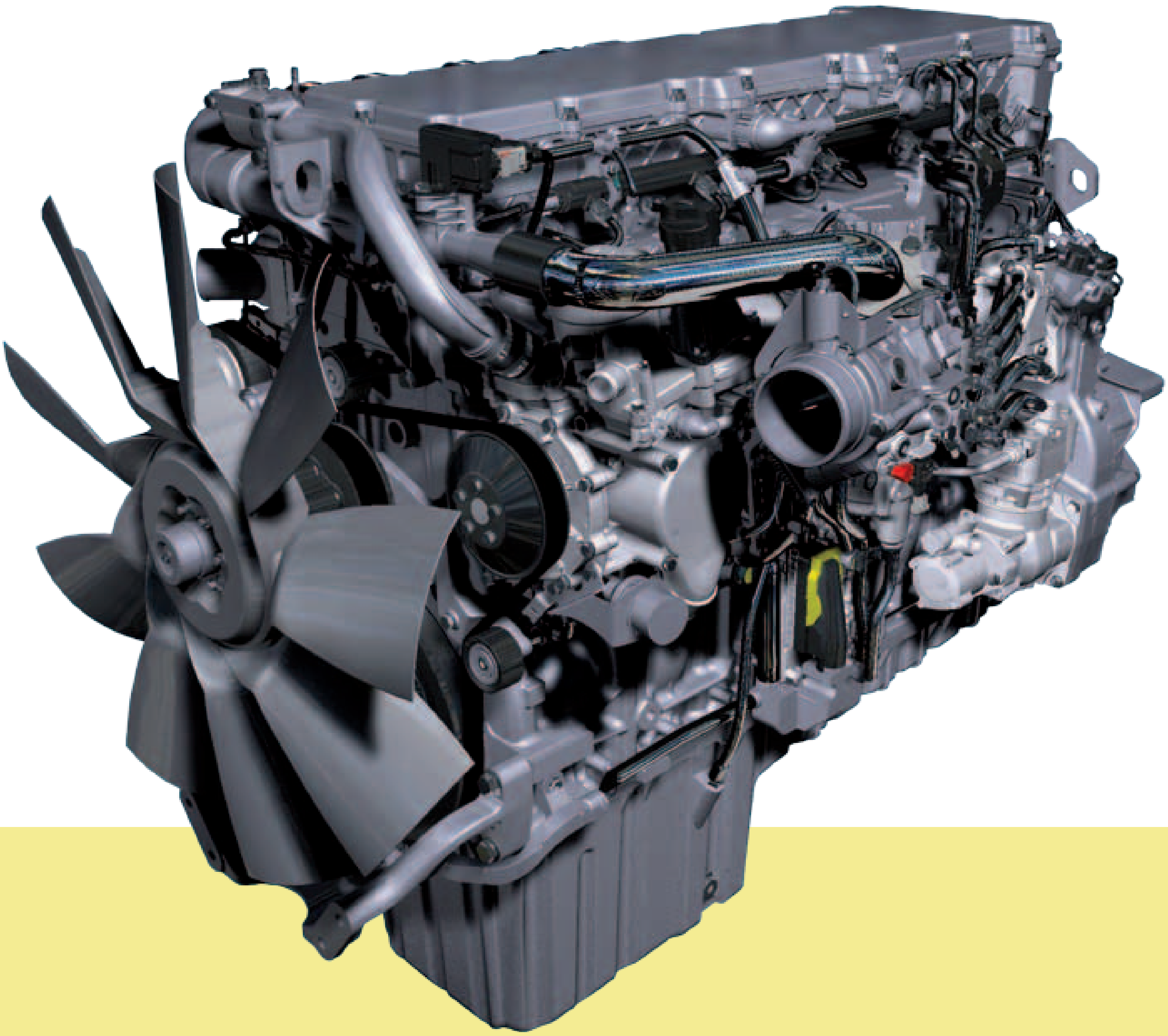
COMBINATION WITH THE ENGINE CONTROL UNIT

Engine control units constitute families. Their main feature is their computing power and their interface function. Demands for new functions do not always coincide with each new control unit generation. In such cases, smart actuators provide a possible answer. The electronics in the actuator take on the task of positioning, as specified by the engine management system, and confirm diagnosis and status. In this scenario, the actuator provides the computing power for positional control and actuator monitoring and also supplies the power; and the engine control unit is relieved of these tasks. This particularly applies to the use of BLDC motors.

OUTLOOK

In future, sophisticated vehicle drive systems will need a variety of mechanisms for adjusting control components. Electrical mechanisms will become pre-eminent. However, the power requirement should be handled sparingly since it will count towards the total energy balance. The primary objective is to restrict the requirements for drives to what is essential and to select the most energy-efficient actuator. Ensuring this requires know-how and an understanding of the application. The manufacturer's task will be to acknowledge that the actuator is a precisely tailored sub-system.

An actuator can be included and adjusted at an early stage as a model in combustion engine simulation environments. Adjustments to new environments can be made with little expense and great reliability. The result is that the application design engineer is given the ideal tool for helping to design the desired function and for reliably ensuring the engine function is maintained once applied.



NEURAL MATHEMATICAL MODEL FOR THE DETERMINATION OF THE COMBUSTION PRESSURE PROFILE

The further optimisation of engine combustion, both to meet statutory emissions standards and for the minimisation of fuel consumption is essential for future engine development. Daimler describes a neural mathematical model for a heavy-duty engine that calculates the combustion pressure profile using sensor and control values available from the ECU. The quality of the model is compared directly to cylinder pressure data measured dynamically.

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INITIAL SITUATION

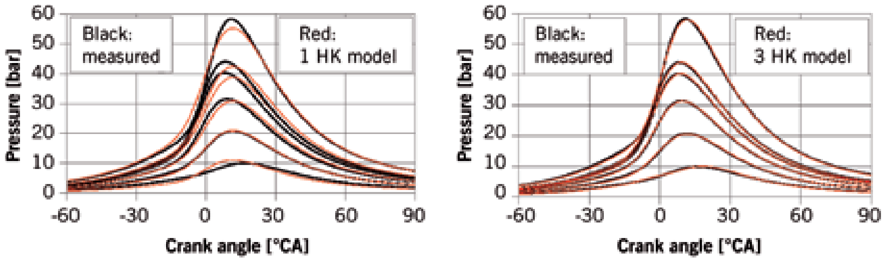
The point in time at which the energy is released influences the efficiency of the engine considerably. Closed-loop control of the center of combustion through in-cylinder pressure transducers achieves this optimally. Furthermore, this technique allows the possibility to calculate exhaust gas emissions physically with information obtained from the cylinder pressure profile. The high cost and low lifetime of the sensors however, has prevented high-volume production of such a system. Future control strategies should therefore be implemented that regulate the combustion in a model-based manner, using sensor information already available in the engine control unit.

MODEL APPROACH

Artificial Neural Networks (ANN) are becoming more and more relevant for the modelling of complex systems [2, 3]. These networks emulate the structure and function of the human brain and are fundamentally different to conventional data processing systems. It will be possible herewith, to solve problems that reach the limits of current conventional analytical processes. Neural networks consist of many interlinked units; the neurons. After a certain amount of previous training, output data can be generated with the network from given input data. This transformation is dependant on the network structure, the number of neurons, and the type and direction of the information flow. The performance or quality of the network can be judged by the difference between the demand and feedback at the output. In order to define the structure of the neural network, decisions have to be made about the quantity of input and output data for the model. For example, for a crankshaft angle based measurement of the cylinder pressure, with a resolution of 0.25°, 2880 values per combustion cycle would be needed. In order to calculate the combustion pressure trace over a crankshaft interval of 90° before TDC to 90° after TDC using ANN from ECU data directly, 720 output values need to be calculated from the input data. Due to this very high number of output values, the complexity of the neural network is greatly increased. Additionally, a large number of datasets for the training are needed. The duration of the training and calculation are also greatly increased. The goal then is to train a small neural network with few input and output values that meets the requirements of real-time capability. Consequently calculation of the combustion pressure profile in a small time frame with a motor-specific neural network could then be implemented in an ECU.

MAIN COMPONENT ANALYSIS FOR THE MODEL OUTPUT

For the determination of the combustion pressure trace from ECU measurement data with a compact ANN and reduced input and output data, a suitable data compression technique with minimal data-loss must be sought. The principal component analysis (PCA) describes a mathematical and statistical analysis technique, whereby the course of measurement signals can be reduced to a smaller number of orthogonal principal components with minimal data loss. The central ingredient of the principal component analysis of time-discrete measurement series, is the principal-axis transformation. For this transformation method, the starting point is the covariance-matrix. From this, all the main components will be extracted that best represent the information contents of the time-discreet



❶ Retransformed pressure profiles with one and three main components

	1 HK		3 HK'EN		5 HK'EN		8 HK'EN		10 HK'EN	
BP	F	S	F	S	F	S	F	S	F	S
1	0.679	22.2	0.053	6.18	0.010	2.67	0.001	0.848	0.0002	0.42
2	0.530	19.6	0.051	6.10	0.008	2.42	0.001	0.823	0.0003	0.50
3	1.538	33.4	0.052	6.16	0.013	3.07	0.002	1.232	0.0004	0.57
4	3.042	46.9	0.085	7.85	0.029	4.59	0.004	1.583	0.0007	0.73
5	3.211	48.2	0.123	9.43	0.035	5.04	0.005	1.809	0.0008	0.76
6	2.970	46.4	0.257	13.6	0.063	6.74	0.007	2.324	0.0015	1.05

❷ Results of the error examination (F = maximum mean quadratic error [bar²], S = standard deviation [bar])

measurement signals and between which no correlation exists, which is described through the orthogonality of the principal values. With increasing correlation, fewer principal components are needed to reconstruct the time-discrete datasets [1]. ❶ shows, for different load conditions of a combustion engine, the comparison of measured data and retransformed pressure profiles (with different numbers of main components). To obtain a qualitative comparison between the measured and retransformed data, an error examination was undertaken, whose results are in ❷. It shows that the measured data can be represented with enough accuracy, with a principal component quantity of eight. For the model implementation here, the number of principal components chosen was ten.

MODEL INPUTS

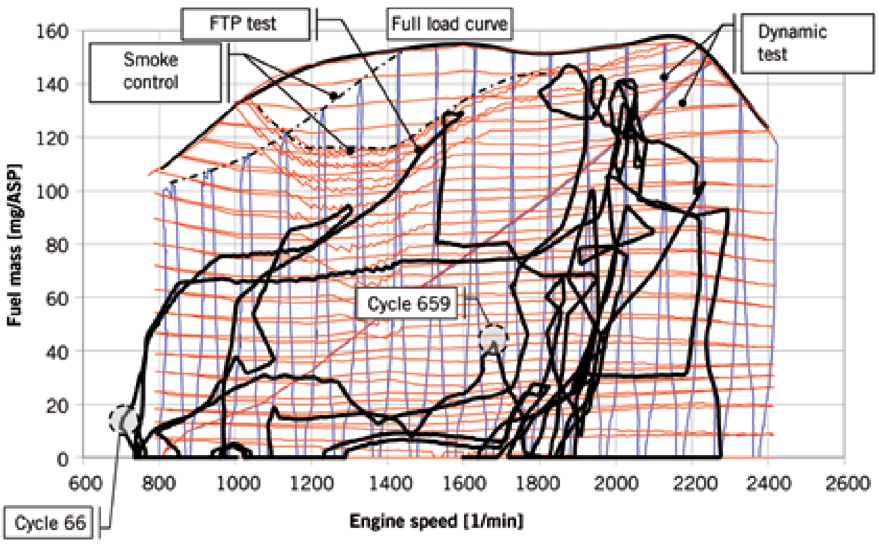
The cylinder pressure describes a physical value that is independent of position in the cylinder of a combustion engine. It characterises the combustion process, which is why it has such great importance for thermodynamic analysis. The definition of the system boundaries in the pressure trace analysis encompasses with closed valves: the cylinder head, the cylinder bore, and the piston. Inside these system boundaries from the point in time where the inlet valve is closed, the boundary conditions for the following combustion are already pre-defined. In dependence of these conditions, in the direct injection process, the corresponding fuel quantity is injected at a chosen crankshaft angle. Based on the knowledge from pressure profile analysis that the combustion process and therefore the combustion pressure are dependant on the conditions at the point in time "Inlet valve closed", the model input values must be able to describe these physical boundary conditions from available sensor data. Using the example of the heavy-duty combustion engine, eleven input values are used: Motor speed, the fresh air intake mass, the cylinder mass, boost pressure, mixture temperature (Fresh air and EGR) plus the injection parameters, start of the control signal for the injection of the main- and pre-injection, duration of the main- and pre-injection and their aspect ratio.

ANN MODEL FOR THE COMBUSTION PRESSURE TRACE

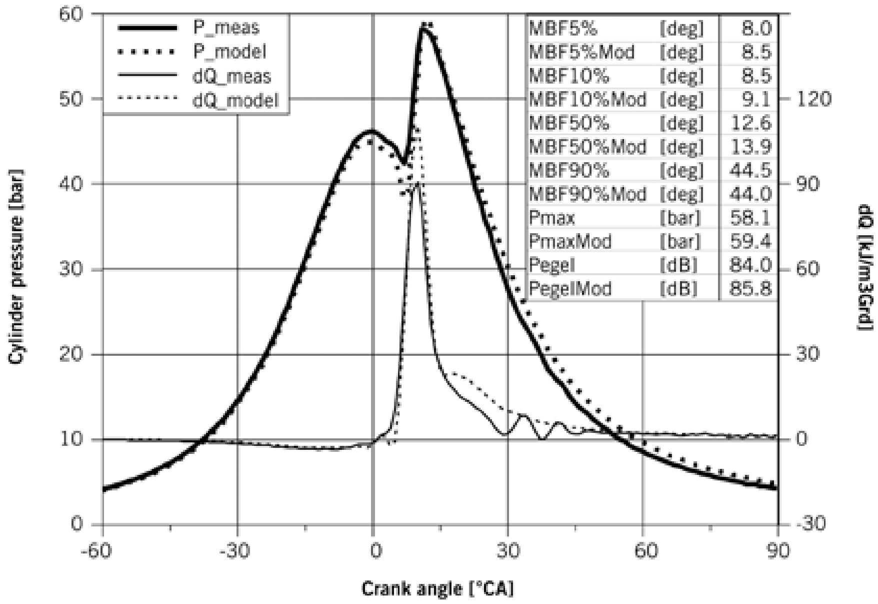
For the generation of the model there are many different possible types of networks, with different learning characteristics and structures. For this model application a multi-layer feed-forward network with back-propagation was chosen. It is defined by eleven input neurons, two hidden layers with twenty neurons each, and ten output neurons. This type of network is especially suited for non-linear problems where an initial quantity should be assigned to a target quantity. A tan-sigmoid function was chosen for the transfer function of the hidden layer and a linear function for the output layer. For fast training, the optimised Levenberg-Marquardt algorithm was used.

GENERATION OF TRAINING AND GENERALISATION DATA

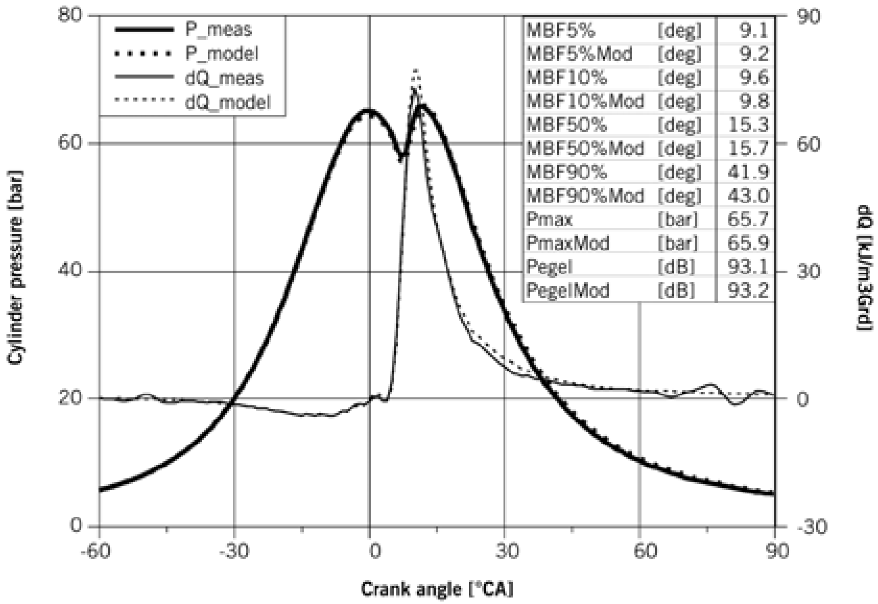
ECU measurement data and the corresponding measured combustion pressure profiles for training the network were recorded on the basis of dynamic test profiles – due to the large number of variables, with which the internal combustion can be influenced. Starting with a production ECU dataset, the training data was generated from dynamic speed and load pro-



❸ „FTP cycle w/o idle" and dynamic speed and load profiles



❹ Comparison of combustion pressure in FTP cycle for combustion cycle 66



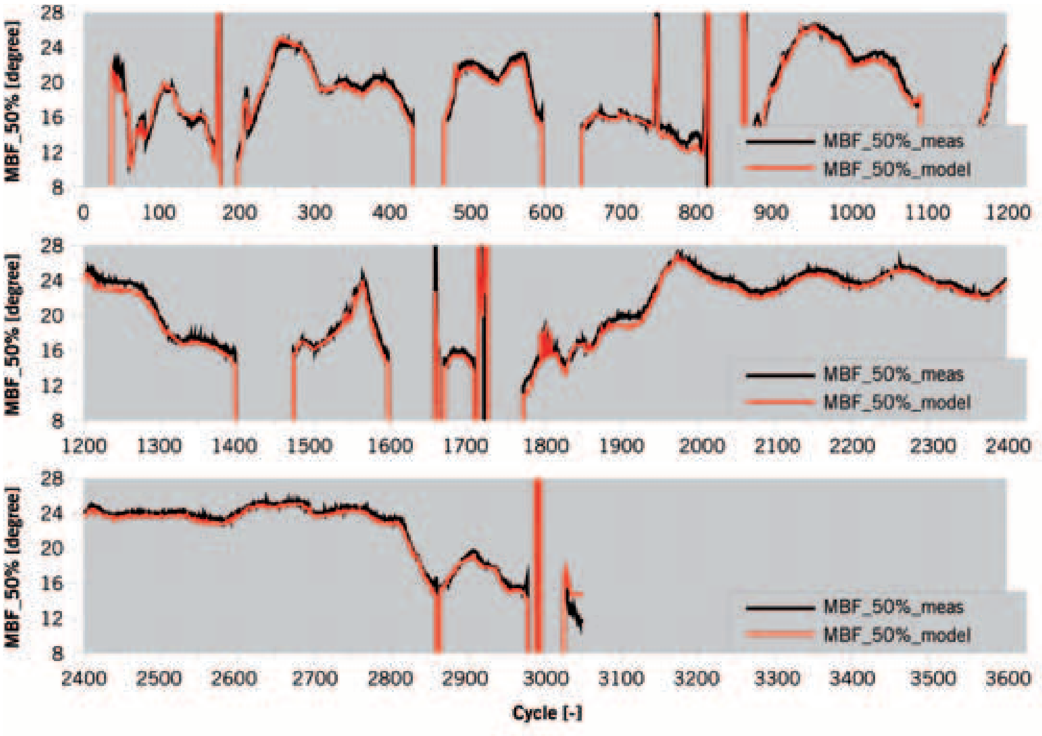
❺ Comparison of combustion pressure in FTP cycle for combustion cycle 659

files – A basis variant, two variants with variation of the start of the main injection, and variation of the position of the EGR valve. To check the generalisability of the ANN model application, the test engine was run with the production dataset in the FTP test cycle, without idle points.

COMPARISON OF COMBUSTION PRESSURE IN THE FTP CYCLE

Converting the operating points of engine speed and injected quantity from the dynamic driving profiles and the "FTP cycle without idle" into an engine map, ❸, it is possible to recognise in advance which areas of the map will supply little or no training data. With this missing data, there is a possibility of having greater errors in the model, but this is not necessarily the case. As can be seen in ③, only the operating points below 780 rpm lie outside the two dynamic test profiles. In the main region of the FTP cycle between 1700 to 2100 rpm, there is a very dense "net" of training information. The principle of operation of the smoke limit is very visible in the region between 800 rpm and 1600 rpm. The maximum fuel quantity, as defined by the full-load curve, is limited; the operating points of the FTP cycle run along the "dents" in the red lines, at which the smoke limit is active.

In ❹ and ❺, for the two highlighted operating points in ③, combustion cycles 66 and 659, the modelled and measured pressure traces are shown. With this overlay, the quality of the cylinder pressure model can be demonstrated. Even though the model inputs engine speed and cylinder mass lie outside the normal input range, a reasonable modelled pressure trace results. The peak pressure in the modelled trace of combustion cycle 66 is 1,3 bar higher than the measured



❻ Center of combustion (MBF 50 %) of modelled (red) and measured (black) combustion pressure in FTP cycle

trace, leading to a generally higher cylinder pressure level in the expansion phase, however, the end of combustion (MBF 90 %) is 0,5° advanced. The start of combustion (MBF 5 %) and the center of combustion (MBF 50 %) are, in spite of the inaccuracies in the model, within an acceptable tolerance of 2°. A small deviation of 1,8 dB(A) is observed between modelled data and measurement for the combustion noise. This is explained by the greater pressure rise in the early combustion phase, where the signal amplitudes are higher, therefore leading to a general increase of sound level.

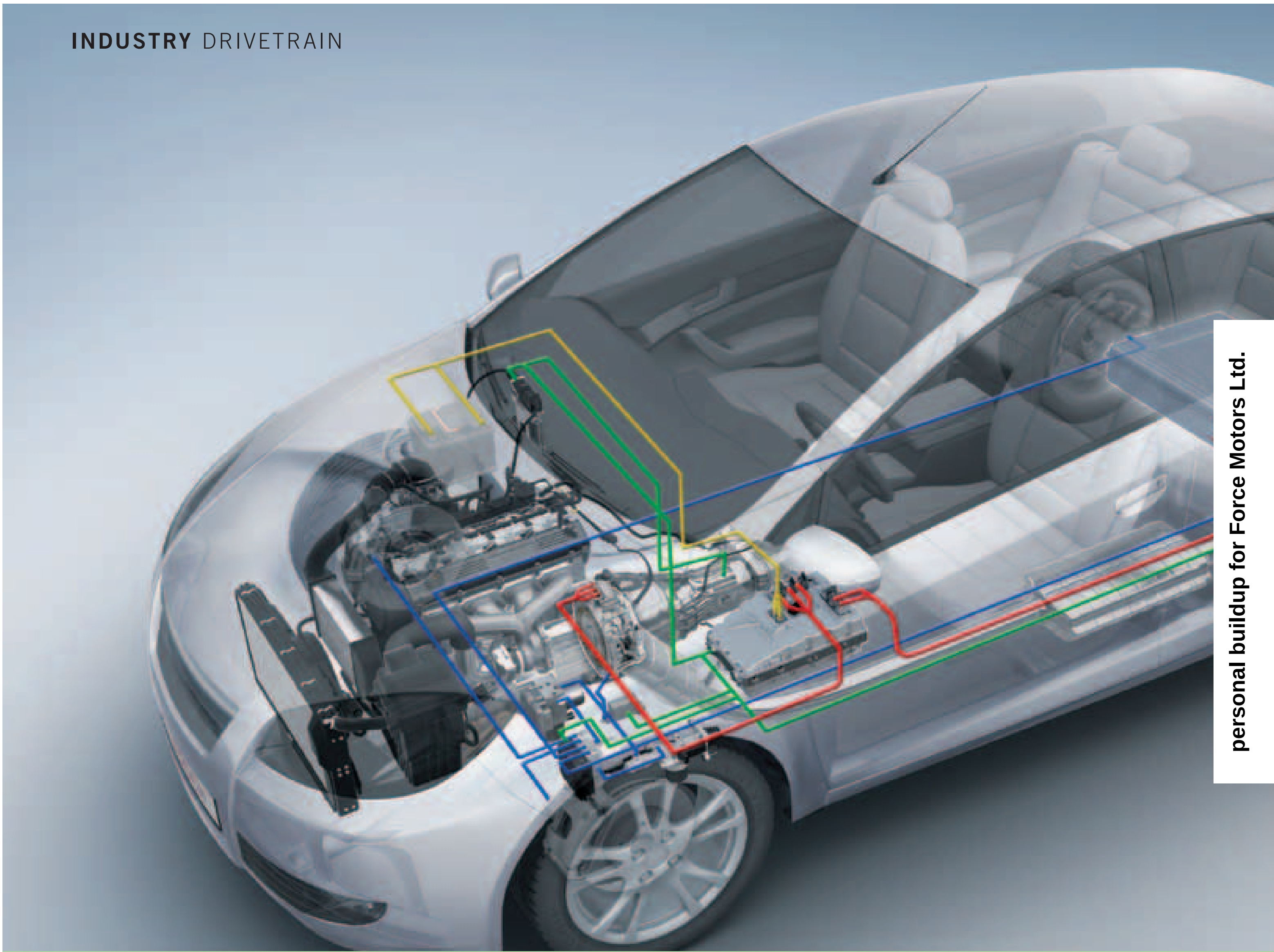
The pressure profiles in combustion cycle 659 lie nearly congruently, which is repeated in the quality of the combustion profile and corresponding thermodynamic data values. Due to the waviness of the energy release profile from the measured cylinder pressure from 60 to 90° after TDC, the modelled cylinder pressure at MBF 90 % has an error of 2.1°. All other thermodynamic values, comparing measured data to modelled data, lie within 1° of crankshaft angle to each other. There is a very good conformity of the two signal profiles in the acceleration and load change phases, (Combustion cycles 0 to 1900), and the static phase (combustion cycles 1900 to 3000). With very few exceptions, which can be explained by the spikes in the load-reversal in coasting in ❻, the median combustion point for the FTP cycle can be calculated using the model with an accuracy of ± 1°. The correlation coefficient between the measured data and the model is 0.992. Through further stationary measurements throughout the engine operating range (which could not be illustrated here), the accuracy of the implementation of the model was also confirmed.

SUMMARY

Future control strategies for combustion position control can be realised without the need for in-cylinder combustion pressure sensors, in which the model-based combustion control calculates the center of combustion position from other available engine sensors. The cylinder combustion pressure model demonstrated here offers the possibility not only to determine the median combustion position with very high precision, but the ability to infer other important thermodynamic parameters, which in turn can also be used as control values in the combustion process.

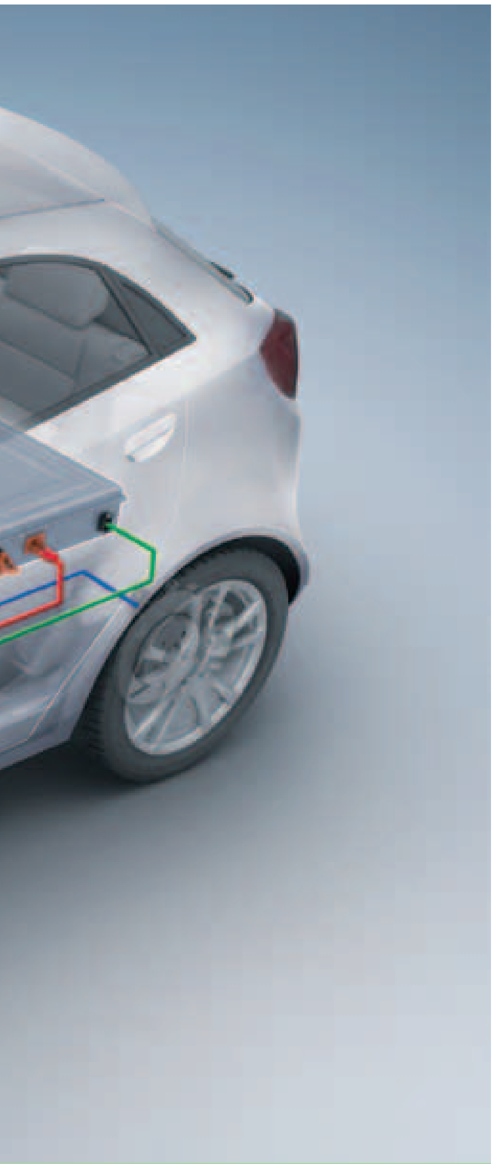
REFERENCES

[1] Marinell, G.: Multivariante Verfahren. Oldenbourg–Verlag, 1995
[2] Zell, A.: Simulation neuronaler Netze. Addison-Wesley, 1994
[3] Brause, R.: Neuronale Netze. B. G. Teubner Verlage, Stuttgart, 1991



OVERALL POWERTRAIN OPTIMISATION OF HIGHLY BOOSTED SPARK IGNITION ENGINES

Currently the driver for innovations concerning the powertrain itself is mainly the reduction of fuel consumption which is directly connected to meeting CO_2 emissions limits. This results in diverse solutions depending on the vehicle market and the engine class. Next to measures concerning a robust combustion concept design for extreme downsized engines, alternative transmission concepts and several hybrid vehicle architectures variants are also discussed. Bosch proposes for powertrains incorporating a spark ignition gasoline engine to build a bridge between the three main domains: internal combustion engine, transmission and electrification.



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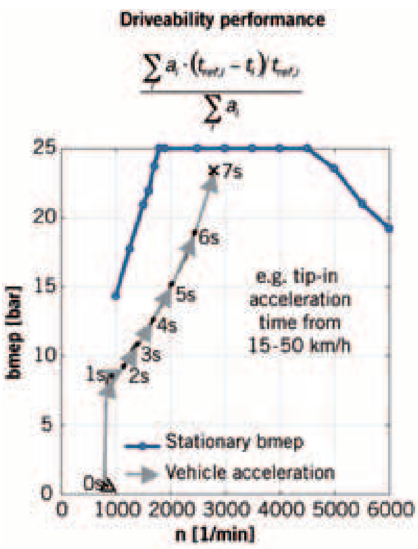
METHODOLOGY

Typical and for the future promising powertrain topologies are compared to each other on a simulation basis. The mutual coupling of measures in each domain show here different potentials of the technology packages. For optimisation purposes the mutual interdependency of these tasks has to be considered. This way a navigation map through the remarkably large number of combinations regarding structural alternatives (e.g. transmission AT, CVT or DCT) as well as with respect to concrete applications of specific parameters (e.g. downsizing-ratio, number of transmission steps or performance parameters of electrical machines).

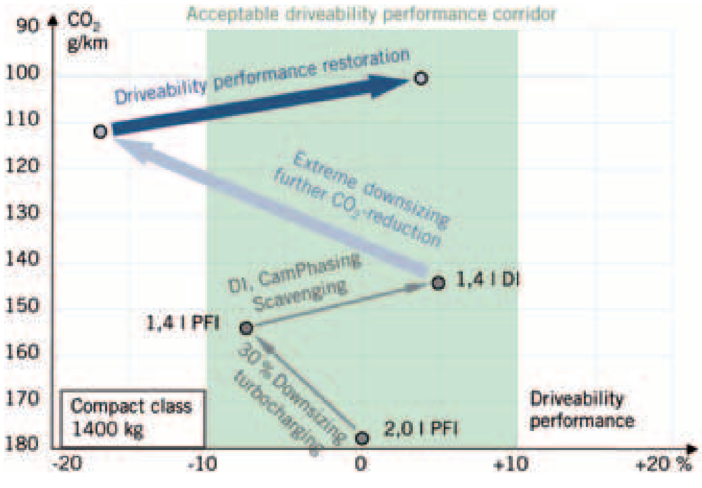
AREA OF COMPROMISE BETWEEN FUEL CONSUMPTION AND EMISSIONS REDUCTION AS WELL AS DRIVEABILITY PERFORMANCE

Clear criteria are needed for the objective evaluation of technology packages regarding CO_2 reduction.

For the determination of the fuel consumption, the velocity profile of the implied drive cycle plays a significant role. Known cycles in emission legislation are NEDC, FTP and JC. Additionally a large number of drive cycles were published claiming to allow for a more realistic benchmark regarding specific applications. In this work a compact class vehicle running in the NEDC will be addressed.



❶ Evaluation criteria for driving performance



❷ Interaction of driving performance with CO_2 emissions

When evaluating driveability of a vehicle, its market and respective manufacturer's demands play a fundamental role. For the sake of clarity, only three acceleration times are considered. Following tip-in accelerations times from a constant vehicle velocity to a higher velocity are considered: 15 to 50 km/h, second gear; 80 to 120 km/h, v_{max} gear; 80 to 120 km/h, third gear. As a scalar evaluation parameter these times are weighted, averaged and then used as a new parameter to quantify the percentage deviation with respect to the basic configuration [1], as shown in ❶. When a vehicle is accelerated from a nominal constant velocity, a clear difference can be recognized between its reachable steady state maximum torque and its dynamic behaviour. Additionally, ❷ shows fuel consumption and driveability performance for portfolio analysis comparison highlighting the interaction between the measures resulting from the different technology packages within the aforementioned domains.

Since there are no existing applications up to date for most of the following configurations the evaluation parameters presented for all configurations in this work will be determined by means of simulations.

As an approach for the fuel consumption simulation the so called reverse simulation [2] is used. For this approach the dependency between cause and reaction is reversed. Starting points are the desired acceleration and the actual vehicle velocity. With adequate models for vehicle, wheel/axle, transmission, clutch, electrical drive, battery and internal combustion engine the required operation points for internal combustion engine and electrical machine can be determined.

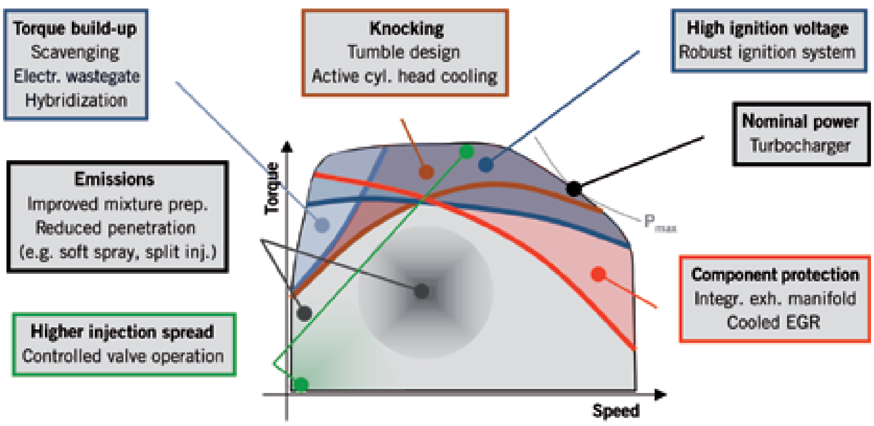
For the evaluation of the acceleration behaviour a simulation based on a description of the powertrain with ordinary differential equations is used which can be called a forward simulation to distinguish this method from the one used for fuel consumption. For engine simulation a 1-d model is used. Since the torque from the internal combustion engine and the electrical machine lead to vehicle motion changes, a driver model is needed.

INTERNAL COMBUSTION ENGINE MEASURES

The thermodynamic split of losses for a stoichiometric operated spark ignition engine shows the following main losses [3]: throttle losses, thermodynamic gas property losses of the air fuel mixture and friction losses. Future potential can be found in the reduction of the wall heat losses and a reduction of the burn duration.

The transition to a boosted engine with simultaneous displacement reduction applying gasoline direct injection and continuous camphasing is the dominant way for the technical realization of the previously mentioned effects. At an attractive cost/benefit ratio, throttle losses and, up to a certain downsizing ratio, the friction losses can also be reduced. Despite reductions in fuel consumption can be achieved, significant challenges with respect to oil dilution and full load efficiency have to be addressed. ❸ shows an overview of the challenges and potential solutions of specific areas within the engine operating map.

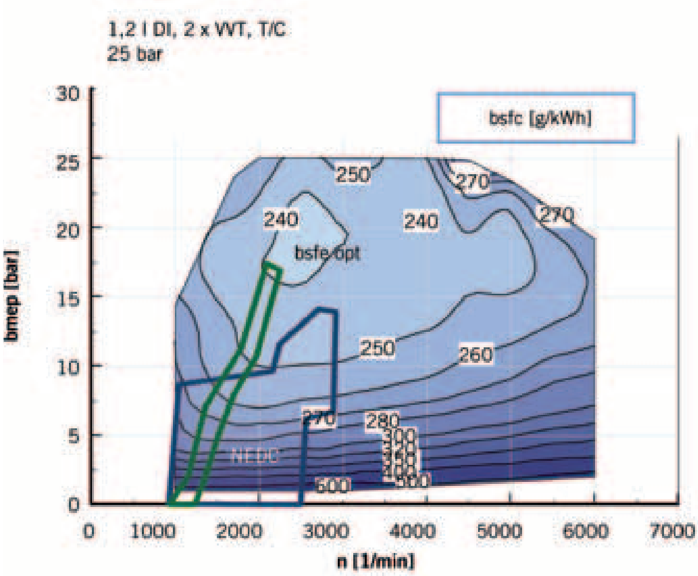
To further increase these potentials, a next step could be to consider a more refined variable valve train (variable valve lift) or a lean mixture strategy in certain operating map areas. There are still some lingering challenges that need to be ad-



❸ Challenges in gasoline engines with high boosting dressed for some topics within stratified lean operation such as exhaust gas after-treatment and useable operating map area.

Today's current measures for downsized concepts in compact class vehicles with downsizing ratios greater than 50 %, still suffer from the negative effect of turbo-charger lag (delayed response) due to their torque dynamics. Even the higher steady state torque levels cannot help mitigate this challenge.

Additional internal combustion engine measures (e.g. two-stage turbocharging, supercharging combined with turbocharging, electrical supercharger and other boosting concepts which are currently in a research stage) compete with technology packages of the transmission and electrification domains of the powertrain.



❹ Operating points in the NEDC with various transmission concepts

TRANSMISSION MEASURES

For comparison purposes of the powertrain combinations a uniform transmission design with respect to first and final gear ratio was chosen. As a criteria for the first gear ratio creeping (driving with idle speed at 6 km/h) and up-hill driving with trailer (10 km/h constant speed with 20 % slope) as typical driving situations were chosen, whereas both criteria have to be satisfied. For highly boosted small engines the power demand for the up-hill drive is the deciding criteria. Due to automatic gear downshift using automated transmissions, an acceleration advantage against conventional manual transmissions can be observed, when high comfort level and short shifting time of modern transmission technologies are taken into account.

The most significant effect with respect to fuel consumption can be realized with automatic transmission through operating point shifting towards higher loads, ❹. The resulting mechanical efficiency of the transmission and the actuation energy necessary to shift will have to be addressed as well. Automated transmissions were designed with respect to maximal transmission ratio spread with an overdrive in the highest gear which results in a significant fuel consumption advantage and an increased comfort level at part load. Another possibility for downspeeding is a hydraulic torque converter as driveaway element (6AT, CVT). This way, a higher available torque allows for the first gear design to be 15 % longer taking into account thermal losses.

ELECTRIFICATION

With the introduction of additional electrical energy storage and an electrical machine as well as the required power electronics, mechanical energy of the combustion engine and brake energy can be converted and stored, as well as converted back to mechanical driving energy later on. Hereby, another efficiency increase of the powertrain can be realized over an internal combustion engine displacement reduction (Torque Boost) and the operation in optimal map areas with respect to fuel consumption. Additionally, the brake energy can be recuperated.

A disadvantage can be found in the increased weight and the higher packaging demands as well as the low energy content of the battery compared to the additional cost.

A description of the different hybrid vehicle architectures won't be presented here since these are already satisfyingly presented in literature [4]. ❺ shows a succession of the different electrification measures within the powertrain from a system as well as from a function point of view. The various possibilities for the arrangement of the aggregates and the choice of the technical parameters (power, torque, energy content, etc.) lead to wide possible solutions for different requirements within each vehicle class and market. Key to a commercial success is therefore an adequate component package set from the supplier; ⑤.

		mHEV	sHEV	PHEV	EV
Power range		60-180 V 5-15 kW	250-400 V 20-60 kW	250-400 V 40-80 kW	≥400 V 30-150 kW
Functionality	ICE Downsizing	0	+	++	n/n
	Recuperation	+	++	++	++
	Torque assist	+	++	++	n/n
	Electric driving	- (– parking)	0 (~2-5 km)	+ (~25 km)	++ (100-150 km)
	Fun to drive	0	+	++	0/+++
Bosch Components	E-machine				
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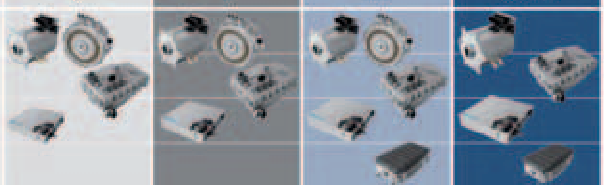
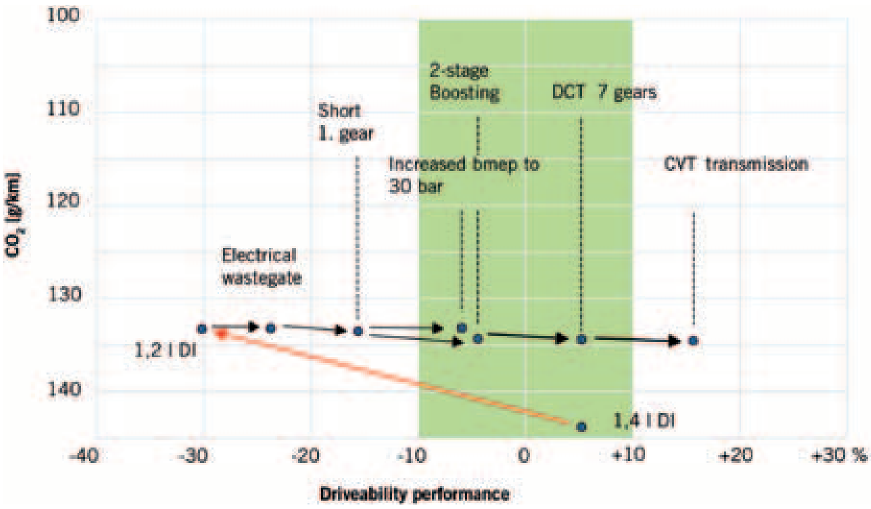
	Battery				
	Charger				



❺ Levels of powertrain electrification measures

❻ Portfolio fuel consumption/driveability performance with internal combustion engine measures and transition to automatic transmission



ATTRACTIVE POWERTRAIN ARCHITECTURES

As already shown in ② the loss in drivability that comes with a downsizing ratio of roughly 30 % for a PFI engine, could be compensated by the use of scavenging with turbocharging (typically with gasoline direct injection and cam phasers). As a next step an engine with a downsizing ratio of 40 % is shown in ❻. It was assumed that this engine is optimized and therefore for example fitted with a cylinder head integrated exhaust manifold and an electrically actuated wastegate. Nevertheless attenuation in drivability performance could occur and the question has to be posed as to how this disadvantage can be resolved without compromising the gained improvements in fuel consumption.

To solve this question a combination of engine, transmission and electrification variants were investigated. Therefore 420 variations were simulated (14 engines, six transmissions and five hybrid variants) [1]. This work constitutes a representative sample of these combinations.

Comparisons of engine, transmission and electrification variants shown in ⑥ do not account for additional measures to reduce fuel consumption such as weight reduction, low friction tires, thermal management, vehicle electrical system management, start stop (already included in hybrid concepts), etc. Therefore, the choice of concept does not relate to currently available market concepts but in return offers a more objective comparison. All statements are referenced to a naturally aspirated (n.a.) four-cylinder with port fuel injection (PFI) with a displacement volume of 2.0 l integrated in a compact class vehicle (1400 kg).

At downsizing ratios of up to 40 %, increasing the mean effective pressure or implementing a two-stage turbocharger, as an internal combustion engine measure, shows a positive impact to reach desired driveability performance. For higher downsizing ratios these methods reach their limits. Additional challenges have to be taken into account, for example for two-stage turbocharging like the gap losses of the small turbocharger as well as the mechanical engine design for higher mean effective pressures and engine cooling and the ability to handle with increasing ignition break down voltage.

Without further measures taken on engine side the original drivability levels can be reached with modern automatic transmissions. ⑥ shows exemplarily the impact of dual clutch transmission (DCT with seven gears, dry clutch, gear spread 6.5) and a continuously variable transmission CVT (gear spread 6.5).

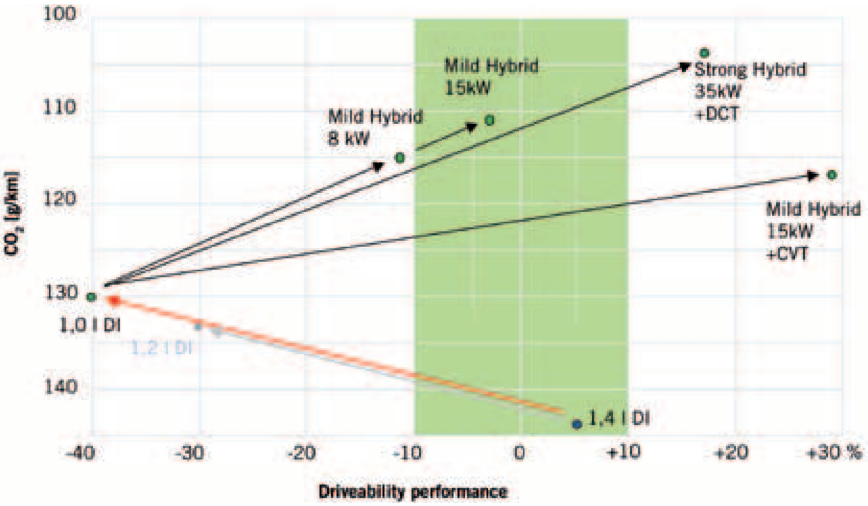
Electrical hybridization with an electrical machine of roughly 15 kW as a Mild-Hybrid is technically possible and immediately attaining satisfactory driveability performance levels. The sheer cost of this technology becomes a challenge despite the substantial CO_2 reduction achievable as observed in ❼. Hereby, boost functions coupled with hybridization could be used to further increase the downsizing ratios of the internal combustion engine in which the features of hybridization measures are based on an internal combustion engine with only 1.0 l displacement volume. Hybridization with 8 kW already delivers a considerable fuel consumption reduction whilst slightly reducing driveability performance.

Even more demanding, and outclassing the reference concept with respect to driveability, is the application of a strong hybrid whereas the use of a 35 kW electrical machine in combination with an automatic transmission (here DCT) was taken into account. Comparing the different hybrid concepts in combination with different transmissions types was investigated. A CO_2 emissions advantage can be found for transmissions with high efficiency while the operating point shifting potential plays a minor role.

Since the combination of mild hybrid with a CVT significantly exceeds the drivability performance requirements, one can pose the question whether a reduction in driveability performance to the reference level, would translate into a cost reduction for this combination. Especially for the mild hybrid with manual transmission significant CO_2 reduction potential can be reached, which otherwise would be less for the version with CVT due to the aforementioned reasons.

CONCLUSIONS

When developing an internal combustion engine downsizing concept, a careful choice is required in selecting the appropriate technology packages to ensure the demanded driveability level is met. Many factors, some



❼ Portfolio fuel consumption/driveability performance with hybridization and extreme downsizing

of which are not entirely predictable, play a very influential role in the future implementation of the technology such as:

: the market with customer preferences, willingness to pay, legislation boundary conditions as well as application specific requirements (City Tolls, amortisation considerations, mileage)
: the typical or the relevant driving cycle respectively
: the vehicle and engine class
: the technological advances in vehicle weight, rolling and aerodynamic resistance.

While these measures are occurring in the background to reduce the cost of hybrid electric vehicles, they compete directly with other technical developments to achieve highly boosted concepts (extreme downsizing) with further displacement reduction. Even if the latter achieves an increase in steady state torque while meeting transient torque requirements, the gains in driveability performance cannot be met for further fuel consumption reduction by means of larger transmission gear ratio spreads. Therefore an optimized combination of technology packages for engine, transmission, and electrification has to be found while continuing to meet the demands and requirements of the vehicle's market segment and class.

Simulations that include CO_2 and driveability as well as cost considerations under the constraint of specific boundary conditions help in the definition of these optimised concepts. Components and systems suppliers, such as Bosch, can provide the much needed base building blocks to handle the variety of required technology packages. Equipped with know-how in engine and transmission controllers as well as electrification, Bosch can certainly shine some light on the solution.

REFERENCES
[1] Berns, S.; Hammer, J.; Benninger, K.; Klenk, M.; Frei, R.: Reduzierung der CO_2-Emissionen bei guter Fahrbarkeit. 31. Internationales Wiener Motorensymposium 2010
[2] Mann, K.; Walter, M; Kustosch, M.; Endres, J.; Huber, T.: Modellgestützte Entwicklung von Komponenten für 4WD-Hybridfahrzeuge. 6. Grazer Allradkongress 2005
[3] Kulzer, A.; Christ, C. ; Rauscher, M.; Sauer, C.; Würfel, G.; Blank, T.: Thermodynamic Analysis and Benchmark of Various Gasoline Combustion Concepts. SAE World Congress 2006, Detroit
[4] Huber, T.; Richter, B: Modellgestützte Hybrid Systementwicklung – Grundlagen und Anwendung. 7. Symposium "Steuerungssysteme für den Antriebsstrang von Kraftfahrzeugen", Berlin 2009

THANKS

The authors are grateful to Thomas Huber and Christian Wiegand for their valuable contribution to the simulation work, as well as Dr. Juergen Hammer for promoting and accompanying this paper.

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12 I 2010 _ December 2010 _ Volume 71
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NEW APPROACH TO TURBOCHARGERS FOR FOUR-CYLINDER GASOLINE ENGINES

At the chair for internal-combustion engines at the Technische Universität Dresden a new turbocharging approach was developed in which variable exhaust valve timing with a dual exhaust manifold is combined with a staged turbocharging system. The essential advantages of this approach are an improvement in the scavenging behavior, a reduction in the specific fuel consumption as well as substantially more spontaneous torque buildup.

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MTZ
PEER REVIEW
RECEIVED 2010-06-30
REVIEWED 2010-08-02
ACCEPTED 2010-09-16
REVIEWED BY EXPERTS FROM RESEARCH AND INDUSTRY.

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1 INTRODUCTION

Turbocharging is one of the key technologies that have crucially influenced the development of the internal-combustion engine in past years. While the enhancement of absolute performance originally stood in the foreground, the current focus is unambiguously on the increase of the specific characteristic data. The decisive motive for the change in design philosophy was and is the demand for low CO_2 emissions. The strategy known as downsizing, which is particularly commonly employed with gasoline engines – unchanged performance with lower engine displacement – offers crucial advantages in comparison to rival approaches. The higher efficiency and consequently the lower fuel consumption/lower CO_2 emissions are evident not only in the certification-relevant driving cycle, but can also be experienced in real operation by users. At the same time, this kind of gasoline engine, because of its unaltered exhaust aftertreatment with a 3-way catalytic converter and its mostly simple system components, maintains the better part of its cost advantage against the diesel engine.

For an optimally efficient decrease in inner friction, the reduction of engine displacement called for by the downsizing approach includes lowering the cylinder count. As a result of this, the four-cylinder will be employed in the future as the engine for all vehicle classes. The widely diverging goals of these applications – from "performance" engines in light compact cars to "eco"-motorization in heavy luxury-class vehicles – inevitably call for differing master plans. If the entire spectrum is to be successfully covered with a single basic engine, then solutions must be found that go beyond differentiation through the turbocharging system. In any case, the technological modular concept must be expanded with systems that consider the idiosyncrasies of the four-cylinder in the gas-exchange process.

2 CORE PROBLEM AND STATE OF THE ART

The underlying principle of turbocharging is the direct coupling of a turbine located along the exhaust stream with a compressor for precompression of the air required by the engine using the residual energy contained in the exhaust gas. The consequences of a purely thermodynamic coupling with the basic engine, which are sufficiently well-known, are on the one hand the delayed boost pressure buildup occurring with a spontaneous load requirement, and on the other hand the retroaction on the gas-exchange process that accompanies buildup of exhaust gas at the turbine. The latter point is particularly crucial for the four-cylinder engine, since the elevated exhaust back pressure not only increases the work necessary to purge exhaust from the cylinder, but affects the efficiency of the charge exchange to a considerable degree.

When one considers the cylinder counts common in passenger car drivetrains, the cases displayed in ❶ can be distinguished on the basis of the resulting firing interval and the opening period of the exhaust valve. The event duration of the corresponding camshaft was here set at 210°, a common compromise between maximum utilization of the power stroke and minimal charge-cycle work.

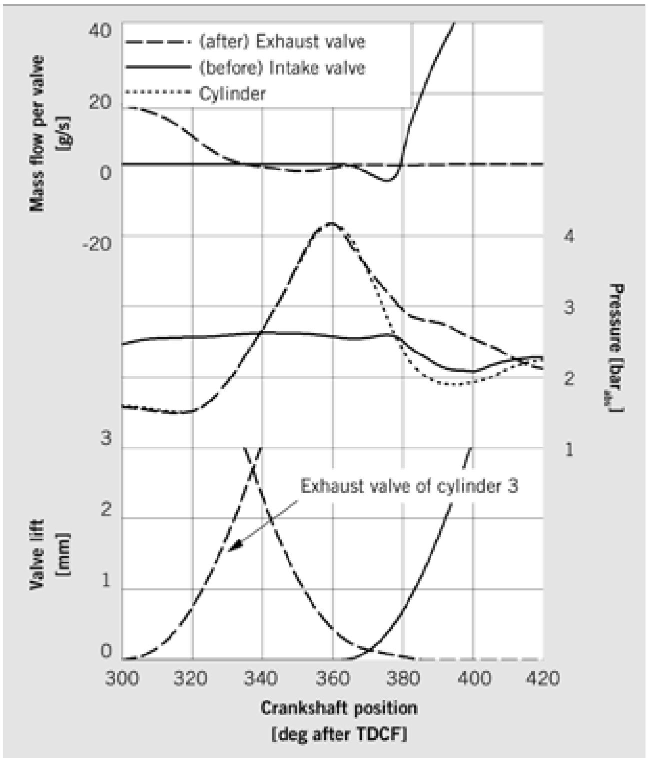
For a cylinder count ≤ 3, due to the lack of overlap in the exhaust strokes of the individual units, no retroaction on the process is to be expected from immediately joining the exhaust streams. All other aggregations display a more or less pronounced overlapping in the exhaust stroke. ❷ makes clear the effect on the cylinder's scavenging behaviour.

As can be gathered from ①, with a conventional design, the four-cylinder engine exhibits an overlap of the exhaust opening of two cylinders of approximately 30° crank angle. As a result, the rise in pressure in the exhaust system owing to the blowdown of the next cylinder in the firing order reaches the scavenging cylinder shortly before the closing of its exhaust valve. The scavenging gradient above the cylinder at this point is strongly negative even in case of a clearly positive intermediate gradient, thus barring active scav-

	UNIT	$\alpha_{FI} > \alpha_{EV,\,1\,mm}$	$\alpha_{FI} < \alpha_{EV,\,1\,mm}$	$\alpha_{FI} << \alpha_{EV,\,1\,mm}$
Cylinder count	[-]	2, 3	4	5, 6
Firing interval	[deg]	360, 240	180	144, 120
Exhaust stroke overlap	[deg]	–	30	66, 90

$\alpha_{EV,\,1\,MM}$: exhaust valve opening duration at 1 mm
α_{FI}: firing interval

❶ Exhaust stroke overlap for various cylinder counts



❷ Exhaust-gas-turbocharged four-cylinder engine – gas-exchange process at low engine speed and high load for cylinder 1 enging of the residual gas. Rather, reverse flows actually come about from the exhaust manifold. As a consequence, in the design of the valve timing, overlapping of inlet and exhaust is forgone or minimally implemented. For a yet greater number of coupled cylinders and the increasing overlap of the individual exhaust strokes that goes along, the effect on the gas-exchange process shows itself somewhat less drastically. Particularly in the case of a five-cylinder engine, this behaviour can still be purposefully influenced over a moderately extended exhaust event [3].

The equation (Eq.) for the brake mean effective pressure shows the direct influence of the volumetric efficiency on this characteristic value of the internal-combustion engine (four-stroke) with a direct proportionality, all other parameters and constraints being equal.

EQ.	$p_{me} = H_u \cdot \eta_e \cdot \frac{\lambda_l \cdot \rho_L}{L_{min} \cdot \lambda}$

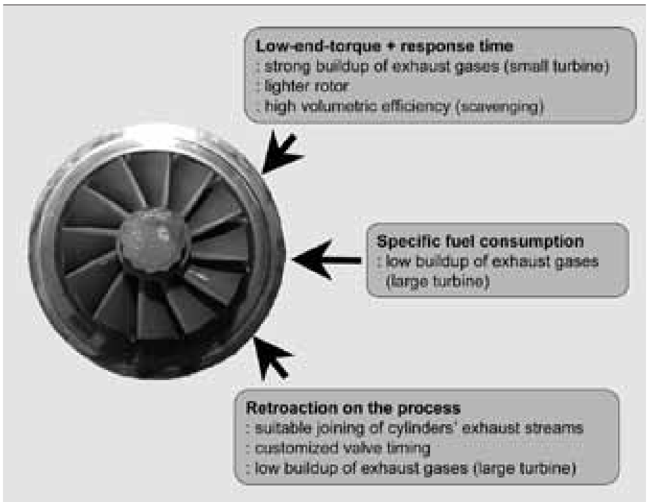
EQ.	
p_{me}	Mean eff. pressure
λ_l	Volumetric efficiency
ρ_L	Air density
H_u	Lower heating value
η_e	Real efficiency
L_{min}	Stoichiometric air requirement
λ	Lambda

In consideration of the demonstrated specific constraints of the four-cylinder, until now two different approaches to a solution have been pursued: on the one hand shortening exhaust-side event lengths (shortened exhaust event [4]), and on the other hand separation into a 2+2 aggregation of the cylinders with a corresponding exhaust manifold/turbine configuration (twin flow [5]). Both techniques, in comparison to the conventional design, have led to a substantially more effective gas-exchange process and, as a further consequence, to improved performance at low engine speeds. Both approaches, however, have the intrinsic disadvantage of more charge-cycle work at high engine speeds, which eventually results in higher fuel consumption. This disadvantage can be avoided if the procedures mentioned are designed suitably to the needs of each case (exhaust-event length [6]/transition between pulse and constant pressure).

It should nevertheless be noted that despite these measures not all the disadvantages of turbocharging in general and in four-cylinder engines in particular have been or can be solved. Particularly against the background of a yet more systematic downsizing, with the higher degree of turbocharging that must necessarily go along, new approaches must be found. This pertains on the one hand to dynamic behaviour, which, as in the naturally aspirated engine, should not exhibit any noticeable delay in the stationary torque buildup, and on the other hand to improving the efficiency of the turbocharging system as a whole, with the goal of decreasing full-load consumption, particularly at rated output.

The fundamental problem of supercharged passenger car engines can be principally attributed to the following three points:

: low torque at low engine speeds, furthermore – depending on operating point – somewhat lethargic dynamic behavior



❸ Requirement profile for turbocharging

: partially lower efficiency of the turbocharging unit with a corresponding effect on the overall efficiency, particularly with the use of a wastegate to control the power of the turbine

: undesired retroaction on the process of the internal-combustion engine.

The design criteria for the turbocharger derivable from the respective points are diametrically opposed to each other, ❸, in some respects. To the above-mentioned points should be added the universally valid requirement of maximal efficiency of each individual component.

Most of the contrary demands outlined here result from the radically different flow rate characteristics of the coupled machine types, the piston engine as a displacement engine and the turbocharger as a combination of two continuous-flow machines, as well as the previously mentioned purely thermodynamic coupling.

3 VARIABLE EXHAUST-VALVE TIMING WITH SEPARATED EXHAUST CHANNELS

The goal of the project was the development and analysis of alternatives for turbocharging of a direct-injection, four-cylinder gasoline engine conforming to the following criteria:

: clear improvement of the scavenging behaviour to increase performance in the lowest range of engine speeds

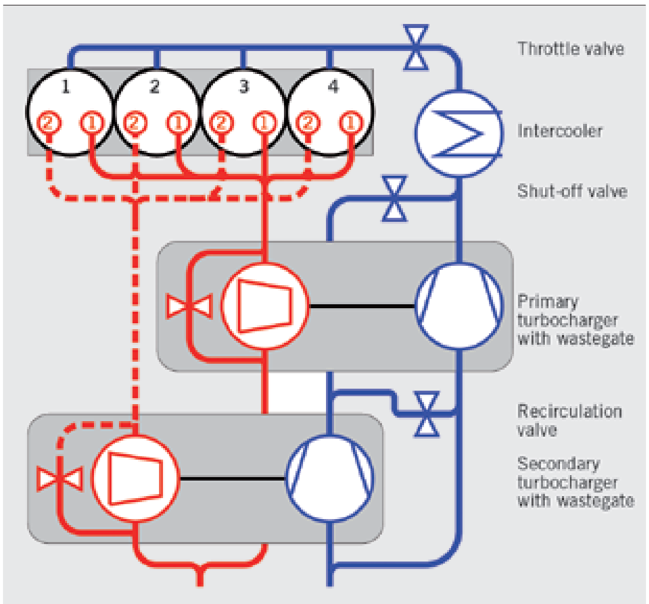
: reduction of the specific fuel consumption at rated power.

The fulfilment of these basic requirements requires measures to configure the gas-exchange process as well as adjusting the turbocharging unit to the particular operating point [1].

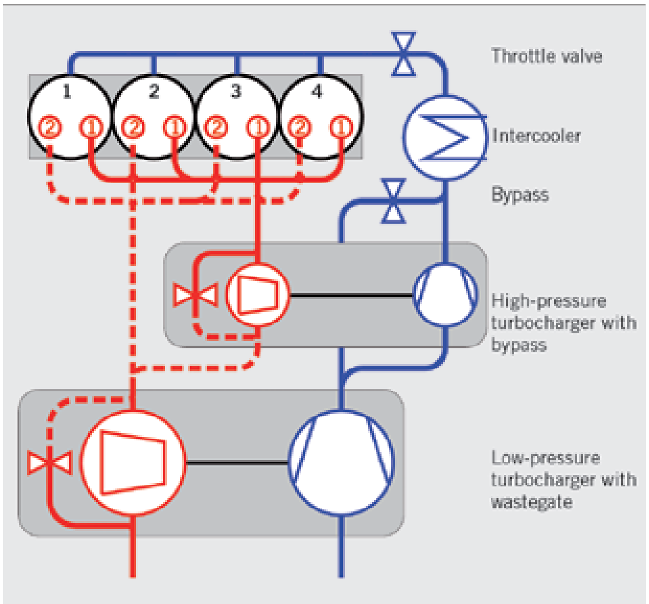
3.1 SYSTEM CONCEPT

As already stated, a shortened exhaust event also facilitates a positive scavenging gradient across the cylinder in a four-cylinder during valve overlapping. However, this solution at the same time increases the work necessary to purge exhaust gas from the cylinder at high engine speeds.

A solution to this conflict is possible only through valve timing with a variable opening duration. The simplest option in this case is simultaneous variation of both exhaust valves running furthermore synchro-



❹ Sequential turbocharging with variable exhaust valve timing and separated exhaust channels – schematic



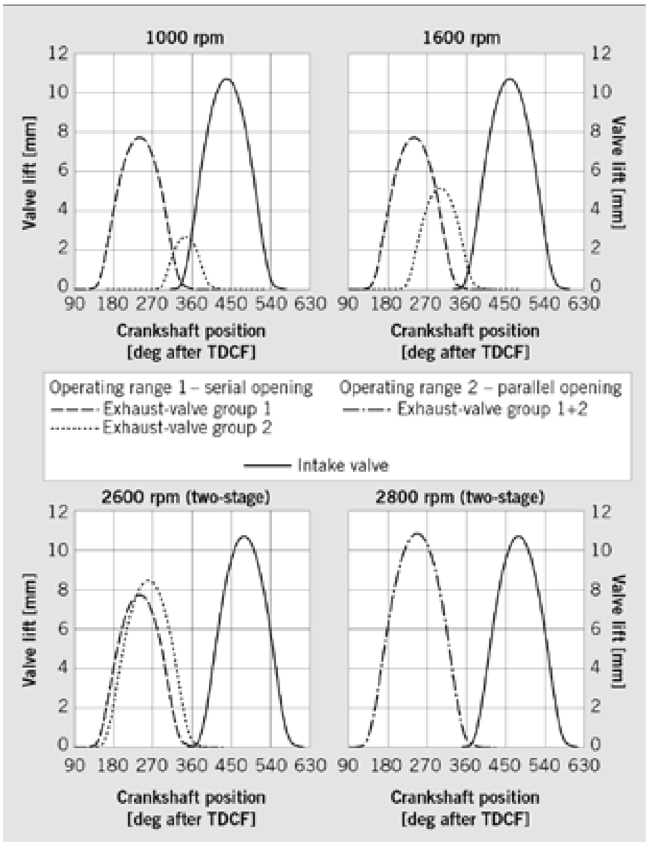
❺ Two-stage-regulated turbocharger with variable exhaust valve timing and separated exhaust channels – schematic

nously [6]. If separated timing of the exhaust valves is also included in this consideration (in combination with separated exhaust lines), further possibilities can be exploited. Aside from the potential separation of turbine admission from the scavenging process, separate and variable adjustment of the exhaust valves allows temporal and quantitative distribution of the exhaust stream and the enthalpy contained in it. This function poses an ideal point of approach for staged turbocharging systems, whose fundamental idea is to adapt the turbocharging unit to the operating point of the engine. The most important quantities here are the exhaust-buildup behaviour of the turbine(s) and the flow-rate of the compressor(s), including the associated efficiencies and polar moments of inertia of the components. The operation of staged turbocharging systems is determined to a considerable extent by the distribution of the available exhaust-gas enthalpy between the turbines. In conventionally implemented approaches (two-stage, sequential) this functionality is realized via a valve in the exhaust-gas system. When one compares the opening characteristics of this valve (closed in the lowest engine speed range, open in the highest) with the operating states of a sequential and variable timing of the exhaust valves (increasing overlap of the opening cross-sections with rising engine speed), the synergy arising from the combination of both processes becomes clear. That is, the quantitative distribution of the exhaust mass flow necessary in staged systems is combined with the time-periodic distribution of serially opening exhaust valves by feeding the exhaust-gas enthalpy flow of both exhaust valves into the staged turbocharging system at differing positions [2].

❹ and ❺ show the system schemata of the staged turbocharging approaches relevant for passenger-car gasoline engines – the parallel operation of two turbochargers in the sequential form, as well as their serial arrangement as a two-stage-regulated turbocharging group.

As can be seen from the schematics, the exhaust mass flows from one cylinder's two exhaust valves are segregated between separate exhaust manifolds, each of which feeds into a turbine. ❻ shows the principal progression of intake- and exhaust-side valve timing.

In the first operating range, beginning with the lowest engine speed (⑥ – 1000 rpm) under stationary engine conditions a maximum of the available exhaust-gas enthalpy is fed to either the primary turbocharger (sequential) or the high-pressure charger (two-stage) until the rated mean pressure is reached by setting a



❻ Variable exhaust-valve timing – valve timing strategy

PROPERTY	UNIT	EXPERIMENTAL VEHICLE/ MODEL ENGINE
Displacement	cm³	1984
Stroke	mm	92.8
Bore diameter	mm	82.5
Compression	–	9.6
Rated output	kW	155
Rated brake mean effective pressure	bar	22
Maximum turbine inlet temperature	°C	1050
Mode of injection	–	homogenous

❼ Experimental vehicle – mechanical characteristic data

minimal valve event length at exhaust-valve group 2. At the same time, a sufficiently large overflow cross-section is available for active scavenging of residual gases over the late valve event position and an early adjustment of the inlet camshaft. An additional advantage of the selected valve timing is that the exhaust mass flow in exhaust manifold 1 is free of scavenge air, which facilitates direct measurement of the effective air-fuel ratio in the cylinder. Both points – effective purging of residual gas as well as the possibility of employing a turbocharger that is small in comparison to the basis – are of fundamental importance for a substantial increase in the stationary as well as the transient performance in the lowest speed range.

When the rated mean pressure is reached, a larger portion of the exhaust-gas enthalpy flow is fed into the second turbocharger (sequential: secondary turbocharger; two-stage: low-pressure turbocharger) (⑥ – 1600 rpm) by opening exhaust-valve group 2 earlier, in order to limit a further increase in torque. Parallel to this, the valve overlap is continuously retracted. This process proceeds continuously with increasing mass flow rate through the engine, until the turbocharger corresponding to exhaust-valve group 2 has reached an operating state that allows a complete (two-stage) or partial (sequential) takeover of the compression work (⑥ – 2600 rpm). A significant difference between the variants arises from the contribution of the switchable charger to the compression of the fresh air taken in by the engine in the first operating range. While the low-pressure stage in the two-stage-regulated layout assumes a continuously rising portion, the secondary compressor in sequential turbocharging plays no role in the compression process in the entire first operating range.

With the transition of the staged turbocharging systems into the second operating mode (sequential: parallel operation of both turbochargers; two-stage: phasing out of the high-pressure stage), the necessity of separate timing of the exhaust valves disappears, so a transition begins at this point to synchronous timing of the exhaust valves (⑥ – 2800 rpm). The selected valve lift curve (event position, event length) corresponds to that occurring in the conventional process. To switchover between the operating modes, on the fresh-air side the secondary compressor in the sequential variant is integrated into the provision of charge air by simultaneously closing the recirculation valve and opening the shut-off valve; in the two-stage-regulated system a parallel bypass is opened to avoid flow losses in the coasting high-pressure compressor. In both variants, regulation of the degree of turbocharging in the second operating range is done, as in the underlying systems, by diverting part of the exhaust-gas enthalpy flow.

3.2 CALCULATION RESULTS

All the findings presented below were calculated with the module GT-Power of the calculating system GT-Suite. The basis of the computation model is a four-cylinder gasoline engine with direct gasoline injection, ❼.

In addition to those modelling assumptions, which are independent of the selected layout principle of the turbocharging unit, fundamentally differing conditions for the dimensioning of the turbocharger arise with sequential and two-stage turbocharging.

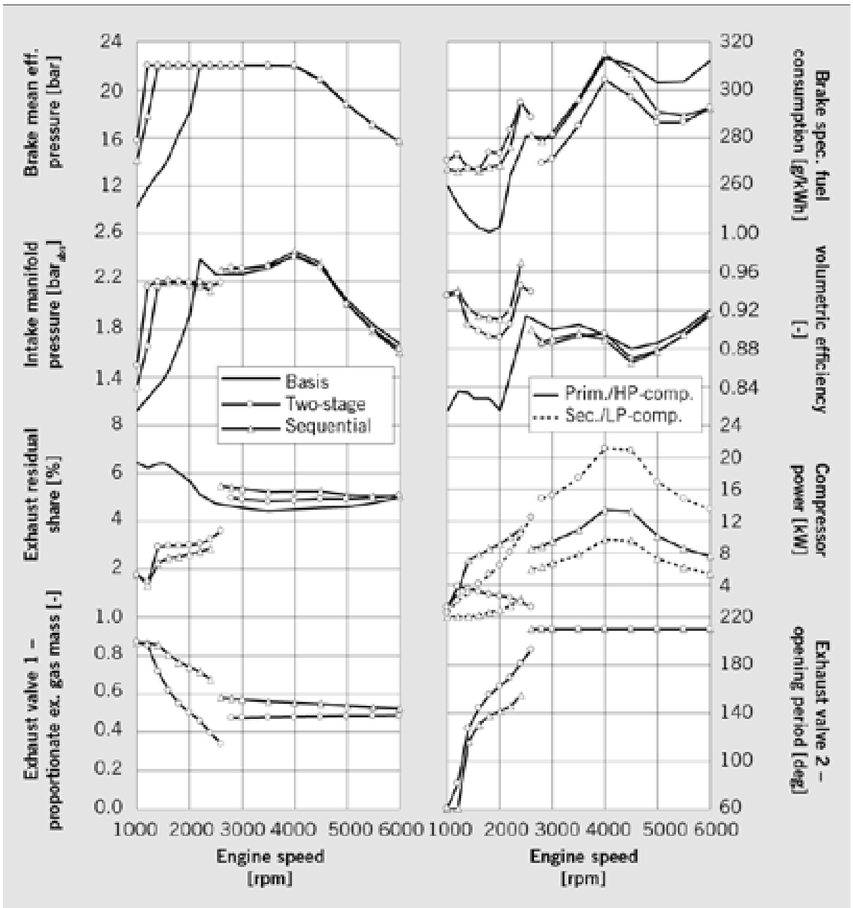
The dimensioning of the corresponding turbochargers with a serial layout (two-stage turbocharging) arises on the one hand from the fact that the high-pressure stage is primarily responsible for dynamic torque buildup in the lowest speed range, and on the other hand that in the upper load range the low-pressure stage fully takes over the compression work. The border for the ratio of min and max flow of the stages is determined in large part by the desired system properties in the middle speed range, in which, particularly in the transient range, there exists a danger of tamping or over-revving the high-pressure turbocharger; this is caused by a delay in running up the low-pressure stage, which is substan-

PROPERTY	UNIT	HIGH-PRESSURE STAGE		LOW-PRESSURE STAGE	
		COMPRESSOR	TURBINE	COMPRESSOR	TURBINE
Relative flow rate of the basis turbocharger	%	60	60	110	140
Outer diameter	mm	41	34	55	53
Polar moment of intertia	kgm²	$1.6*10^{-6}$	$2.7*10^{-6}$	$7.3*10^{-6}$	$2.2*10^{-5}$

❽ Two-stage turbocharging – characteristic data of the turbochargers

PROPERTY	UNIT	PRIMARY STAGE		SECONDARY STAGE	
		COMPRESSOR	TURBINE	COMPRESSOR	TURBINE
Relative flow rate of the basis turbocharger	%	70	70	50	50
Outer diameter	mm	44	37	37	32
Polar moment of inertia	kgm²	$2.4*10^{-6}$	$3.9*10^{-6}$	$1.0*10^{-6}$	$1.7*10^{-6}$

❾ Sequential turbocharging – characteristic data of the turbocharger



❿ Full-load curve – sequential vs. two-stage vs. basis

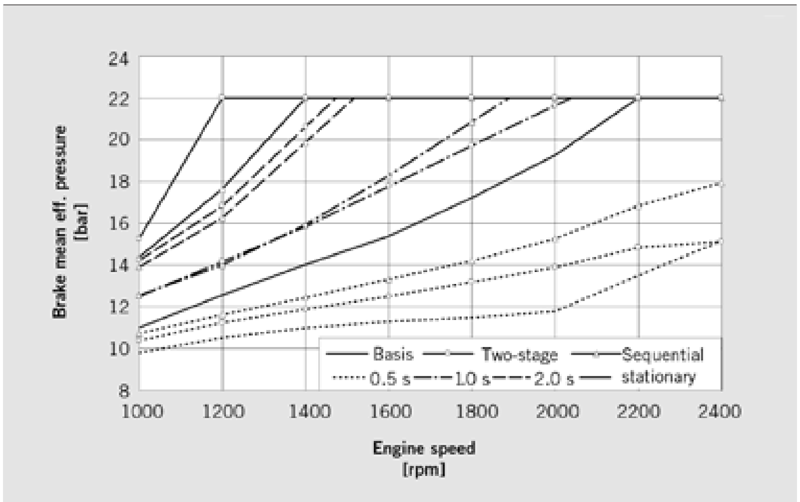
tially more sluggish [7]. To further investigate the two-stage-regulated variant, the dimensioning of the continuous-flow machines presented in ❽ was chosen.

Substantially more difficult than the design of the two-stage variant's turbochargers is the definition of suitable component sizes for the sequential layout. The particular challenge of this variant is the realization of a harmonious transition from 1-turbocharger to 2-turbocharger operation. Here one must also ensure that under transient operating conditions and in air recirculation mode, the switchable secondary turbocharger can be accelerated to the required speed for connection by the time the transition point is reached, and that after being connected it arrives at an operating point sufficiently far from the pumping limit. In the course of design calculations it was ascertained that a transition without any disadvantage to the torque delivered by the engine is not realizable with a symmetrical layout of the turbocharger. For further consideration of this variant, the asymmetrical layout of primary and secondary turbochargers presented in ❾ was chosen.

❿ shows a juxtaposition of the calculated full-load characteristic data of both variants in comparison with the basis.

As the curve of the brake mean effective pressure shows, both variants facilitate a substantial increase in the lowest speed range. The sequential variant reaches the target value of 22 bar at an engine speed of 1400 rpm, a decrease of 800 rpm compared to the basis. The two-stage variant affords an even more marked improvement in the stationary low-end torque, reaching the target value at 1200 rpm, i.e., another 200 rpm sooner. The further increase compared to the sequential variant can be partially ascribed to the high-pressure turbocharger's turbine, which dams up more gas, while the downstream low-pressure turbine also facilitates further utilization of the enthalpy still contained in the exhaust gas after the high-pressure stage. The advantage in the brake mean effective pressure available under stationary conditions to the new variants, at 1000 rpm compared with the basis (10.3 bar), amounts to, respectively, 37 % (sequential: 14.1 bar) and 52 % (two-stage: 15.7 bar). This considerable gain can only partially be attributed to the higher intake manifold pressure (sequential: +18 %; two-stage: +34 %), but is also the result of a substantially more efficient gas-exchange process.

As engine speed rises (once rated mean pressure is reached), an increasing portion of the exhaust gas mass is conducted through the valves in group 2 by increasing the opening period of these valves, in order to limit the boost pressure (two-stage: direct to the low-pressure turbine; sequential: the secondary turbocharger turbine). As a result of the turbocharger layout, as engine speed increases in the first operating range, sequential turbocharging displays continuously rising power requirements at the primary compressor, which is assigned to exhaust valve group 1 (exhaust manifold 1). In contrast, the power requirements of the high-pressure compressor in the two-stage variant sink with increasing engine speed, because an ever-greater portion of the compression work is taken over by the low-pressure stage. In the two-stage



⓫ Sudden load increase – sequential vs. two-stage vs. basis, brake mean effective pressure dependent on time and engine speed

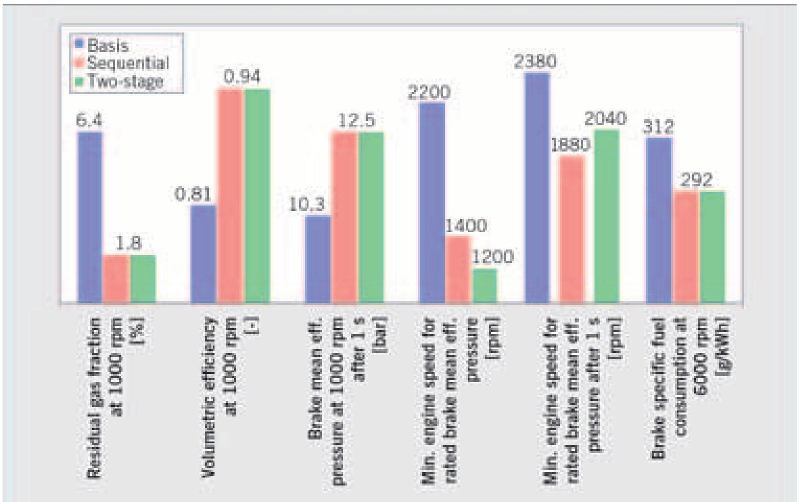
variant, therefore, considerably more exhaust gas mass is conducted through exhaust valve group 2 as engine speed increases than in the sequential variant.

The brake specific fuel consumption of a gasoline engine is essentially determined by the accumulation of fuel-air mixture necessary for component protection. An essential property of the new variants is the employment of two exhaust manifolds, with the higher of the temperatures ascertained in the two lines being decisive for the required accumulation. An immediate consequence of this (necessary) strategy is a fuel-consumption disadvantage that grows with the difference in the temperatures of the two lines, since on average the permissible temperature potential of the components is not exhausted. If one contrasts the variants with each other, unequivocal advantages in favour of the two-stage variant appear in this respect. In the entire second operating range, this variant achieves a fuel-consumption advantage of -4.4 % on average compared with the basis (maximum: -6.4 % at 6000 rpm), resulting from a lower airflow rate as well as the lower accumulation, with the lower accumulation having twice the effect of the lower airflow. With the sequential variant a lower brake specific fuel consumption is only detectable at over 4500 rpm (maximum: -6.4 % at 6000 rpm).

The results for both variants under transient conditions (sudden load increase) from brake mean effective pressure of 2 bar) are summarized in ⓫.

The advantage of both variants is evident immediately after the opening of the throttle valve (t = 0.5 s). The term spontaneous or naturally aspirated torque is of limited applicability, because at this time, a discernable turbocharge pressure buildup occurs with both variants even at an engine speed of 1000 rpm. That the two-stage variant irrespective of its supposedly more responsive high-pressure stage (lighter rotor assembly, turbine with lower absorptivity), does not exhibit any advantage at this point compared with the sequential variant can in essence be traced back to the sequential layout of the high- and low-pressure turbochargers. In accordance with the layout, the two turbochargers differ considerably from each other in their polar moments of inertia (factor 6.8). Owing to the delayed run-up of the low-pressure stage in transient operation, the shares of the compression work taken on by the two compressors differ from those in the stationary state. Despite the small share of the boost-pressure buildup taken by the low-pressure stage, it forms a bottleneck along the exhaust line, on account of which the enthalpy gradient at the high-pressure turbine is lower. These two factors, as properties intrinsic to the system of two-stage turbocharging, cannot be compensated by sequential exhaust valve opening used. As a result the sequential variant exhibits a substantial advantage in brake mean effective pressure compared to the two-stage approach after 0.5 s, on average 4 % up to 1600 rpm, climbing to nearly 16 % (+2.8 bar) at 2400 rpm. After 1 s the sequential as well as the two-stage variant achieve similar brake mean effective pressures at engine speeds up to 1400 rpm, at approximately 13 % over the stationary maximum of the basis. Over that speed the sequential approach exhibits a gain that grows with engine speed (1800 rpm: +21.1 %), while for the two-stage variant the advantage found continued to remain nearly steady. The disadvantage found in the two-stage approach compared to the sequential variant is the result of a decreasing share of the compression work taken by the low-pressure stage as engine speed increases, i.e., as the gradient of boost pressure buildup increases. The sequential variant exhibited response behaviour similar to a naturally aspirated engine with regard to rated brake mean effective pressure (pme = 22 bar within 1 s) beginning at 1880 rpm, 500 rpm sooner than with the basis. The two-stage system does not provide the corresponding system dynamics until 2040 rpm, on account of the disadvantages mentioned. If the interval considered is extended to 2 s, still an acceptable period, a limiting speed of 1520 rpm (680 rpm lower than the basis) emerges for the sequential variant. As a result of the overall higher turbocharging potential of the two-stage turbocharging group and the progressive run-up of the low-pressure stage, an advantage for the two-stage variant over the sequential variant shows itself for the first time at this point, since the corresponding value at 1460 rpm is once more 60 rpm lower.

With regard to their practical implementation it should be stated that sequential und two-stage turbocharging are comparable concerning their complexity. A practically relevant difference between the turbocharging systems under consideration – sequential and two-stage – is their transferability to smaller engine sizes.



⓬ Overview of selected findings

Since in the former case the smallest representable turbocharger size (30 mm turbine wheel diameter) is already implemented in the secondary stage in the design investigated, the displacement can be lowered (< 2.0 l) only by accepting deterioration of the specific characteristic values in the lowest speed range (low-end torque, response behavior). In contrast, the two-stage variant shows potential for lowering the engine size by approximately 20 %, i.e., to 1.6 l, resulting from the possibility of further decreasing the turbine wheel diameter of the high-pressure stage, which in this context is decisive.

4 SUMMARY

In downsizing of gasoline engines, particularly for exhaust-gas-turbocharged four-cylinders, limits in simultaneous maximization of low-end torque and power density have crystallized. These can be overcome only while taking into account the idiosyncrasies of gas exchange in four-cylinder engines.

In the course of this project, full variable exhaust valve timing with dual exhaust manifolds was developed and combined with staged turbocharging systems (sequential turbocharging and two-stage turbocharging). The variants investigated make possible a clear improvement in scavenging behavior in the lowest speed range, a reduction in the brake specific fuel consumption in the rated output range, as well as substantially more spontaneous torque buildup.

The new systems were investigated concerning their properties under full load with the help of a calculation model and a sudden load increase at constant engine speed. Essential characteristic values in comparison to the basis are summarized in ⓬.

All in all, the sequential and two-stage variants exhibit a similar level with respect to the characteristics found, with the sequential approach showing tendential advantages in dynamic operation, while the two-stage system shows more continuous system behaviour overall.

For a final system evaluation of exhaust valve timing with dual exhaust manifolds, further factors will without doubt have to be considered. Among these are particularly: costs, availability, influence on engine emission behavior, and establishment of suitable automatic control strategies. Furthermore, the question of what levels of complexity in the design of a variable valve drive are ultimately necessary, appropriate and optimal remains to be resolved.

REFERENCES

[1] Roß, T.: Neue Konzepte zur Abgasturboaufladung eines direkteinspritzenden Vierzylinder-Ottomotors. Fortschritt-Berichte VDI Reihe 12 Nr. 687, Düsseldorf: VDI-Verlag, 2008

[2] TU Dresden (2010): Verfahren zum Betreiben eines Mehrzylinder-Ottomotors mit Abgasturboaufladung. Deutsches Patent. 10 2008 036 308, 04.02.2010.

[3] Böhme, J.; Müller, H.; Ganz, M.; Marques, M.: Der neue R5 TFSI im Audi TTRS, eine aufladetechnische Herausforderung. 11. Aufladetechnische Konferenz, Dresden 2009

[4] Szengel, R.; Middendorf, H.; Pott, E.; Theobald, J.; Etzrodt, T.; Krebs, R.: Der TSI mit 88 kW – die Erweiterung der verbrauchsgünstigen Ottomotorenlinie von Volkswagen. 28. Internationales Wiener Motorensymposium, Fortschritt-Berichte VDI Reihe 12 Nr. 639, Düsseldorf: VDI-Verlag, 2007

[5] Groff, E.; Königstein, A.; Drangel, H.: Der neue 2,0 l Hochleistungs-Turbomotor mit Benzindirekteinspritzung von GM Powertrain. 27. Internationales Wiener Motorensymposium, Fortschritt-Berichte VDI Reihe 12 Nr. 622, Düsseldorf: VDI-Verlag, 2006

[6] Wurms, R.; Budack, R.; Böhme, J.; Dornhöfer, R.; Eiser, A.; Hatz, W.: Der neue 2.0 l TFSI mit Audi Valvelift System für den Audi A4 – die nächste Generation der Audi Turbo FSI Technologie. 17. Aachener Kolloquium Fahrzeug- und Motorentechnik, Aachen 2008

[7] Zellbeck, H.; Roß, T.; Guhr, C.: Der hochaufgeladene Ottomotor mit Direkteinspritzung. In: MTZ 68 (2007), Nr. 07-08, S. 546-555

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TAILOR-MADE FUELS FROM BIOMASS – POTENTIAL OF BIOGENIC FUELS FOR REDUCING EMISSIONS

Fuels made from biomass offer an enormous potential for use in vehicle applications by greatly reducing carbon dioxide emissions affecting the climate as well as engine pollutant emissions such as soot or nitrogen oxides. Scientists in the Cluster of Excellence "Tailor-made Fuels from Biomass" are researching production as well as the engine's combustion of innovative biofuels. The long-term objective is to optimize the entire process chain from producing the biomass down to the engine's combustion without competing with the food chain to the extent possible. Initial fuels with adapted properties confirm the potential of a nearly soot-free combustion in diesel engines.

THE SEAL OF APPROVAL FOR SCIENTIFIC ARTICLES IN MTZ.
MTZ
PEER REVIEW
RECEIVED 2010-05-30
REVIEWED 2010-07-02
ACCEPTED 2010-08-23
REVIEWED BY EXPERTS FROM RESEARCH AND INDUSTRY.

1 INTRODUCTION

To be able to meet the needs of future societies of continued mobility tomorrow, we need to look for and provide alternatives for fossil energy sources. Fuels made from biomass open up a high potential for also considerably reducing pollutant emissions such as soot or nitrogen oxides in addition to reducing CO_2 emissions. Developing new types of fuels while at the same time using recoverable resources has proven to be a research area with a high need for interdisciplinary cooperation between natural and engineering sciences.

The Cluster of Excellence "Tailor-Made Fuels from Biomass" (TMFB) at RWTH Aachen University was set up in 2007 as part of the Excellence Initiative in order to research new, biomass-based, synthetic fuels for use in vehicle applications. For this purpose, 21 institutes from the RWTH, as well as two external companies (Fraunhofer Institute for Molecular Biology and Applied Ecology, Aachen and the Max-Planck Institute for Coal Research, Mülheim) from the fields Biology, Chemistry, Process Engineering, and Combustion Engineering have coming together in the "Fuel Design Center". The long-term objective of this Cluster of Excellence is to determine the best possible combination of fuel components, whose properties will be determined by the requirements of future combustion processes. The vision of the Cluster of Excellence here is to optimize the entire process chain from the biomass down to the combustion engine, if possible without competing with the food chain.

2 PRODUCTION OF BIOFUELS

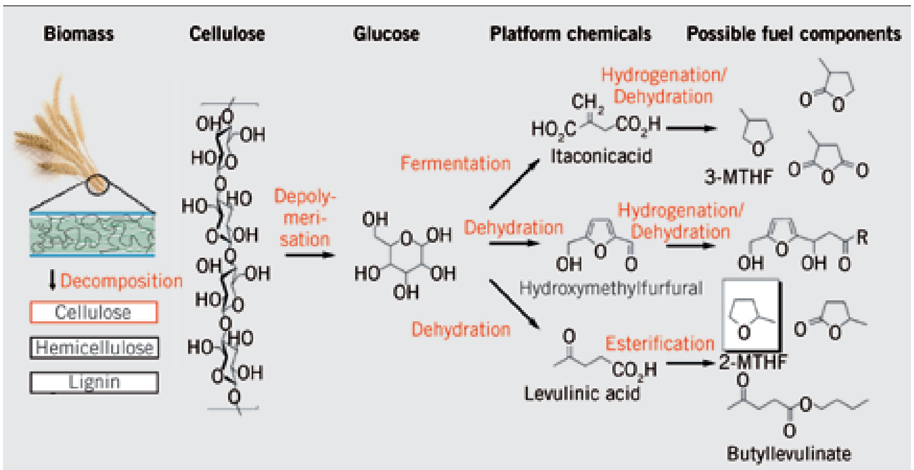
The presently used industrial method for producing fuel from biomass results in the so-called first generation biofuels. To make these, vegetable oils are made into biodiesel (B100) and starch as well as sugar are made into bioethanol. However, cultivating crops for making fuel stands in direct competition with the production of food and feed plants, plus only the fruit and seeds of the plants are being used. To increase the yield per area, future concepts include the complete utilization of plant material that is not used as a food source.

Plant material such as grasses, stalks, or wood consists of lignocellulose. This is a complex composite material consisting of the three biopolymers cellulose, hemicellulose, and lignin. Methods using lignocellulose are producing the so-called second generation biofuels. The production of these fuels are currently still at the state where they undergo technical testing. In the so-called Biomass-to-Liquid (BTL) method, the highly complex structures of the plant material are split up into a mixture of carbon monoxide and hydrogen (syngas) at very high temperatures. The syngas is converted in catalysts into hydrocarbons with longer chains (Fischer-Tropsch synthesis), which can be directly used as fuels or further processed in refinery processes [1].

The TMFB Cluster of Excellence is pursuing an innovative research approach of using nature's synthesis capability and converting and modifying biopolymers only as far as is needed for using them as fuel. To do this, methods are developed in the Cluster of Excellence for the specific chemical conversion of biomass. First of all, the lignocellulose must be split up into its components cellulose, hemicellulose, and lignin. Reaction media such as ionic liquids are used to break up these components. Ionic liquids are compounds that – such as table salt – are made up of positively and negatively charged ions, but have a melting point of below 100 °C due to their molecular structure. Using various catalytic conversion methods, the individual components can then be converted into the desired fuel molecules. ❶ shows possible methods of converting organic matter into possible fuel components.

3 ENGINE TESTS

Closely related to fuel production is the new or further development of engine concepts with improved performance and emission characteristics. Previous first and second generation biofuels (B100 and BTL) imitate the properties of conventional diesel fuels,



❶ Methods of converting new types of biofuels in the TMFB Cluster of Excellence

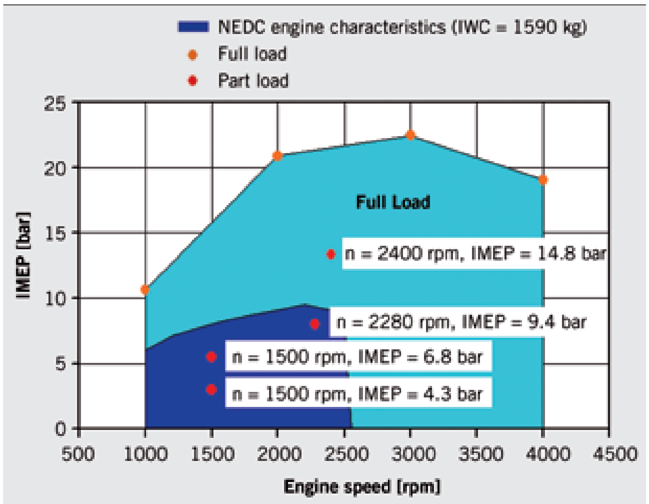
		EN 590 DIESEL	B100	BTL/GTL	2-MTHF
BOILING TEMPERATURE	°C	180-350	352-357	175-329	80
CALORIFIC VALUE	MJ/kg	42.9	36.29	45.86	33.5
DENSITY	kg/m³	833	883	764	867
CETANE NUMBER	-	56.5	51.8	78	~15
OXYGEN CONTENT	m-%	0.14	10.8	0	18.6
AROMATICS CONTENT	m-%	24.9	0	0	0

❷ Properties of the analyzed biofuels

thus calling for only minor engine adjustments. To use the entire potential in terms of emissions and consumption, the properties of future fuels can differ considerably from those of conventional fuels, making it necessary to tailor the combustion systems that are being used to these special properties. In a single-cylinder diesel research engine, the engine behavior of new types of biofuels is being analyzed in order to determine the potential of the TMFB research approach.

	UNIT	DIESEL ENGINE
DISPLACEMENT	cm³	390
STROKE	mm	88.3
BORE	mm	75
COMPRESSION RATIO	–	15
VALVES/CYLINDERS	–	4
MAX. CYLINDER PRESSURE	bar	220
FUEL INJECTION SYSTEM	–	Bosch Piezo Common Rail System
MAX. INJECTION PRESSURE	bar	2000
TURBOCHARGING	–	max. 3.8 bar abs.

❸ Parameters of the single-cylinder diesel research engine



❹ Analyzed load points for fuel tests

4 REQUIREMENTS FOR TAILOR-MADE BIOFUELS

First of all, the production of innovative tailor-made biofuels requires that we define the desired fuel properties. Based on this, the entire potential of the fuel design process can be utilized during the subsequent optimization of the combustion process. Studies conducted in the past have shown that it is possible to clearly reduce the emissions in conventional diesel engines by merely adjusting the fuel properties. This resulted in further criteria for future biofuels in addition to requirements for oxygen content in the fuel and the cetane number. The requirements for the new types of fuels can be summarized as follows [2, 3, 4]:

- oxygen content in the fuel of approximately 20 mass-% as a compromise between the reduction in calorific value and the potential for soot reduction
- longer ignition delay for an improved mixture formation
- fuel free of aromatics
- good evaporation properties while observing the critical flash point in diesel fuels (> 55 °C corresponding to the EN 590 standard)
- material compatibility with the polymer materials used
- ensuring adequate lubricity if necessary through additives.

4.1 BIOFUELS ANALYZED

To assess the potential of the different biofuels, three fuels similar to diesel fuel as well as one fuel made using the TMFB approach were compared to each other. In addition to EN 590 diesel fuel used as a reference fuel, rapeseed oil methyl ester (B100, biodiesel) and one GTL fuel (Gas-to-Liquid) were analyzed in the engine. The GTL and BTL fuels are identical due to the production method (Fischer-Tropsch synthesis), even if methane is used as base material instead of biomass. 2-methyltetrahydrofurane (2-MTHF) is a biofuel component that can be directly made from cellulose using catalytic steps and that corresponds to the requirements placed with regard to reduced ignition quality and increased oxygen content. ① shows the schematic production chain of the tailor-made biofuel 2-MTHF [5].

The most important properties of the analyzed fuels are shown in ❷.

4.2 TEST OBJECT AND ANALYZED LOAD POINTS

The single-cylinder engine used for the diesel engine tests with a swept volume of 0.39 l was designed to have lowest emission levels while at the same time featuring high fuel efficiency. A com-

	POINT OF 50 % ENERGY CONVERSION	PILOT OFFSET	ACTIVATION DURATION OF PILOT	RAIL PRESSURE	BOOST PRESSURE	CHARGE AIR TEMPERATURE	EXHAUST GAS BACK PRESSURE
	°CA BTDC	°CA	µs	bar	bar	°C	bar
n = 1500 rpm, IMEP = 4.3 bar	-6.6 at 0.5 g/kWh ISNOx	10	180	720	1.07	25	1.13
n = 1500 rpm, IMEP = 6.8 bar	-5.8 at 0.5 g/kWh ISNOx	11	140	900	1.50	30	1.60
n = 2280 rpm, IMEP = 9.4 bar	-9.2 at 0.5 g/kWh ISNOx	20	120	1400	2.29	35	2.39
n = 2400 rpm, IMEP = 14.8 bar	-10.8 at 1.0 g/kWh ISNOx	28	120	1800	2.60	45	2.80

❺ Engine calibration

pression ratio of 15:1 was selected in order to be able to represent acceptable peak pressures in spite of the increased charge density. The combustion system reaches a specific output of 80 kW/l at maximum peak firing pressures of 190 bar. A common rail system with a maximum fuel injection pressure of 2000 bar is used as injection system. To optimize the flow characteristics, one intake port is designed as a filling port, the second one as a classic swirl port. Creating charge movement is supported by seat swirl chamfers on both intake valves. The combustion chamber geometry is designed with a conventional recess shape, which was further optimized together with the nozzle geometry (8-hole, ks = 1.5) in order to achieve the best possible air utilization. Reducing the compression, using a higher maximum cylinder and injection pressure, as well as improved EGR cooling makes lowest particulate emissions possible, and as a result the research engine meets the Euro 6 standard. ❸ shows a summary of the parameters of the test engine used. Additional information on the single-cylinder research engine can be found in various publications [6, 7].

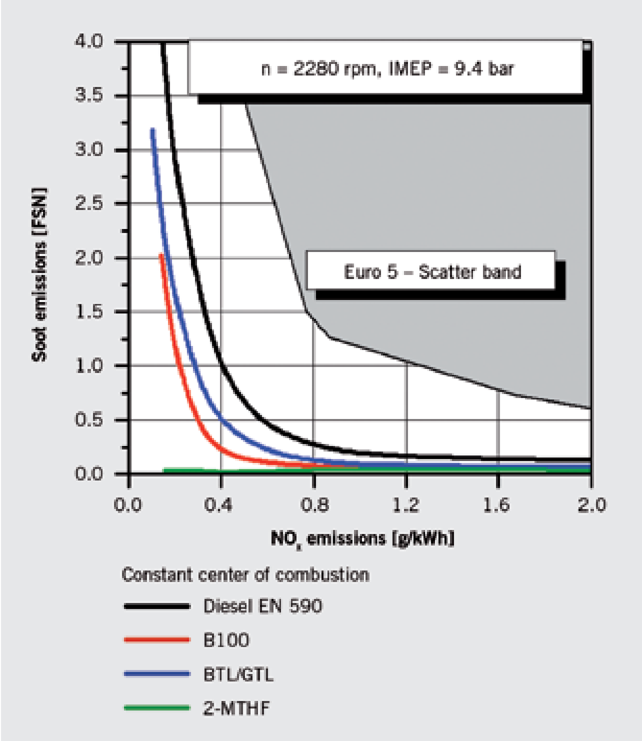
Analyzing new types of biofuels calls for selecting suitable materials for the polymer components supplying fuel. Traditional sealings made from Viton and NBR are suitable for new types of biofuels to a limited extent only, since the biofuels that are being used can lead to swelling and finally to leaks in the system during constant use. Corresponding studies have shown that materials made from Teflon are suitable for constant use.

All fuels were analyzed with a pilot injection and at a constant center of combustion, whereby in each case the start of injection was adjusted accordingly. The other calibration parameters such as intake manifold pressure, fuel injection pressure, and charge air temperature had been optimized in earlier studies considering realistic boundary conditions in order to comply with the Euro 6 standard, [6]. The fuels were analyzed in four load points, three of which are within the NEDC range for an inertia weight class of 1590 kg, ❹. ❺ shows the corresponding calibration for the different load points.

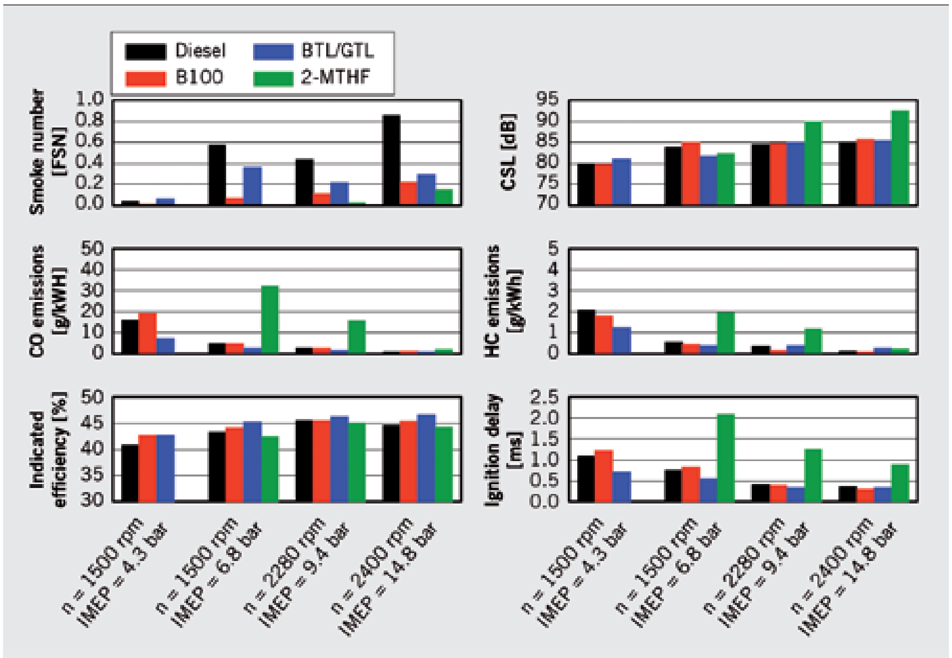
5 RESULTS

❻ shows the particle/nitrogen oxide tradeoff for the four analyzed fuels in the middle load point at n = 2280 rpm, IMEP = 9.4 bar. Even with lowest nitrogen oxide emissions, output of the engine's soot emissions can be almost entirely prevented with 2-MTHF. In comparison to conventional diesel fuel based on crude oil, biodiesels and the GTL fuel also exhibit a significant soot reduction potential. The oxygen content of biodiesel (B100) leads to a decrease of the engine's soot emissions. The advantage of BTL fuels is due to the long-chain, aromatics-free paraffin structures. However, the soot reduction potential of both components is limited compared to 2-MTHF.

❼ shows the emission levels and the efficiency of all load points, whereby estimated nitrogen oxide values were used corresponding to a possible Euro 6 standard, [6]. When using 2-MTHF, the intake pressure in the load point n = 1500 rpm, IMEP = 6.8 bar was increased from 1.5 to 2.1 bar in order to simulate a compression ratio of 19. It was possible to ensure stable ignition as a result of this measure. In the lowest partial load point of n = 1500 rpm, IMEP = 4.3 bar, it was not possible to achieve a stable ignition even for a simulated compression ratio of up to 21. This is why no



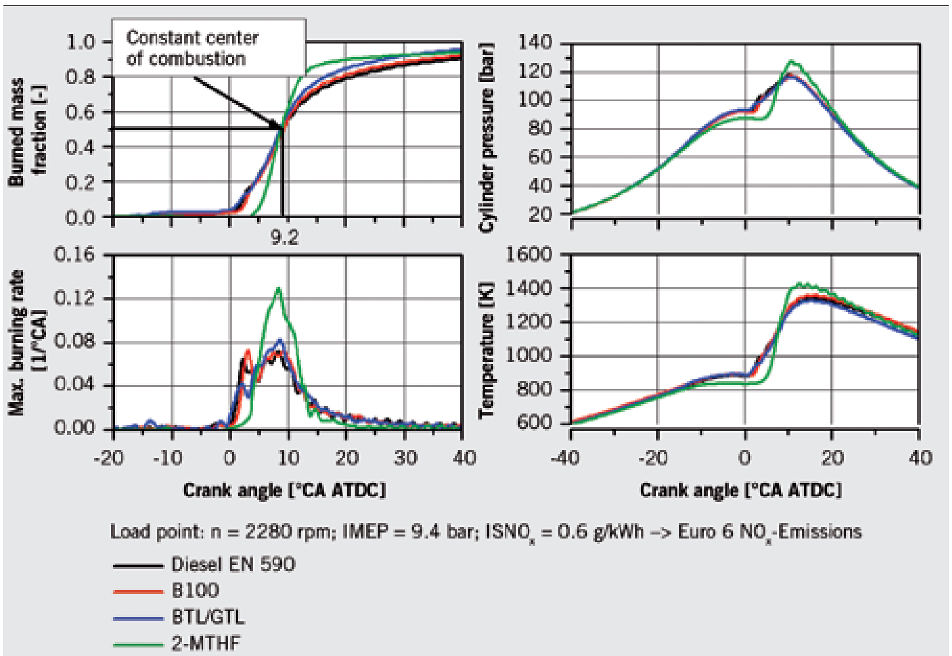
❻ Nitrogen oxide/particle curves of different biofuels



❼ Emission behavior of different biofuels

results are shown in this load point for 2-MTHF. The different fuels were analyzed at a constant center of combustion. The noise emissions are listed as CSL values (Combustion Sound Level). The CSL value combines the rate of cylinder pressure with weighted functions for direct and indirect combustion noise excitation while taking mechanical and flow-related noises into account [8]. The potential of 2-MTHF with respect to soot emissions can be seen throughout the entire map area. The reason for this is a better mixture homogenization due to the longer ignition delay. However, the results also show the disadvantages concerning other emissions and efficiency. The very low cetane number and the resulting, considerably longer ignition delay lead to clearly increased HC and carbon monoxide emissions, in particular at lower loads. The longer ignition delay leads to a higher portion of premixed combustion with 2-MTHF and thus to clearly increased noise emissions.

❽ shows burning curves, rates of combustion, and temperature profiles of the analyzed fuels. In spite of different ignition delay times, it is possible to represent a combustion similar to that of diesel fuel with the help of an adjusted start of injection both with B100 as well as with GTL. If the location of the center of combus-



❽ Burning curves, rates of combustion, and temperature profiles of different biofuels tion is kept constant, the combustion of these fuels is nearly identical. The reasons for the lower maximum heat release rate at the beginning of the combustion of GTL is the lower ignition delay time (CN = 78) and the therefore smaller portion of premixed combustion. The high portion of premixed combustion in 2-MTHF leads to higher pressure increases, a higher maximum burning rate, as well as faster combustion and finally, as a result, to increased peak combustion temperatures. Additionally, the lower temperatures in the expansion cycle make it more difficult to completely oxidize carbon monoxide. The higher peak combustion temperatures in 2-MTHF also lead to a lower isentropic exponent, resulting in higher cycle losses. Besides, because of the longer ignition delay, a lower portion of fuel is converted in the optimum location of center of combustion (near TDC), which additionally explains the lower efficiency of 2-MTHF we see in ④.

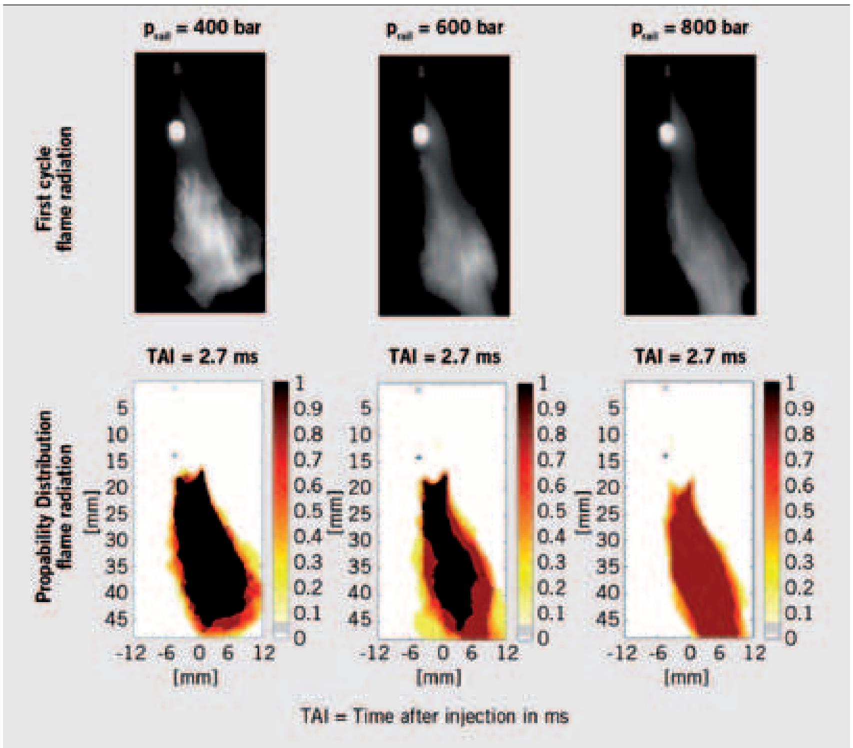
6 OUTLOOK

In the lowest load point of n = 1500 rpm, IMEP = 4.3 bar, ignition was not possible with 2-MTHF. This is why specially adapted combustion systems are required in the low partial load range as well as for reliable cold start behavior. To be able to ignite alternative fuels with considerably longer ignition delay times in diesel engines even at low pressures and temperatures, first glow ignition studies were conducted on ethanol in a high-pressure combustion chamber, which was purchased with funds from the Cluster of Excellence. In the process, a ceramic glow plug with a maximum surface temperature of 1200 °C was used. Ethanol features an even lower ignition quality than 2-MTHF and was used for these initial studies because it has a much better material compatibility with the sealings used in the high-pressure chamber. ❾ shows the boundary conditions of the high-pressure chamber tests. The chamber temperature (800 K) and the chamber pressure (50 bar) correspond to the lowest part load point (n=1500 rpm, IMEP=4.3 bar).

❿ shows first results from measurements with the glow plug used. At constant chamber temperatures and pressures, fuel injection pressures varying between 400-800 bar were used; the injection period was kept constant at 450 µs. The flame radiation was recorded in the visible range 2.7 ms after injection. In the figures at the top, you can in each case see the glow plug as well as the illuminated injection spray. Altogether, five injection processes were recorded in each case, whereby the figures at the bottom show the probability distribution of the flame radiation. The colors shown describe the percentage of injection processes where flame radiation was recorded at the particular point. Areas where flame radiation occurs are marked in black, whereas where flame radiation has never been recorded are marked in white. At an injection

	UNIT	HIGH-PRESSURE CHAMBER
CHAMBER TEMPERATURE	K	470 (max. 1000)
CHAMBER PRESSURE	bar	40 (max. 150)
GLOW PLUG		Ceramic
GLOW PLUG SURFACE TEMPERATURE	°C	1200
INJECTION PRESSURE	bar	400 – 800
INJECTION PERIOD	µs	450
ANGLE OF ROTATION BETWEEN INJECTION SPRAY AND GLOW PLUG	°	10

❾ Boundary conditions in high-pressure chamber tests



❿ High-pressure chamber studies

pressure of 400 bar, combustion takes place after every injection process. Moreover, the actual flame propagation can be reproduced very well. There are only small areas where combustion does not take place after every injection. Even at 600 bar, the glow plug can induce combustion after every injection, but it is now less likely that the flame propagation can be reproduced. At 800 bar, one in five injections will not lead to a combustion process. Overall, we see a poorer reproducibility of flame propagation with increasing fuel injection pressure. However, using a correctly designed glow plug and correspondingly adjusted calibration, it is possible to induce reliable combustion of fuels with a very long ignition delay in diesel engines.

7 CONCLUSION

Tailor-made fuels made from biomass have the potential of significantly reducing local emissions as well as global carbon dioxide emissions. Utilizing nature's synthesis capability allows us to make these fuels of the future available with energy efficient means. With 2-methyltetrahydrofurane (2-MTHF), it was possible to identify a tailor-made fuel that meets the defined requirements for the most part. Because of the oxygen in the fuel, in combination with longer ignition delays, particle emissions can be avoided almost entirely even at greatest EGR rates. Disadvantages are higher noise, HC and CO emissions. The purpose of innovative combustion systems with glow ignition is to ensure reliable ignition of the fuel/air mixture even at low temperature and thus to contribute to complete oxidation.

REFERENCES

[1] Bohlmann, J.: Biokraftstoffe der zweiten Generation: Herstellungsoptionen, Stand der Technik, Effizienz, Kosten. Mobil mit Biomasse, Tagung vom 27. September 2006, Stuttgart, Germany, 2006

[2] Pischinger, S.; Muether, M.; Janssen, A.: Maßgeschneiderte Biokraftstoffe – Ergebnisse aus dem Exzellenzcluster der RWTH Aachen University. 31. Int. Wiener Motorensymposium, 2010

[3] Janssen, A.; Muether, M.; Pischinger, S.; Kolbeck, A.; Lamping, M.; Koerfer, T.: Tailor-Made Fuels for Future Advanced Diesel Combustion Engines. SAE 2009 Fuels and Lubricants Meeting, Florenz; 2009-01-1811, 2009

[4] Janssen, A.; Muether, M.; Pischinger, S.; Kolbeck, A.; Lamping, M.: Tailor-Made Fuels: The Potential of Oxygen Content in Fuels for Advanced Diesel Combustion Systems. SAE 2009 Fuels and Lubricants Meeting, San Antonio; 2009-01-2765, 2009

[5] Geilen, F. M. A.; Engendahl, B.; Harwardt, A.; Marquardt, W.; Klankermayer, J.; Leitner, W.: Selective and flexible transformation of biomass-derived platform chemicals by a multifunctional catalytic system. Angew. Chem. Int. Ed. doi:10.1002/anie.201002060, 2010

[6] Muether, M.; Lamping, M.; Kolbeck, A.; Cracknell, R.; Rickeard, D.; Ariztegui, J.; Rose, K.: Advanced Combustion for Low Emissions and High Efficiency; Part 1: Impact of Engine Hardware on HCCI Combustion. 2008 SAE International – Powertrains, Fuels and Lubricants Meeting, Rosemont, Illinois/USA, 2008-01-2405, 2008

[7] Adolph, D.; Busch, H.; Pischinger, S.; Kolbeck, A.; Lamping, M.; Körfer, T.: Modernes Dieselbrennverfahren zur Darstellung günstiger Motorrohemissionen bei verbessertem Verbrauchsverhalten. In: MTZ 01/2008, Seite 42-50, 2008

[8] Alt, N.; Nehl, J.; Heuer, S.: Combustion Sound Prediction within Combustion System Development. SIA "Vehicle Comfort", 2004

THANKS

This work was performed as part of the Cluster of Excellence "Tailor-Made Fuels from Biomass", which is funded by the Excellence Initiative by the German federal and state governments to promote science and research at German universities.
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