personal buildup for

Force Motors Ltd.

MTZ worldwide 11/2010, as epaper released on 13.10.2010
http://www.mtz-worldwide.com

content:

www.MTZonline.com

MTZ WORLDWIDE

11 November 2010 | Volume 71

REDUCTION of Particulate Number Emissions through Calibration Methods

COOLANT PUMP with Electric and Mechanical Drive

TESTING of Technologies for Exhaust Aftertreatment under Real Conditions

AIR PATH MODELS for Gasoline Engines with Extended Valve Train Variability

GAS EXCHANGE IN TURBOCHARGED ENGINES

CONTENTS

COVER STORY

GAS EXCHANGE IN TURBOCHARGED ENGINES

4, 10 I In naturally aspirated engines, the design of the gas exchange is aimed primarily at achieving high specific power outputs. In turbocharged engines on the other hand, it can also contribute to reducing CO_2 emissions, as it can enable high torques to be developed at low engine speeds. In BMW four-, six- and eight-cylinder engines, the separation of the gas flow that is important for the gas exchange is achieved by a dual-line design. In addition to the design of the turbocharger units, the car maker also describes the different measures and methods applied in designing the gas exchange. At the Institute for Internal Combustion Engines and Automotive Engineering of the University of Stuttgart, a comprehensive development toolbox was used to examine the effects of air scavenging in a turbocharged spark-ignition engine at full load.

COVER FIGURE BMW
FIGURE ABOVE Mercedes-Benz

OPTIMISM

Dear Reader,

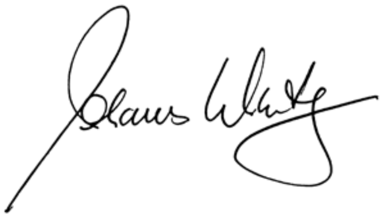
I returned from the IAA Commercial Vehicles in Hannover full of enthusiasm. Enthusiastic about the positive mood in that section of our industry that was hit most severely by the crisis. But also enthusiastic about the many meetings with engineers who refuse to be disheartened by the huge challenges they are facing.

The commercial vehicle sector is a particularly tough and competitive industry. Purchasing decisions are generally made on a rational basis, and it is almost impossible to conceal technical weaknesses behind a design or a famous brand. Competition in the growth markets is essentially characterised by higher market shares of the domestic manufacturers there. Already today, of the world's five biggest manufacturers of heavy trucks, three are Chinese and one is Indian. And the growing requirements regarding low-CO_2 powertrains are more difficult to fulfil because the focus for commercial vehicles has always been on minimising fuel consumption – after all, it is this that determines the running costs for the operator.

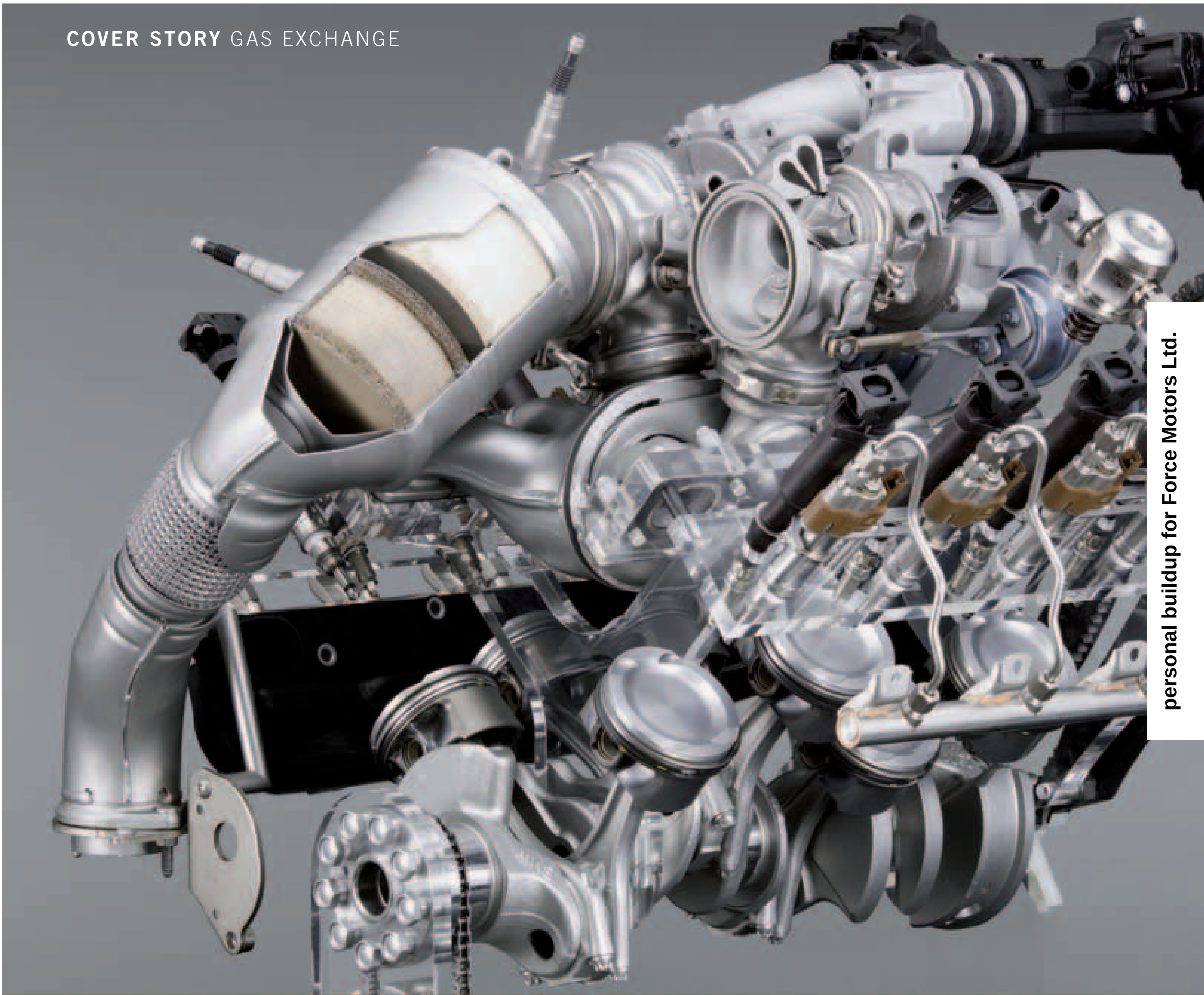
So why am I nevertheless optimistic? Because it also became clear at the IAA just how great the technological lead of the Europeans, both manufacturers and suppliers, actually is. This applies not only but in particular to engine technology. It is therefore by no means unreasonable for the head of MAN, Dr. Georg Pachta-Reyhofen, to announce that he intends to significantly expand the company's business selling industrial engines. And in the field of hybridisation, especially for urban buses and distribution vehicles, German companies like Daimler are clearly a length ahead.

Whether and to what extent European high-tech commercial vehicles will be successful on the developing markets in Asia one day will depend to a considerable degree on whether greater affluence will also result in higher demands for product quality. Internal combustion engines, whether electrified or not, may then become a similar export hit to our machine tools today.

Not all of this can be applied to passenger car powertrain technology. But our optimism, based on our faith in European engineering expertise, certainly can.



JOHANNES WINTERHAGEN, Editor-in-Chief
Frankfurt/Main, 23 September 2010



REDUCTION OF CO_2 EMISSIONS WITH GAS EXCHANGE

With the voluntary commitment that the ACEA has made for itself in 2008, fuel consumption and CO_2 emissions of vehicles have become the focus of politics and customers. With its Efficient Dynamics strategy, BMW has developed CO_2 technologies at an early stage, which has led to results that are below the agreed values. In turbocharged engines the gas exchange design takes on a significant role for CO_2 reduction.

AUTHORS

DR.-ING. CLAUS REULEIN
is Manager Powertrain Development Gasoline Engines CAE Gas Exchange at the BMW Group in Munich (Germany).

DR.-ING. CHRISTIAN SCHWARZ
is Department Manager Powertrain Development Gasoline Engines Thermodynamics and Exhaust System at the BMW Group in Munich (Germany).

REQUIREMENTS

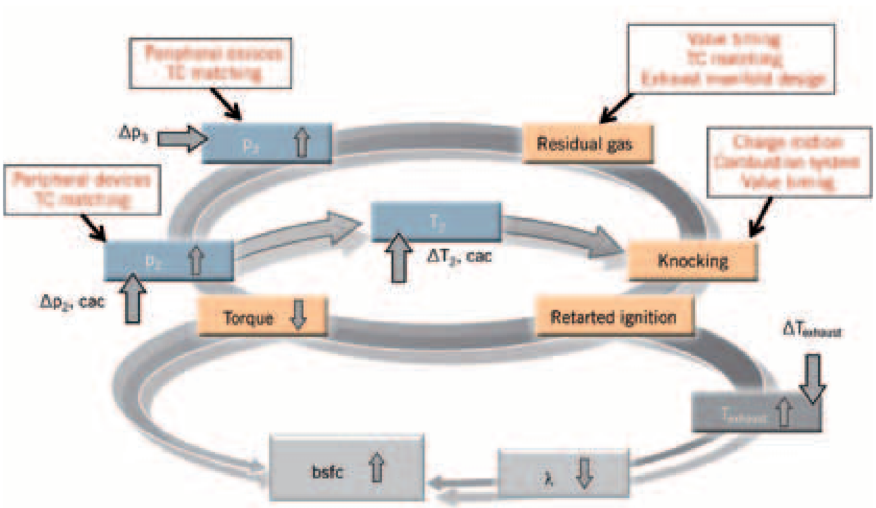
In 2005, it already became apparent that the requirements for CO_2 reduction and emissions limits would become the same in all markets. Because there are very different qualities of fuel in the individual markets, consistent implementation of the stratified lean-burn technology for spark-ignition engines is only possible with great efforts and a new strategy was required.

This development was aided by the ideal combination of turbocharging and direct injection with central injector position, which on the one hand includes an increase in power and torque and on the other hand – almost in contrast – includes downsizing as a means of reducing fuel consumption. Combined with Valvetronic, a technology that was already introduced in 2001 and has therefore been established for a number of years, this TVDI system provides potential for CO_2 emissions that is comparable to that of lean-burn technology. At the same time, a conventional exhaust gas treatment for worldwide use has been made possible.

EFFECTS OF THE GAS EXCHANGE DESIGN

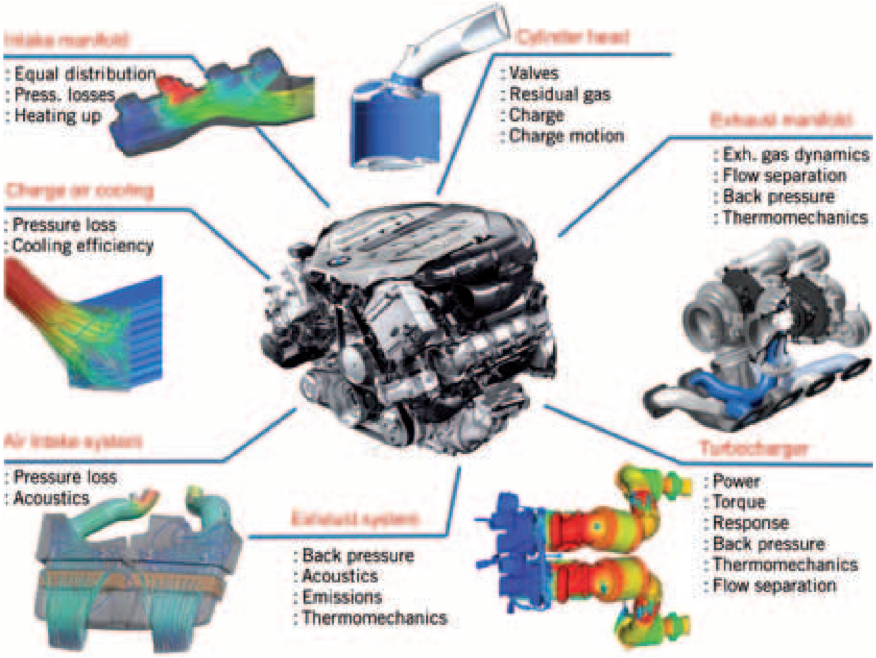
Whilst the gas exchange design for naturally aspirated engines is primarily useful to achieve high specific power outputs, it takes on a significant role for CO_2 reduction in turbocharged engines. High torque at low revs contribute to high specific loads, due to the high gearing that is possible as a result of relocated operating points; this is significant for CO_2 reduction. The high demands placed on BMW drivetrains in terms of power and response require corresponding measures to ensure that boost is generated very well, even at the basic design stage.

At the same time, the gas exchange design also has indirect repercussions on the CO_2 emissions of the internal combustion engine. In ❶ the interactions between the exchange gas design and parameters relevant to the combustion process are shown. The picture makes it clear how pressure losses before and after the compressor influence the behavior of the engine under full load due to the intake manifold temperature set by the charge air cooler and the exhaust back-pressure, which thus has an effect



❶ Interactions in turbocharged gasoline engines

❷ Gas exchange components and assemblies



on fuel consumption, specific torque and specific power.

The result of a balanced design is an exhaust gas temperature that is manageable in terms of cost and a compression ratio that is as high as possible, which then contributes to great potential for CO_2 emissions in the fuel consumption cycle. When combined with fully variable valve operation (Valvetronic), a high compression ratio is of particular significance, because the potential can be increased considerably as a result of improved combustion quality using phasing and masking. Under full load, the tendency to knock or pre-ignition with very high pressure peaks in the combustion chamber, can be considerably reduced by an optimized gas exchange as a result of this, there are also benefits to the degree of effectiveness due to beneficial 50 % mass fraction burned properties and little need for enrichment. Not least, an exhaust back pressure that is as low as possible with a corresponding reduction of gas exchange work contributes significantly to reduced CO_2 emissions.

All in all, an optimum gas exchange design can considerably reduce the gap between the customer's fuel consumption and the official fuel cycle consumption, which leads to the downsizing approach being better accepted by the customer. However, this optimization must be holistic and without compromise; otherwise it will not achieve its objective. We will now go into the measures taken and methods used at BMW in more detail below.

❷ shows an overview of all gas exchange components and shows the potential for optimization for each individual component. The air intake system has, as it does with a naturally aspirated engine, significant potential for improving low end torque (LET) and power through low losses in pressure; however the conflict of objectives between the existing package and acoustic requirements must be taken into account as well as the requirements of the different operating conditions and additional requirements that result from emissions requirements (such as HC fleece for Zero Evap). The high sensitivity of a charged spark-ignition engine to the state of the compressor makes 3D CFD optimization absolutely necessary. This applies in particular for a flow into the compressor that is as free of pre-swirling as possible, which is usually difficult to implement due to packaging requirements.

As already described in ①, the intercooler with a spark-ignition engine contributes considerably to good overall engine characteristics, especially in relation to other conditions for combustion. Here, the conflict of objectives between a loss of pressure and possible recooling rates must be solved as a basis for optimum combustion. Particular attention must be paid to the design decision of whether to have a direct or indirect intercooler. There are different advantages for either design, depending on the packaging and cooling capacity available.

With the intake manifold, which plays a subordinate role in a turbocharged engine, the main focus is an even distribution between the cylinders in terms of providing blow-by and tank ventilation. The pipe length setup required for naturally-aspirated engines to exploit the dynamic effects of gas are rather counterproductive for turbo engines.

Due to the high degree of filling with a turbo engine, the cylinder head is of considerable importance due the corresponding design of the inlet valves. If it is possible to create a sufficient level of charge motion, both the formation of the mix and the combustion can be significantly improved. This leads to an earlier 50 % MFB and thus to lower exhaust gas temperatures and lower fuel consumption under full load. As a result of this, the gap between cycle fuel consumption and customer fuel consumption can be reduced considerably. If it is possible to implement long exhaust valve timing in conjunction with optimized flow separation in the exhaust gases, fuel consumption under part-load can be reduced using the existing potential of Valvetronic by a further 2 %, without a reduction of low end torque potential.

In doing so, charge motion measures also play a decisive role with low valve lift via phasing and masking as well as an optimization of the intake and exhaust valve diameters and ports.

The design of the exhaust manifold plays a decisive role for the overall optimization of the gas exchange design. The use of exhaust gas dynamics by means of CFD-optimized pipe cross-sections and routings is of equal significance to consistent flow separation. To also be able to avoid the disadvantages of a turbocharged engine (heat sink) when setting up emissions characteristics, the surface/volume relationships and the flow through the waste gate must be optimized using CFD calculations.

TURBOCHARGER DESIGN

The most significant component for a CO_2-optimized design of the engine is the turbocharger. With the compressor, the main focus is a very good degree of effectiveness over a wide range of the compressor map. The here from resulting low requirement for compressor power makes good low end torque characteristics possible with good potential for engine power at the same time. It is primarily the interaction between the compressor housing and the compressor wheel where the flow is optimized using 3D CFD optimisation. Thus, a recirculation valve connection that is not optimal may cost up to three percentage points in effectiveness.

The same applies for the turbine design as well as the compressor design. In addition to the flow of turbines in conjunction with the manifold and its construction shape that has already been mentioned, the main focus here is on increasing the degree of effectiveness. The separation of gas flows, which is very important for gas exchange, is made possible in four, six and eight cylinder engines by the use of a twin-flow design (twin-scroll technology). As a result of this, there is a gap in the crankshaft angle of at least 240° between consecutive gas exchange events. All measures lead to a considerable reduction of exhaust back pressure with positive effects in relation to fuel consumption under full load and under part load as well as manageability of combustion in the low end torque area and the nominal power output area. Inertia torque that is as low as possible due to a small wheel diameter or a specially chosen material improves the spin-up time and is a prerequisite for the amount of downsizing being accepted, due to outstanding response characteristics.

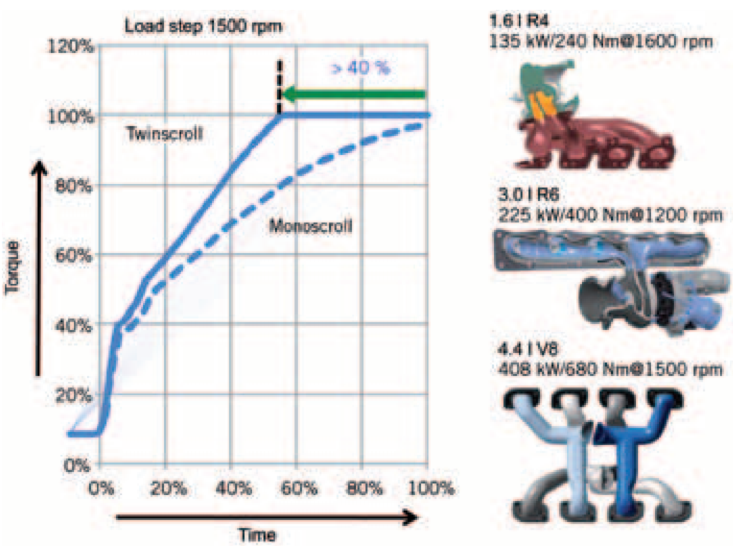
With the rear exhaust system, there is a conflict of objectives between packaging, emissions requirements, acoustics and costs in terms of minimizing back-pressure. In the process, 1D and primarily 3D CFD calculations have become indispensable in the development process. As the degree of engine charging increases, overall optimization becomes ever more important, because the back pressure that appears at the exhaust valve and thus the emissions-related exhaust back pressure is determined by multiplying the rear exhaust back-pressure by the turbine pressure ratio. This is – as already described – of great importance for part load and full load, as well as response characteristics.

ADOPTION AND EFFECTS IN SERIAL APPLICATION

The BMW Efficient Dynamics concept combines a significant reduction in CO_2 due to downsizing with very good dynamic properties, despite higher gearing. These characteristics can only be implemented by means of a consistent gas exchange design with no compromises using twin-scroll technology, as a result of which a wide spread between high specific output of up to 100 kW/l and low end torque of up to 200 Nm/l at revs under 1300 rpm is made possible.

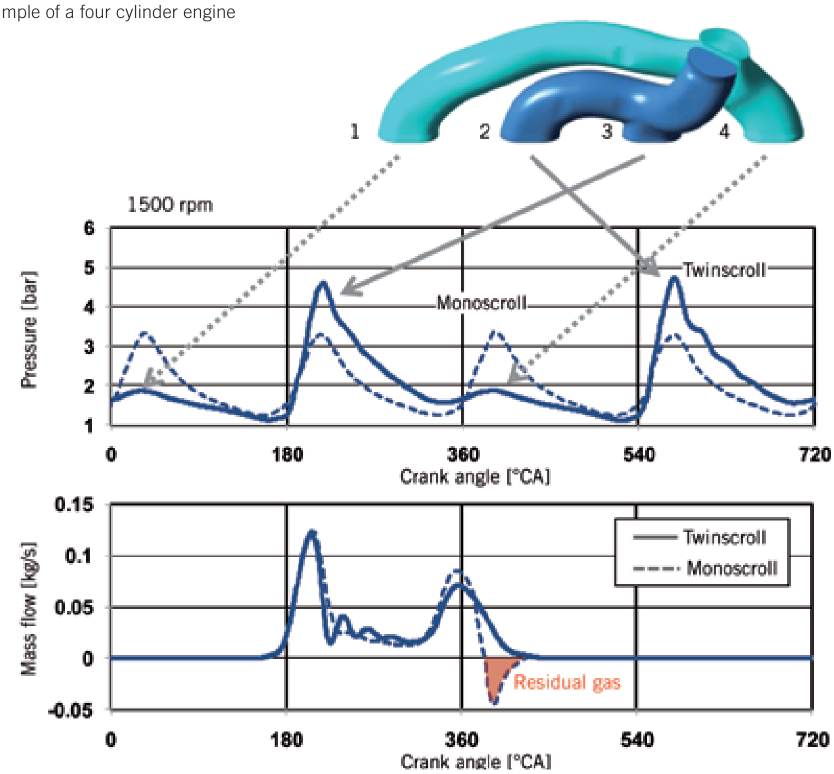
This technology has now been consistently implemented in a wide range of BMW engines (V8, in-line six and in-line four). As a result of the use of twin-scroll turbines, it is possible, as shown in ❸, to increase the required time to create boost by over 40 % in comparison to conventional turbines.

What is particularly impressive is its application in a V8 for the M engines in the X5M and X6M, where an exhaust



❸ BMW Group twinscroll applications

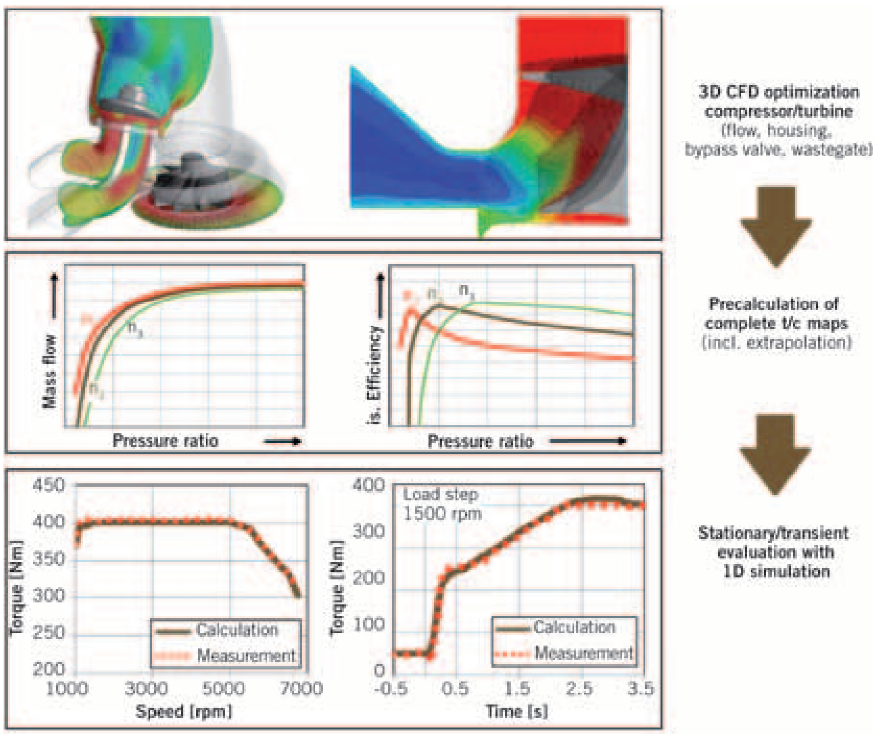
❹ Functionality of twinscroll turbines as example of a four cylinder engine

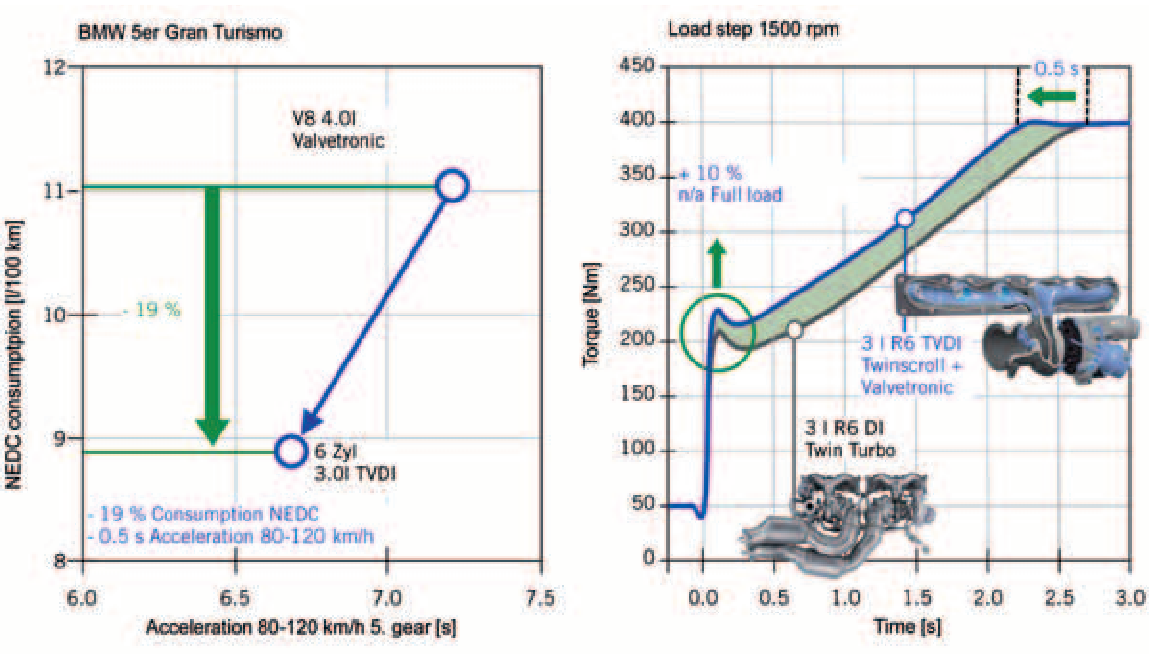


manifold covering the rows of cylinders is used in the V of the motor to keep even firing intervals of 360° of crankshaft angle in the individual flows of the twin-scroll turbocharger. Only in this way can the optimum gas exchange be achieved.

❹ shows, using a four cylinder engine as an example, the twin-scroll effect in comparison to a monoscroll turbine. The monoscroll turbine shows four smaller pulses with relatively little gas dynamics. However, the interaction of exhaust

❺ Turbocharger matching process at BMW





❻ Functional results for the BMW 3 l six cylinder Twin Power engine

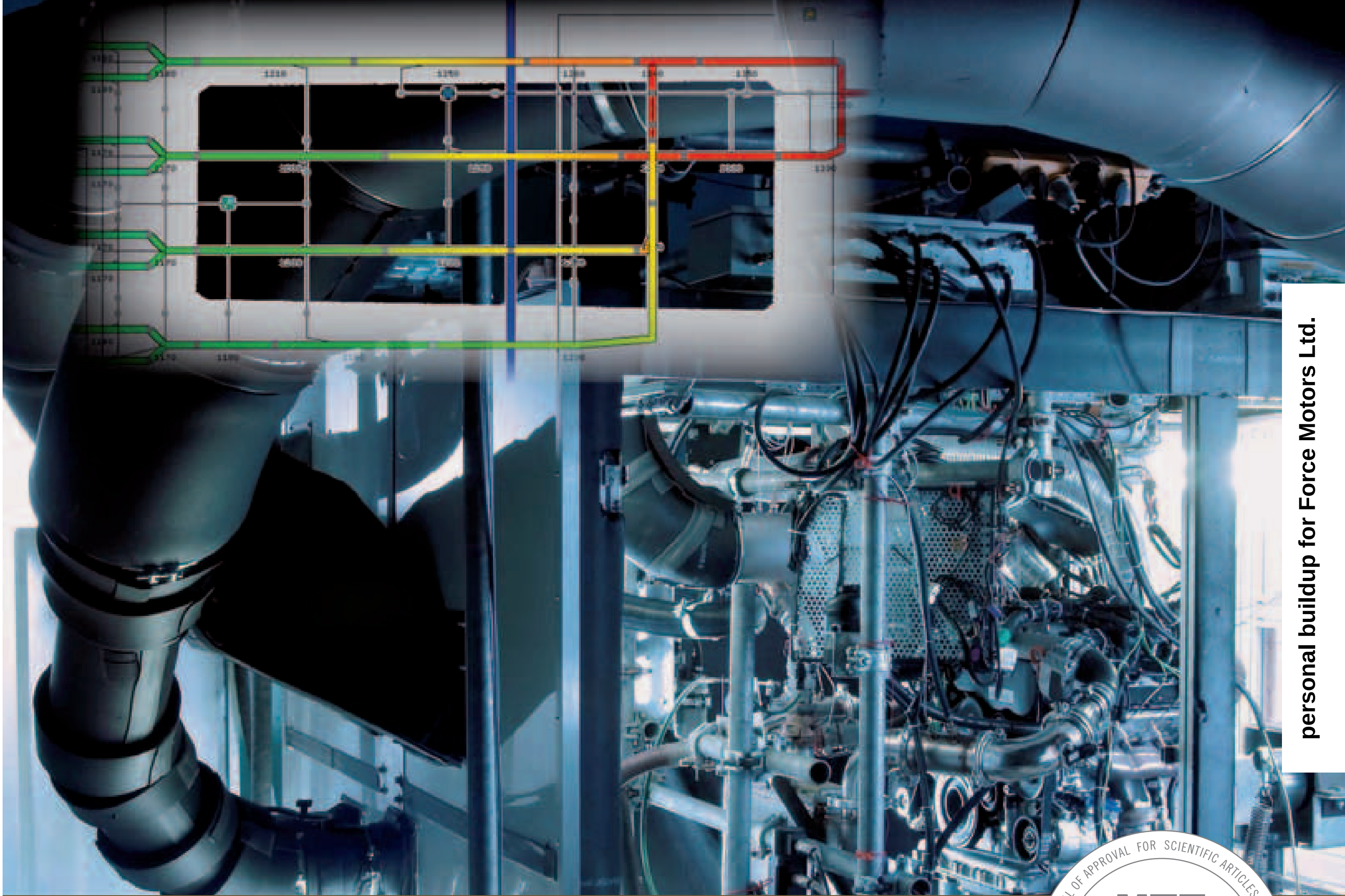
pulses with the same camshaft timing due to a flushing slope leads to a significant amount of exhaust gas being pushed back into the cylinder due to the pressure wave of the next cylinder to be ignited. This leads to a considerable increase in residual gas in the cylinder and makes it impossible to operate the motor with high torque at low revs. To weaken the effect, a considerably reduced exhaust valve timing is required for the low end torque area only. If the engine is to show high specific power outputs, variable switching of the timing is required.

With the twin-scroll turbine, the 180° crankshaft angle only influences the cylinders that are to be subsequently ignited to a small degree, due to a leakage of the two separated flows when they meet in the turbine wheel. However, this course of pressure does not have a detrimental impact on the residual gas characteristics. In addition, the considerably increased dynamic properties of the two flows that are separated by a 360° crankshaft angle is evident. This all leads to the positive effect on transient characteristics that is described in ③.

❺ shows a simplified illustration of the individual process steps when evaluating and designing the turbochargers. The requirement for the highest dynamic properties requires a design where all details of the compressor and turbine are optimized. In addition to the optimizations that have already been mentioned, an initial calculation of the parameters is made using 3D CFD calculations, based on CAD data from the compressor and turbine housings and the geometries of the rotor disc. The precision of the calculation for engine-related areas is approximately 2 %. The 3D calculation also make predictions for the parameters possible; these cannot be measured on the test bench.

With this data, both stationary and transient characteristics can be forecast with sufficient precision using 1D CFD calculation. In the process, the twin-scroll effect is portrayed by a special process in the 1D calculation exactly as it would be in reality; a comparison between measurement and calculation in the lower section shows this impressively, using an inline six engine with twin-scroll technology (Twin Power Turbo) as an example.

❻ shows, in the right section, the interaction of the full variable valve lift control with the twin-scroll turbocharger using the full-load acceleration described in ③ as an example. As a result of consistent optimization of valve lift with transient driving maneuvers, naturally aspirated full-load was increased considerably. This leads to a further considerable reduction of the time taken to achieve the maximum torque; approximately 0.5 s. This effect on full load can only be achieved in combination with twin-scroll turbocharging, Valvetronic and direct injection. All in all, this technology provides the prerequisites to reduce fuel consumption significantly around the world, as the diagram on the left section of the picture shows impressively, using the 5 Series GT as an example. It is thus possible to achieve an improvement of acceleration from 80 to 120 km/h by approximately 0.5 s and reduce fuel consumption, in conjunction with an eight speed gearbox, by 19 % in comparison to a naturally aspirated V8 engine with the same power.



THE SEAL OF APPROVAL FOR SCIENTIFIC ARTICLES IN MTZ.
MTZ PEER REVIEW
RECEIVED 2010.06.30
REVIEWED 2010.07.05
ACCEPTED 2010.08.09
REVIEWED BY EXPERTS FROM RESEARCH AND INDUSTRY.

TRANSIENT SIMULATION WITH SCAVENGING IN THE TURBO SPARK-IGNITION ENGINE

A promising approach to decrease fuel consumption in spark-ignition engines involves supercharging with an exhaust-gas turbocharger, by way of which load point shifting and/or longer transmission ratios can be shown. Direct injection provides additional benefits, when combined with exhaust-gas turbocharging and the use of devices for manipulating the phasing of inlet and exhaust control times. On account of the many degrees of freedom this mode of operation offers, the use of computational simulations to design the gas exchange is extremely advantageous, to reduce both time and expense during development. To this end, the effects of scavenging in a turbo SI engine at full load were thoroughly examined at the Institute for Internal Combustion Engines and Automotive Engineering (IVK) with the aid of a comprehensive development toolkit.

AUTHORS

DIPL.-ING. ANDREAS SCHMID
previously employed in Internal Combustion Engines at the Research Institute of Automotive Engineering and Vehicle Engines Stuttgart (FKFS); currently Research Assistant at the Institute of Internal Combustion Engines and Automotive Engineering (IVK) of the University of Stuttgart (Germany).

DR.-ING. MICHAEL GRILL
Project Manager, Internal Combustion Engines, at the FKFS in Stuttgart (Germany).

DIPL.-ING. HANS-JÜRGEN BERNER
Head of Department, Thermodynamics and Combustion Systems in the Internal Combustion Engine department at the FKFS in Stuttgart (Germany).

PROF. DR.-ING. MICHAEL BARGENDE
Member of the Managing Board of the FKFS and Chair in Internal Combustion Engines at the Institute of Internal Combustion Engines and Automotive Engineering (IVK) of the University of Stuttgart (Germany).

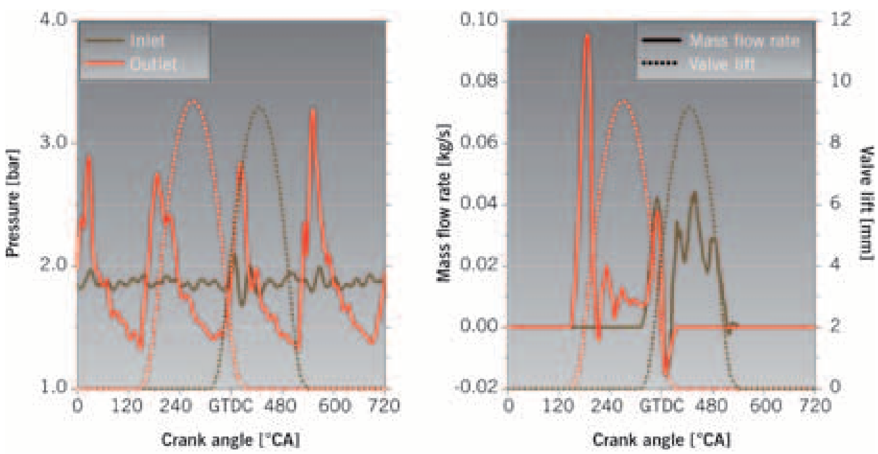
PRINCIPLES OF SCAVENGING

In a turbocharged SI engine, scavenging can be applied in dethrottled operation as soon as the boosting system provides sufficient boost pressure. To achieve this, a positive scavenging gradient must occur during the valve overlap phase. This provides the possibility of expelling the working gas directly from the intact tract into the exhaust tract, in a process known as "scavenging." In direct-injection SI engines, this method can be used to particular advantage, since here a transfer of fuel into the outlet duct can be prevented by choosing an appropriate injection point. The required positive scavenging gradient is strongly influenced by gas-dynamic effects, particularly the exhaust pulse, on account of the intermittent operation of the internal combustion engine. In engines with exhaust-gas turbocharging, this effect is especially pronounced during pulse turbocharging, since here the exhaust gas backpressure increases only momentarily and can therefore be considerably lower than the pressure in the intake tract in the overlap phase. The effect of the positive scavenging gradient on a four-cylinder SI engine with a conventional boosting system and a single-scroll turbine is shown in ❶. It is apparent that a positive pressure gradient exists over large areas during the overlap phase. This enables a significant mass flow to be "scavenged" directly from the intake tract to the exhaust tract.

Through this effect, in addition to an increase in the fresh air content due to residual-gas reduction, a decrease in charge temperature is also achieved, thereby lessening the knock tendency. This in turn makes it possible to use earlier ignition points with commensurately positive effects on the efficiency level. If fresh air is passed through the combustion chamber without taking part in the combustion process, the global air/fuel ratio differs from the air/fuel ratio in the cylinder during combustion. And if a rich air/fuel mixture is necessary at full load to reduce the knock tendency, a globally stoichiometric air/fuel ratio can still be shown via scavenging. If the not yet completely oxidized exhaust gases in the "rich" combustion process react with fresh scavenged air prior to the turbine, an increase in the exhaust-gas enthalpy available to the turbine can be achieved. Overall, the effects of scavenging mean that responsiveness is improved at low rpms and a higher maximum torque can be achieved at lower rpms, thereby mitigating the major shortcomings of a supercharged SI engine.

SIMULATION MODELS

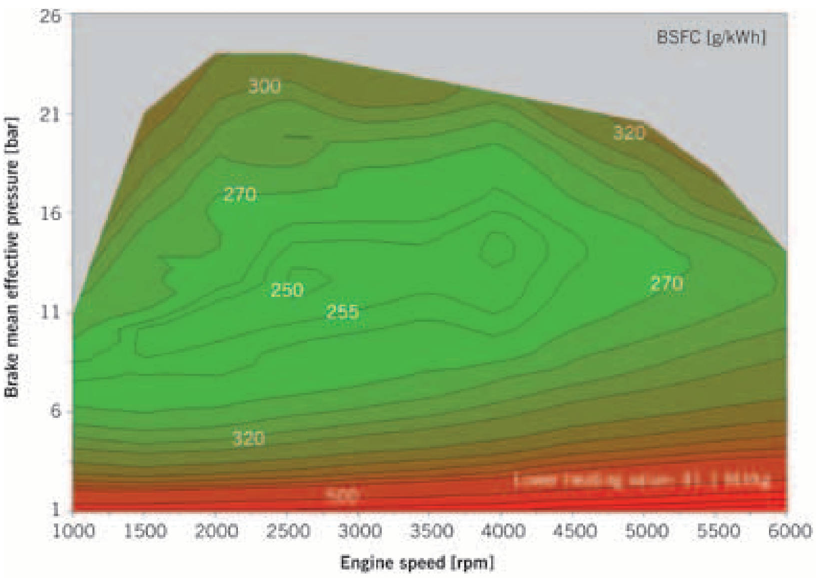
Combustion with scavenging in an SI engine is simulated using a combination of a quasidimensional combustion model and a knock model, which have been integrated into a commercial 1D flow-simulation software application by Gamma Technologies [2]. The simulation environment of the high pressure process with the corresponding calorics [3,4] for the high pressure simulation was presented in [1]. The quasidimensional entrainment combustion model [1,2] used takes into account the effects of cylinder geometry, residual gas,



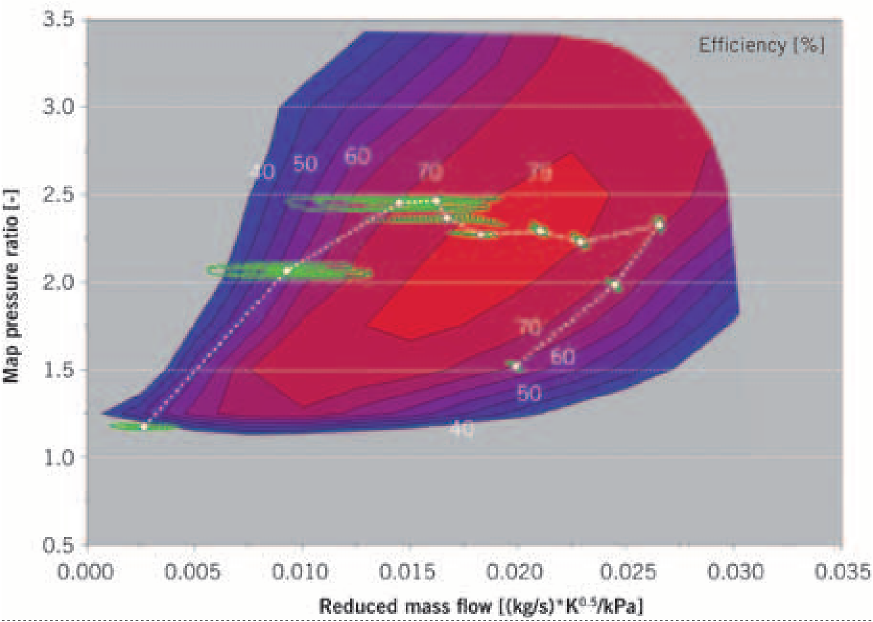
❶ Scavenging during the valve overlap phase in cylinder 1

	VALUE	UNIT
NUMBER OF CYLINDERS	4	[-]
DISPLACEMENT	1597	cm^3
BORE	74.1	mm
STROKE	92.6	mm
MAXIMUM POWER OUTPUT	140	kW
MAXIMUM TORQUE	305	Nm
COMPRESSION RATIO	10 : 1	[-]
TURBINE TEMPERATURE LIMIT	950	°C
VEHICLE WEIGHT, INCL. DRIVER	1540	kg
C_w*A	0.564	m^2

❷ Basic simulation model data



❸ Consumption map of the simulation model under consideration



❹ Compressor map at full load

turbulence, and mixture composition, thus providing a nearly complete description of SI engine combustion. Basic parameters were adjusted to the corresponding models on the basis of measuring data. The influence of the knock tendency is simulated using an innovative approach that takes into account the influence of turbulence and hot spots on the autoignition status; this was likewise compared against measuring data. To simulate gas dynamics, a 1D flow model calibrated over large areas is used. The engine model including combustion is integrated into a whole vehicle model to allow simulation of transient acceleration procedures. This ensures feedback between real driving operation and the boundary conditions of the flow simulation and hence also combustion. The basic data of the combined engine and vehicle model are presented in ❷. To simulate the scavenging effects, a simulation of the complete gas dynamics and combustion of each individual working cycle during the entire acceleration process is carried out. Also represented is an anti-knock controller that provides ignition timing alteration when the autoignition threshold is exceeded, thus giving a realistic simulation of the knock limit. The engine model used is coupled with an exhaust-gas turbocharger with no flow separation.

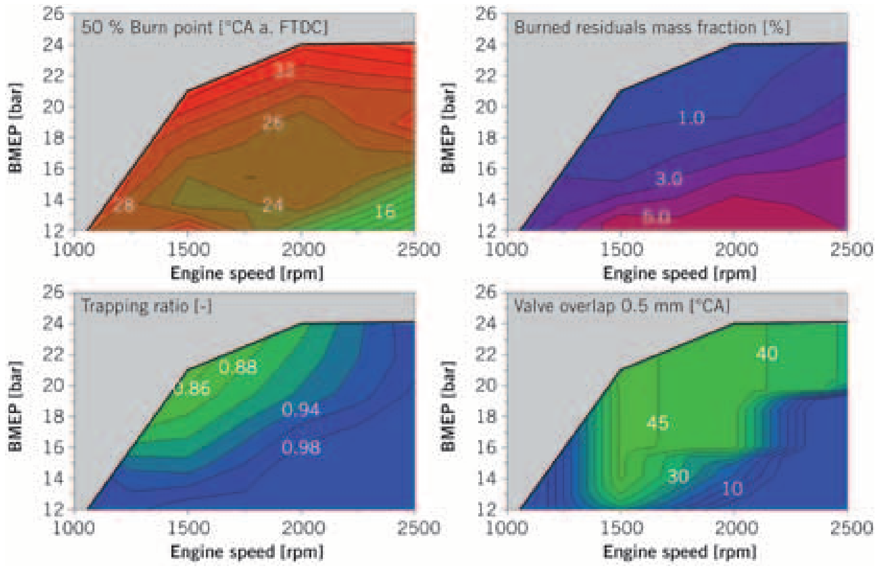
STATIONARY SIMULATION OF SCAVENGING

Overall, the engine concept uses a stoichiometric air/fuel mixture, provided a richer mixture is not required to protect the components. The specific fuel consumption in the operating range of the simulated engine is represented in ❸. ❹ shows eleven stationary full-load operating points (1000 to 6000 rpm, Δn = 500 min rpm) on the compressor efficiency map. For each operating point, the combinations of reduced mass flow and pressure ratio occurring during a working cycle are shown (in green), along with their mean values (in white).

In ❺, the valve overlap times appropriately placed on the characteristic map and the resulting trapping ratio are illustrated. The trapping ratio is defined as the available trapped fresh mixture during the combustion divided by the supplied fresh charge per cycle. Scavenging produces a significant reduction in hot residual mass, thereby sharply decreasing the knock tendency. This allows for a drastic increase in maximum torque in the low rpm range. The figure also shows the 50 % burn point that occurs when a constant knock tendency is regulated. The positive influence of the significantly reduced residual mass is particularly apparent at the 50 % burn points. At very low speed, scavenging is very difficult to implement in a four-cylinder engine with a single-scroll turbine, since an exhaust pulse occurs directly in the overlap phase on account of the geometry-dependent operating times. Given the present exhaust geometry, this cannot be prevented by manipulating the valve timing; for this reason, valve timings with no valve overlap were chosen at 1000 rpm. In the engine under investigation, scavenging is possible up to approximately 2500 rpm. Beyond this, its use is essentially limited by two effects.

- The efficiency of the boosting system is decreased by the boost-pressure limiting via the wastegate during blow-off, which necessitates a higher average exhaust pressure as the engine speed increases in order to achieve the same boost pressure.
- In addition, when examined based on the crank angle, the exhaust pulse becomes longer as the engine speed increases; this decreases the scavenging window and as a result, a reliable application of scavenging is no longer possible.

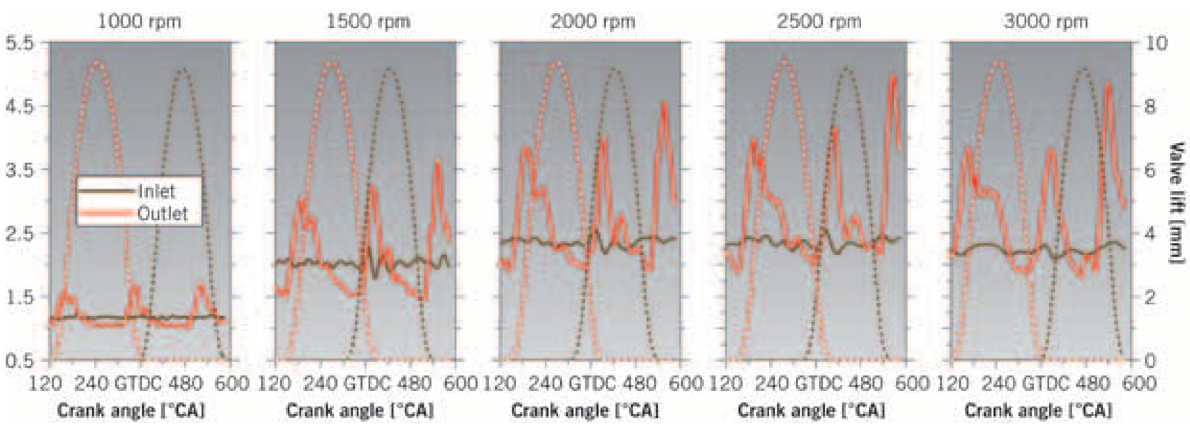
The pressure profiles upstream of the intake valve and downstream of the exhaust valve are shown in ❻ for speeds between 1000 rpm and 3000 rpm with the valve timing used. It must be noted in regard to this representation that each exhaust pulse is heavily determined by the exhaust valve timing.



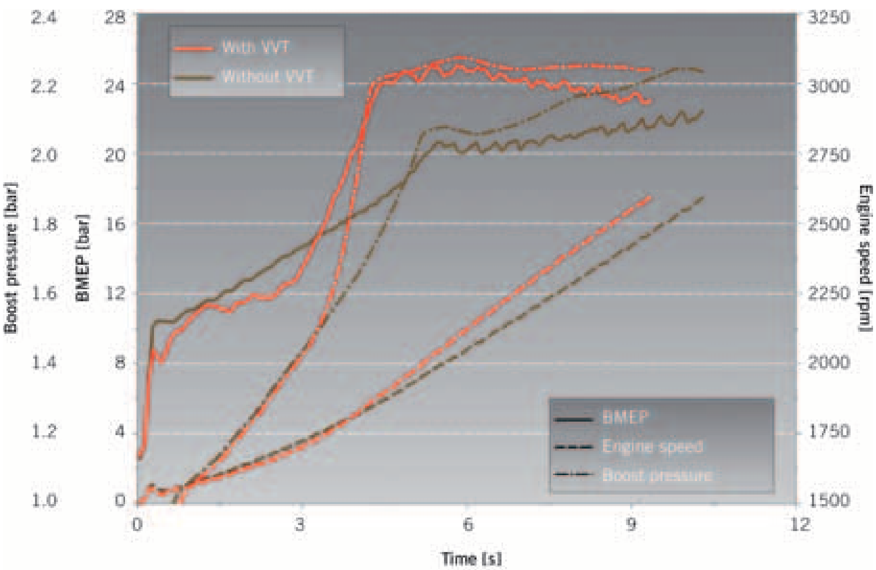
❺ Burn point, residual mass, trapping ratio, and valve overlap in the scavenging area

TRANSIENT SIMULATION OF SCAVENGING

Scavenging for performance improvement is generally implemented in the vehicle's acceleration phases. A purely stationary examination of scavenging is therefore impractical, as control and application influences cannot be meaningfully reproduced in a stationary model. Therefore, in addition to the stationary observation, non-stationary engine operation with a virtual vehicle will be examined to simulate scavenging. In principle, transient simulations can be performed with map-based engine models, with mean-value models of the internal combustion engine, or with complete transient gas dynamics. A calculation with purely map-based engine models is not useful for turbocharged engines, as the boosting system response cannot be reproduced. While it is possible to show this response using mean-value models, the scavenging phenomenon cannot be reliably reproduced even in these models. For this reason, at IVK [5], scavenging simulations are carried out with complete examination of the non-stationary gas dynamics. In this case, basic functions of the ECU, such as lambda control, wastegate control, and anti-knock control, but above all the manipulation of the valve overlap, must be reproduced in a transient-capable fashion. The simulation computing time increases significantly with this type of simulation. However, for the sudden load increases shown below, the computing time is still less than 60 minutes on a standard PC. When defining the set-point values for the phase adjusters



❻ Gas dynamics of cylinder 1 at full load; overlap phase shown



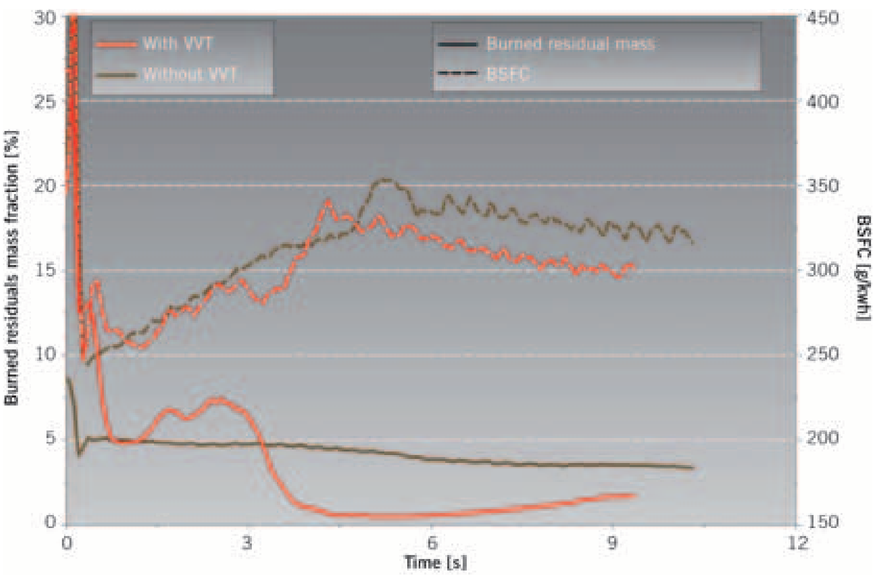
❼ Brake mean effective pressure, boost pressure, and engine speed during the acceleration process

(variable valve train), it must be kept in mind that if a sufficient boost pressure has not yet built up, the residual mass may increase if a significant valve overlap is used. This has an extremely negative effect on the knock tendency. If the valve overlap is not chosen until just after a sufficient boost pressure for scavenging has been built up, the result is an extremely sudden increase in engine torque. This can lead to problems with driveability and control. In addition, the transition range between a negative and positive scavenging gradient is quite narrow and difficult to determine in real driving due to a wide range of boundary conditions. Thus an ideal control of the phase adjusters is not possible in real applications. For this reason, a small negative scavenging gradient due to the valve overlap was tacitly accepted for the transient simulations in order to achieve the most realistic simulation results possible.

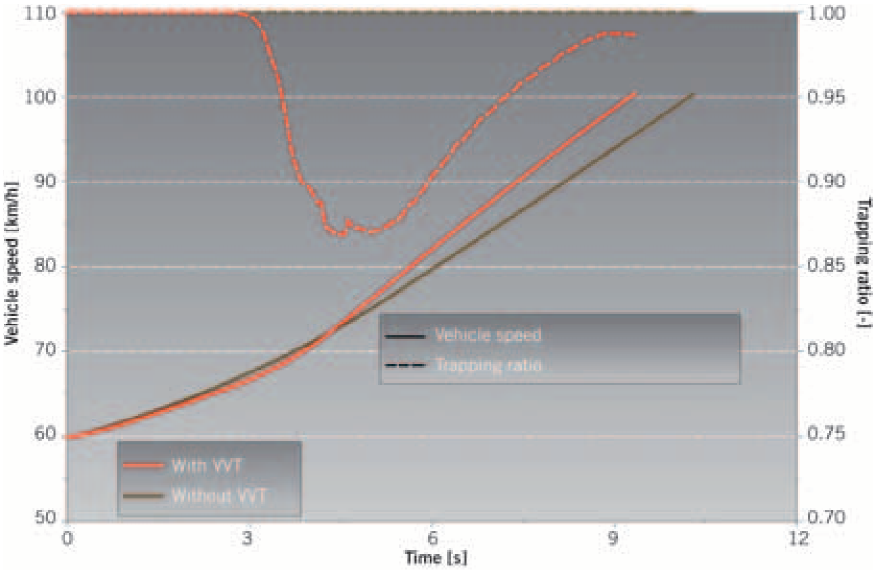
ACCELERATION 60–100 KM / H

To illustrate the effect of scavenging, an acceleration simulation from 60 km/h to 100 km/h in fifth gear will be examined below. A simulation with fixed valve timing and no valve overlap (without VVT) will be compared against a simulation with the variable valve timing (with VVT) presented. The maximum boost pressure was adapted in such a way that no critical states were achieved as a result of the increased knock tendency without valve overlap. In addition, in order to maintain the same compression ratio, an acceleration enrichment was applied to ensure that sufficient engine torque could be made available. Because no thermal convergence occurs in transient simulations, the simulation results are dependent on the chosen starting conditions. Therefore, for the simulations presented, constant-speed driving for 12 seconds was simulated at the initial speed before the sudden load increase. For dethrottling at low loads, optimized timing for the generation of internal EGR is used during the simulation with a variable valve train, with negative effects initially in a sudden load increase. The characteristic of the brake mean effective pressure, the boost pressure, and the engine speed in the acceleration phase with and without a variable valve train is shown in ❼. It is evident that with VVT the initial full-load torque is slightly lower at first due to the residual gas in the intake manifold. The subsequent increase in brake mean effective pressure at the beginning occurs somewhat more slowly due to the partially negative scavenging gradient and the globally stoichiometric concept, resulting in a lower boost pressure as well as engine speed increase. As soon as there is sufficient boost pressure and clear scavenging effects come into play – in accordance with the valve timing – the brake mean effective pressure with VVT increases very rapidly over the brake mean effective pressure with constant timing and achieves significantly higher maximum values, thus accomplishing shorter acceleration times overall.

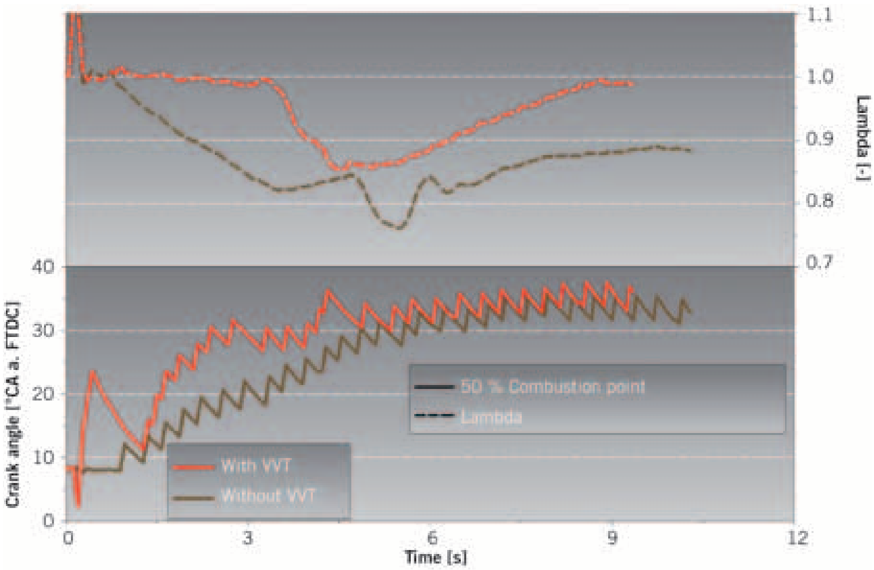
The residual mass fraction and specific consumption during the acceleration process are shown in ❽. The residual gas initially present in the intake manifold during the sudden load increase can be discerned. A relevant advantage in consumption at low loads is achieved by dethrottling via internal EGR. Up to approximately three seconds after the sudden load



❽ Residual mass and specific fuel consumption during the acceleration process



❾ Trapping ratio and vehicle speed in the acceleration process



❿ 50 % combustion point and air ratio in the cylinder during the acceleration process

increase, when the residual mass is still quite high, no significant advantages in consumption are achieved by not enriching the mixture during acceleration. The disadvantage in consumption that should result from the acceleration enrichment is compensated in this area by a diminished knock tendency. As soon as the residual mass decreases, however, this results in significant advantages in consumption due to the stoichiometric concept with scavenging.

In ❾, the point at which fresh air becomes available in the exhaust tract as a result scavenging is discernible based on the trapping ratio. The vehicle speed characteristic is also shown. The shorter acceleration time (approximately 1 second) with the use of VVT is clearly discernible.

In the concept presented here, an acceleration enrichment was not used when a variable valve train (VVT) was used. A rich air/fuel mixture for combustion is only provided to protect the turbine or on account of scavenging. The air/fuel ratio during combustion and the 50 % burn points resulting from the knock controller are shown in ❿. The chosen acceleration enrichment without VVT is clearly discernible. It is also apparent, in looking at the 50 % burn points, that similar ignition timings can be achieved by means of engine operation with scavenging – despite a significantly higher brake mean effective pressure and a lean air/fuel ratio.

SUMMARY

Quasidimensional simulation models combined with 1D flow simulation open up the possibility of reproducing complex combustion processes and comparing engine concepts. In combination with a vehicle model, the interactions between gas dynamics, combustion, and vehicle response can be reproduced. This information can then be used to make predictions relating to accelerating performance, fuel consumption, and driveability.

REFERENCES

[1] Grill, M.: Objektorientierte Prozessrechnung von Verbrennungsmotoren. Dissertation, Universität Stuttgart, 2006, http://elib.uni-stuttgart.de/opus/volltexte/2006/2725/

[2] Grill, M.; Billinger, T.; Bargende, M.: Quasi-Dimensional Modeling of Spark Ignition Engine Combustion with Variable Valve Train. SAE-Paper: 2006-01-1107

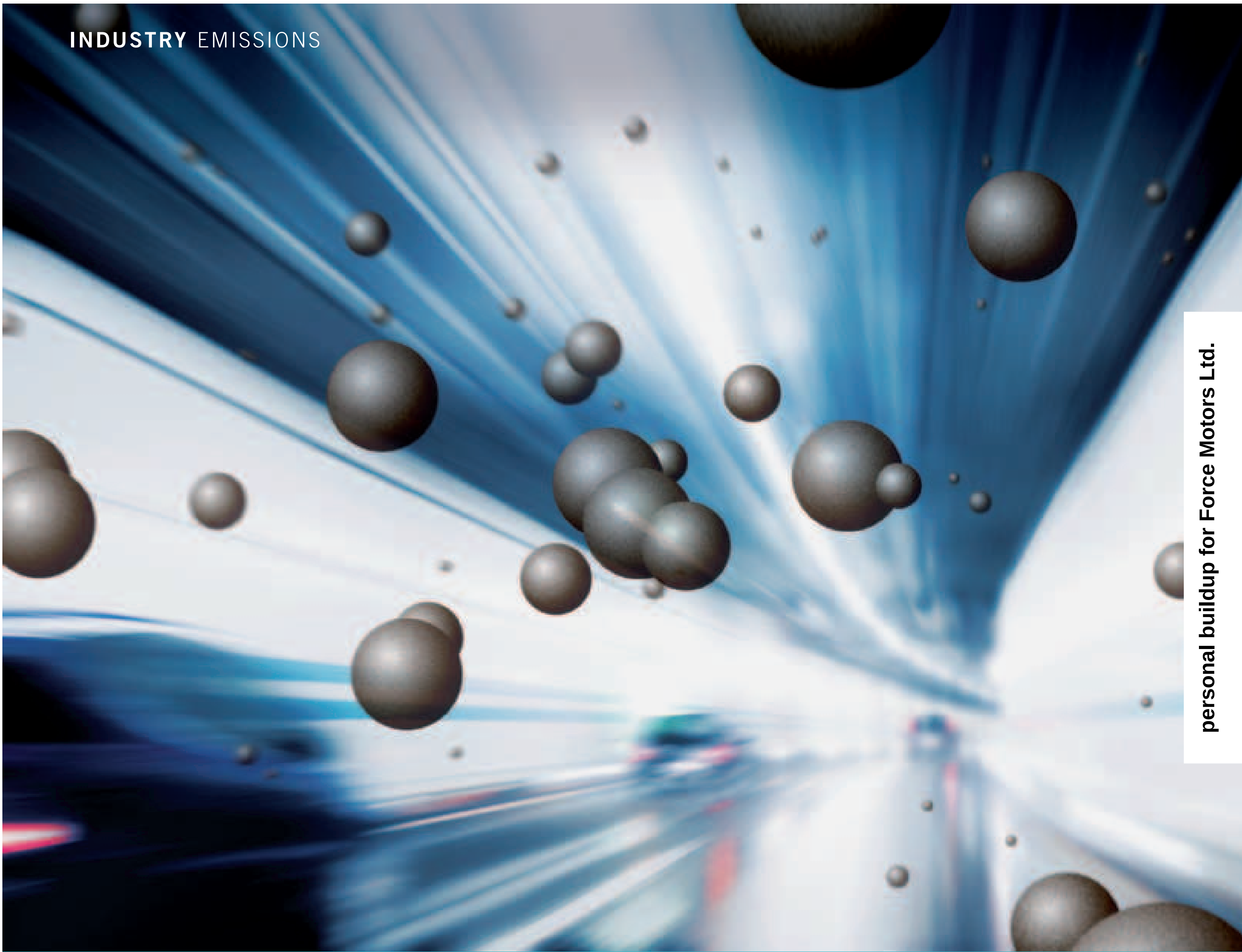
[3] Grill, M.; Schmid, A.; Chiodi, M.; Berner, H.-J.; Bargende, M.: Calculating the Properties of User-Definied Working Fluids for Real Working-Process Simulations. SAE-Paper: 2007-01-0936

[4] Grill, M.; Chiodi, M.; Berner, H.-J.; Bargende, M.: Berechnung der thermodynamischen Stoffwerte von Rauchgas und Kraftstoffdampf beliebiger Kraftstoffe. In: MTZ 68 (2007), Nr 5, S. 398 - 406

[5] Schmid, A.; Grill, M.; Berner, H.-J.; Bargende, M.: Transient and Map-Based Driving Cycle Calculation with GT-Suite. GT-User Conference 2009, Frankfurt, Tagungsband

THANKS

The authors would like to thank the Research Association for Combustion Engines (FVV, Frankfurt/Main) that founded the research project (no. 994) with capital resources. Furthermore the authors would like to thank the initiator of the project, Dr. techn. Dipl.-Ing. Herwig Richter, Porsche AG, and the chairman of the project Dr.-Ing. André Kulzer, Robert Bosch GmbH, for the excellent assistance. A special thank goes to the entire FVV research group consisting of industry representatives for the helpful suggestions and impulses.



REDUCING PARTICULATE EMISSIONS
NEW CHALLENGE FOR SPARK-IGNITION ENGINES WITH DIRECT INJECTION

As exhaust-emission legislation tightens across the globe, a general reduction in the particulate-matter limit value for spark-ignition direct-injection engines is for the first time being combined with plans to introduce a limit on the particle number for spark-ignition engines. This is pushing the reduction of particulate emissions further into the focus of developers. IAV (Ingenieurgesellschaft Auto und Verkehr) has analyzed the various influences at play and extended its production-level calibration process to cover particulate emissions, confirming this process on the basis of a mass-production concept.

AUTHORS

DIPL.-ING. OLAF KANNAPIN
is Head of the SI-Engine Calibration Department in the Powertrain Mechatronics Business Area at IAV GmbH in Berlin (Germany).

DIPL.-ING. THOMAS GUSKE
is Project Engineer for SI-Engine Calibration in the Powertrain Mechatronics Business Area at IAV GmbH in Berlin (Germany).

DIPL.-ING. MICHAEL PREISNER
is Team Manager of SI-Engine Calibration in the Powertrain Mechatronics Business Area at IAV GmbH in Berlin (Germany).

DIPL.-ING. MATTHIAS KRATZSCH
is Vice President of SI-Engine Calibration in the Powertrain Mechatronics Business Area at IAV GmbH in Berlin (Germany).

MOTIVATION

Aimed at improving air quality – particularly in city centers – legislation has been in place for quite a while to limit particulate emissions in the diesel-engine segment. A major milestone has been the introduction of the diesel particulate filter, drastically reducing the particle number (PN) and particulate matter (PM) emitted for diesel concepts.

In the past, however, only marginal attention has been paid to particulate emissions from spark-ignition engines. Some of today's laws on exhaust emissions from spark-ignition engines already provide for restrictions on particulate matter [1]. Future legislation must be expected to cover far stricter criteria, particularly also in relation to particle number emitted, ❶.

PARTICLE FORMATION

Mechanisms of particle formation are basically familiar in professional circles [2, 3]. This article focuses less on the chemical principles underlying the way in which particles are formed but rather on the principles that actively influence their formation.

Particles essentially result from incomplete combustion and poor mixture formation, with excessively high fuel concentrations occurring locally in the combustion chamber. In particular, fuel wetting cold combustion-chamber walls is responsible for intensive particle formation which is why the main focus of measures to reduce particulate emissions must rest on cold engine operation. In addition to optimizing combustion-chamber and injection geometry, the parameters of charge cycle, with valve timing and any applicable valve lift, as well as the injection parameters must be selected so as to avoid wall wetting as far as possible and to optimize assistance for fuel vaporization and distribution [4].

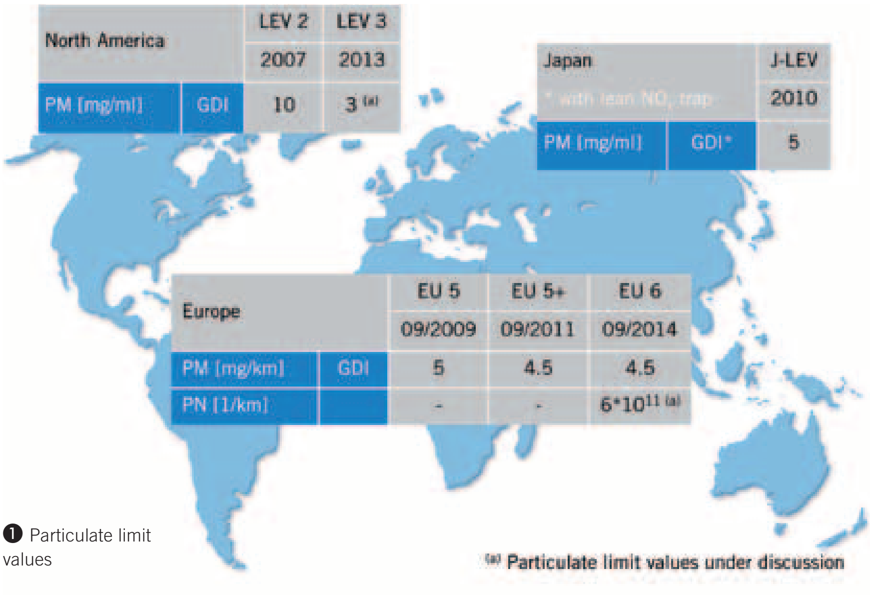
MEASUREMENT TECHNIQUE FOR COUNTING PARTICLES

IAV simultaneously used two DMS500 systems from Cambustion Ltd. operating on the basis of electrical mobility classification, and one APC489 system from AVL on the basis of condensation particle counting.

It should be realized that counting particles in exhaust gas of just a few millionths of a millimeter in size by and large pushes the limits of the instruments that can be viably produced with today's technical capabilities. Reliable results can only be obtained by averaging a large number of measurements. This method ensures statistical certainty. In addition, it evens out the cycle-stage related behavior of the engine with its associated particle-number fluctuations.

TEST SETUP AND REDUCING PARTICULATE EMISSIONS

The investigations were conducted on a supercharged four-cylinder direct-injection spark-ignition engine. They did not involve varying the combustion-chamber shape, injectors, injector position or other hardware components. For the purpose of this study, the test specimen was operated on an air-conditioned dynamic engine test bench with exhaust and particulate emission measuring facilities. In addition, NEDC exhaust-emission tests

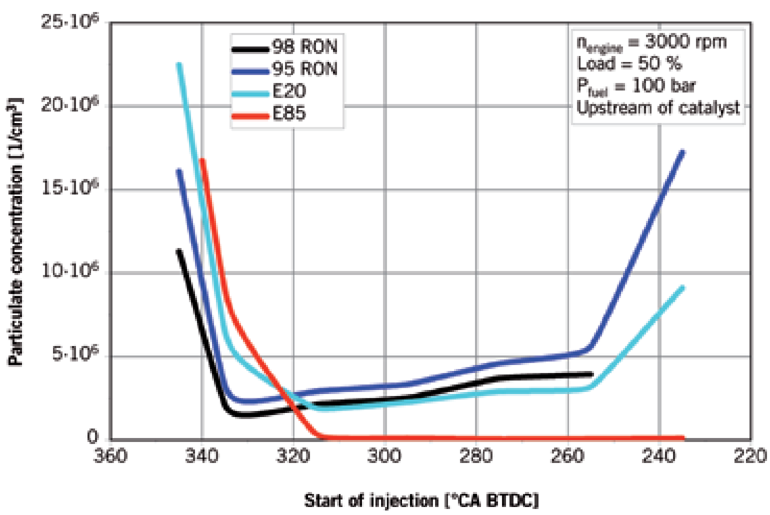


❶ Particulate limit values

were carried out with a vehicle running on an exhaust-emission roller dynamometer. These were broken down into several parts in line with IAV's calibration process adapted on the particulate side – starting with a steady-state part at operating temperature for basic calibration, a forced-cooled steady-state part with coolant at a temperature of 20 °C for calibrating the catalyst heating phase and a dynamic part applying the complete NEDC cycle with associated variable conditioning. Adapted DoE methods were used for each of the two steady-state optimization processes [5].

For the purpose of collectively confirming all of the findings gathered from individual investigations, NEDC exhaust-emission tests were conducted on a vehicle running on an exhaust-emission roller dynamometer with optimized data.

All investigations provided the capability of influencing the ECU parameters, with fuel pressure, overall injection timing and injection splitting being instrumental and varied for optimizing particulate emission. The purpose of these analyses was to reduce particulate emissions without any negative influence on fuel consumption, classic emissions as well as engine smoothness and drivability. In addition to this, four different fuels containing differing amounts of ethanol were examined with the aim of identifying the influence of fuel specially.



❷ Particulate concentration in relation to angle at start of injection

INFLUENCE OF FUEL

Different fuel qualities have a major influence on particulate emissions from spark-ignition DI engines. In addition to the chemical composition of fuels, characterized by their basic formulation as well as different additives, the ethanol component has a marked influence on particulate emission. An elevated ethanol component produces a far higher supply of OH radicals during the combustion phase, resulting in increased particle oxidation and ultimately in a lower level of particulate emission.

For the purpose of the investigations, previously analyzed fuels were used in filling-station quality; the E20 blend was produced for the measurements. For an optimum particle result, each fuel requires an adapted set of ECU parameters with changed injection timing over those for specification 98 RON fuel, ❷. Despite the additional 30 % injection volume coming from E85 and resultant increase in the duration of injection, E85 produces clear benefits through adjusting injection timing. The formulation of 98 RON versus 95 RON fuel is also shown to have an effect that cannot be influenced by calibration methods. All other investigations were carried out using the fuel specified for the engine used.

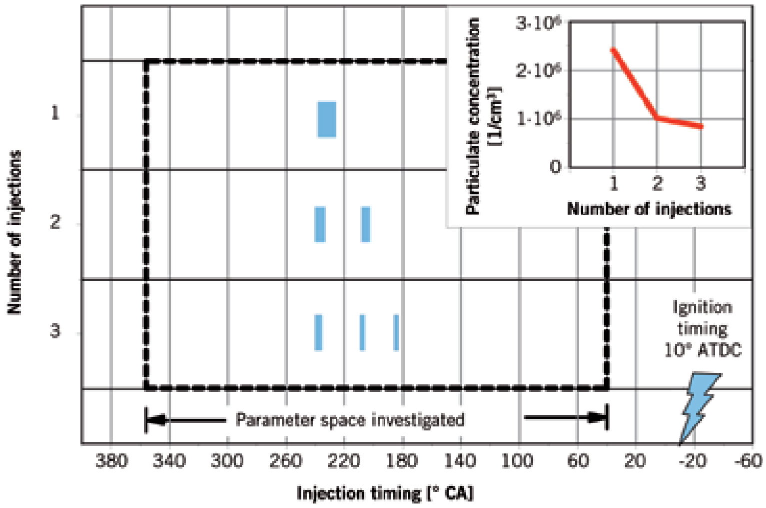
FUEL PRESSURE

Particularly in the case of direct-injection engines, the best possible preparation of the air/fuel mixture provides the basis for optimum combustion. With defined engine hardware, i.e., without the possibility of varying components or geometries, particulate emissions can be reduced across a scatter band from 20 % to 50 % (depending on fuel grade and boundary system condition) by significantly raising fuel pressure while at the same time optimizing the start of injection (SOI).

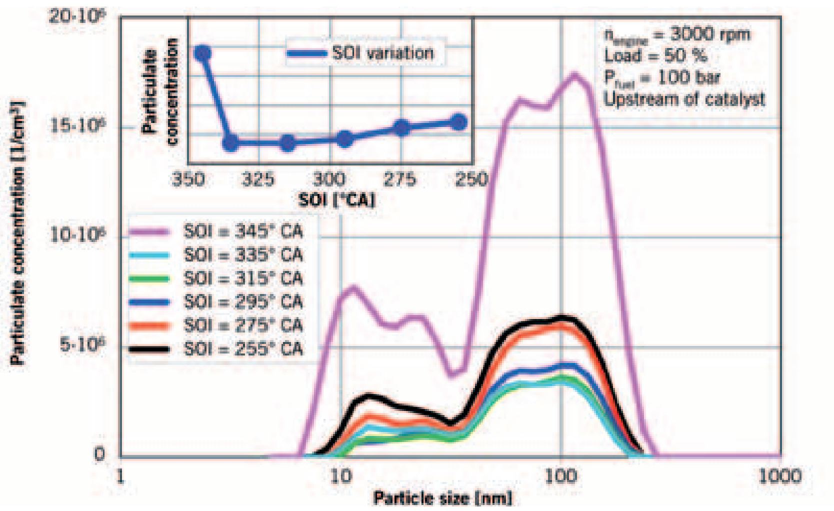
INJECTION TIMING AND SPLITTING

Optimizing the mixture formation process involved, varying injection pressure while adjusting the parameters that define injection timing. Doing so, the angle at the start of injection is optimized in a way that minimizes fuel wetting the piston head or combustion-chamber wall. Given the underlying concept, a further advantage can be gained by splitting the injection quantity into a rapid succession of several partial quantities per work cycle. As a result of the interaction of injection timing, charge motion and combustion-chamber geometry, particularly in the case of a cold engine, this significantly reduces particulate emission. A typical load point was selected for the purpose of investigating the optimum injection parameters in relation to catalyst heating and optimized using DoE on a steady-state cold engine operating at lambda = 1 with constant engine timing of 10 °ATDC. In this context, the single, dual as well as triple-stage injection was modelled within the test space in terms of partial quantities and injection timing.

It was shown that the splitting during the catalyst heating phase provides various benefits. In addition to stable combustion with a variation coefficient suitable for practical use, low exhaust-gas emissions as well as reduced consumption, it was possible to achieve a significant reduction in particulate emissions. Shown in ❸, the results of particle-optimized parameters must be seen in relation to general exhaust-emission calibration in production concepts.



❸ Multiple injection catalyst heating load point (1300 rpm, 50 % relative load)



❹ Size spread on varying start of injection

CHANGES IN SIZE AND MODE

Electric mobility classification not only provides the capability of determining the pure particle number but also of making statements on size distribution and on the two particulate modes occurring in the exhaust gas. Volatile particles are ascribed to the nucleation mode that is not evaluated conforming to the Particle Measurement Program (PMP) in emission legislation. Non-volatile particles are ascribed to the accumulation mode that is evaluated in conformity with PMP.

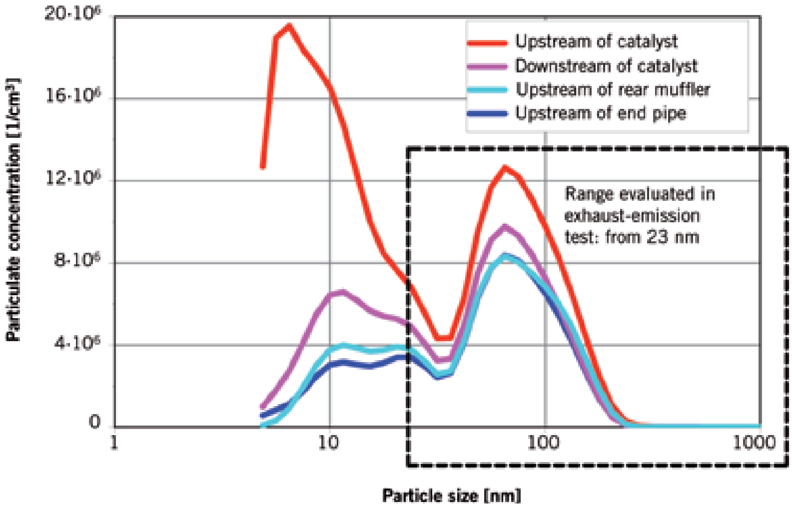
The quantitative change of particle concentration by varying the SOI is presented in ❹. Here, it can be seen that in spite of concentrations varying, both the ratio of nucleation mode and accumulation mode as well as spread remains virtually unchanged in both modes.

Low engine temperature has a disproportionately strong influence on particle concentration and affects the ratio of both modes into the operating temperature range. Engine lambda, engine speed and load also affect mode ratio. The investigations revealed no appropriate way of using calibration-based methods for achieving any qualitative changes in size distribution.

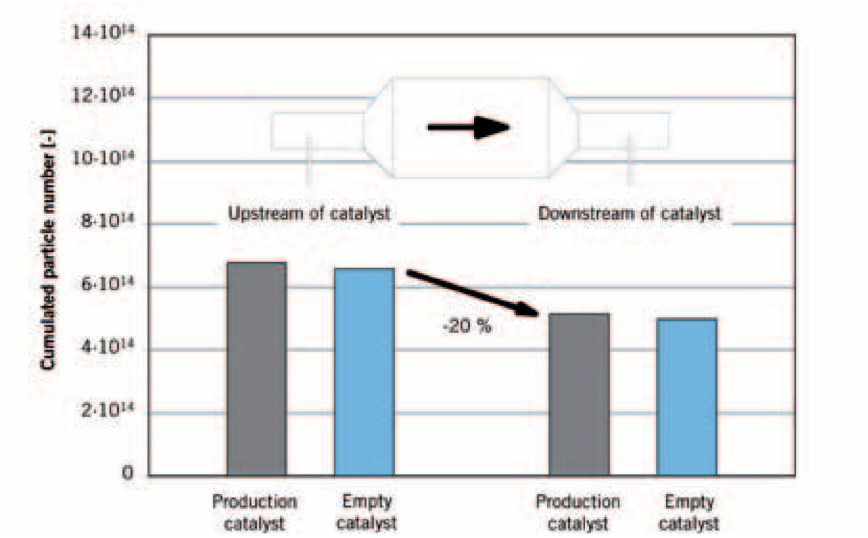
EFFECTS OF THE EXHAUST SYSTEM

Depending on the sampling point, identical untreated particulate emissions produce different results both in terms of particle concentration as well as in the distribution of modes, ❺. In particular, the close-coupled catalyst gives rise to a sharp reduction in the nucleation mode. Particulate reduction then tails off further down the exhaust system. The rear muffler no longer produces any measurable reduction range (23 to 2500 nm) evaluated in conformity with PMP. As part of further investigations, reference measurements were conducted on catalysts and empty casings without monoliths. Measuring the differences between the sampling point upstream and downstream of the catalyst returned a mean reduction of approximately 20 % from the catalysts as well as the empty casing.

This shows that the catalysts examined produce no identifiable chemical effect in relation to reducing particulate emission, ❻. This is confirmed by the experience that compared with new catalysts, ones that have aged have no appreciable influence on the particle result. In other words, timing and temperature are sufficient at the hot end to oxidize particles under the conditions prevailing in the exhaust-gas mass flow.



❺ Average size spread in NEDC at different sampling points



❻ Influence of catalyst in NEDC

OVERALL IMPLEMENTATION IN A VEHICLE

In keeping with IAV's calibration process adapted to optimizing particulate emission, the individual findings gathered from the various sub-investigations were evaluated, brought together in an optimized overall dataset and, following precision optimization, examined in exhaust-emission tests on a vehicle. These measures were shown to produce a positive effect, particularly in the familiar problem zones, cold starting/catalyst heating, drive-off cycles and high vehicle speed. The NEDC revealed a 65 % reduction in the particle number. A reduction in the particle number was associated with lower particulate matter in all exhaust-gas cycles. Further examination of these data in the FTP75 showed a reduction in particulate matter by 50 %.

SUMMARY AND OUTLOOK

The stages planned across the globe in future to limit particulate emission demand hardware modifications in a number of engine concepts and sweeping optimization of calibration in all of them.

The various influences fuel grades are shown to have are primarily the result of fuel-formulation weighting and oxygen content. Each examined variation of injection parameters or fuel generally demands an adjustment to the start of injection in ECU data. Increasing fuel pressure leads to a considerable fall in the particle number. Splitting the injected fuel quantity into separate injection events while adjusting timing not only enhances mixture preparation but also produces a reduction in the particle number. The greatest reduction is achieved by switching from single to dual injection.

A newly aligned, model-based process for optimizing particulate emission has enabled IAV to achieve a significant reduction in the particle number on a concept engine. This puts the company in an ideal position to resolve the challenges encountered in developing tomorrow's production vehicles.

REFERENCES

[1] Mercedes-Benz: Emissionen, Kraftstoffverbrauch. In: Vorschriften, Testverfahren und Grenzwerte Pkw und leichte Nutzfahrzeuge. Stuttgart, September 2008

[2] Konieczny, R.; Brück, R.; Zaldua-Moreno, N.: Die Herausforderung der zukünftigen Partikelanzahlgrenzwerte; Partikelentstehung und Partikelreduktion. 6. Internationales Forum Abgas- und Partikel-Emissionen, Ludwigsburg, 9.-10. März 2010

[3] Pflaum, S.; Wachtmeister, G.; Mackovic, M.; Frank, G.; Göken, M.: Wege zur Rußbildungshypothese. 31. Internationales Wiener Motorensymposium 2010

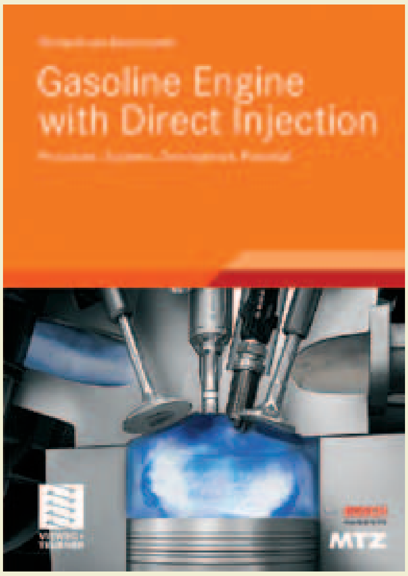
[4] Rossbach, M.; Velji, A.; Wagner, U.; Spicher, U.; Suntz, R.; Bockhorn, H.: Investigations of the formation and oxidation of soot inside a direct injection spark ignition engine. SIA Conference The Spark Ignition Engine Of The Future, Strasbourg, 2.-3.12.2009

[5] Röpke,K.: DoE-Design of Experiments. Verlag Moderne Industrie, 2005

[6] Kratzsch, M.; Kannapin, O.; Liebsch, S.; Preisner, M.; Guske, T.: Reducing the Particle Number in DI Gasoline Engines. 10th International Stuttgarter Symposium 2010

WWW.VIEWEGTEUBNER.DE

Gasoline Engines are the answer to the challenges of future



Richard van Basshuysen

Gasoline Engine with Direct Injection

Processes, Systems, Development, Potential

2009. xviii, 437 pp. With 399 Fig. Hardc. EUR 49,00

ISBN 978-3-8348-0670-3

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

authors | editors

Dr.-Ing. E. h. Richard van Basshuysen was Head of Development for premium class vehicles and for engine and transmission development at Audi. Today, he is editor of the magazines ATZ and MTZ. The editor was supported by a distinguished team of authors consisting of 22 experts and scientists from industry and universities.

Please send me

____ Copies
Gasoline Engine with Direct Injection
ISBN 978-3-8348-0670-3
EUR 49,00
(+ shipping and handling)

Fax +49(0)611.7878-420

Company ____ Last name | First name ____ 321 09 001

Department ____

Street name ____ Postcode/Zip | City, Country ____

Date | Signature ____

Please supply at retail price via bookshop or directly from Vieweg+Teubner Verlag. Subject to change without notice. 1|2009.
Managing Directors: Dr. Ralf Birkelbach, Albrecht F. Schirmacher. AG Wiesbaden HRB 9754

TECHNIK BEWEGT.

VIEWEG+TEUBNER

REDUCTION OF PARTICULATE NUMBER EMISSIONS THROUGH CALIBRATION METHODS

The ever-stricter emission limits for passenger cars are continually posing new challenges for engine development. The European Union with the introduction of Euro 5 has for the first time a gasoline emission standard for particulate mass, with Euro 6 in addition to particulate mass there will be a limit for the particulate number. AVL has investigated the possibilities of reducing internal engine particulate emissions, developed concepts for future emissions and tested the potential for the reduction of particulates by calibration methods.

AUTHORS

DR. PAUL KAPUS
is Manager for Gasoline Engine Development in the Field of Engineering and Technology Drive at AVL List GmbH in Graz (Austria).

DIPL.-ING. HELMUT JANSEN
is Manager for Calibration Gasoline Engines in the Field of Engineering and Technology Drive at AVL List GmbH in Graz (Austria).

DIPL.-ING. MARTIN OGRIS
is Technical Specialist Calibration Gasoline Engines in the Area of Engineering and Technology Drive at AVL List GmbH in Graz (Austria).

DIPL.-ING. PETER HOLLERER
is Lead Engineer for Emission Calibration, Calibration Gasoline Engines in the Area of Engineering and Technology Drive at AVL List GmbH in Graz (Austria).

CHALLENGE

The limitation of particulate emissions requires not only a rethink to the principles of the basic thermodynamic interpretation of the modern gasoline engine but also additional instrumentation. To achieve stable emission results for production, new calibration strategies will be necessary. AVL has investigated the possibilities of reducing internal engine particulate emissions and developed strategies for future emission concepts. The proposed limits for Particulate matter (PM) for Euro 5 currently does not require a high amount of additional calibration effort in order to reach the targets. The challenge is to meet the proposed Euro 6 targets with the current production gasoline engines with respect to particulate number. This technically raises the question of whether, to achieve the future limits, it is necessary to use exhaust gas treatment with a particulate filter (PF), or that it is possible to reduce the particulate emissions from the engine as required with internal engine changes, more specifically calibration changes. So, while public debate on fine particulate emissions has not yet considered the gasoline engine, and to avoid any extra costs for additional components for exhaust gas treatment, there exists within the car manufacturers a strong motivation to achieve the particulate limits by purely internal engine measures. The Euro 6 limit is not yet decided. The value for particulate number (PN) of gasoline engines will be decided in 2011. Within the following discussion, the Euro 6 diesel limit with 6.0 $x10^{11}$ particulates/km will serve as a benchmark against which the results will be compared.

COMPARING CONTEMPORARY TECHNIQUES FOR LEVELS OF PARTICULATE NUMBERS

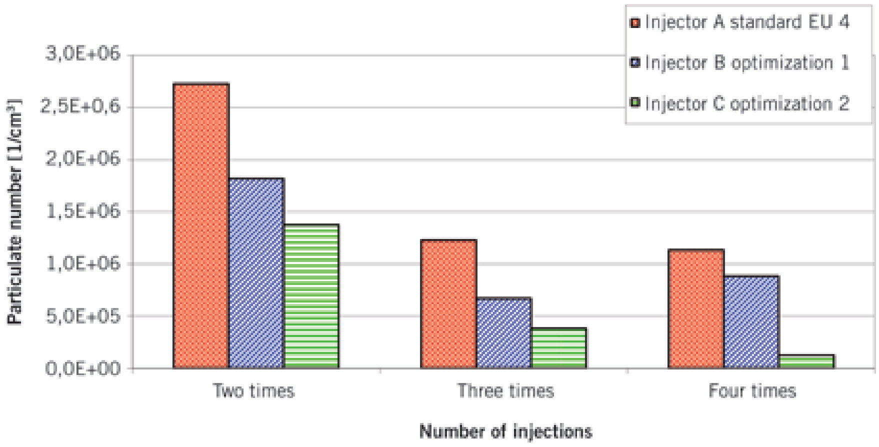
Only diesels with an after treatment of a diesel particulate filter (DPF) can reach the Euro 6 PN limit for diesel vehicles. When comparing the market, and techniques for passenger car engines, gasoline engines with external mixture formation show a comparable level of particulates as diesel engines with a DPF, but gasoline engines with internal mixture formation are by factors of 3 to 10 over this limit. For this reason the report refers to only direct injection gasoline engines.

CALIBRATION STRATEGY FOR EURO 4 AND EURO 5

A rich mixture in the local combustion zone, by the direct injection of fuel into the combustion chamber is in Euro 4 and Euro 5 calibrations part of the emission strategy. The deliberate stratification by double injection is a design criterion for the catalytic converter to achieve efficient rapid light-off behavior; this is realized with late ignition angles and achieves the lowest possible raw emissions. This proven combustion system leads to a stable emission concept with low misfire occurrences during catalyst heating idle speed, the resulting particulates are due to this adverse operating point.

PARTICULATE FORMATION

Sooty combustion is the result of inadequate treatment of the air-fuel mixture. Low fuel atomization or rather fuel droplets on cold



❶ Effect of injector and injection strategy on the particulate number

combustion chamber walls are the main causes of high particulate concentrations. Unlike a gasoline engine with port injection there is little time to homogenize and vaporize the injected fuel in the DI engine. This homogenization depends on the interaction of the atomization of injected fuel with the air charge movement due to tumble. To achieve a good mixture, the highest possible level of fuel atomization is required, this depends largely on the quality and design of the injector.

EURO 6 INJECTOR

The location of the injector has to be decided; a decision between the side and central installation position hast to be made. The side mounting position is said to be a wall-guided system and the central installation position leads to an air guided mixture formation. Both systems, with regard to particulate formation, face the same challenges, namely the prevention of wall wetting while optimizing the spray quality.

The injector is defined by:

: Number and shape of the injection holes
: spray pattern, position of the holes to each other and injection angle
: stationary flow rate Q_{stat} [g/min]
: minimal injection time
: production methods.

As further design criteria, the injector will not only be designed to meet performance, fuel, oil dilution, and the gaseous emissions, but also particulates. The Euro 6 compatible injector must meet high standards regarding quality spray whilst avoiding interactions of the injected fuel jets. Too high penetration depth leads inevitably to wall wetting and thus to the risk of oil dilution and/or particulate formation. On the other hand too low fuel pressure leads to fuel droplets and to particulate formation. Current studies show that the optimization of the injector can contribute significantly to the reduction of particulate number (PN), ❶. An optimized combustion spray pattern can be achieved by the injector method of manufacture, ①, or by optimization of specific steps with which the particulate emissions are significantly reduced. The piezo injector in a central installation position achieves good results for particulates; it operates well with a high fuel injection pressure and multiple injections, but at high cost and only works for a central injection position.

MEASUREMENT TECHNIQUES AND METHODS

For a successful Euro 6 production calibration, the methodology and the measuring system used during the combustion development must be considered together, ❷. The particulate measurement supports the development process and visualizes the particulate effects.

In thermodynamic engine development phase, the transparent engine [1, 2] is used to evaluate the mixture formation in standardized tests using visual measurement equipment to optimize the combustion and component layout. Standardized methodology for cold-start and catalyst heating are developed in parallel on the transparent engine and the engine test bed ("Virtual Vehicle Engine Testbed", VVETB), ❸.

❷ Development cycle and measurement

Additionally simulating the emission test cycle on the engine testbed investigating the cold dynamics and warm up behavior, using specific measurement tools such as “Particulate counter” [3], “Microsoot Sensor” and “Emission Bench” are carried out. The VVETB has established itself as an ideal test environment for the PN optimization, where the benefits of faster control and engine conditioning can be used, ②. Furthermore; the VVETB can easily simulate hybrid drive systems and be used to optimize the particulate emissions. During the thermodynamic phase, the layout of the basic design of the engine completed so that the potential for a successful Euro 6 calibration is high.

CALIBRATION OPPORTUNITIES FOR REDUCING PARTICULATE EMISSIONS

To illustrate the possibilities for calibration, a production 1.6 l turbo engine with gasoline direct injection was used, it already complied to Euro 4 standards. In the reference measurements over the NEDC, It reached a PN value of 2.5 $x10^{12}$ to 3$x10^{12}$ particulates/km, which corresponds to approximately 350 % to 450 % of the Euro 6 limit values for diesel vehicles. From this level it would mean that the particulate number for gasoline engines, with direct injection must be reduced by 80 % to 90 %, to reach the limit for diesel. For calibration, this means that the following functions are to be re-calibrated, ❹:

: cold start: 10 % reduction
: catalyst heating: 10 % reduction
: cold dynamic, starting a cold engine: 60 % reduction
: stationary and dynamic effects in hot engine: 20 % reduction.

The functions in the ECU software must now be, with the knowledge of the abovementioned mechanisms, calibrated and optimized further for the reduction of particulate formation. The key features were modified and the following explains the calibration quality that was achieved.

COLD START

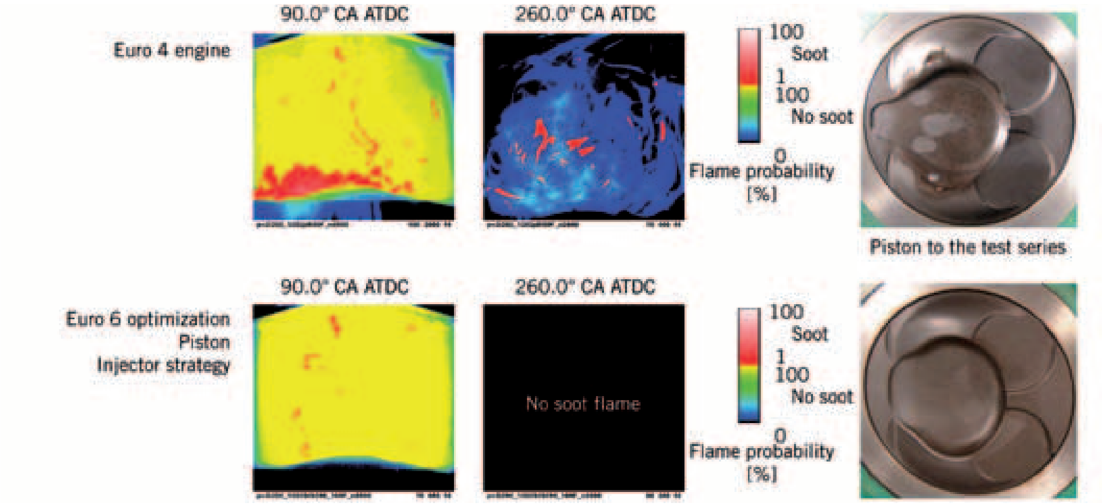
A better engine start under all possible environmental conditions such as cold, hot and altitude with different fuel qualities is the prime objective of each production calibration. Up to now, the start calibration for the emission start was only with respect to stable and misfire free engine speed acceleration, but in the future, the optimization effort has to consider the particulate emissions. The very

TEST ENVIRONMENT	MEASUREMENT EQUIPMENT	METHOD/TEST PROCEDURE	TASK
TRANSPARENT	Laser technology Indicating system Emissions AVL DIX Visio technology	Cold start Catalyst heating (AVL-CHIT@1200rpm) Part load/Full load	Component predefinition Strategy predefinition Qualitatively mixture evaluation
ENGINE TEST BED	Indicating system Emissions (bench) Particulate Counter Micro soot sensor Opacimeter (Visio technology on demand)	Catalyst heating (AVL-CHIT@1200rpm) Part load	Component predefinition Target catalogue evaluation Pre calibration
DYNAMIC ENGINE TEST BED (AVL-VVETB)	Indicating technology Emission (Bench) Particulate Counter Microsoot Sensor Opacity (Visio technology on demand)	Catalyst heating Dynamics and load Emission cycle simulation	Component predefinition Target catalogue evaluation Simulation of emission tests Pre calibration Robustness evaluation
VEHICLE	Fast FID Particulate Counter	Cold start Cold start warm-up	Optimization Evaluation Refined calibration
CHASSIS DYNO	CVS Particulate Counter Micro Soot Sensor	MVEG, ... FTP, ... Special tests Cold start warm-up	Optimization Statistics Refined calibration

❸ Test environment, methods and application

CALIBRATION WORK PACKAGE	MECHANISM OF PARTICULATE ACCRUEMENT	CALIBRATION ACTION	PARTICULATE INFLUENCING HARDWARE
COLD START	Insufficient mixture preparation Droplet accruement Wall wetting	High pressure start Multiple injection Optimised injection settings	Piston geometry Injector High pressure pump Volume of fuel rail Geometry of combustion chamber Fast recognition wheel pattern
CATALYST HEATING	Insufficient mixture preparation Wrong injection timing Wall wetting Stratification effects	PN optimised injection strategy Multiple injection Rail pressure Camshaft position Ignition angle	Piston geometry Injector High pressure pump Geometry of combustion chamber Camshaft
COLD DYNAMIC	Insufficient mixture preparation Wrong injection timing Wall wetting Stratification effects Too rich air/fuel ratio	Dynamic calibration of injection settings Multiple injection Rail pressure Camshaft position Ignition angle	Piston geometry Injector High pressure pump Geometry of combustion chamber Camshaft
STEADY STATE AND DYNAMIC EFFECTS AT HOT ENGINE	Insufficient mixture preparation Wrong injection timing Wall wetting Stratification effects To rich air/fuel ratio	Steady state injection strategy Multiple injection Rail pressure	Piston geometry Injector High pressure pump Geometry of combustion chamber Switchable water pump

❹ Influence of calibration and hardware on particulate results

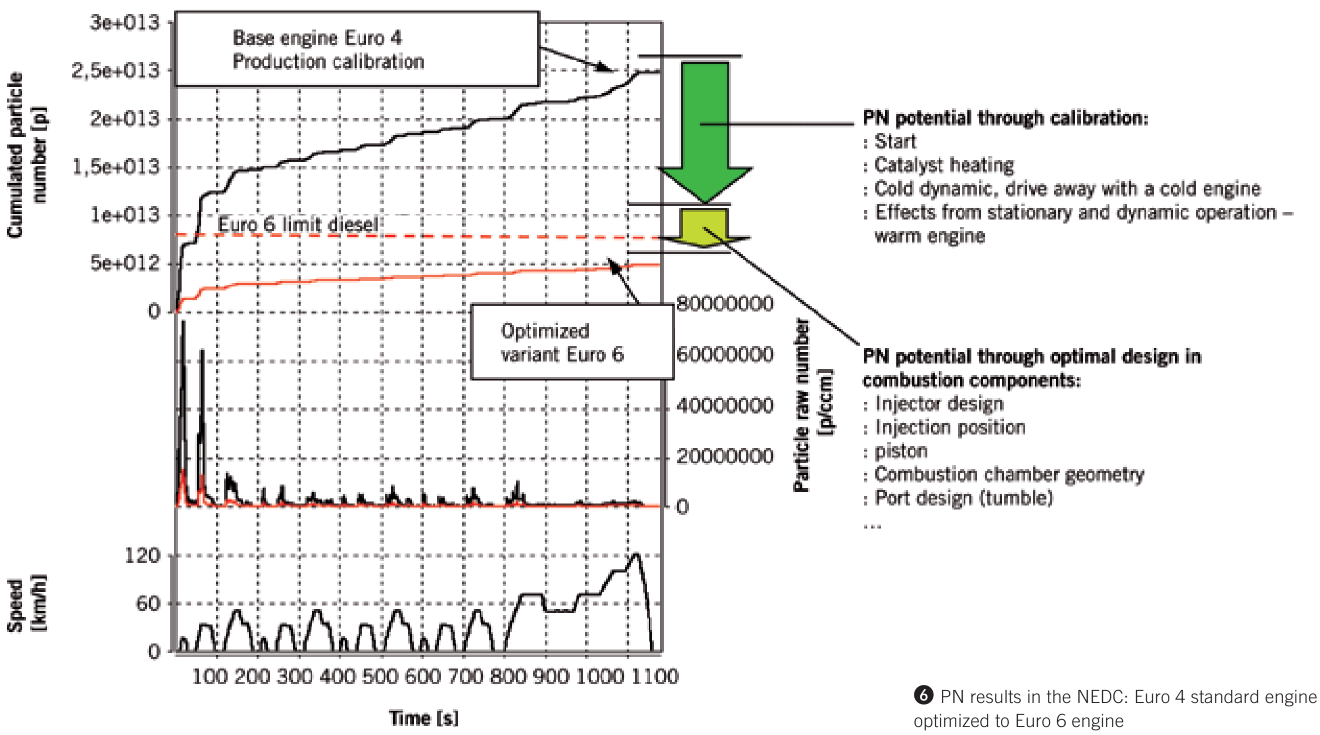


❺ Glass engine results: Euro 4 standard engine optimized to Euro 6 engine

first start injection has to be of the highest quality for mixture formation. In principle, it is important to start with as high as possible fuel rail pressure and with an optimized injection strategy so that in the combustion chamber there is minimal wall wetting against the cold combustion chamber walls. Wall wetting can be largely prevented by an intelligent injection strategy (e.g. multiple injections). To avoid having a longer start time due to a delayed injection, a matching of injection hardware is recommended.

CATALYST HEATING

In catalyst heating mode a late ignition angle causes much of the combustion energy to dissipate as heat in the exhaust, causing the combustion chamber to heat up slowly. Fuel droplets, which at this stage come into contact with the cold combustion chamber walls, cannot evaporate completely, thus leading to high particulate emissions. The same principles are applied here as in the start criteria, keep the fuel in suspension and to provide the highest possible ignitable mixture at the spark plug. For this type of engine layout there are currently two different injection strategies for catalyst heating with double injection. For Euro 5 applications, the wall-guided system with later second injection position (70° KW to 40° KW before TDC) has been established, in future specifically Euro 6 it may become necessary, to advance the second injection and thereby reduce the particulate concentration. With an earlier second



❻ PN results in the NEDC: Euro 4 standard engine optimized to Euro 6 engine injection less fuel reaches the cold piston, by reflection of the fuel injection off the rear of the piston bowl an additional atomization of the fuel can be achieved. This calls for an intelligent choice of piston geometry. Injection timing, number of injections, ①, fuel rail pressure and distribution factors need to be modified accordingly for Euro 6. There are more than two injections necessary. The transparent engine and catalyst heating test (AVL ChIT @ 1200rpm) predefine the injection parameters on the engine test bed with an optimized DOE method, ❺.

DRIVE-AWAY WITH A COLD ENGINE

Load and speed transients in the NEDC contribute to approximately 60 % of the overall PN emissions, a large proportion arise during driving of the first three hills of the test cycle, within the first 200 s of the test, ❻. In the subsequent hills, the PN curve flattens off significantly, an effect of the warmer combustion chamber. During acceleration of the engine, a high load is required; at this point, the particulate raw emissions are the highest. The mechanisms are similar to the start and catalyst heating; the calibration challenge is to optimize the particulate emissions while not sacrificing the other emissions or driveability of the engine during transient operation. Calibration possibilities within transient operation can also incorporate multi-injection strategies as well as a combination of fast engine warm up methods, for example the use of a switchable electric water pump.

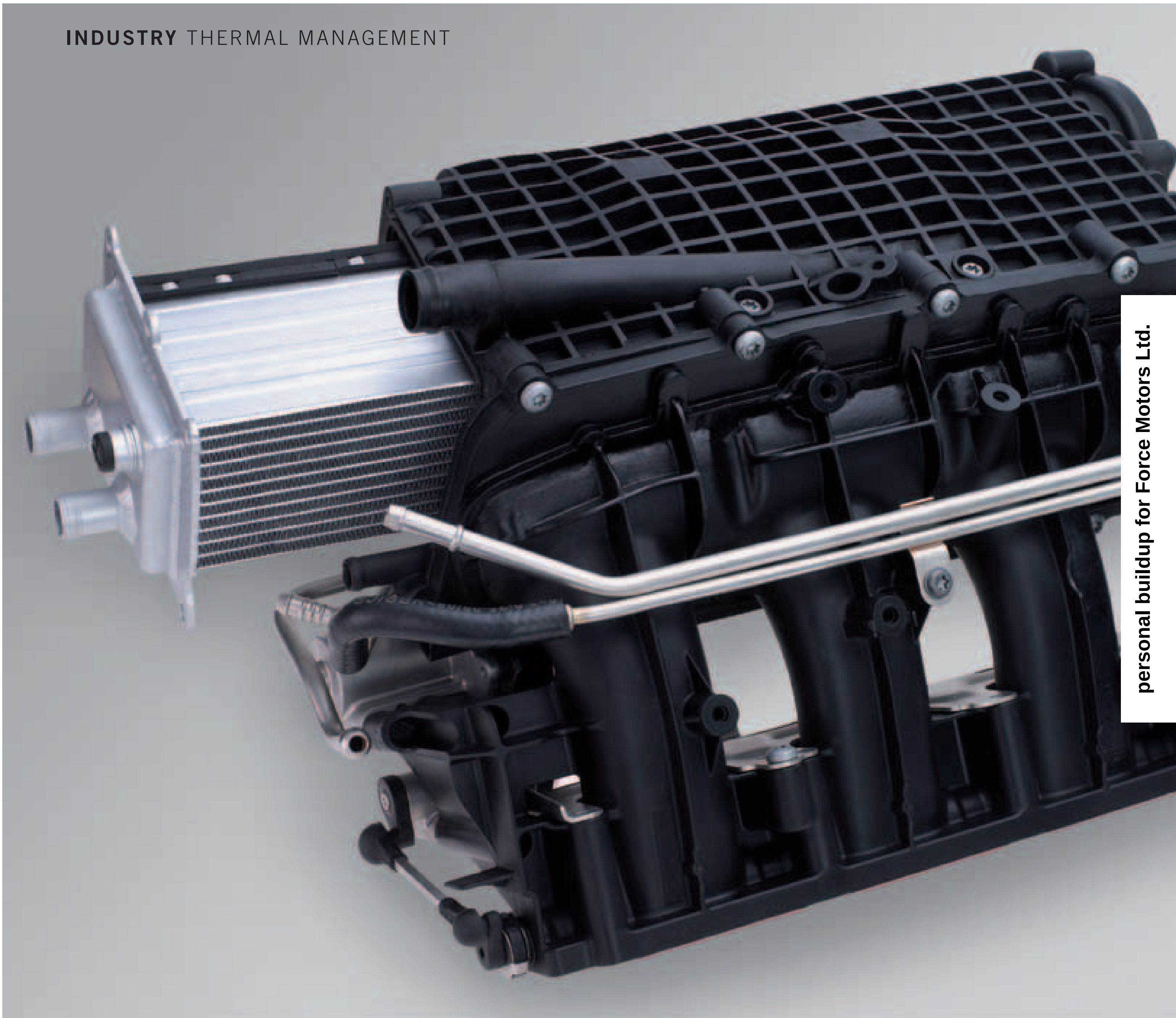
STATIONARY AND DYNAMIC EFFECTS IN A WARM ENGINE

In the EUDC (extra urban drive cycle of the NEDC), there are higher vehicle speeds and longer acceleration phases than during the city cycle. The above-described mechanisms for particulate formation are engine operating conditions, however, when the engine is warm these have only a reduced influence, so calibration methods used during cold operation for PN reduction are not necessarily fully required. However, it is also worth finding the optimal injection strategy for the calibration and implementing these injection parameters into the engine calibration.

SUMMARY

In comparison to the Euro 4 starting point, gasoline engines without the use of a PF can, with calibration methods, demonstrate clear improvements to the particulate emissions but only when the optimal components for the calibration are considered. The engine must be designed for Euro 6 with the optimum thermodynamic layout in mind, utilizing the test environments and instrumentation available, such as transparent engine, Particulate Counter or Microsoot Sensor. These are necessary and in combination with calibration and new methodology, is the basis for a successful calibration. Within the ECU new functionality and strategies for cold start, catalyst heating, dynamic and warm-up must be developed. A large proportion of the particulate number emissions can be reduced with an Euro 4 system by means of intelligent calibration strategy, further clear improvement can be achieved with optimized engine technology and an optimized combustion process.

REFERENCES
[1] Fuchs, H.; Hopfner, M.; Kapus, P.; Winklhofer, E.: Methods and criteria for integration into fuel injector boosted gasoline direct injection engines. Injection systems for IC engines, IMechE London, 13-14 May 2009
[2] Fuchs, H.; Hirsch, A.; Ogris, M.; Winklhofer, E.: Mixture Quality Evaluation For Transient Mode Gasoline Engine Calibration. ICEF2010, San Antonio, USA September 12-15, 2010
[3] Giechaskiel, B.; Cresnoverh, M.; Jörgl, H.; Bergmann, A.: Calibration and accuracy of a particle number measurement system. Measurement Science and Technology 21 (2010) 045102 (13pp)



INTAKE MODULE WITH INDIRECT INTEGRATED CHARGE AIR COOLER

Within a common project of Behr and Mann+Hummel a comparison between direct and indirect charge air cooling integrated in the intake manifold was carried out by vehicle measurements in the wind tunnel. In comparison with the series-production solution based on direct charge air cooling, a prototype intake module (intake manifold with integrated indirect charge air cooler) significantly reduced charge air pressure loss, provided a highly uniform temperature distribution in the runners, and reduced the charge air volume between the compressor and point of entry into the engine.

AUTHORS

DIPL.-ING. KARL-ERNST HUMMEL is Project Manager Advanced Development Thermodynamics at Mann+Hummel in Ludwigsburg (Germany).

DR.-ING. BERNHARD HUURDEMAN is Head of Advanced Development Thermodynamics at Mann+Hummel in Ludwigsburg (Germany).

DIPL.-ING. JOHANNES DIEM is Project Manager Advanced Engineering Engine Cooling Passenger Cars at the Behr GmbH & Co. KG in Stuttgart (Germany).

DR.-ING. CHRISTIAN SAUMWEBER is Head of Advanced Engineering Engine Mounted Components Engine Cooling Passenger Cars at the Behr GmbH & Co. KG in Stuttgart (Germany).

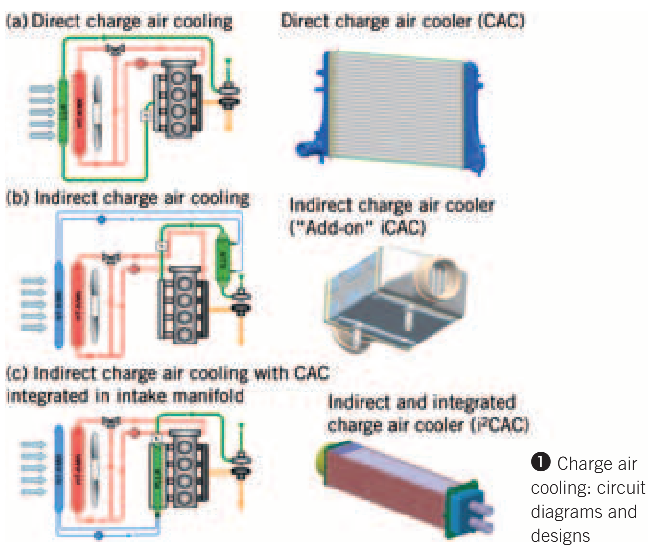
LAYOUTS OF CHARGE AIR COOLERS

The current debate on the need to reduce CO_2 emissions will lead to a noticeable increase in the proportion of turbocharged engines in the next few years, with a clear trend towards “downsizing”. More efficient combustion at all operating points and higher boost pressures require further enhancements in charge air cooling and intake systems. The current state of the art for charge air coolers (CAC) is direct CACs, as an air-cooling device that is installed in the vehicle front end. However, indirect CAC technology is also gaining ground. This type of CAC, which can be located virtually anywhere in the engine compartment, uses a cooling medium that is re-cooled via an air-coolant cooler (known as a low-temperature radiator, or LTR). While the additional heat exchanger and secondary cooling circuit increase the complexity of the cooling system, this approach offers some very attractive advantages in comparison with direct charge air cooling.

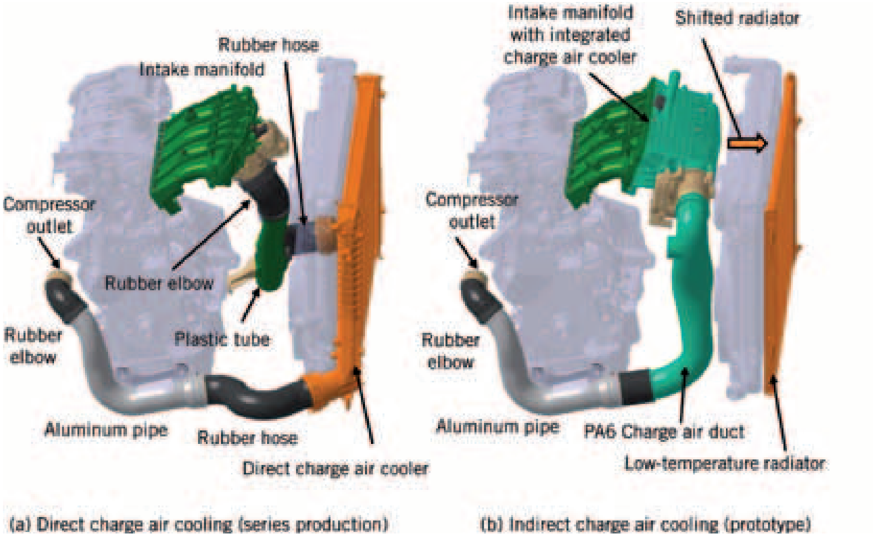
: An indirect CAC can be a very compact unit, allowing installation close to the engine. This substantially shortens the distance between the compressor and CAC, and between the CAC and the engine.
: Significantly lower charge air pressure losses increase volumetric efficiency.
: Significantly reduced charge air volumes between the compressor and engine inlet improve the engine response.
: The more compact size of an LTR in comparison with a direct CAC of equivalent performance makes more package space available in the front end.

These advantages can be further leveraged by integrating the indirect CAC directly in the intake manifold, rather than fitting it as an add-on component between the compressor and the throttle body. The circuit diagrams and designs of the coolers for the various types of cooling system are shown in ❶.

In the past Behr and Mann + Hummel were involved in joint advanced engineering projects with an automaker for the development of a charge air cooling system integrated in the intake manifold. In 2008 another joint collaboration project was started to compare a series-production direct charge air cooling system



❶ Charge air cooling: circuit diagrams and designs



❷ Installation comparison for direct and indirect charge air cooling

with indirect charge air cooling. The tests were mainly focused on thermodynamic and flow comparisons between the two charge air cooling variants. A deliberate decision was made not to adapt the engine control system for indirect charge air cooling. The benefits of an indirect charge air cooling system directly integrated in the intake manifold are described and quantified in the remainder of this paper.

INITIAL SITUATION AND BACKGROUND TO THE COMPARISON

Since Behr and Mann + Hummel are series-production suppliers of the engine cooling module and intake manifold for the VW Passat 2.0 l TFSI, it was a logical decision to use this vehicle as the basis for the comparison between direct and indirect CAC.

The transverse 147-kW TFSI engine has a turbocharger on the exhaust manifold before the firewall, with the intake manifold placed on the opposite side of the engine, near the engine cooling module, ❷ (a). The compressor output of the turbocharger exits via a rubber elbow into an aluminum pipe, which is connected via a rubber hose to the full-surface charge air cooler located in the front end before the engine radiator. From the CAC, the air is delivered through a rubber hose, plastic pipe, rubber elbow, and the centrally located throttle body into the intake manifold.

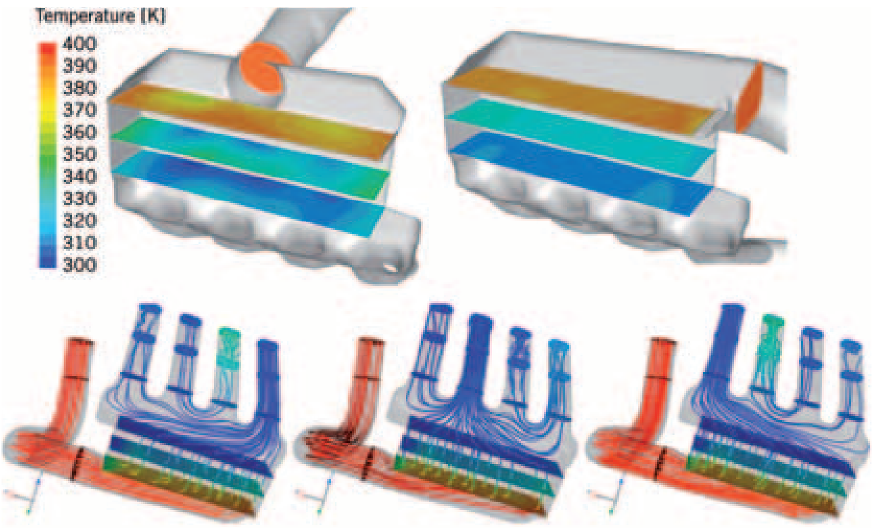
For the conversion to indirect charge air cooling, along with the integration of the CAC in the intake manifold it was also necessary to carry out a complete reconfiguration of the charge air ducts except the series-production aluminum charge air duct.

DESIGN CONCEPT SELECTION

The prototype was developed on the basis of extensive packaging analyses, CAD models of several concept variants, and flow simulations. Replacing the 32-mm-deep series-production charge air cooler in the front end with a 16-mm-deep LTR meant that the engine cooling module could be moved further forward, thereby increasing the intake manifold package space, ② (b). The centrally positioned throttle body in the production vehicle was moved to the side to achieve a more favorable lateral flow to the CAC and a uniform temperature distribution in the runners. A comparison of the stationary simulation results for the two throttle body positions, ❸ shows the thermodynamic advantages of lateral inflow, with a very uniform temperature distribution at the charge air cooler outlet for the prototype geometry. Further non-stationary simulations incorporating the firing order also confirmed a favorable evenness of air mass and temperature distribution between individual cylinders, ③ (below).

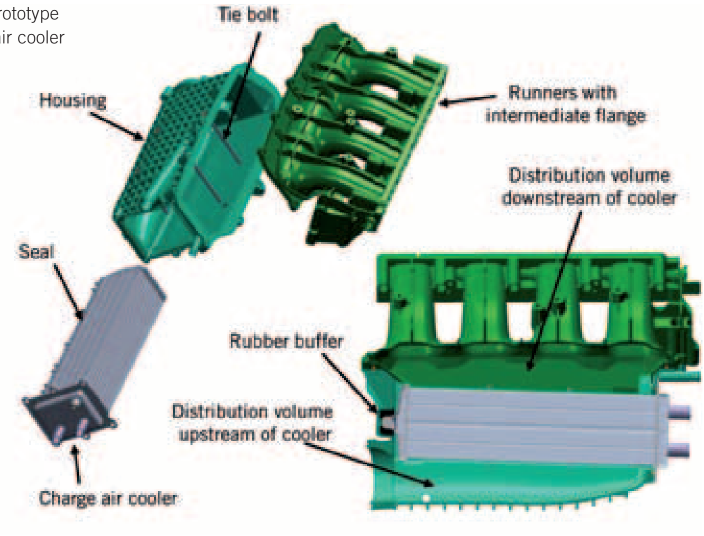
CHARGE AIR COOLER AND LOW-TEMPERATURE COOLANT CIRCUIT

The direct charge air cooler was replaced with a separate low-temperature (LT) coolant circuit, comprising an LTR, an indirect CAC integrated in the intake manifold, ① (c), a surge tank, and an electric water pump. Advance design calculations for the LTR core were carried out with a one-dimensional cooling system simulation, so that together with the integrated CAC it is equivalent to the direct CAC in terms of cooling performance. The design chosen for the integrated indirect CAC was a cross counter-flow cooler with a fin-and-tube matrix, charac-



❸ Flow simulation for intake manifold with integrated charge air cooler – top: temperature distribution at inlet, in middle, and at outlet of integrated charge air cooler for middle and lateral flow into cooler; bottom: path line snapshots with temperatures for non-stationary simulation

❹ Intake manifold prototype with indirect charge air cooler



terized by a very uniform charge air temperature distribution in the outlet cross section, and also low charge air pressure losses, very compact dimensions, and considerable flexibility.

AIR DUCTING THROUGH THE INDIRECT SYSTEM AND INTAKE MANIFOLD

In the indirect charge air cooler, ② (b), the air leaves the series-production aluminum duct via a new PA6 charge air duct designed to the available package space. The air flow then passes through the throttle body located at one side of the intake manifold into the plenum, into the integrated cooler, and then into the runners of the intake manifold.

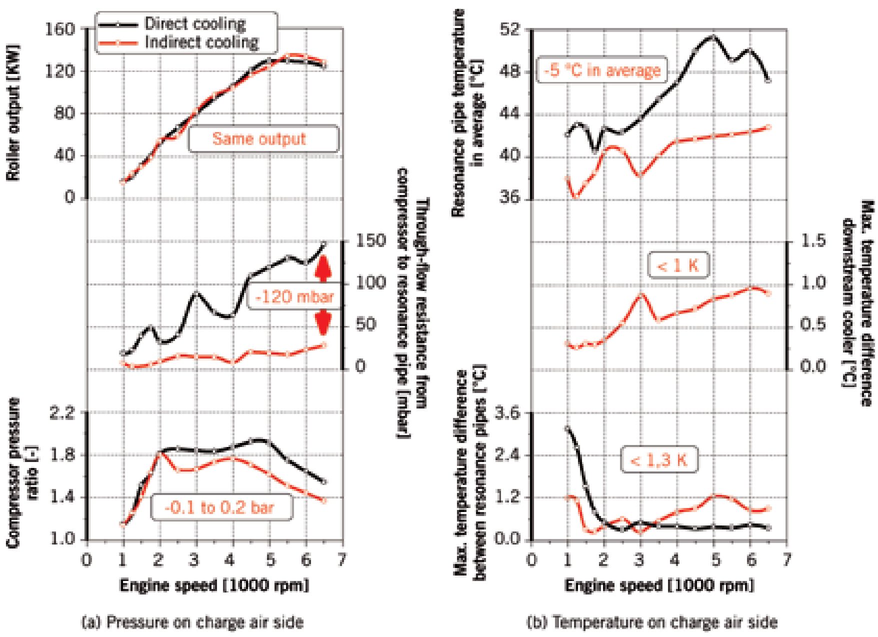
For reasons related to the mounting location and assembly operation, the intake manifold was designed as two separate parts, ❹. The first part contains the intake manifold runners, bellmouths, and cylinder-head flange of the series-production intake manifold. A fiberglass-reinforced PA6 housing was made using a silicone cast process for the new prototype with integrated CAC. A decision was made to insert the cooler into the side of the intake manifold to minimize the flange surface area, the number of bolting points, and the sealing length. The cooler was assembled into the manifold with a rubber grommet on the opposite side of the mounting flange. A high priority was given to the internal seal between cooler and inside wall of the housing in order to minimize bypass flow, which could lead to an uneven temperature distribution between individual cylinders. Four tie-bolts were fitted to reduce any expanding of the housing under high boost pressures, the charge air leakage around the cooler, and to improve burst strength.

Because the new intake manifold concept is heavier than the series-production version, it requires a stiffer bracket to reduce the vibration load. Vibration test bench measurements showed that the acceleration at the throttle body could be reduced to the same level as the series-production manifold. The shorter duct lengths of charge air ducts and the smaller size of the charge air cooler integrated in the intake manifold reduced the charge air volume (compressor outlet to cylinder head inlet) by approximately 30 % in comparison with the series-production design. A reduction of up to 50 % could be possible in transversal mounted engines if the entire system was redesigned.

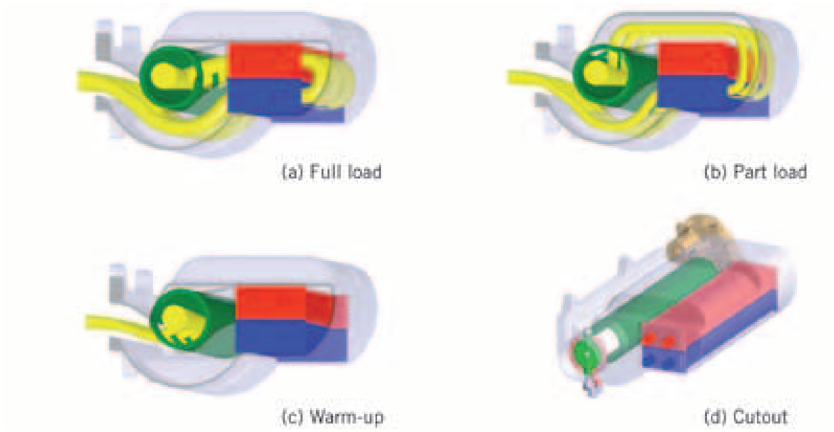
VEHICLE MEASUREMENT RESULTS

To provide the basis for comparable and reproducible measurement results for the cooling system in real-life vehicle conditions, the vehicle was first measured in the Behr wind tunnel in the series-production version with direct CAC. Afterwards, the conversion to charge air cooling integrated in the intake manifold was carried out, and a second set of wind tunnel measurements was generated. For that purpose, the vehicle was fitted with pressure sensors along the charge air line from the air filter to the intake manifold, and with low-pressure indication in the intake manifold and exhaust manifold. Temperature sensors were applied to the cores in the engine cooling module, the runners, the exhaust line, and at the inlet and outlet (six sensors in each case) of the integrated CAC. The wind tunnel measurement results are shown in ❺:

The vehicle could be driven with the same power output without any change to



❺ Wind tunnel results for full load conditions



❻ Principal swichting options of U-flow HT- and LT-charge air cooling with rotary valve

the engine control system. There were significant advantages in terms of charge air pressure losses. The pressure loss from the compressor to the runners was reduced by 80 %. The required boost pressure was reduced by up to 0.2 bar. This advantage can be used either to boost power (i.e. higher intake manifold pressure) or to improve response through the use of a smaller turbocharger.

The temperature sensors in the runners displayed excellent evenness of distribution (temperature difference < 1.3 K), with a temperature up to 5 K lower than in the series-production system. The temperature difference directly behind the cooler between the six measurement points placed at the outlet was less than 1K. This led to an improvement in knocking behavior, creating the possibility for consumption-reducing optimization of ignition timing.

OUTLOOK: COMPACT COOLING DESIGN CONCEPT FOR HIGHER POWER REQUIREMENTS

Highly supercharged small engines with a small cylinder displacement volume and high-performance engines will prompt a demand for higher power ratings for charge air cooling. In both cases, improvements to overall cooling systems in terms of package space requirements, cost, and weight can be achieved by moving to a new “cascade cooling” system. This involves indirect cooling of the charge air in two stages:

: a first charge air cooler is used to extract heat using coolant from the high-temperature (HT) engine cooling circuit
: a second LT coolant-cooled CAC then brings the charge air temperature down to the required end value.

Cascade charge air cooling reduces the thermal requirement on the LT circuit, allowing the use of smaller LT pumps, which may not even require automatic control, and is therefore less expensive.

For small engines, the use of cascaded HT-LT charge air cooling could provide the basis for more compact indirect charge air coolers integrated in the intake manifold. The main factor determining the cooling performance of the charge air cooler is the air-side flow length through the cooler, which therefore has to remain constant. Placing the HT and LT cooler cores one upon the other and reducing the cross-section area of the coolers results in a U-shaped through-flow configuration in the cooler on the charge air side, which is very compact and has only very slightly increased charge air pressure losses. One particularly promising option is the combination of a charge air cooler with a U-flow configuration with a rotary valve, ❻. The three different switching control options allow adjustment of the charge air temperature over a very wide range:

: full-load mode: flow through both the HT CAC and the LT CAC, ⑥ (a)
: partial load mode: the HT CAC is bypassed (to prevent the charge air being heated by the hotter coolant) and the charge air is cooled only in the LT CAC, ⑥ (b)
: during the engine warm-up phase, both coolers are bypassed, ⑥ (c), which is made particularly easy by the U-shaped arrangement of the coolers.

High-pressure exhaust gas recirculation can be restricted with the rotary valve. A cutout function for diesel engines can be provided with an additional rubberized flap, ⑥ (d), which could for example be activated via a simple mechanism by the rotary valve.

This combination of U-flow HT and LT CACs and a rotary valve provides the basis for a very light, compact, and cost-effective charge air cooling system.

SUMMARY

Vehicle measurements were carried out to compare direct charge air cooling with indirect charge air cooling integrated in the intake manifold. First, measurements on the series-production vehicle with direct charge air cooling were performed. Afterwards, the air path, cooling module in the front end, and the intake manifold were redesigned using a variety of CAE methods, in order to create an indirect charge air cooling system integrated in the intake manifold. In a direct comparison with the series-production solution, the changeover to indirect charge air cooling integrated in the intake manifold resulted in lower charge air pressure losses up to 80 %, a very uniform temperature distribution in the runners (< 1.3 K), and a reduction of approximately 30 % in the charge air volume between the compressor and engine inlet.

For future highly charged and high-performance engines, the changeover to cascaded indirect charge air cooling is likely to offer further advantages in terms of package space and system costs. A design concept has been presented for a cooler unit with a U-shaped flow configuration on the charge air side, which in combination with a rotary valve would lead to an extremely compact and powerful intake module allowing demand-oriented charge air cooling.

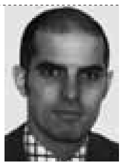
HYBRID COOLANT PUMP WITH ELECTRIC AND MECHANICAL DRIVE

The new hybrid coolant pump from BorgWarner combines the advantages of electrically and mechanically driven pumps in a single system. In electric mode, the pump can be flexibly controlled, while in mechanical mode it operates with a high efficiency. Extensive simulation by BorgWarner has demonstrated the potential of the hybrid pump for achieving further improvements in fuel economy.

AUTHORS

DR. PHILIP KELLER
is Technical Specialist for Numerical Simulation at BorgWarner Engine Systems Group in Auburn Hills (USA).

DR. WOLFGANG WENZEL
is Program Manager at BorgWarner Engine Systems Group in Ludwigsburg (Germany).



DR. MICHAEL BECKER
is Director Advanced R&D at BorgWarner Inc. and Head of the Engine Systems Group in Ludwigsburg (Germany).

JOSHUA ROBY
is Lead Engineer Thermal Innovation at BorgWarner Thermal Systems in Marshall (USA).

VARIABLE COOLANT PUMPS INCREASE EFFICIENCY

Thermal Management of combustion engines becomes more and more important due to the rising demand to reduce CO_2 emissions. Improved cooling circuit components, especially variable coolant pumps allow the reduction of parasitic losses from auxiliary components and also the controlling of the engine temperature for best engine efficiency. To evaluate different system and component concepts with regard to their influence on fuel consumption, simulation offers a wide range of opportunities, [1, 2].

HYBRID COOLANT PUMP CONCEPT

A key direction of thermal engine management is variability of cooling components. Variable coolant pumps can adapt the coolant flow to the requirements of an engine. The simplest version of a variable pump is the on-off pump, which improves fuel consumption during engine warm up. Pumps of this type are already in series production. Fully variable electric pumps offer much more potential for effective thermal management, allow cooling after engine stop and support start stop applications. They are also in series production. The main issue for this kind of pump is the limitation of the electric power in the usual 12/24 V automotive electrical system and the poor efficiency at high pump loads.

A hybrid coolant pump (HCP) featuring both mechanical and electrical drives combines the advantages of all other pump concepts such as: cooling after engine stop, compatibility to start-stop systems, control of pump power in engine part load for efficiency and low power consumption at high engine loads due to the highly efficient direct mechanic drive.

COOLING SYSTEM REQUIREMENTS

Current coolant pumps often require up to 2 kW driving power at full load. Pressure loss optimized coolant circuits might help to reduce the pump power but on the other hand additional cooling functions or new engine layouts, e. g. LP-EGR systems on gasoline engines, add new thermal load to the cooling circuit as well as water-cooled exhaust manifolds and turbine housings, [3, 4, 5]. Such concepts also require after-cooling after engine stop. In addition, powertrain electrification leads to new requirements for vehicle cooling systems.

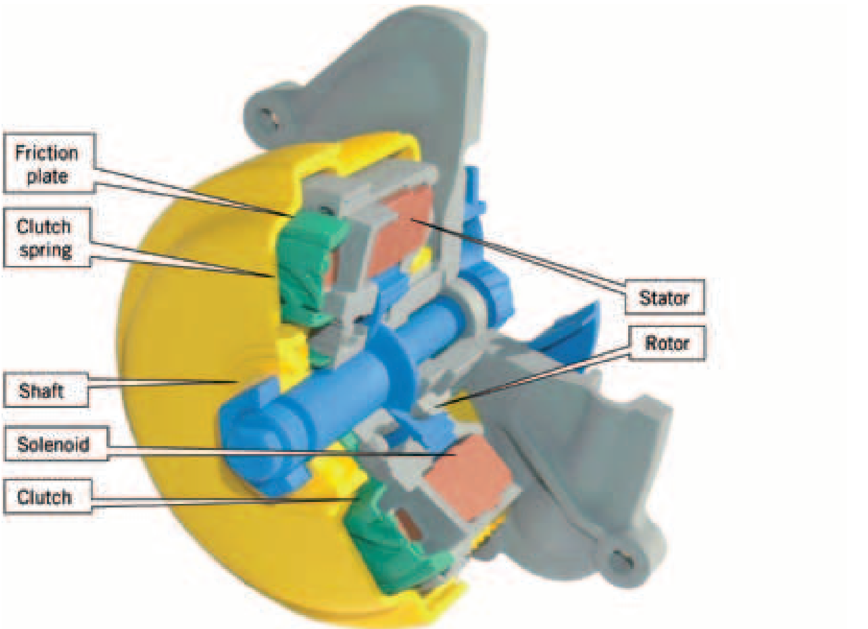
By investigations of BorgWarner on various vehicle types and driving cycles it was found out that a conventional coolant pump operates primarily in two prominent load ranges: one at rather low load and the other at high load. Based on these considerations the HCP concept was developed to handle the low power operation area with electrical drive and the high power operation mode with mechanical drive. This approach offers high variability in the low to mid power area and an efficient drive at high power requirements.

FUNCTIONAL CONCEPT

The standard coolant pump for a 3.0-l-V6-diesel engine for a light duty truck was chosen as a base system. The sizing and power of the HCP was defined to fit the base system with a speed range from 0 to 7000 rpm and a maximum driving power of up to 2 kW. In electrical mode the speed can be set to zero or adapted to the required coolant flow. In the mechanical mode, which is also the fail safe mode, the speed depends directly on the engine speed.

The main components of the HCP are shown in ❶. The shaft (blue) carries the impeller wheel from the original pump. The housing to adapt the pump to the engine (grey) has the same dimensions as the original part. In electric mode the e-motor, consisting of rotor and stator (red), powers the impeller as the solenoid (red) is actuated and the dry friction clutch is opened against the clutch spring. In mechanic mode the solenoid is turned off, the clutch is engaged and so the outer pump housing (yellow) which also carries the pulley is connected to the pump shaft. The e-motor is disabled, so almost no electrical losses occur, and the shaft rotates synchronous to the crank speed (according to the belt ratio).

It was high priority during the development to maintain the relevant package envelope of a conventional pump and to keep the cost below a fully electric pump. Package and interface design of the HCP are very close to the reference pump so a replacement of the reference pump to the HCP on the target engine was possible



❶ Layout of the HCP

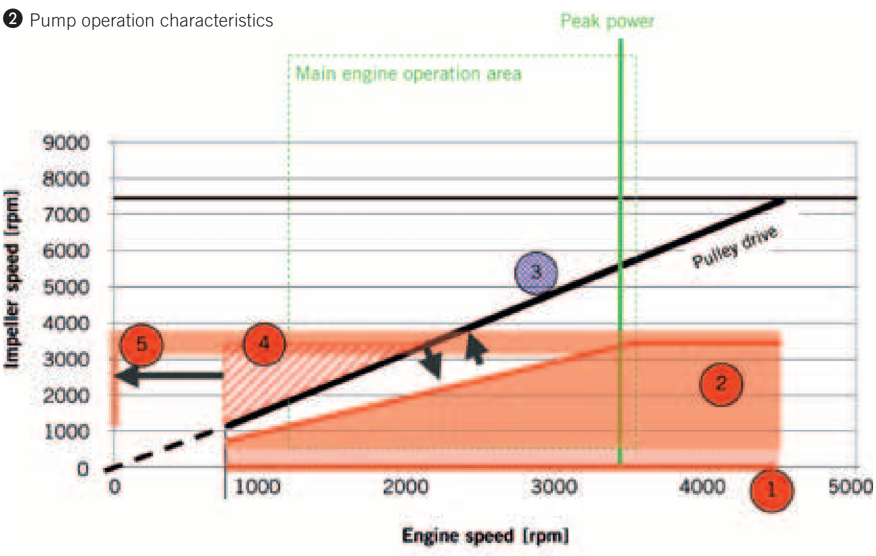
without any further changes. The cost could be kept low because of the smaller size e-motor and the simplified yet effective clutch mechanism.

OPERATION STRATEGY AND PARAMETERS

The general switching strategy is shown in ❷ where impeller speed is plotted versus engine speed. The pump characteristics can be described as follows:

: Mode 1: Pump is stopped for cold start and engine warm up.
: Mode 2: Main electric operation area. Pump speed can be set independent from engine speed with potential for optimized thermal management. The upper limit of this range is given by the maximum power output of the e-motor (an important design parameter) which is not a fixed value due to the range of flow resistance of the cooling circuit and the changing coolant viscosity over temperature.
: Mode 3: From a certain pump power on the mechanical mode is more effective than the electrical mode. This upper limit for main electric area is given by the mechanical drive curve with a fixed belt ratio, represented in ② by the thick black line (No. 3). The gap between both modes is linked to the efficiency chain difference. In the mechanical mode, only the belt friction and the bearing friction have to be taken into account while in electric mode, the alternator and e-motor efficiency have to be added to the chain. The gap (represented by the arrows) is to be avoided by the switching strategy.
: Mode 4: If required for low engine speeds, the pump power in electrical mode can be set above the limit for mechanical power. This function can support passenger compartment heating and provides additional degree of freedom for the pulley ratio.
: Mode 5: At engine stop, the pump can run with variable speed on electric mode for after-cooling.

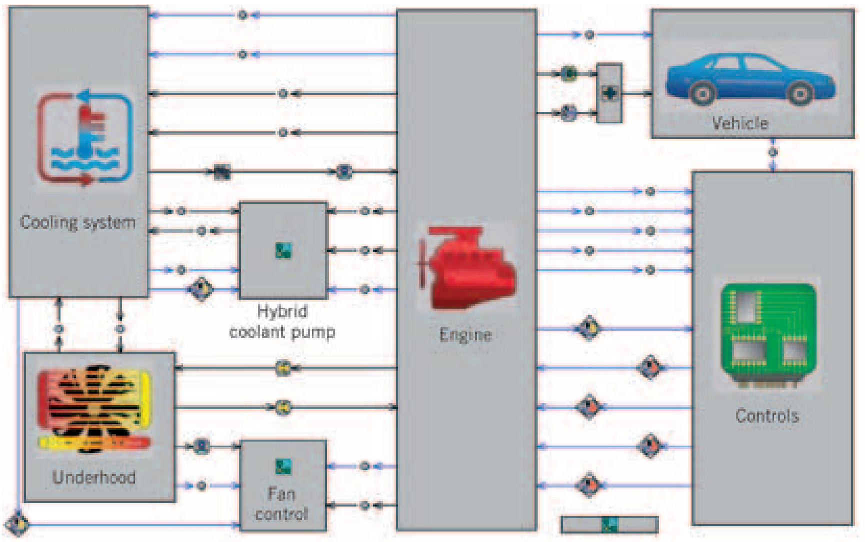
❷ Pump operation characteristics



A main driver for the cost and the package is the e-motor size. There are practical limits for the sizing. If the size is too small, the fully variable range of the pump operation map is very small. For cabin heating at low engine speed it is necessary that the electrical drive is able to provide enough coolant flow at the cold start condition when the thermostat is closed and the coolant viscosity is relatively high. Another requirement is related to providing high enough coolant flow after engine stop. If the e-motor is too big, a negative impact on the package requirements and component cost must be expected.

The definition of the mechanical driving area depends on the efficiency chain of both the electrical and mechanical sides. If the total efficiency chain between impeller and crank is taken into account, the overall efficiency of the electric drive is usually lower. The use of starter generators and recuperation of brake energy improves the efficiency in the board net and can increase the area of electric driving.

In addition to the mechanical design parameters the switching strategy needs to be optimized. The pump is operated by a pump controller which can be a separate module or is part of the ECU. In any case the pump needs to be added to the overall thermal management of the engine (if existing) to ensure a proper interaction of the components.

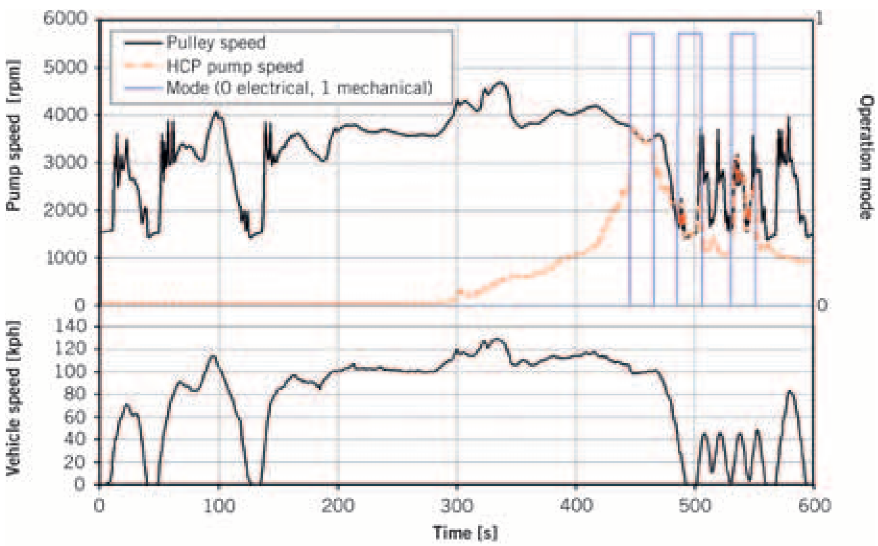


❸ Model structure

SIMULATION ENVIRONMENT

The evaluation of the HCP in the cooling circuit environment is carried out with a simulation model which was developed using the software "GT-Suite" from Gamma Technologies Inc. The model is described in detail in [2]. It consists of sub models for the vehicle, cooling circuit, front-end cooling package and engine with a simplified oil circuit. The engine model is actually a mean value model which uses a neural network analysis to reduce simulation duration. The engine load is defined by the cycle and the vehicle driving resistance. The vehicle model is a light duty delivery truck. The air-coolant interface at the front-end is modeled in a quasi 3-D environment. All models are coupled and, therefore, allow realistic feedback from one subsystem to another. The overall model structure is shown in ❸.

In contrast to the non variable pump to the HCP requires a controller which calculates the pump speed as a function of the thermal engine condition. The fan is driven by the crankshaft speed controlled by a viscous clutch. The HCP simulation model consists of a controller part which calculates the required pump speed and a second part which decides what mode should be used.



❹ Vehicle speed (bottom) and results of the simulation with a 200 W E-motor (top) for the US06 emission cycle

PARAMETER OPTIMIZATION

The US06 emission test cycle, ❹, was chosen to compare results of the HCP. With an average speed of 77.3 kph and a maximum speed of 129 kph, it is fairly aggressive. For the performance evaluation, two initial temperature conditions are considered: cycle start conditions where all temperatures are set to 25 °C and a fully warmed up condition where the metal, coolant and oil temperatures are all set to 85 °C.

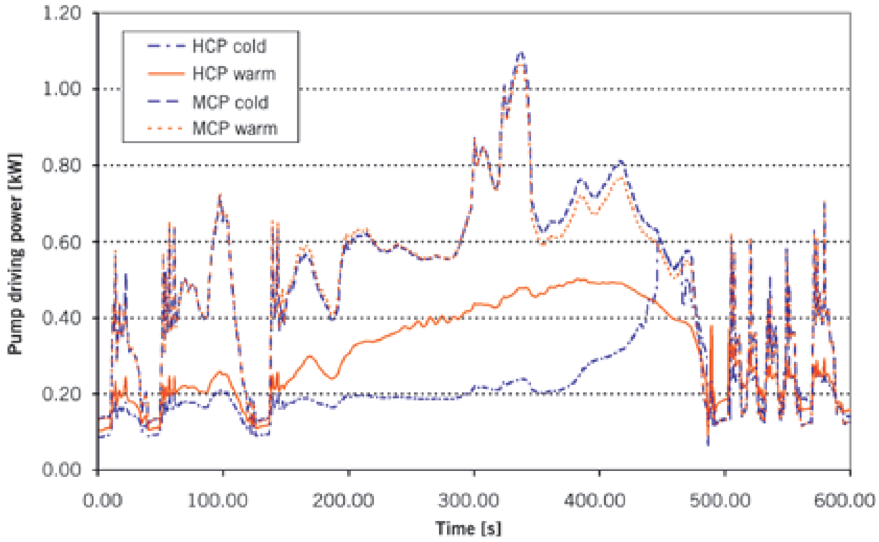
As previously described the power of the e-motor is one of the most important design parameters for the pump with the pressure loss in the cooling system as an important factor. Simulations were carried out with e-motor sizes from 150 to 350 W. The results show that a pump power of 200 W is the optimal setup under the given boundary conditions. Such a small e-motor size is also very beneficial for the package envelope and the cost of the HCP. The pump characteristic is shown in ④. It can be seen that the switch from electric to mechanical mode occurs three times. The first and the third switch are made because the limiting pump speed was reached whereas the second switch was demanded by the controller due to efficiency reasons.

With a payload of 1000 kg the average power (or work) of the pump rises but there is no more frequent switching. In the warm start case the average power rises further but again not more than three switches occur. So even for this case the 200 W e-motor power is sufficient. It should be investigated, however, whether this power is also sufficient for fast passenger compartment heating (mode 4).

CYCLE EVALUATION AND BENCHMARK

To evaluate the benefits, the HCP was compared to a belt driven mechanical pump (MP) in the US06 cycle at the two different starting temperatures, ❺, and 1000 kg additional vehicle load.

One of the main advantages of the HCP can be seen when the average power consumption of the pumps is compared. With an average of 0.22 kW the HCP consumes significantly less power than the MCP with 0.49 kW. Just this decrease in parasitic losses already leads to a fuel consumption benefit of 0.7 % in the cycle. In



❺ Comparison of both pump concepts, warm and cold condition

reality lower friction losses and less injected fuel due to the faster heating up of the engine add a major saving potential to this. In the warm case the HCP shows very good results with 0.31 kW compared to 0.49 kW of the MCP.

SUMMARY AND OUTLOOK

The Hybrid Coolant Pump (HCP) combines the benefits of fully electric and mechanical pump concepts with a cost target below the fully electric version. The small package envelope of the HCP often allows the replacement of a conventional pump by the HCP without any further modifications to the target engine. Comprehensive simulation efforts resulted in optimized parameter setting and showed significant potential for fuel consumption reduction in the whole engine map, especially during engine warm-up. In addition the HCP supports start-stop operation and after-cooling after engine stop. To gain the full benefits the HCP needs to be embedded in a comprehensive thermal management approach of the whole vehicle.

REFERENCES

[1] Morel, T.: Integrated Simulation of Engine, Vehicle and Cooling for System Analysis and Component Selection. Tagung Virtual Powertrain Creation 2008, MTZ International Congress, October 2008
[2] Keller, P.; Wenzel, W.; Becker, M., Roby, J.: Coupled Thermal-Engine Simulation for a Light Duty Application. SAE Paper No. 2010-01-0806, 2010
[3] Ito, N.; Ohta, T.; Kono, R.; Arikawa, S.; Matsumoto, T.: Development of a 4-Cylinder Gasoline Engine with a Variable Flow Turbo-charger. SAE Paper No. 2007-01-0263, 2007
[4] Kuhlbach, K.; Mehring, J.; Borrmann, D.; Friedfeld, R.: Zylinderkopf mit integriertem Abgaskrümmer für Downsizing-Konzepte. MTZ 70 (2009) Nr. 4
[5] Roth, D.; Sauerstein, R.; Becker, M.; Meilinger, R.: Applikation einer Hybrid-Abgasrückführung bei aufgeladenen DI-Ottomotoren. MTZ 71 (2010) Nr. 4

THANKS

The authors would like to thank Thomas Buchholz, Jürgen Roth, Timm Kiener and Thomas Kritzinger for their support during the HCP development.

The reference book for adhesives practitioners

The Adhesive Technology Compendium for the trade magazine adhesion

Content:

Topical background information on:

- Adhesives suppliers
- Equipment and plant manufacturing companies
- Research and development companies

Guide to German laws and regulations

Overview of European standards and test methods

Sources

Statistical overview: Production and markets

Adhesive Technology Compendium 2009
304 pages. paperback.
Euro 25.90
ISBN 978-3-8348-0912-4

Order

Yes, I wish to order ____ copies

Adhesive Technology Compendium 2009
304 p., paperback, € 25.90
ISBN 978-3-8348-0912-4

Subject to modifications.
Available from bookstores
or from the publishers.

Fax: +49(0)611.78 78-440 or at www.adhaesion.com

Full name

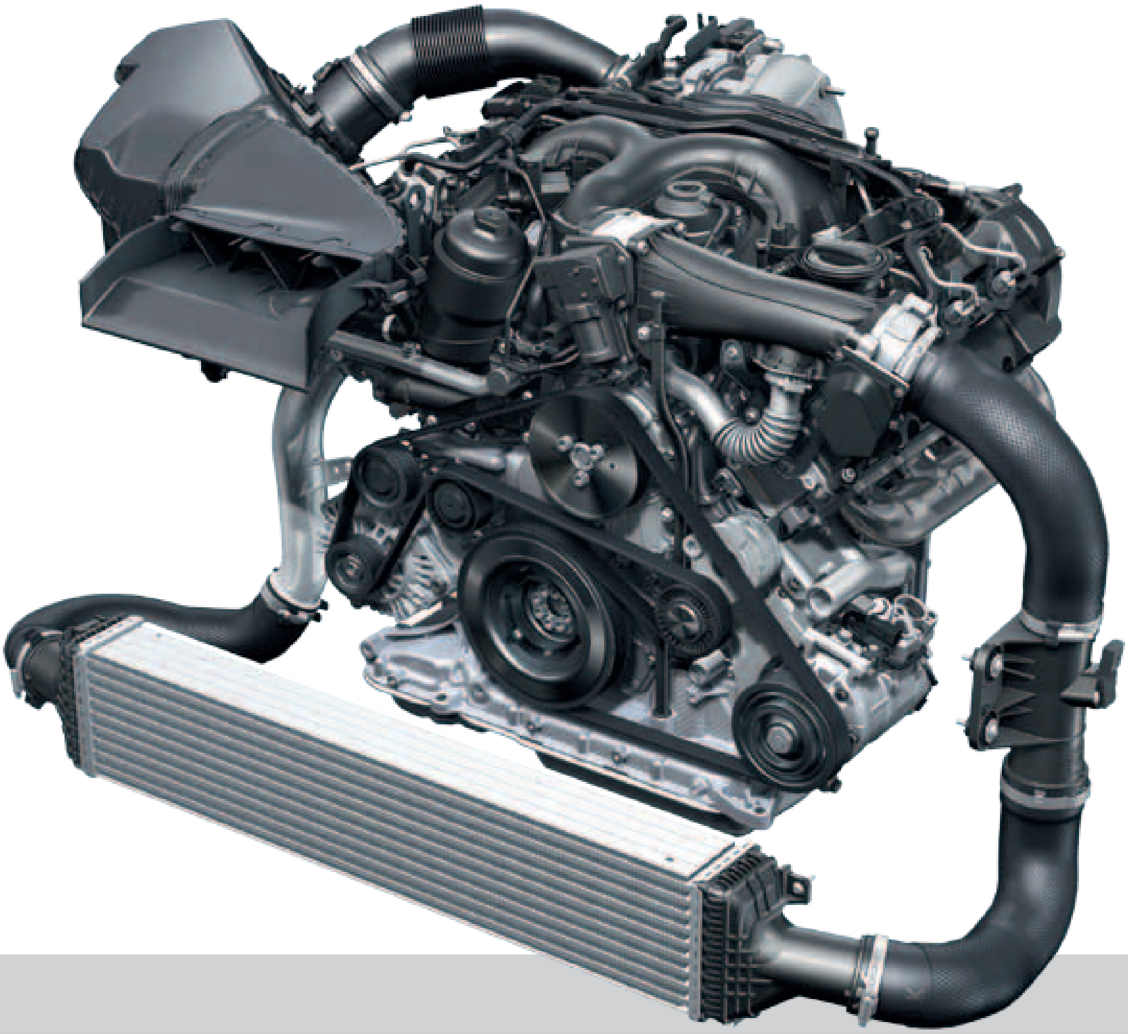
Street (no PO box) | Post code/City

Date | Signature

Abraham-Lincoln-Str. 46 • D-65189 Wiesbaden • Phone: +49 (0) 611.78 78-0 • www.viewegteubner.de
CEO: Dr. Ralf Birkelbach. Company registered at AG Wiesbaden HRB 9754

VIEWEG+TEUBNER

THE NEW GENERATION OF THE AUDI 3.0 L V6 TDI ENGINE



PART 2 – THERMODYNAMICS, APPLICATION AND EXHAUST EMISSION CONTROL

The new 3.0 l V6 TDI engine with four valves per cylinder, piezo injectors, common rail fuel injection system, turbocharging with variable turbine geometry (VTG), swirl action control and dual coolant circuit with intelligent thermal management and integrated exhaust gas recirculation (EGR) cooling develops – depending on the intended application – between 150 kW and 184 kW and has a maximum torque of between 400 Nm and 550 Nm. It complies with Euro 5 exhaust emission limits and has very low CO_2 emissions. In the following thermodynamics, application and exhaust emission control of the new engine are described. The first part of the article in MTZ 10 already depicted the design and the mechanics of the new engine.

AUTHORS

DIPL.-ING. JÖRN KAHRSTEDT
is Head of Thermodynamics and Diesel Engine Applications at Audi AG in Neckarsulm (Germany).

DIPL.-ING. STEFAN ZÜLCH
is Head of Thermodynamics and Diesel Engine Applications at Audi AG in Neckarsulm (Germany).

DIPL.-ING. CHRISTOPH STRENG
is Team Coordinator of Diesel Engine Thermodynamic Development at Audi AG in Neckarsulm (Germany).

DIPL.-ING. RALPH RIEGGER
is Team Coordinator of Diesel Engine Applications at Audi AG in Neckarsulm (Germany).

MODULAR CONSTRUCTION

The new second-generation 3.0 l V6 TDI engine is a further systematic development of Audi's successful diesel strategy. A high standard of refinement is combined with sporty performance, but the strictest exhaust emission limits are also complied with and extremely low fuel consumption figures are achieved: these were the main development objectives for the new-generation V6 TDI engine, ❶.

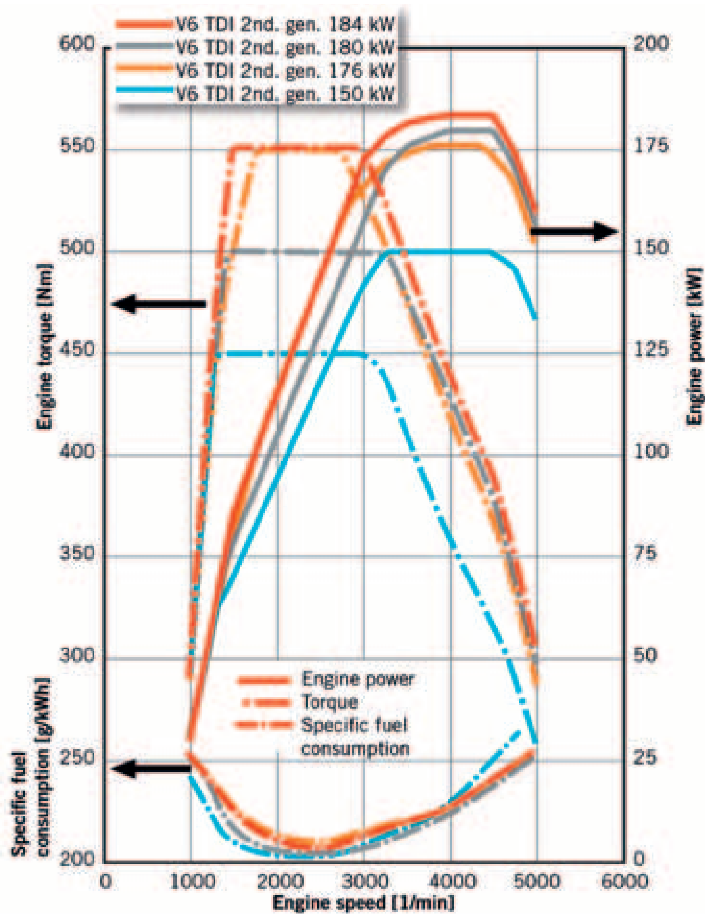
Since the Audi V6 TDI engine is suitable for a wide range of applications in various vehicle categories, great value was attached during its development to modular construction and an identical parts strategy, in order to achieve maximum synergy between the various applications and have an optimally configured engine available for each of them, ❷. The basic engine is always the same, but by choosing between two camshaft patterns, three turbocharger settings and two fuel injection systems it can be given the thermodynamic characteristics that yield the best results in each case, ❸. Further details of the components used and their specific characteristics are given below.

DISPLACEMENT	2.967 l
STROKE/BORE	91.4 mm/83 mm
FIRING ORDER	1 – 4 – 3 – 6 – 2 – 5
COMPRESSION RATIO	16.8:1
FUEL INJECTION SYSTEM	Bosch CRS 3.18/ CRS 3.20
TURBOCHARGER	HTT GT2056 / GT2260 with VTG and electronic adjuster
NOMINAL POWER OUTPUT	150 – 184 kW
NOMINAL TORQUE	400 – 550 Nm
EMISSION CATEGORY	EU 5

❶ Technical data of the second-generation V6 TDI

PIEZO COMMON RAIL SYSTEM

All versions of the second-generation V6 TDI are equipped with piezo common rail



❷ Full load V6 TDI 2nd generation

	V6 TDI 184 KW, 550 NM IN AUDI A8	V6 TDI 180 KW, 500 NM IN AUDI A7	V6 TDI 176 KW, 550 NM IN AUDI Q7	V6 TDI 150 KW, 400/450 NM "EFFICIENCY" ENGINE
FUEL INJECTION SYSTEM	CRS 3.18	CRS 3.18	CRS 3.20	CRS 3.20
MAX. INJECTION PRESSURE	1800 bar	1800 bar	2000 bar	2000 bar
HYDRAULIC FLOW AT NOZZLE	350 ml/30 s	350 ml/30 s	300 ml/30 s	300 ml/30 s
TURBOCHARGER VERSION	GT 2260	GT 2260	GT 2260	GT 2056
COMPRESSOR TRIM	Trim 58	Trim 55	Trim 55	Trim 55
EXHAUST CAMSHAFT EVENT PERIOD	202° at crank-shaft	202° at crank-shaft	202° at crank-shaft	176° at crank-shaft

❸ Modular construction of engine family

systems from Bosch. In each case the "CP4.2" two-plunger pump with a stroke of 5.625 mm is used, whether the maximum rail pressure is 1800 bar or 2000 bar. Common to all applications are forged rails of reduced length and volume compared with the earlier versions.

The engine versions for the A7 and A8 use a piezo injector system with a maximum rail pressure of 1800 bar in conjunction with a hydraulic flow rate of 350 ml/30 s; these values represent the optimal compromise between technical complexity and hydraulic performance.

Low throttling losses in the intake system, charge air flow path and exhaust system on new-generation Audi vehicles are preconditions for the required power outputs of 176/180 or 184 kW and torques of 500 or 550 Nm to be reached.

For use in the Audi Q7 a piezo injector system with a maximum rail pressure level of 2000 bar was chosen; this permits injectors with a hydraulic flow rate of 300 ml/30 s to be used.

The reduced nozzle flow improves mixture formation and enables even a vehicle in the Q7's category to achieve extremely low emission and fuel consumption values (EU5 with 195 g/km CO_2 in the NEDC) in conjunction with a maximum power output of 176 kW and 550 Nm torque.

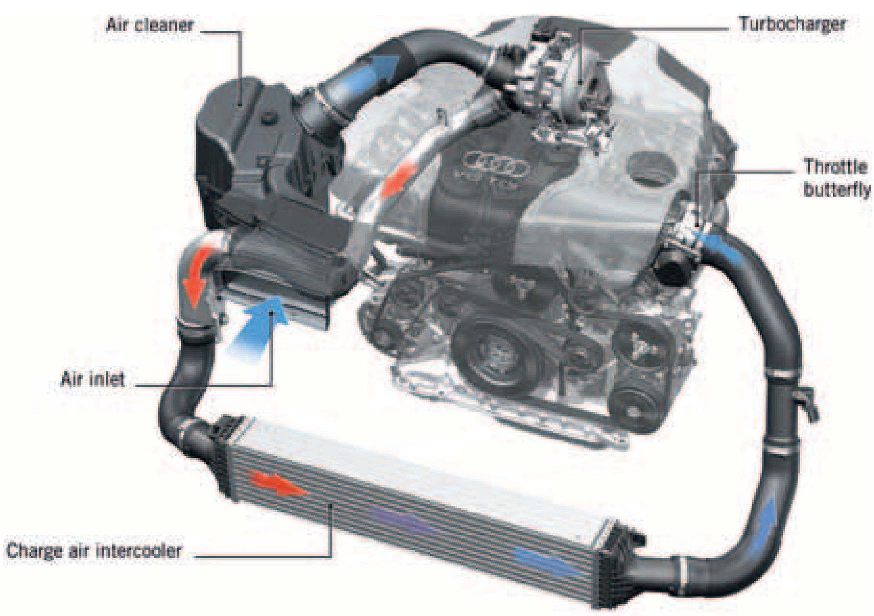
The new "efficiency" version of the V6 TDI, which takes the place of the first-generation 2.7 TDI, achieves outstanding fuel consumption figures. This new version makes use of the shared basic engine from the modular V6 TDI system with a displacement of 3.0 l. It was decided that discontinuing the 2.7 TDI would increase synergies in the identical parts strategy and during the engine's development phase, without the need to sacrifice the qualities of the 2.7 TDI. This good take off behaviour for which this engine was known has been achieved on the 3.0 l efficiency engine as well, by the choice of optimal settings for the lower-powered version.

This new next-generation V6 TDI also uses the 2000-bar fuel injection system. The reduced exhaust valve opening period chosen for this engine has the advantage of permitting a higher level of usable expansion work and thus assuring higher efficiency. The optimal spray pattern from the injector operating at 2000 bar injection pressure ensures excellent full-load fuel consumption figures right up to high engine speeds. In the same way, the small amount of swirl needed at part load means that the swirl flap does not have to be operated so frequently; this reduces the throttling effect on the engine's intake side and boosts part-load efficiency.

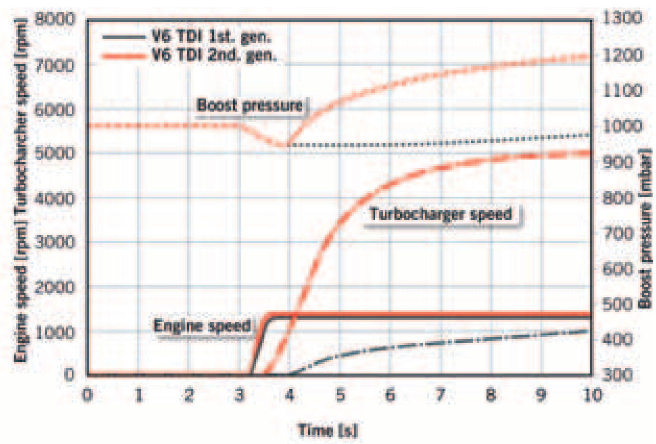
TURBOCHARGING SYSTEM

The turbochargers for these engines, ❹, are selected according to the power-output and torque targets that have to be reached. For the 184 kW version in the Audi A8 the large turbocharger with compressor trim of 58 is used. The 176/180 kW version has a compressor trim of 55, and for the 150 kW efficiency engine a smaller turbocharger was adopted.

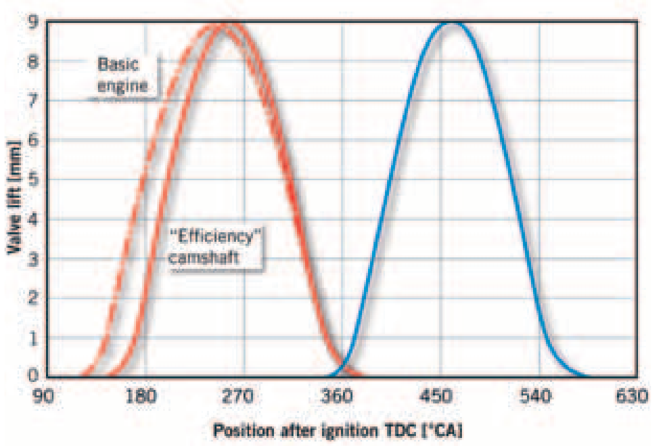
Both the large turbocharger for the higher power outputs and the smaller one for the efficiency engines have a further developed and significantly optimised plain bearing system with considerably lower friction losses than the systems previously used. This optimisation is especially evident in turbocharger response after starting a cold engine, and of course in particular when oil temperatures are very low. ❺ is an example of boost pressure build-up at engine idle speed after a cold start at -15 °C outside temperature, compared with a turbocharger of the same size from the previous generation. The more rapid increase in speed and the associated turbocharger boost pressure build-up can be clearly seen.



❹ Turbocharging system



❺ Boost pressure build-up at idle speed after a cold start at -15 °C



❻ Valve timing

A further step in optimisation of the turbocharging system on the second-generation V6 TDI is a developed version of the turbocharger compressor wheel. The new compressor wheel operates at higher efficiency and has a broader compression characteristic. This reduces any tendency towards turbocharger pumping action at high boost pressures and low engine speed, and results in reduced system sensitivity to the occurrence of unpleasant flow noise from the charge air path. Even on the 184 kW version of the V6 TDI, a second pressure-side flow damper was not needed despite the higher compressor trim.

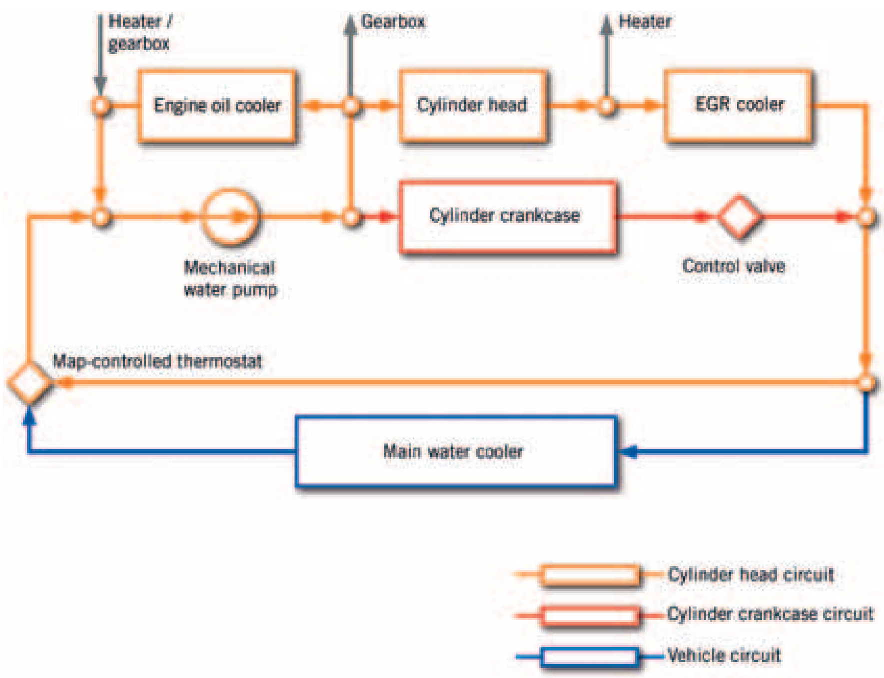
GAS EXCHANGE CYCLE / VALVE TIMING

The new dual-branch plastic intake manifold with central swirl flap plays an important part in reducing the new V6 TDI engine's fuel consumption and increasing its performance potential. In conjunction with intake ports on which basic swirl level and flow rate have been optimised, the new intake manifold optimizes the volumetric efficiency of the engine. Compared with the previous engine increased swirl by using the centrally located swirl flap is needed less often. This reduction of pressure loss optimises fuel consumption when the engine is operating at part load. As already mentioned, the "efficiency" version of the new engine has exhaust valve timing optimised for minimum fuel consumption. Reducing the exhaust valve event time from 202° to 176° of crankshaft angle improves specific fuel consumption by 2 % as a result of the longer expansion phase. ❻ shows the valve lift graphs for the basic engine version with the valves open through 202° of crankshaft rotation, and the "efficiency" camshaft with an event period of 176° of crankshaft rotation.

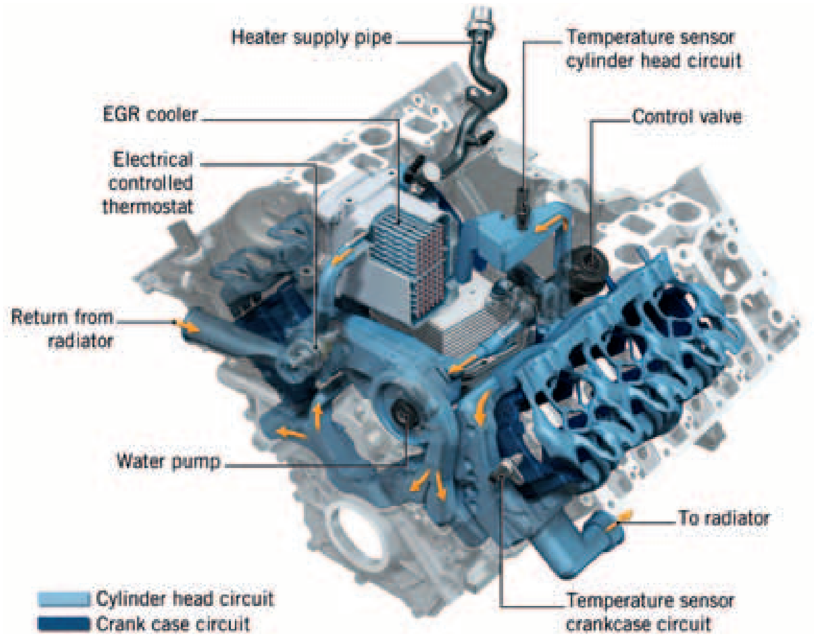
THERMAL MANAGEMENT

Special attention was devoted during development to the engine's thermal operating conditions as a means of boosting efficiency. In addition to the quickest possible engine warm-up while the coolant is not circulating in this phase, the aim was to utilise the advantages of thermal management in all engine operating ranges. The new Audi V6 TDI therefore uses a "split cooling" concept: coolant flows through the engine block and cylinder heads in two separate circuits arranged in parallel. This is to ensure that the most suitable temperature level can be selected for the cylinder heads and for the engine block even if the engine is at its regular operating temperature, ❼.

Positioned at the front of the V between the two cylinder banks, the water pump delivers coolant continuously to the



❼ Schematic cooling circuit



❽ Thermal management with separate cylinder head and engine block cooling

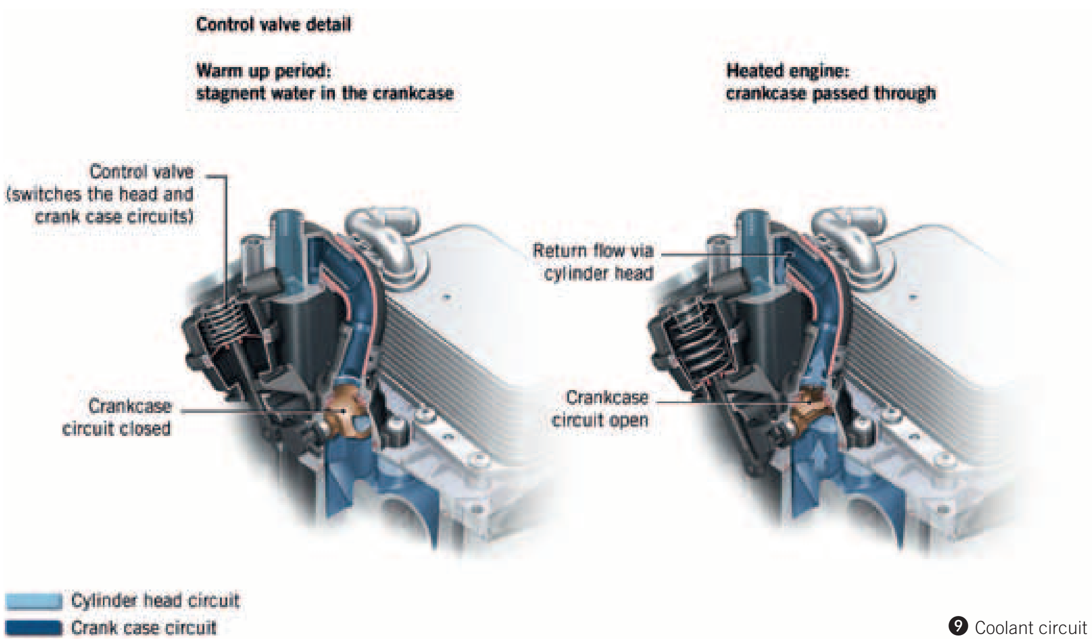
exhaust sides of the engine block. At that point the flow is divided: some of the coolant flows to the cylinder heads, the remainder to the engine block. After passing through these areas the two part-flows return to the suction side of the water pump.

CYLINDER HEAD CIRCUIT

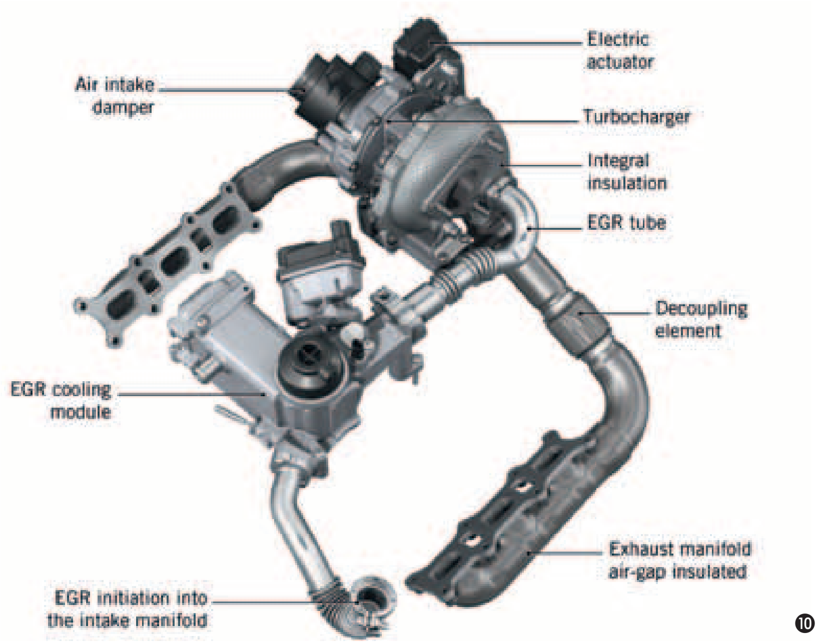
The continuous-flow cylinder head circuit consists primarily of the water cavities in the two cylinder heads, the oil and EGR cooler, the car's heating and transmission oil heat exchangers and the main radiator, ❽.

The temperature in the cylinder head circuit is controlled by a mapped-characteristic thermostat with a wax expansion element that can be heated. During engine warm-up there is no flow through the thermostat; it only opens above 90 °C. Until this temperature is reached, no heat energy is supplied to the main radiator. Warm coolant is used to heat the transmission and, if required, for the car's interior heater. When the flow passes through the expanding wax element, the temperature in the cylinder head circuit can be lowered, subject to the physical limit represented by the main radiator. The following conditions apply:

: protection of the cylinder head if exposed to extreme loads
: maximum EGR cooling requirement
: transmission cooling requirement.



❾ Coolant circuit regulating valve



❿ Exhaust manifold, turbocharger and EGR module

ENGINE BLOCK CIRCUIT

Coolant enters the engine block on the exhaust sides of the cylinder blocks. A non-return valve is provided at each entry point to prevent the coolant from flowing in the wrong direction. The engine block circuit is initially shut down by a vacuum-controlled ball valve at the coolant exit point. With no coolant flow, the engine warms up more rapidly and friction losses are therefore reduced. After the engine has been fully warmed up, the temperature in the engine block circuit is regulated to approximately 105 °C at the ball valve, so that the reciprocating parts can operate in the optimal friction-loss range, ❾.

A major development priority was the provision of "stationary coolant" in the engine block circuit. A ball valve with very low leakage throughout its operating life was therefore chosen as an actuator. The non-return valves ensure that coolant flows in the correct direction through the engine block circuit. This part-circuit also has a separate breather. The cylinder-block water jackets are connected via the cylinder head gaskets with a collector rail in each cylinder head. This ensures that air bubbles can escape from the engine block circuit at the highest point, even if the coolant is stationary. Bleed lines run from the collector rails to a bleed valve that connects the permanent cylinder head and engine block breather systems together. The bleed valve has a ball-shaped float that seals off one part-circuit against the other. Even when the engine block circuit has been bled, this prevents any heat energy from leaving the circuit via the permanent breather.

To encourage rapid warm-up still more, the concept includes an oil cooler bypass on the oil side of the circuit. An expanding wax element opens a bypass duct at oil temperatures above 103 °C and diverts the main oil flow past the oil cooler.

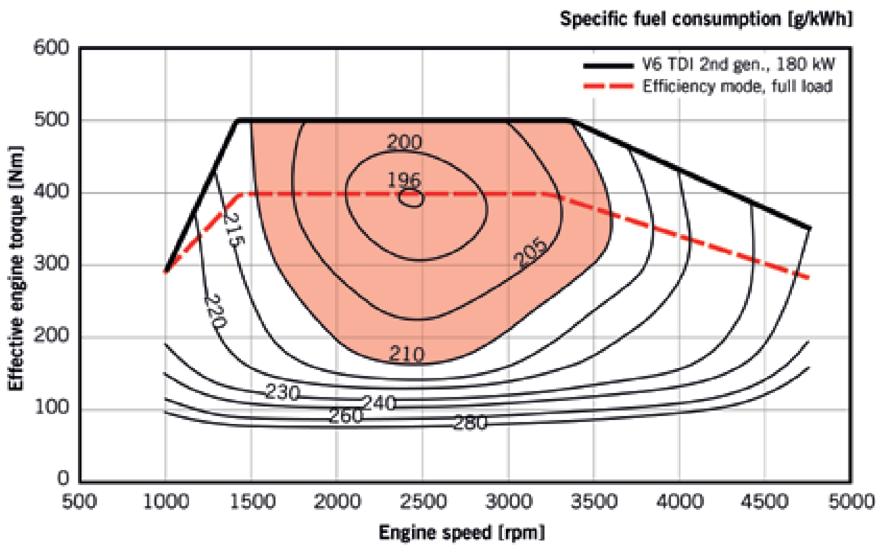
Audi's Innovative Thermal Management (ITM) principle as used on the new V6 TDI engine with its "split cooling" concept provides an independent flow from the cylinder head circuit to the interior and transmission-fluid heaters even when there is no coolant flow through the engine block. Since the two part-circuits both operate in the optimal temperature range for minimum friction, further reductions in friction losses are possible even when the warm-up phase has been completed.

EXHAUST GAS RECIRCULATION

The cooling performance of the EGR system, ❿, has been optimised to such an extent that it equals that of the system on the previous engine although no separate EGR low-temperature coolant circuit is provided. Since the EGR cooler is now permanently connected to the cylinder head circuit on the coolant side, and no longer supplied with cold water directly from the main radiator, the temperature at which coolant enters the EGR cooling system is higher, but by uprating the EGR cooler the cooling performance of the complete system has been slightly increased. An advantage of the new EGR cooling system is its much-reduced complexity, due to deletion of the EGR low-temperature circuit via the coolant-side connection to the cooler cylinder head circuit of the new dual-circuit cooling system; weight is therefore also significantly reduced.

A further primary objective of further development work on the EGR system was to reduce pressure drop in the EGR line from the turbocharger take-off point to the EGR entry point at the intake manifold. In this way high levels of exhaust gas recirculation can be achieved without assisting the scavenging effect by using the throttle butterfly. The resulting drop in gas exchange losses improves fuel consumption.

Introduction of an EGR bypass flap valve that makes a tight seal ensures that there is no unwanted flow through the EGR cooler at wall temperatures that are too low. This improves the long-term reliability of the EGR cooling system and, by reducing the system's total cooling performance in the bypass operating mode, leads to a drop in CO emissions.



⓫ Fuel consumption characteristic at full load in “efficiency” mode

ENGINE MANAGEMENT

Use of the Bosch “CP44” means that all applications of the V6 TDI engine now have a single type of engine management control unit. The control unit is driven in all cases by the EDC17 engine management software generation.

Many multi-task functions are handled by the engine management control unit as well as those that only concern specific engine functions. For example, heating, transmission and engine components involved in thermal management are given priority by a central actuating logic in the engine management control unit according to momentary interior heating, transmission cooling or engine warm-up needs.

The intelligent engine start-stop system circuitry, which has to take air conditioning, engine management, brake system, transmission and engine functions into account, is also centrally located in the engine management control unit.

APPLICATIONS

All engine measures are influenced by the start-stop system and the “efficiency” mode driver option available for the first time on the new A7. When efficiency is selected, there is a moderate reduction in engine power and torque, ⓫. This reduces full-load fuel flow and also displaces the operating points into optimal-efficiency areas of the mapped characteristic.

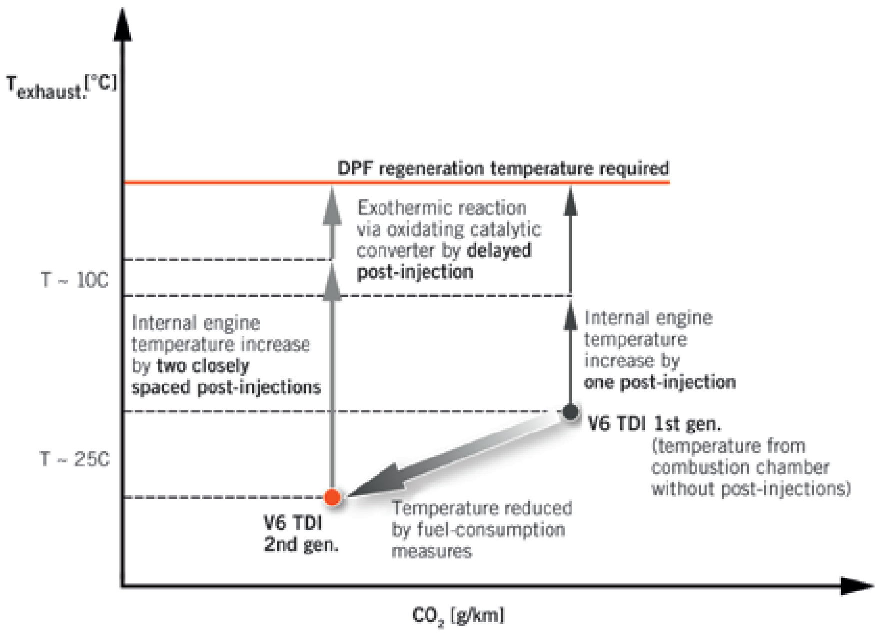
For the new Audi A7 the new-generation V6 TDI engine is available in two performance versions. The 180 kW version is joined by an efficiency version developing 150 kW. With front-wheel drive and automatic transmission, the 150 kW engine in the new A7 has a fuel consumption of only 5.3 l per 100 km according to the NEDC. The 180 kW version with quattro driveline records a figure of 6.0 l/100 km. Both these figures are outstanding for this car segment. Compared with the today´s Audi A6 they represent improvements of up to 17 %, from cars with levels of performance that only sports cars would have reached a short time ago, ⓬.

EXHAUST EMISSION CONTROL

The oxidating catalytic converters used with all Audi V6 TDI engines and the coated diesel particulate filter (DPF) were further optimised for the new-generation engines. The oxidating catalytic converter

	ACTUAL AUDI A6	NEW AUDI A7	ACTUAL AUDI A6	NEW AUDI A7
TRANSMISSION	Automatic, FWD	Automatic, FWD	Automatic, quattro	Automatic, quattro
EMISSION CATEGORY	EU 5	EU 5	EU 5	EU 5
MAX. POWER OUTPUT (KW)	140	150	176	180
MAX. TORQUE (NM)	380	400	500	500
FUEL CONSUMPTION (L/100 KM) (MVEG)	6.4	5.3	7.1	6.0
ACCELERATION (S) (0–100 KM/H)	7.9	7.8	6.8	6.3
TOP SPEED (KM/H)	227	238	250 (governed)	250 (governed)

⓬ Performance figures and fuel consumption for the A7 with V6 TDI engine



⓭ Active DPF regeneration with three post-injections at lower part load

has a volume of 1.0 l and the diesel particulate filter of 3.7 l. A considerable increase in regeneration intervals has been achieved with aluminium titanate as the DPF substrate; it has proved the most suitable in numerous tests.

A feature introduced for the first time on the new V6 TDI is three-phase post-injection during particulate filter regeneration as a means of raising the temperature during low-load operation, ⓭. This ensures safe, rapid soot burn-off in all driving conditions, especially in stop-and-go traffic. Most of the necessary thermal energy is generated by two early post-injections in close proximity and therefore capable of burn-off. The third, late post-injection, intended to cause an exothermic reaction via the oxidating catalytic converter, uses smaller quantities of injected fuel. This three-phase post-injection procedure makes DPF regeneration reliable even if the exhaust gas is at a lower temperature on account of low fuel consumption. At the same time it minimises oil dilution and improves the oxidating catalytic converter's resistance to ageing.

THANKS

In the preparation of this paper additionally have collaborated:
Dipl.-Ing. Richard Bauder, Head of Diesel Engine Development at Audi AG in Neckarsulm (Germany).
Dipl.-Ing. Andreas Fröhlich, Head of V6 Diesel Engine Design at Audi AG in Neckarsulm (Germany).
Dr.-Ing. Christian Eiglmeier, Head of Diesel Particulate Filter (DPF) Applications for V6, V8 and V12 Diesel Engines at Audi AG in Neckarsulm (Germany).

SYNOPSIS

The second-generation V6 TDI engines have been successful in achieving ambitious performance targets for the Audi vehicle range, extremely low fuel consumption and efficient exhaust emission control. Their modular construction enables the engine to be optimally matched to various vehicle concepts and, despite the many applications for the engine in different body styles and with various transmissions, dynamic, high-efficiency settings were always achieved without the need for compromises.

REFERENCES

[1] Bauder, A.; Clos, R.; Hatz, W.; Hoffmann, H.; Pölzl, H.-W.; Reichert, H.J.: The new Audi V engines. Vienna Engine Symposium, 2002
[2] Bauder, R.; Reuss, T.; Hatz, W.; Pölzl, H.-W.: The new V6 TDI from Audi. Vienna Engine Symposium, 2004
[3] Anton, C.; Bach, M.; Bauder, R.; Franzke, G.; Hatz, W.; Hoffmann, H.; Ribes-Navarro, S.: The new 3.0-litre V6 TDI engine from Audi, Part 1: Design and mechanical elements. In: MTZ 65 (2004) No 7/8
[4] Bauder, R.; Brucker, D.; Hatz, W.; Lörch, H.; Macher, A.; Pamio, Z.-G.; Reuss, T.; Riegger, R.; Schiffgens, H.-J.: The new 3.0-litre V6 TDI engine from Audi, Part 2: Thermodynamics, application and exhaust emission control. In: MTZ 65 (2004) No 9
[5] Bauder, R.; Bach, M.; Köhne, M.; Streng, C.; Hofmann, H.; Rossi, D.: The new 3.0-litre V6 TDI engine in the Audi Q5. In: ATZextra The New Audi Q5 (2008)
[6] Bauder, R.; Bach, M.; Fröhlich, A.; Hatz, W.; Helbig, J.; Kahrstedt, J.: The new-generation 3.0 TDI engine from Audi. Vienna Engine Symposium, 2010
[7] Bauder, R.; Bach, M.; Fröhlich, A.; Helbig, J.; Ortwein, J.; Rossi, D.; Seifried, G.: The New Generation of the Audi 3.0 l TDI Engine, Part 1: Design and Mechanical Elements. In: MTZ 71 (2010) No 10

MOTIVATION

In 2009, new rail emissions legislation (EU IIIA) reduced the limits for nitrogen oxides (NO_x) and particulate matter (PM) from the levels set by the UIC II standard. And, as ❶ shows, a further significant lowering of the permissible levels is due to come into force in 2012 (EU IIIB) [1]. The advent of Stage IIIB in particular makes new in-engine and external engine technologies necessary. MTU sees the best solution for rail applications in the combination of exhaust gas recirculation (EGR) and diesel particulate filtration (DPF). The aim of this project, however, was to test out exhaust gas aftertreatment (EGA) technologies in general. These are not yet universally established in all applications in the off-highway sector and for that reason, MTU and Deutsche Bahn (DB) conducted their joint research project with the aim of testing out these new technologies at an early stage. The "LOCEX" (Locomotive with Clean Exhaust) project provided an ideal platform for trialing MTU's internally developed and produced prototype EGA system under the tough conditions of everyday rail duty. The system was tested under real operating conditions such as variable ambient temperatures, shock exposure etc. This gave MTU the opportunity to gain important knowledge for the development of future compact and efficient EGA systems, while DB was able to benefit by gaining extensive experience with the technology early on.

The sections that follow describe the concept and the design implementation of the EGA system. Compliance with the stringent emission limits is to be verified on the engine test stand and in field trial analyses.

CONCEPT

For concept development, the type of use and class of locomotive were first determined. To facilitate testing under the most stringent conditions possible, a Class 294 shunting locomotive was chosen. DB Schenker Rail provided locomotive

AUTHORS

DIPL.-ING. (BA) CLAUDIA WEIGEL
is Exhaust Aftertreatment Designer in the Department for New Technologies at MTU Friedrichshafen GmbH (Germany) and "LOCEX" Project Manager since 04/2009.

MBA, DIPL.-ING.(FH) GUIDO SCHÄFFNER
was "LOCEX" Project Manager in the Department for New Technologies until 03/2009 and is now Exhaust Aftertreatment Function Team Leader for the Series 4000 in the Development Department at MTU Friedrichshafen GmbH (Germany).

DR.-ING. PETER KATTWINKEL
is Application Engineer for Commercial Vehicle Applications in the Automotive Catalysts Department of Umicore AG & Co. KG in Hanau-Wolfgang (Germany).

DIPL.-ING. PETRA VIEHWEG
is Diesel Engine Technology Subproject Manager in the Technology, System Integration and Services Department, Deutsche Bahn (Germany) and Manager of the "LOCEX" Project at DB AG since September 2006.

TECHNOLOGIES FOR EXHAUST AFTERTREATMENT TESTING UNDER REAL CONDITIONS

The use of exhaust aftertreatment technology in off-highway applications is not yet universally established. For this reason, MTU Friedrichshafen GmbH and Deutsche Bahn AG conducted a joint research project involving early testing of various basic technologies for exhaust gas aftertreatment (EGA) under real conditions. The knowledge gained in the rail applications sector will be transferrable to other sectors and it will also be possible to combine the basic technologies involved in different ways.

number 294 635-8, which had been repowered as standard with an MTU Type 8V4000R41 engine in 2008 [2, 3]. The engine has a power output of 1000 kW and is UICII-compliant.

The aim of this project was to use various EGA technologies to achieve compliance with the Stage IIIB exhaust emission limits well in advance of the 2012 target date. This involves a minimum reduction in NO_x emissions of 63 % and a lowering of particulate emissions by at least 84 %. Compliance with the limits has to be demonstrated in accordance with the ISO-F cycle, a static testing cycle involving idling, partial load and full load phases.

The concept selected was a system comprising diesel oxidation catalyst (DOC), diesel particulate filter (DPF) and selective catalytic reduction (SCR) catalyst. Combined systems of this type have already undergone comprehensive examination both for on-road and non-road applications [4, 5, 6].

An aqueous urea solution (AUS) was used as the reducing agent. To avoid the possibility of ammonia (NH_3) release, a slip catalyst was integrated downstream of the SCR catalyst. ❷ shows the basic structure of the EGA system. The soot in the DPF was to be oxidized by means of passive, NO_2-based regeneration. A diesel oxidation catalyst (DOC) was employed upstream of the DPF for that purpose. This converts some of the NO emitted by the engine into NO_2, which oxidizes the soot at a lower exhaust temperature than is the case with a purely oxygen-based reaction. To avoid overloading the filter and consequent overheating of the engine, the concept provides for the additional installation of a diesel burner for active regeneration. The burner can heat up the exhaust in order to enable the soot to be burned off by oxygen. Wherever possible, however, the project sought to avoid the use of active regeneration methods.

As the space available in locomotives is very limited, the EGA system had to be fitted in place of the silencer and had to take over the silencing function itself. At the same time, the maximum permissible exhaust backpressure due to the EGA system was set at 130 mbar in order to prevent an excessive increase in fuel consumption. To be able to meet the backpressure requirements, the catalysts and the DPF had to be dimensioned with a large-area inflow cross-section. For a large-scale diesel engine capable of 1000 kW, the catalyst and DPF volume needed can no longer be achieved with only a single carrier substrate in each. Multiple substrates have to be fitted in parallel. In order to meet all these requirements, an innovative concept was required to deal

DR.-ING. MARC HEHLE
is Exhaust Aftertreatment Team Leader in the Department for New Technologies at MTU Friedrichshafen GmbH (Germany).

DR.-ING. DIRK BERGMANN
is Head of the Department for New Technologies at MTU Friedrichshafen GmbH (Germany).

with substrate configuration and flow routing within a single housing.

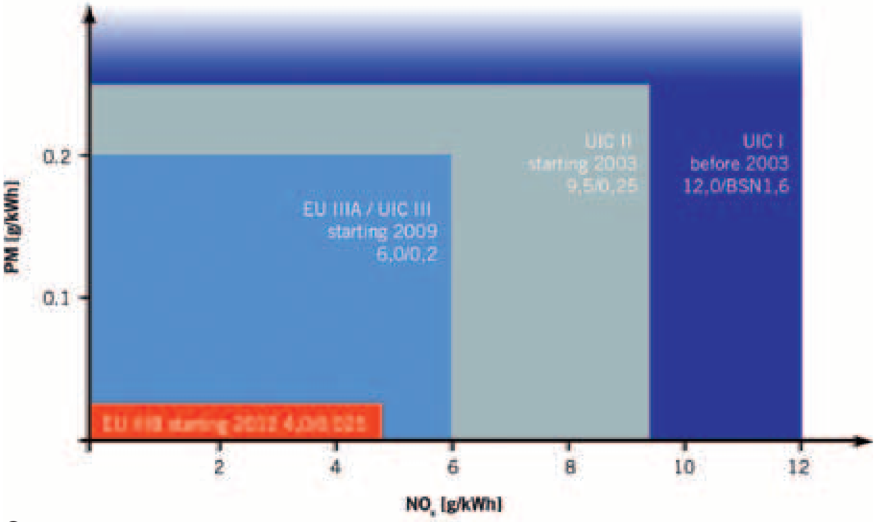
However, the challenges were not limited to catalyst configuration. MTU also had to break new ground on urea (AUS) feed technology. Airless systems had to be used for urea dosing because the rail sector's stringent safety regulations meant that the existing compressed-air supply for the braking system could not be used and there was no space available for an additional compressor. At the time the project was launched, however, there were no airless dosing systems capable of handling the quantities required for a 1000 kW engine. Consequently, two pumps and two injectors were specially adapted to meet project requirements. MTU also had to develop a concept for temporary AUS storage in the locomotive. The solution involved fitting a purpose-designed tank with a capacity of approximately 150 l in place of a reserve diesel fuel tank. As AUS freezes at temperatures < -11 °C, a tank heating system running off the engine coolant was fitted and to prevent the urea pipes freezing, they were routed alongside the cooling system piping.

In addition to the technical specifications for the EGA system, DB directives for conversion of the locomotive also had to be followed. Modifications to the shunter during the course of the conversion had to be notified to the German Federal Railways Office in accordance with TEIV (Trans-European Interoperability Regulations). Particular emphasis was placed on the overall weight of the locomotive and the associated maximum permissible wheelset loads as well as on compliance with the noise emission limits.

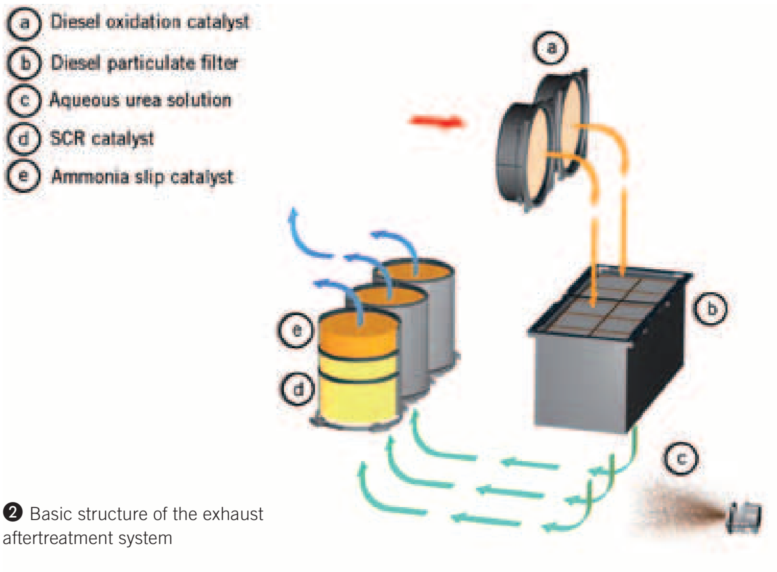
The locomotive operator, DB Schenker Rail, also specified a further condition for EGA system design: EGA installation was not to alter any of the locomotive's driver controls or its maintenance intervals.

DESIGN OF THE EXHAUST AFTERTREATMENT COMPONENTS

The design was developed jointly with the Umicore company and reflected the typical duty profile of a shunting locomotive. Among the particular challenges involved were the load profile with its idling component of > 70 %, and the average exhaust temperatures of < 300 °C.



❶ Progressive tightening of EU emission limits for locomotives

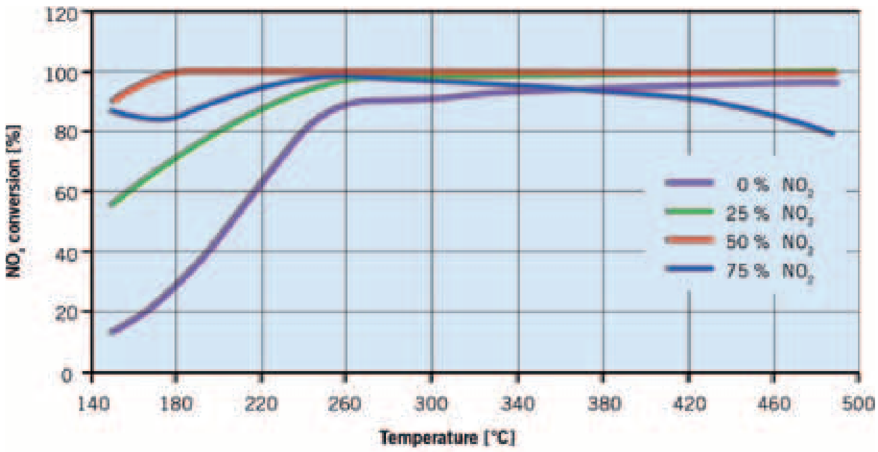


❷ Basic structure of the exhaust aftertreatment system

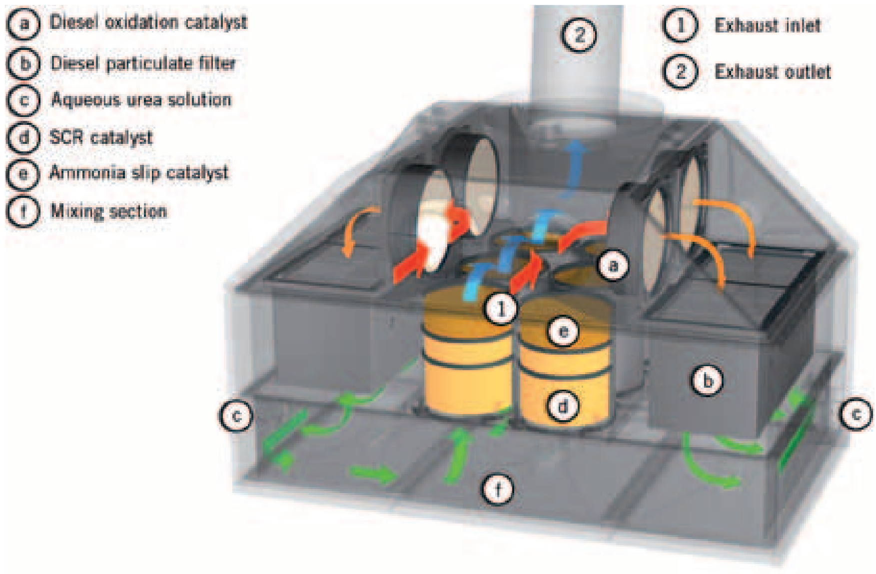
The particulate filter had to be suitable for active regeneration using a diesel burner as well as for passive NO_2-based regeneration. A DPF made of highly porous silicon carbide was thus selected. Dimensioning was based on simulation calculations and component tests aimed at meeting demands for the lowest possible exhaust backpressure and a high ash storage capacity.

Reliable passive filter regeneration requires a sufficiently high exhaust temperature as well as a high NO_2 concentration in the exhaust. To achieve that, a DOC as well as a catalytically coated DPF were used.

Selection of the SCR catalyst, which is located downstream of the DPF, must reflect the possible risk of transient excessive temperatures during active filter regeneration. For that reason, a zeolite-based SCR catalyst that is stable at high temperatures was chosen. In order to obtain high NO_x conversion whilst also retaining a compact catalyst volume, equal proportions of NO and NO_2 in the exhaust are crucial. ❸ illustrates the interrelationship by plotting the NO_x conversion rates established on the model gas test stand against the NO_2/NO_x ratio. It can be seen that, especially at low temperatures, an increase in NO_2 content significantly improves NO_x conversion. If the NO_2 constituent rises above 50 %, the conversion rate dips slightly again. A further point for consideration is that NO_2/NO_x ratios of > 50 %, can create N_2O (laughing gas) as an undesired byproduct. The DOC and coated DPF must therefore be suitably



❸ NO_x conversion rate as a function of temperature and NO_2/NO_x ratio



❹ Cross-section through the exhaust aftertreatment system

dimensioned to prevent an excessively high NO_2 concentration upstream of the SCR catalyst. N_2O emissions in the critical temperature range can also be limited by lowering the AUS dosing rate although this necessarily involves a corresponding reduction in the NO_x conversion rate.

DESIGN

In order to integrate the technology in the locomotive, the space available had to be utilized as efficiently as possible. At the same time, however, the system had to be easy to maintain and straightforward replacement of individual components was essential. A modular design was therefore selected. The welded housing consists of three levels, each of which is fitted with the catalysts. ❹ shows flow routing. The mixing section, which is integrated in the housing, deserves particular mention. This section is designed to allow the AUS injected into the exhaust as much time as possible for evaporation, mixing and thermolysis before it flows through the SCR catalysts.

Large diameters were used in order to accommodate the catalyst volume as compactly as possible within the space available. In the case of the DPF, even square-section substrates were used to achieve optimum use of space. However, reliable fixing of the ceramic catalysts inside the sheet-metal housing, known as "canning", is then significantly more difficult than with round substrates as the required even surface pressure cannot be produced as easily. MTU developed and patented its own canning process for reliably fitting square substrates in the housing [3]. This process also means that the filter substrates can be removed quickly, easily and non-destructively from the "can" and because this canning method has a number of additional advantages, the round DOC and SCR catalyst substrates were also fitted on the same principle.

The process involves wrapping the monoliths in a ceramic-fiber mat and pressing them into the outer casing with an extra, specially formed sheet-metal jacket, ❺. The sheet-metal jacket, in conjunction with a special fitting device, means that no axial force has to be exerted on the catalysts during insertion and extraction so that they remain undamaged.

In combination, all of these measures, i.e. arrangement and geometry of the catalysts, integrated AUS mixing section and specially developed canning process, led to the creation of an extremely compact solution. Although the system is heavier than the silencer it replaces, the locomotive nevertheless remains within overall weight and wheelset load limits.

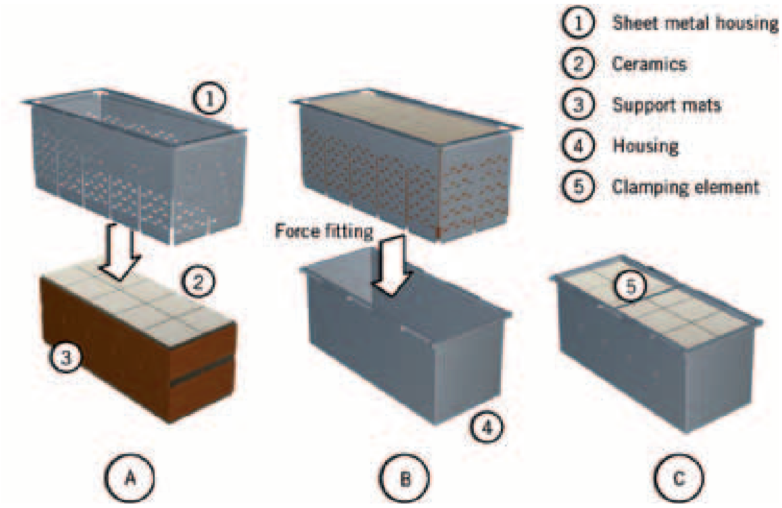
On completion of the design phase, the EGA system was manufactured, assembled and tested on the stand at MTU.

TEST STAND RESULTS

The EGA system underwent intensive examination on the engine test stand. The process involved defining the regeneration strategy for the DPF, configuring the internally developed controller and verifying compliance with the legal limits for particulate matter, NO_x and noise emissions. Special attention was also paid to avoiding unwanted secondary emissions such as NH_3 and N_2O.

An essential part of the investigations was devoted to validating DPF regeneration. Soot starts to react with NO_2 at exhaust temperatures between 250 °C and 300 °C. An operating cycle based on earlier load profile records for an identical shunting locomotive was simulated and run over a period of 50 h on the test stand. An average exhaust temperature of approximately 240 °C was produced. However, no significant rise in exhaust backpressure was identified during the simulated operating cycle. In the course of further investigations into soot accumulation, the engine settings were temporarily altered so that large quantities of soot

❺ New canning process



were emitted at exhaust temperatures < 250 °C. Despite a soot accumulation period of over 8 h, once again no significant increase in exhaust backpressure was identified. On the basis of these passive DPF regeneration results, it was decided that the EGA system required no active DPF regeneration functionality. The investigations also showed that particulate matter emissions were well below the limit specified for EU IIIB.

Various control and calibration scenarios were tested for validating the SCR system. The AUS dosing rate was controlled using an NO_x sensor. As a basic principle, dosing was only enabled at exhaust temperatures > 250 °C in order to prevent urea deposition. With an ammonia-NO_x-ratio (ANR) = 1 (ANR = $nNH_3/(nNO + nNO_2)$), conversion rates of > 90 % were measured at low to medium exhaust volume flows. At higher exhaust volume flows, the conversion rates dropped so that the NO_x reduction rate was around 85 % at full load. Ammonia slip was virtually eliminated by using an ammonia slip catalyst. Test stand measurements showed NH_3 concentrations below 10 ppm downstream of the EGA, even with ANRs between 1 and 1.2. The results also showed that ammonia was not oxidizing into nitrogen oxides and was being completely converted into N_2.

During initial intensive testing of the EGA system on the engine test stand, N_2O was detected at temperatures below 400 °C. In order to suppress the formation of this undesired reaction by-product, ANR was adjusted according to the engine's operating status.

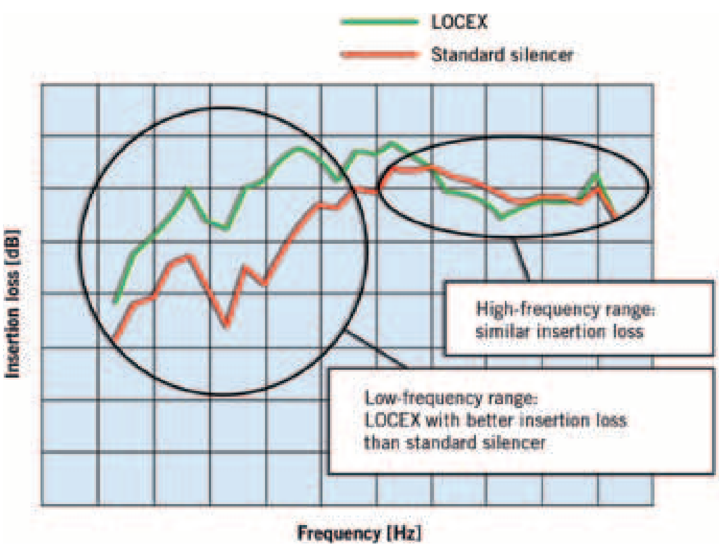
In the acceptance test, an average NO_x conversion rate of > 70 % was recorded over the ISO-F cycle so that NO_x emissions were well below the level of 3.5 g/kWh demanded by EU IIIB. At the same time, NH_3 and N_2O emissions were kept to an absolute minimum.

According to Directive 2001/16/EC, noise emissions from the locomotive must not increase as a result of replacement of a subsystem. It was therefore necessary to demonstrate that the EGA system had at least the same silencing capability as the standard silencer and a comparative test was carried out for that purpose. The results are shown in ❻. In the event, the silencing effect of the EGA system was actually shown to be significantly better.

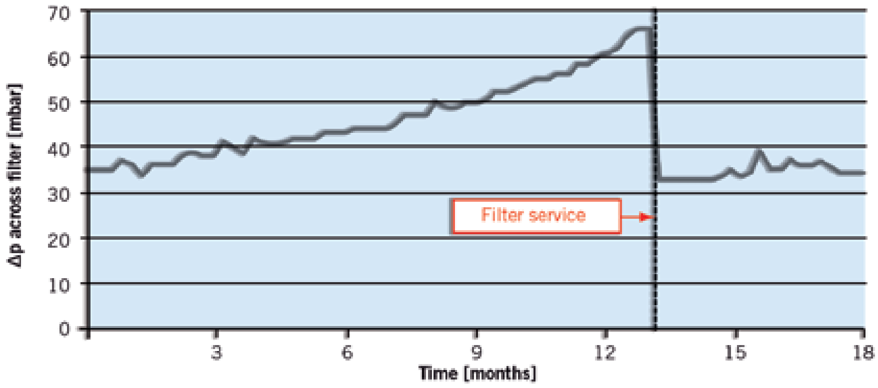
INTERIM FIELD TRIAL RESULTS

Following successful testing and approval on the test stand, the EGA system was installed in the locomotive at the DB locomotive maintenance depot in Cottbus. It has been in operation on daily shunting duties at Bietigheim-Bissingen rail yard since the middle of 2008 with a work schedule of around 3000 h per year. As DB AG had never used AUS for railway transportation in the past, an AUS filling station was set up especially for this locomotive at the DB refueling depot in Kornwestheim.

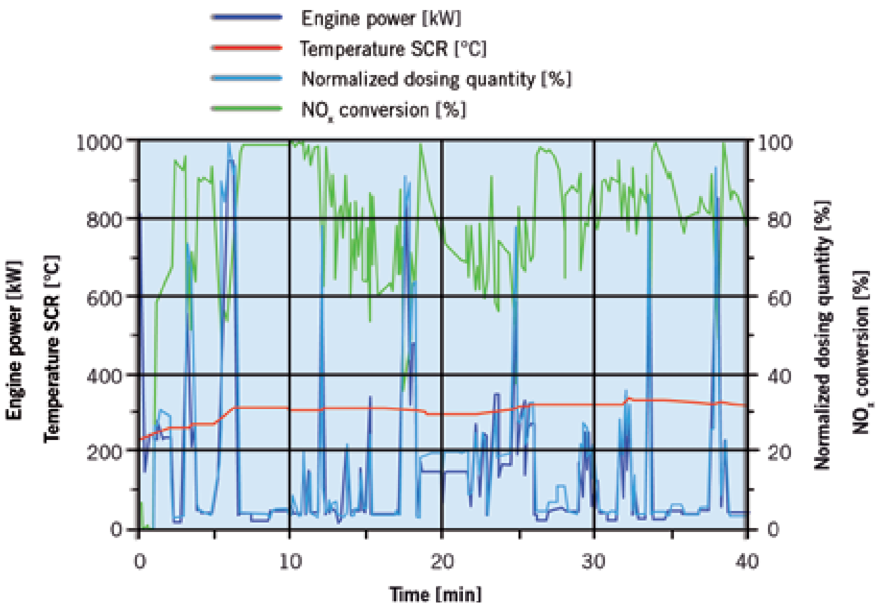
The entire operation can be rated very positively for both MTU and DB. The system has proved fit for purpose and durable under all circumstances. The locomotive



❻ Results of the noise emission test



❼ Exhaust backpressure



❽ Transient NO_x conversion

was available for duty at all times. There were no problems with maintenance or locomotive availability and no unscheduled visits to the maintenance depot.

As expected, analysis of the load profile logged showed an idling component > 70 %. It also showed average exhaust temperature was even lower than 240 °C, which poses a major challenge for continuous regeneration of the DPF. Nevertheless, the DPF exhibited no irregularities over the entire period of the trials. The level of purely passive regeneration taking place was adequate. One reason for this is the occurrence of short but frequent full load phases involving high exhaust temperatures.

After 3000 h of duty, a scheduled filter service was carried out during which the accumulated ash was removed. This involved removing the filter, cleaning it out with compressed air and refitting it. After the filter was serviced, its initial backpressure levels were achieved once again. ❼ shows the backpressure progression over time.

Examination of field trial data on the SCR system, ❽, shows that conversion rates of > 90 % can also be achieved in the field. Precise control means that the system also has good transient load response. Although the average exhaust temperature upstream of the system is below the limit for enabling AUS dosing, urea solution can still be injected more than 50 % of the time. This is because its heat retention capacity allows the system to heat up quickly at full load due to the high exhaust enthalpy flow and to cool slowly during subsequent idling phases.

The results of the field trials were equally positive from the design viewpoint. The mechanical strength of the housing was verified under continuous duty conditions and the modular design of the housing proved a definite advantage when it came to filter servicing and inspections. The internally developed canning process also functioned reliably. During the two winter periods involved, no AUS freezing occurred, either in the tank/pipes or at the refilling station.

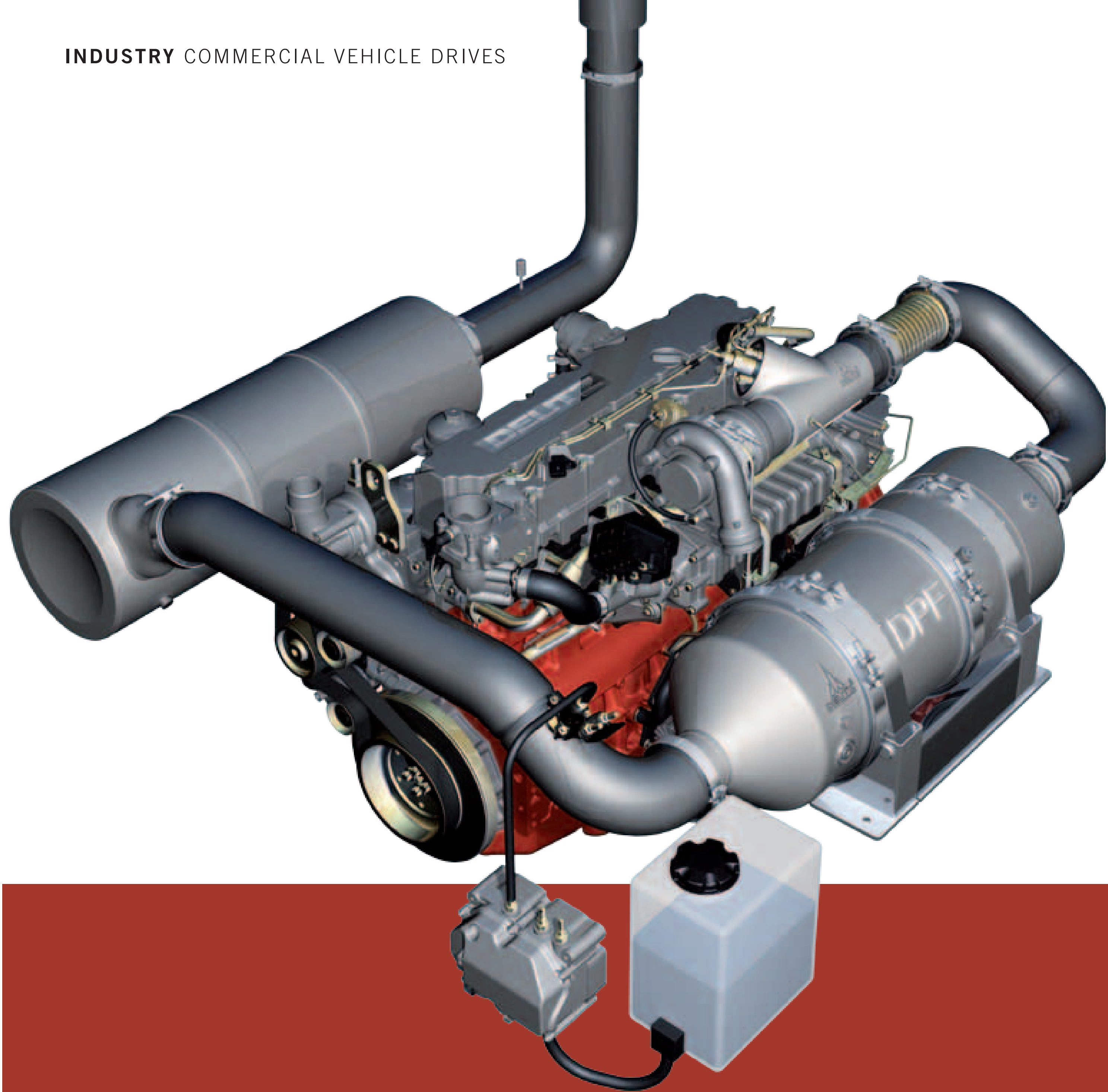
SUMMARY

In the LOCEX research project, MTU was able to put an internally developed and manufactured EGA system, designed jointly with Umicore, to the test under real daily working conditions as operated by DB. There were no problems with the locomotive and no unscheduled visits to the maintenance depot. DB's experience with AUS as an additional consumable new to rail applications was also positive. Compliance with statutory emission limits as per EU IIIB was verified both on the engine test stand and in the field trials.

MTU and DB were able to gain valuable experience and knowledge both of DPFs and a SCR system. For rail applications in accordance with EU IIIB specifications, however, MTU regards a combination of EGR and DPF as more effective because, among other factors, it requires no additional AUS infrastructure. The insights gained from the research project are already being utilized for all applications.

REFERENCES

[1] EU Directive 97/68/EG (as amended by 2004/26/EG)
[2] Kurreck, M.; Remmels, W.; Eckstein, M.; Bücheler, O.; Wachter, V. Die neue Generation der MTU- Motorbaureihe 4000. In: MTZ 68 (2007), Nr. 5, S. 356-362
[3] Patent Application by MTU Friedrichshafen GmbH: „Abgaskomponente"; DE 102008004435 A1
[4] Soeger, N.; Mussmann, L.; Sesselmann, R.; Leippe, G.; Gietzelt, C.; Bailey, O.; Hori, M. Impact of Aging and NOx/Soot Ratios on the Performance of a Catalyzed Particulate Filter for Heavy Duty Diesel Applications. SAE Paper 2005-01-0663
[5] Mussmann, L.; Müller, W.; Soeger, N.; Spurk, P. Combined DPF/SCR Concepts for Future Heavy Duty Emission Limits. 1. MinNOx Tagung 2007, Berlin
[6] Lappas, I.; Geisselmann, A.; Reith, C.; Müller, W. Katalysatortechnologien für Nutzfahrzeuge in On-Road- und Non-Road-Anwendungen. 7. FAD-Konferenz 2009, Dresden



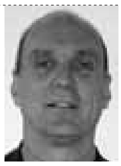
THE NEW DEUTZ TIER 4I ENGINES
PART 2 – PERFORMANCE AND EMISSIONS

With a new engine generation Deutz faces the increasingly tightened exhaust gas regulations. In the MTZ issue 10 the aspects construction and fatigue durability were presented in the first part of this article. The second part discusses the operating and emission performance of the engines in the power class above 130 kW as well as the associated technological measures. At the same time, the sometimes very different requirements of the application areas for agricultural and construction machinery will be taken into account.

AUTHORS

DR.-ING. HEINER BÜLTE
is Head of Technology Development at the Deutz AG in Cologne (Germany).

DR.-ING. HANS-JOSEF SCHIFFGENS
was Head of Research & Development at the Deutz AG in Cologne (Germany).



DIPL.-ING. MARTIN LUTAT
is Head of Electronics at the Deutz AG in Cologne (Germany).

DIPL.-ING. HARTMUT SIEVERDING
is Head of Thermo Dynamics at the Deutz AG in Cologne (Germany).

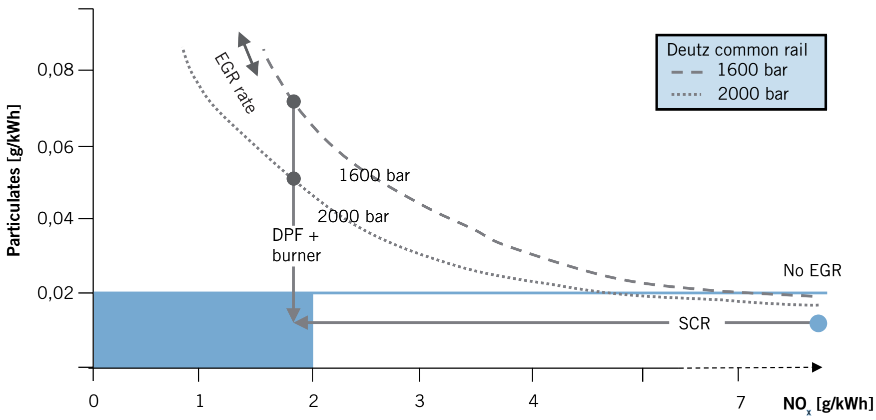
WAYS TO EMISSION DECREASE

Emission legislation remains an essential technology driver for engine development. This also applies for industrial engines, the limit values and introduction schedules of which are staggered according to the engine power. Emissions level EU III B or US Tier 4 interim for an engine power greater than 130 kW with the limit values shown in 1 for NO_x and particulates will become effective on January 1, 2011 and now also requires the wide-scale introduction of exhaust gas aftertreatment for industrial engines. The stricter exhaust legislation not only involves lower limit values but also the test methods. In contrast to the current legislation EU Stage III A or US Tier 3 with the 8-point stationary cycle, a dynamic cycle (NRTC = Non-Road Transient Cycle) also applies as well as the so-called NTE (Not-To-Exceed) range, in which the limit value must not be exceeded by more than 50 % at any operating point.

The two technology paths shown in 1 basically apply for compliance with the legislation: The first path involves an engine-internal optimization of the particulates and fuel consumption in conjunction with a representation of the NO_x limit value with an SCR system already tested in the EU 4 and EU 5 on-road legislation. The second path results from an engine-internal demonstration of the NO_x emissions with an externally cooled exhaust gas recirculation and achievement of the particulate limit value with a diesel particulate filter (DPF). Deutz has introduced a two-stage burner for regeneration of the DPF under all environmental and operating conditions. The particulate matter emissions were reduced by optimizing the combustion process, whose core technology is the 2000 bar common rail system. As already outlined in the first part [1], introduction of the new emission level calls for a radical revision of the engine concept.

EMISSIONS AND PERFORMANCE

On the basis of the exhaust legislation of Stage III B or Tier 4 interim applicable from January 1, 2011, the series production launch of the engines in the power group above 130 kW is set for the autumn of 2010. Deutz has streamlined the product spectrum considerably for this. In this power segment, all engines with a Bosch common rail system of generation 3.3 will be provided with maximum 2000 bar injection pressure. Only the power versions of the TCD 6.1 L6 to 160 kW for construction machinery or up to 180 kW for agricultural applications are equipped with the common rail system of generation 2 currently in series production today with maximum 1600 bar injection pressure. The four-valve system will also be introduced uniformly. The 2V versions of series 2012 and 2013 will not developed for Stage III B/ Tier 4 interim. This results in the four basic engines TCD 12.0 V6 and TCD 16.0 V8 as a further development of the V-engine series 2015 as well as the TCD 6.1 L6 as a successor to the TCD 2012 L06 and the TCD 7.8 L6 as a successor to the TCD 2013 L06.



1 Technology options for fulfilment of the exhaust gas emission level US Tier 4 interim or EU Stage III B for the power class 130 to 560 kW

V-ENGINES TCD 12.0 V6 AND TCD 16.0 V8

The V-engines have probably undergone the most extensive revision to the engine concept. The engines still provided with an electronically controlled PLD system for Stage III A/Tier 3 will now be converted to common rail injection. At the same time, the very positive production experience from the series engines in the 4 to 7 litre class could be utilized. Divergent from the in-line engines, the pressure is generated via a high-pressure pump of generation CPN5.2, which also allows a further increase in the injection pressure of up to 2200 bar without any changes to the design.

In contrast to the in-line engines, the V-engines are equipped with an SCR system as standard. A significant consumption advantage results over the EGR option. Furthermore the cost assessment for SCR systems on the one hand, and for controlled exhaust gas recirculation in conjunction with a particulate filter on the other hand, clearly favour the SCR system in the case of engines of this power category.

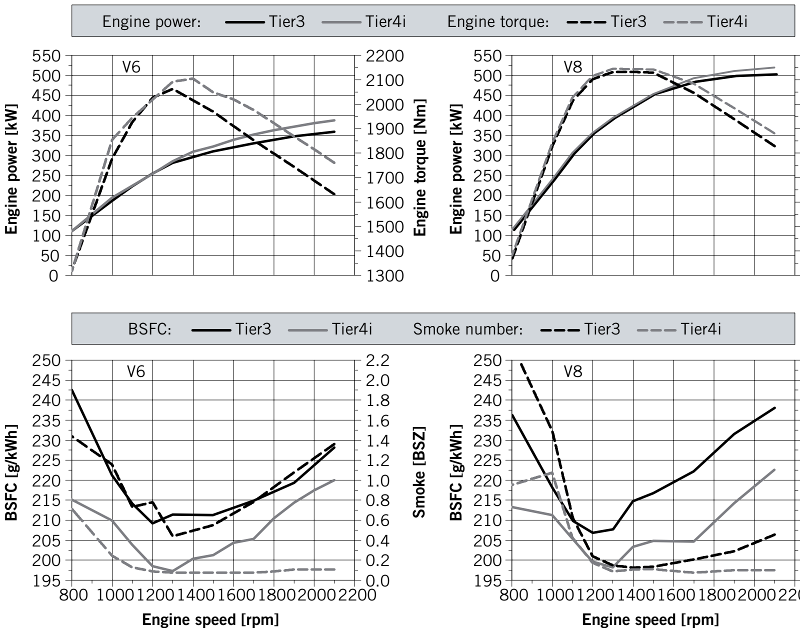
With the SCR system, the optimization of the combustion was concentrated on minimizing the particulates and fuel consumption as well as increasing the torque curve and improving the engine dynamics. In particular through the injection system and by cutting out the EGR, an increase in power from 360 to 390 kW could be achieved in the TCD 12.0 V6 and from 500 to 520 kW in the TCD 16.0 V8, ❷.

The particulate matter emissions reached a level where there is no need for exhaust aftertreatment to achieve the limit values, this accompanied by a reduction in fuel consumption of up to 6 %. The number of injection holes was increased from seven to eight in conjunction with a reduction in the hydraulic flow rate from 620 to 570 cm³/60 s.

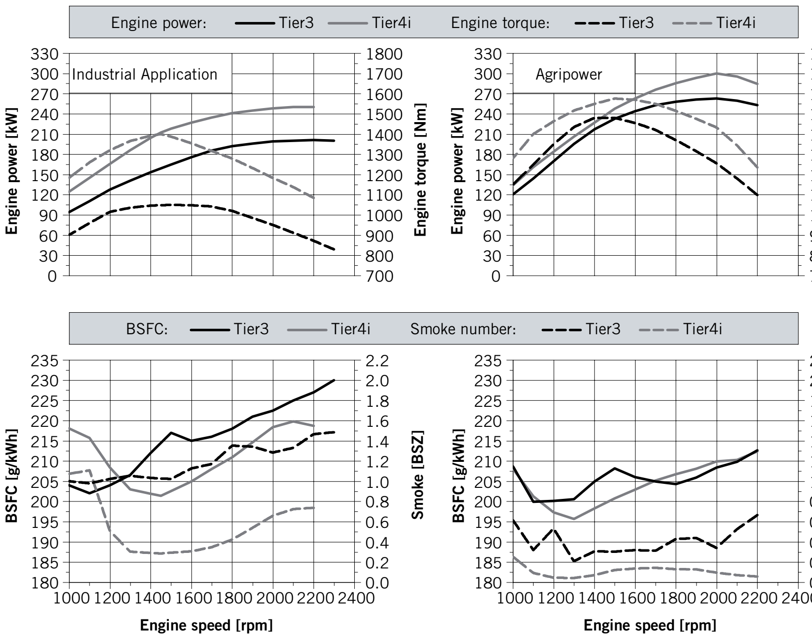
SERIES ENGINES TCD 7.8 L6 AND TCD 6.1 L6

Use of the four-valve technology in the series engines for construction machinery as well means that the two key market segments agricultural machinery and construction machinery [2] are now served by the same basic engine. Here too, the injection system of generation 3.3 with 2000 bar will be introduced. It was possible to continue with the compact unit pump concept despite the increased pressure without widening the tappet, as the higher system pressure on account of the minimized leakage volumes could be overcompensated by using a plunger diameter reduced from 9 to 7.5 mm.

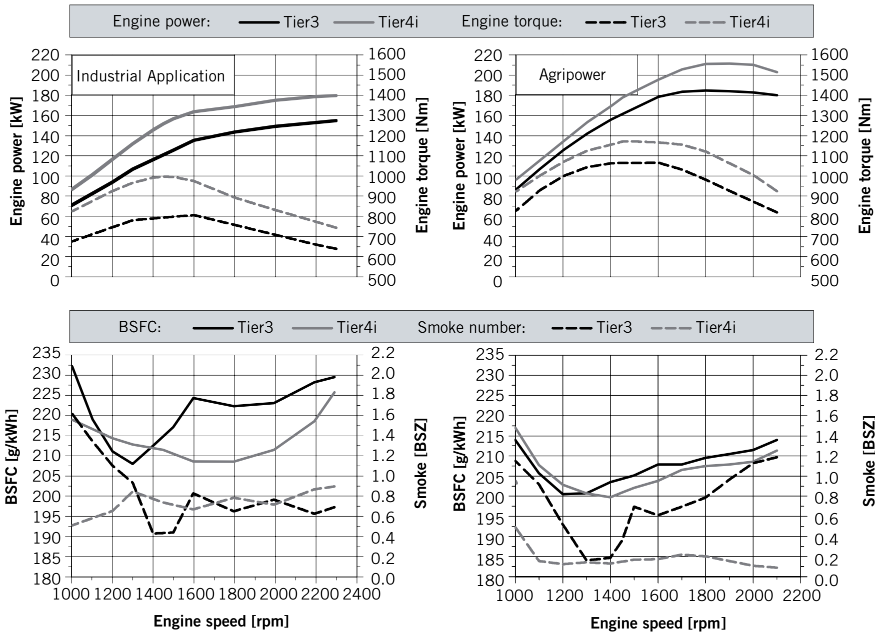
Essential measures for optimizing the combustion and power data in these engines are also the introduction of an eight-hole nozzle as well as the reduction of the hydraulic flow rate from 780 to 660 cm³/60 s in the TCD 7.8 and from



❷ Power, torque, fuel consumption of TCD 12.0 V6 and TCD 16.0 V8



❸ Power, torque, fuel consumption for the TCD 7.8 L6 agricultural with engine protection function and the TCD 7.8 L6 industrial



❹ Power, torque, fuel consumption for the TCD 6.1 L6 agricultural and industrial

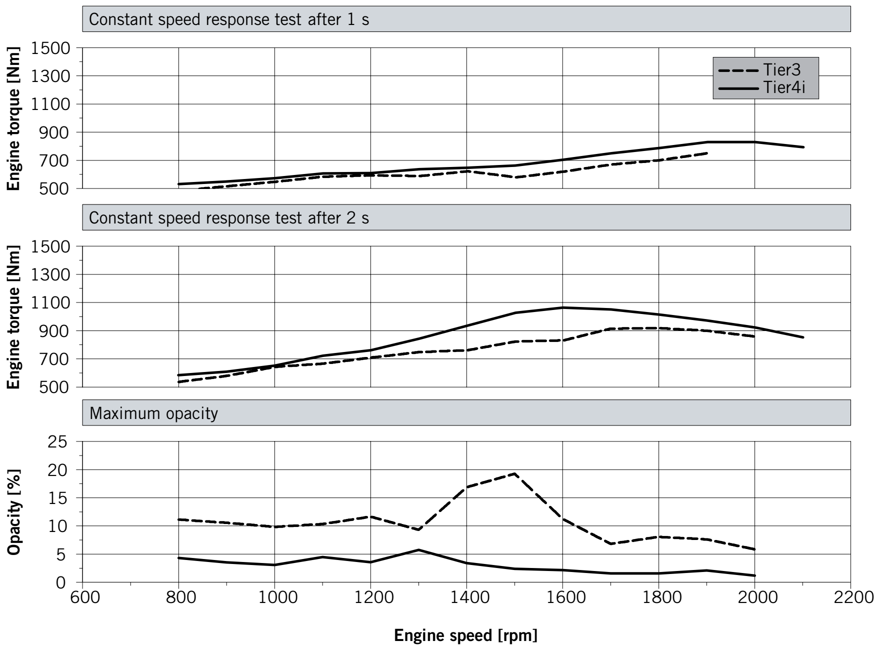
690 to 500 cm³/60 s in the TCD 6.1. Furthermore, the turbocharger efficiency could be increased through the introduction of generation 2 from BorgWarner. The concept of the previous series 2013 also allowed an increase in the cubic capacity for the TCD 7.8 from 7.2 to 7.8 l for [1], which was crucially utilized for increasing the power, ❸.

Two different emission technology concepts are basically applied for the market segments agricultural and construction machinery. In agricultural machinery, particulate filters with thermal regeneration are regarded as critical for reasons of installation space and on account of the hot exhaust gas during the regeneration. As already mentioned for the V-engines, the SCR system provides the option of optimizing fuel consumption and high power density accompanied by soot emissions at the detection limit. In comparison to the Stage III A or Tier 3 engine with EGR, the customer benefit could also be enhanced considerably here by reducing the cooling power.

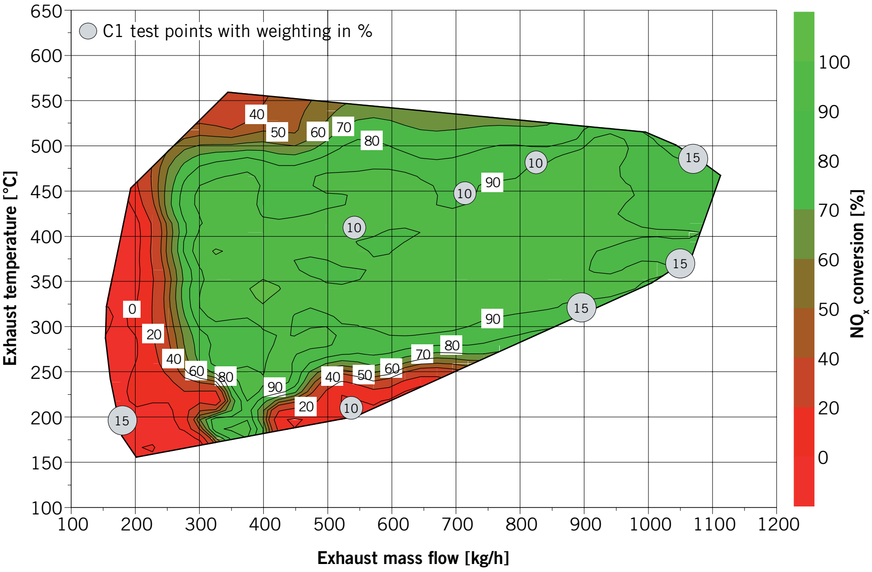
For construction machinery, the use of particulate filters enables applications in specially regulated zones such as tunnel or underground projects, compliance with the Swiss VERT directive as well as separate standards in numerous urban environmental zones. Furthermore, the independence from the AdBlue supply not yet fully implemented in the early phase of the SCR system introduction played an important role for decentralized use of the construction machinery. Replacing the internal EGR by the technically more complex externally cooled and controlled EGR achieved positive results in the power, consumption and soot performance.

③ and ❹ show the power and torque curve as well as the fuel consumption and the soot value at full load for the two engines in direct comparison with the predecessor engine. An engine protection function is realized, which enables a permanent power increase in relation to the intake air temperature and exhaust temperature upstream turbine, setting it apart from conventional power boost concepts, where the extra power is only available for a short time. The TCD 6.1 therefore achieves a rated power of 203 kW with an overpower, typical for agriculture engines, of 211 kW at 2000 rpm, while the TCD 7.8 reaches up to 285 kW corresponding with an overpower of 299 kW to a specific power of 38.6 kW/l in conjunction with a maximum brake mean effective pressure of 25.4 bar at 1500 rpm. The power increase achieved with the revised concept in the TCD 6.1 L6 to 180 kW for construction machinery applications also includes the option for downsizing, as this engine can now serve applications whose requirements in Stage III A are only fulfilled by the 7.2 l engine.

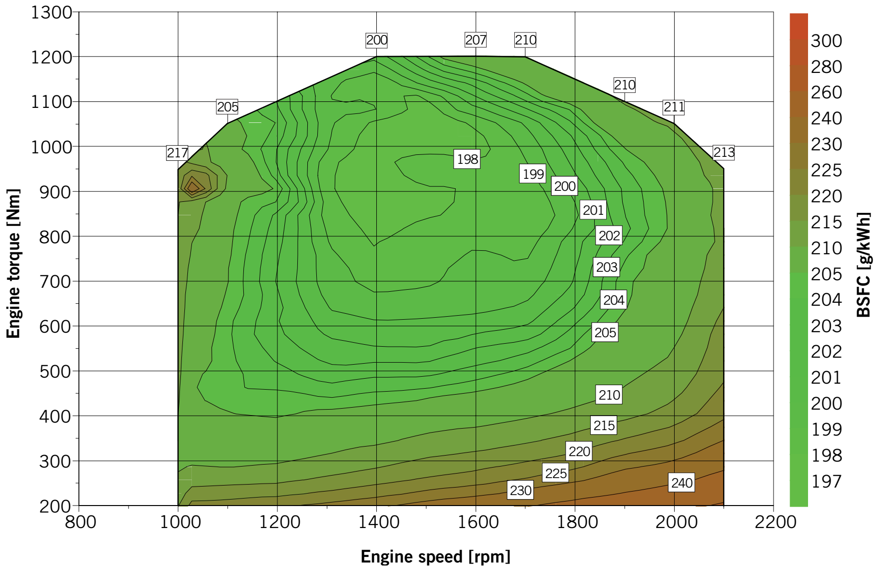
The very good consumption data, ③ and ④, could also be achieved by using a DoE-based data optimization. In doing so, the focus of the fuel optimization was on the customer cycle representative for each application. The reduced consumption in



❺ Comparison of the load response for Tier 4 interim engine compared to Tier 3 with the example of the TCD 6.1 L6 industrial



❻ SCR efficiency of the TCD 6.1 L6 versus exhaust temperature and space velocity



❼ Fuel consumption of the TCD 6.1 L6 agricultural with SCR

comparison to the Tier 3 engines is 4 to 7 %. The power reduction in operation at altitude could be limited considerably by optimization in the altitude chamber. Here the potential of the technology concept could be explored that way, that also at 3000 m altitude, a significantly improved torque curve could be demonstrated.

Use of the controlled external cooled EGR also enabled a significant improvement in the engine dynamics besides the above improvement effects of the stationary values shown. ❺ displays the torque achievable during load pick-up from motored operation to full load at constant speed. Thanks to the optimized supercharging and controlled EGR, a faster torque build-up could be achieved, this in conjunction with opacity values that are below the visible range even without DPF.

SCR FOR AGRICULTURAL MACHINERY APPLICATIONS AND V-ENGINES

The Vanadium-based coating of the SCR catalyst optimized further since introduction of the EU 4 and EU 5 truck engine enables a maximum efficiency of 95 %, as shown in ❻ with the example of the TCD 6.1 L6. The high exhaust gas temperature at the turbine inlet of up to 590 °C accompanied by a power density of almost 35 kW/l also causes a significant efficiency loss in contrast to the optimum operating point. In particular, the high weighting of the rated power point in the 8-point test (NRSC) posed a special challenge here. Nevertheless, a carefully optimized metering strategy could be utilized to achieve SCR efficiencies of 75 to 80 % in the crucial full load area, thereby enabling the consumption data shown in ❼. The diagram of the efficiency versus exhaust temperature and space velocity shows that the exhaust gas temperature is the essential influencing variable for the efficiency, and an increase in efficiency cannot be expected from a substrate enlargement.

DPF WITH BURNER FOR CONSTRUCTION MACHINERY

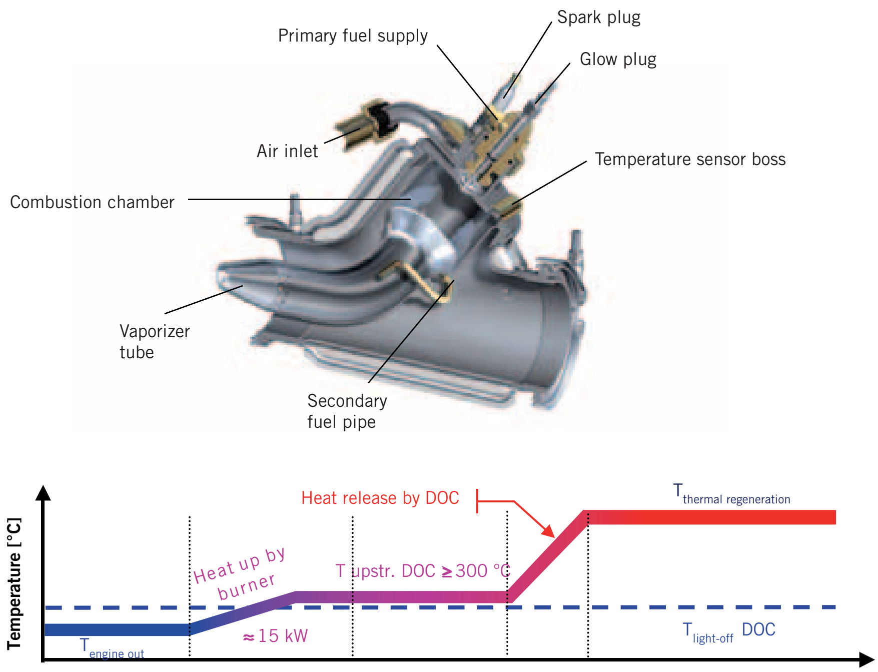
As already explained in [2] and [3], an NO_x/soot ratio greater than 50 and an exhaust gas temperature above 250 °C is required for reliable passive regeneration of a DPF. For engines with exhaust gas recirculation in the power class above 130 kW, this is not provided in the entire map especially due to the low NO_x engine out emissions. Furthermore, a regeneration independent of the application and operating cycle is expected for the mainly very high-quality machines in this power class. This also applies for very low ambient temperatures and operation at altitude. Deutz has developed a burner [1], ❽, for the cyclical thermal regeneration of a DPF, which corresponds to these requirements.

The burner works in two stages. In the first stage, fuel is burned by a stoichiometric combustion with a thermal output to 15 kW. The air pump provides the combustion air for this. For exact metering of the air mass, this is measured and controlled using a hot film sensor. The fuel is supplied via an extended metering module (MU 2) from Bosch Departronic, an additional metering valve being installed for the fuel supply of the first burner stage. The glow plug also allows ignition of the burner under Artic ambient conditions. The spark ignition stabilizes the burner operation. A swirl atomizer is used to achieve a homogenous stabilized stoichiometric combustion (blue soot-free flame), which prevents soot formation through the burner from prolonging the regeneration time. In the second stage, fuel is dosed into the exhaust gas of the first stage, so that this evaporates in the very hot low-oxygen exhaust gas, but is hardly oxidized. However, the evaporation enables an ideal treatment for the catalytic oxidation. At the outlet from the evaporator pipe of the burner, the burner gas is mixed with the engine exhaust gas, following by oxidation of the evaporated hydrocarbons in the oxidation catalyst (DOC). The dosage of the second stage serves as a control variable for the regeneration temperature.

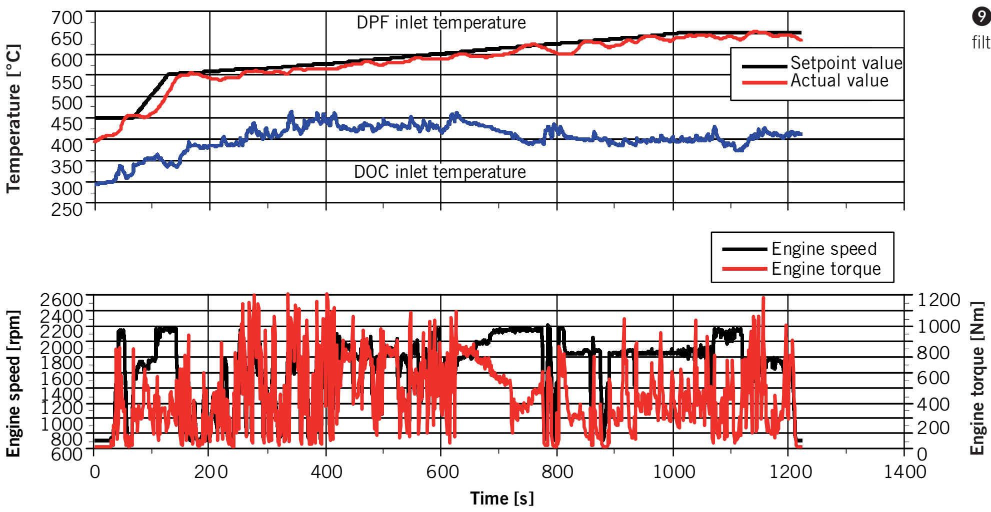
In the development phase, considerable attention was placed on avoiding auto-ignition during entry of the burner gas into the engine exhaust, as this also leads to unwanted heat losses during the regeneration in addition to the thermal loading of the exhaust gas pipes and impairs the controllability of the temperature during the regeneration, whereby temperature peaks in the filter substrate can occur, which cause material damage to the filter in the least favourable case. The coating of the DOC has been specially optimized for this exothermal reaction. Routing the burner air through the housing cover considerably reduces the thermal load on the spark plug seat, while preheating the burner air so as to stabilize the combustion at the same time.

The start of the burner and the temperature control are shown by the diagram in ⑧. A low volume of fuel is evaporated on the hot glow plug surface in the start phase and then burned in a diffusion flame. A stabilization of the flame in the form of a premixed stoichiometric combustion is achieved after a few seconds. Afterwards, the first stage can be increased in a ramp-like manner up to the maximum heating power of 15 kW. The fuel metering in the second stage is possible as soon as the catalyst has reached its light-off temperature of approx. 300 °C. This is always possible, irrespective of the engine operating point and ambient conditions.

With the help of a model-based approach, the second stage controls the regeneration temperature, this ensuring a gentle regeneration of the filter provided with a cordierite substrate thanks to a ramp-shaped start up. The quality of the temperature control in the transient engine operation is demonstrated by ❾ based on the non-road transient cycle. Extensive field testing in various applications verified robust operation of the burner.



❽ Cross section of the burner and temperature control



9 Temperature curve during the particulate filter regeneration in the NRTC test cycle

Cold chamber tests demonstrated the fault-free starting of the burner and a complete regeneration of the particulate filter at –25 °C at high idle operation. The challenge of high idle operation involves the combination of low exhaust gas temperatures with a relatively high exhaust gas mass flow rate, which can be used to test attainment of the light-off temperature of the catalyst under extreme conditions. In the altitude chamber, the regeneration was tested for use above 3000 m. Even with an oxygen content in the exhaust gas of only 2 %, a reliable regeneration in less than 25 minutes is ensured.

ELECTRONICS CONCEPT

A modular concept for sensors and actuators based on the control unit platform Bosch EDC 17 has been created for the various applications. The monitoring and control of the SCR system is on the basis of two NO_x sensors; the stoichiometric combustion in the burner requires an accurate metering of air and fuel, which is accompanied by numerous pressure and temperature sensors. The EGR is controlled with a Venturi sensor, which is used to measure the EGR mass; a potential drift is compensated with the NO_x sensor. The burner DPF system was integrated by Deutz Development, i.e. the associated monitoring of the DPF loading and control of the DPF regeneration is treated as core expertise. A software sharing was agreed with Bosch for this, in which certain modules can be integrated as Deutz-own object code. Interfaces were created by Bosch and memory spaces reserved for the Deutz software. Besides the monitoring and regeneration software for the DPF, Deutz also provides the code for the air path control and special application-dependent machine functionalities for the end customers. The testing and validation of the entire software is carried out on HiL systems, on the engine test bench and in the application by Deutz.

SUMMARY

Deutz has utilized the further development of the engine series of power class 130 to 560 kW for the deadline January 1, 2011, to significantly enhance customer benefit amongst other factors in the criteria of power density, dynamics and fuel consumption. The cross-platform exhaust gas aftertreatment strategy has found a way to achieve a maximum amount of flexibility at reasonable expense which copes with the diverse range of applications. In particular, the different requirements of the market segments agricultural and construction machinery were able to be taken into account. The core of the technology portfolio is the burner, which ensures a reliable regeneration of the particle filter under all operating and environmental conditions, without the operational availability of the machine being impaired by this. The expertise thereby gained in regard to the control concept was able to be secured as part of a software sharing approach.

The engines presented here with introduction of the Tier 4 interim/Stage III B in the power class above 130 kW in January 2011 will be continued with the engines of the power class from 56 to 130 kW in January 2012. This concerns the further development of the TCD 2012 L04 as TCD 4.1 L4 as well as the market launch of the new series TCD 3.6 L4 and TCD 2.9 L4.

REFERENCES

[1] Bick, W.; Feuser, W.; Köhne, M.; Pape, U.; Pister, M.; Schiffgens, H.-J.; Steppat, H.: Die neuen Tier-4i-Motoren von Deutz. Teil 1: Konstruktion und Dauerhaltbarkeit. In: MTZ 71 (2010), Nr. 10

[2] Bülte, H.; Schiffgens, H.-J.; Broll, P.; Beberdick, W.: Technologiekonzept von Deutz zur Einhaltung der Grenzwerte der Abgasstufe US EPA Tier 4 und EU Stufe IV für Motoren in Mobilen Arbeitsmaschinen. 17. Aachener Kolloquium Fahrzeug- und Motorentechnik 2008

[3] Bülte, H.; Schiffgens, H.-J.; Broll, P.; Schraml, S.: Abgasnachbehandlungskonzepte für Motoren in mobilen Arbeitsmaschinen gemäß der Gesetzgebung US Tier 4 und EU Stufe IV – Technologien und Applikationen. 18. Aachener Kolloquium Fahrzeug- und Motorentechnik 2009

THANKS

We gratefully thank Dr.-Ing. Peter Broll, Head of Series Development Exhaust Aftertreatment, and Dr.-Ing. Stephan Schraml, Head of Advanced Engineering Exhaust Aftertreatment, in making this article.

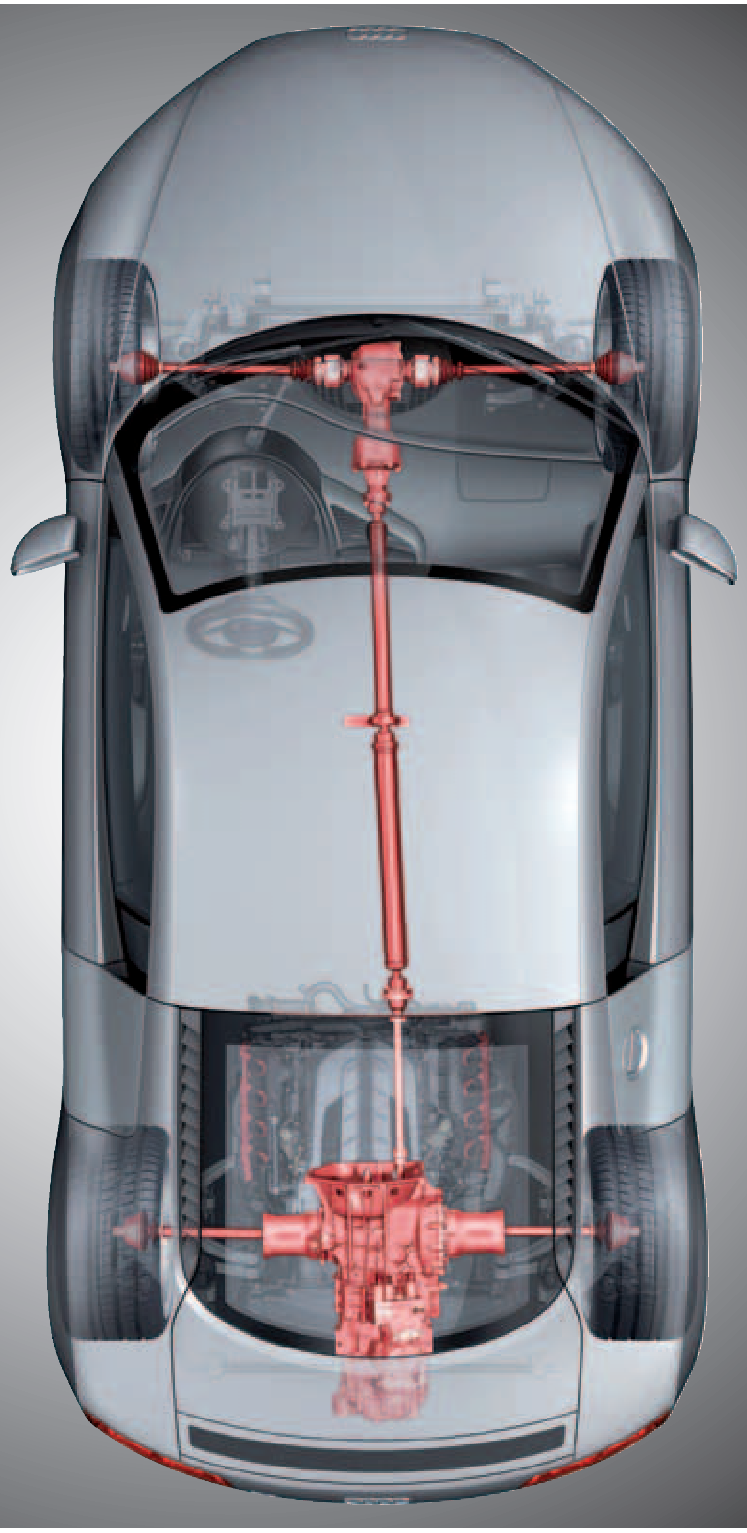
www.ATZlive.de

ATZ live

OPTIMISED PROCESS CHAIN
Efficiency for the powertrain of tomorrow

SIMULATION OF THE POWERTRAIN AND ITS COMPONENTS
Reduced fuel consumption and emissions, improved dynamics and driving pleasure

VIRTUAL PRODUCT DEVELOPMENT
More quality at lower costs



Virtual Powertrain Creation 2010

AVL FEV

12th International Conference

30 November and 1 December 2010
Munich | Germany

PROGRAM AND REGISTRATION
www.ATZlive.de

PEER REVIEW ATZ | MTZ

PEER REVIEW PROCESS FOR RESEARCH ARTICLES IN ATZ AND MTZ

STEERING COMMITTEE

Prof. Dr.-Ing. Lutz Eckstein	RWTH Aachen	Institut für Kraftfahrzeuge Aachen
Prof. Dipl.-Des. Wolfgang Kraus	HAW Hamburg	Department Fahrzeugtechnik und Flugzeugbau
Prof. Dr.-Ing. Ferit Küçükay	Technische Universität Braunschweig	Institut für Fahrzeugtechnik
Prof. Dr.-Ing. Stefan Pischinger	RWTH Aachen	Lehrstuhl für Verbrennungskraftmaschinen
Prof. Dr.-Ing. Hans-Christian Reuss	Universität Stuttgart	Institut für Verbrennungsmotoren und Kraftfahrwesen
Prof. Dr.-Ing. Ulrich Spicher	Universität Karlsruhe	Institut für Kolbenmaschinen
Prof. Dr.-Ing. Hans Zellbeck	Technische Universität Dresden	Lehrstuhl für Verbrennungsmotoren

ADVISORY BOARD

Prof. Dr.-Ing. Klaus Augsburg
Prof. Dr.-Ing. Bernard Bäker
Prof. Dr.-Ing. Michael Bargende
Prof. Dipl.-Ing. Dr. techn. Christian Beidl
Dr.-Ing. Christoph Bollig
Prof. Dr. sc. techn. Konstantinos Boulouchos
Prof. Dr.-Ing. Ralph Bruder
Dr. Gerhard Bruner
Prof. Dr. rer. nat. Heiner Bubb
Prof. Dr. rer. nat. habil. Olaf Deutschmann
Dr. techn. Arno Eichberger
Prof. Dr. techn. Helmut Eichlseder
Dr.-Ing. Gerald Eifler
Prof. Dr.-Ing. Wolfgang Eifler
Prof. Dr. rer. nat. Frank Gauterin
Prof. Dr. techn. Bernhard Geringer
Prof. Dr.-Ing. Uwe Grebe
Prof. Dr.-Ing. Horst Harndorf
Prof. Dr. techn. Wolfgang Hirschberg
Univ.-Doz. Dr. techn. Peter Hofmann
Prof. Dr.-Ing. Bernd-Robert Höhn
Prof. Dr. rer. nat. Peter Holstein
Prof. Dr.-Ing. habil. Werner Hufenbach
Prof. Dr.- Ing. Roland Kasper
Prof. Dr.-Ing. Tran Quoc Khanh
Dr. Philip Köhn
Prof. Dr.-Ing. Ulrich Konigorski
Dr. Oliver Kröcher
Dr. Christian Krüger
Univ.-Ass. Dr. techn. Thomas Lauer
Prof. Dr. rer. nat. Uli Lemmer
Dr. Malte Lewerenz
Dr.-Ing. Markus Lienkamp
Prof. Dr. rer. nat. habil. Ulrich Maas
Prof. Dr.-Ing. Martin Meywerk
Prof. Dr.-Ing. Klaus D. Müller-Glaser
Dr. techn. Reinhard Mundl
Prof. Dr. rer. nat. Cornelius Neumann
Dr.-Ing. Lothar Patberg
Prof. Dr.-Ing. Peter Pelz
Prof. Dr. techn. Ernst Pucher
Dr. Jochen Rauh
Prof. Dr.-Ing. Konrad Reif
Prof. Dr.-Ing. Stephan Rinderknecht
Dr.-Ing. Swen Schaub
Prof. Dr. sc. nat. Christoph Schierz
Prof. Dr. rer.-nat. Christof Schulz
Prof. Dr. rer. nat. Andy Schürr
Prof. Dr.-Ing. Ulrich Seiffert
Prof. Dr.-Ing. Hermann J. Stadtfeld
Prof. Dr. techn. Hermann Steffan
Dr.-Ing. Wolfgang Steiger
Prof. Dr.-Ing. Peter Steinberg
Prof. Dr.-Ing. Christoph Stiller
Dr.-Ing. Peter Stommel
Prof. Dr.-Ing. Wolfgang Thiemann
Prof. Dr.-Ing. Helmut Tschöke
Dr.-Ing. Pim van der Jagt
Prof. Dr.-Ing. Georg Wachtmeister
Prof. Dr.-Ing. Jochen Wiedemann
Prof. Dr. techn. Andreas Wimmer
Prof Dr. rer. nat. Hermann Winner
Prof. Dr. med. habil. Hartmut Witte

Scientific articles of universities in ATZ Automobiltechnische Zeitschrift and MTZ Motortechnische Zeitschrift are subject to a proofing method, the so-called peer review process. Articles accepted by the editors are reviewed by experts from research and industry before publication. For the reader, the peer review process further enhances the quality of the magazines' content on a national and international level. For authors in the institutes, it provides a scientifically recognised publication platform.

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

In 2008, the peer review process utilized by ATZ and MTZ was presented by the WKM (Wissenschaftliche Gesellschaft für Kraftfahrzeug- und Motorentechnik e. V. / German Professional Association for Automotive and Motor Engineering) to the DFG (Deutsche Forschungsgemeinschaft / German Research Foundation) for official recognition.

IMPRINT

MTZ WORLDWIDE

www.MTZonline.com

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

Organ of the Fachverband Motoren und Systeme im VDMA, Verband Deutscher Maschinen- und Anlagenbau e.V., Frankfurt/Main, for the areas combustion engines and gas turbines
Organ of the Forschungsvereinigung Verbrennungskraftmaschinen e.V. (FVV)
Organ of the Wissenschaftliche Gesellschaft für Kraftfahrzeug- und Motorentechnik e.V. (WKM)
Organ of the Österreichischer Verein für Kraftfahrzeugtechnik (ÖVK)
Cooperation with the STG, Schiffbautechnische Gesellschaft e.V., Hamburg, in the area of ship drives by combustion engine

11 | 2010 _ November 2010 _ Volume 71
Springer Automotive Media | Springer Fachmedien Wiesbaden GmbH
P. O. Box 15 46 · 65173 Wiesbaden · Germany | Abraham-Lincoln-Straße 46 · 65189 Wiesbaden · Germany
Amtsgericht Wiesbaden, HRB 9754, USt-IdNr. DE811148419

Managing Directors Dr. Ralf Birkelbach (Chairman), Armin Gross, Albrecht Schirmacher | **Senior Advertising** Armin Gross | **Senior Marketing** Rolf-Günther Hobbeling
Senior Production Christian Staral | **Sales Director** Gabriel Göttlinger

EDITORS-IN-CHARGE
Dr.-Ing. E. h. Richard van Basshuysen
Wolfgang Siebenpfeiffer

EDITORIAL STAFF

EDITOR-IN-CHIEF
Johannes Winterhagen (win)
phone +49 611 7878-342 · fax +49 611 7878-78462
johannes.winterhagen@springer.com

VICE-EDITOR-IN-CHIEF
Ruben Danisch (rd)
phone +49 611 7878-393 · fax +49 611 7878-78462
ruben.danisch@springer.com

CHIEF-ON-DUTY
Kirsten Beckmann M. A. (kb)
phone +49 611 7878-343 · fax +49 611 7878-78462
kirsten.beckmann@springer.com

SECTIONS
Electrics, Electronics
Markus Schöttle (scho)
phone +49 611 7878-257 · fax +49 611 7878-78462
markus.schoettle@springer.com
Engine
Ruben Danisch (rd)
phone +49 611 7878-393 · fax +49 611 7878-78462
ruben.danisch@springer.com
Online
Katrin Pudenz M. A. (pu)
phone +49 6172 301-288 · fax +49 6172 301-299
redaktion@kpz-publishing.com
Production, Materials
Stefan Schlott (hlo)
phone +49 8191 70845 · fax +49 8191 66002
Redaktion_Schlott@gmx.net
Research
Dipl.-Ing. (FH) Moritz-York von Hohenthal (mvh)
phone +49 611 7878-278 · fax +49 611 7878-78462
moritz.von.hohenthal@springer.com
Service, Event Calendar
Martina Schraad (mas)
phone +49 611 7878-276 · fax +49 611 7878-78462
martina.schraad@springer.com
Transmission
Dipl.-Ing. Michael Reichenbach (rei)
phone +49 611 7878-341 · fax +49 611 7878-78462
michael.reichenbach@springer.com

PERMANENT CONTRIBUTORS
Richard Backhaus (rb),
Dipl.-Reg.-Wiss. Caroline Behle (beh),
Prof. Dr.-Ing. Stefan Breuer (sb),
Jürgen Grandel (gl), Ulrich Knorra (kno),
Prof. Dr.-Ing. Fred Schäfer (fs),
Roland Schedel (rs)

ENGLISH LANGUAGE CONSULTANT
Paul Willin (pw)

ADDRESS
P. O. Box 15 46, 65173 Wiesbaden, Germany
redaktion@ATZonline.de

MARKETING | OFFPRINTS

PRODUCT MANAGEMENT AUTOMOTIVE MEDIA
Sabrina Brokopp
phone +49 611 7878-192 · fax +49 611 7878-78407
sabrina.brokopp@springer.com

OFFPRINTS
Martin Leopold
phone +49 2642 9075-96 · fax +49 2642 9075-97
leopold@medien-kontor.de

ADVERTISING

AD MANAGER
Britta Dolch
phone +49 611 7878-323 · fax +49 611 7878-78140
britta.dolch@gwv-media.de

KEY ACCOUNT MANAGEMENT
Elisabeth Maßfeller
phone +49 611 7878-399 · fax +49 611 7878-78140
elisabeth.massfeller@gwv-media.de

AD SALES MANAGER
Sabine Röck
phone +49 611 7878-269 · fax +49 611 7878-78140
sabine.roeck@gwv-media.de

AD SALES
Frank Nagel
phone +49 611 7878-395 · fax +49 611 7878-78140
frank.nagel@gwv-media.de

DISPLAY AD MANAGER
Petra Steffen-Munsberg
tel +49 611 7878-164 · fax +49 611 7878-443
petra.steffen-munsberg@gwv-media.de

AD PRICES
Price List No. 53 (10/2009)

PRODUCTION | LAYOUT
Heiko Köllner
phone +49 611 7878-177 · fax +49 611 7878-78464
heiko.koellner@springer.com

SUBSCRIPTIONS
VVA-Zeitschriftenservice, Abt. D6 F6, MTZ
P. O. Box 77 77, 33310 Gütersloh, Germany
Renate Vies
phone +49 5241 80-1692 · fax +49 5241 80-9620
SpringerAutomotive@abo-service.info

YOUR HOTLINE TO MTZ
Editorial Staff
☎ +49 611 7878-393
Reader's Service
☎ +49 611 7878-151
Advertising
☎ +49 611 7878-395

SUBSCRIPTION CONDITIONS
The eMagazine appears 11 times a year at an annual subscription rate of 269 €. Special rate for students on proof of status in the form of current registration certificate 124 €. Special rate for VDI/ÖVK/VKS members on proof of status in the form of current member certificate 208 €. Special rate for studying VDI members on proof of status in the form of current registration and member certificate 89 €. The subscription can be cancelled in written form at any time with effect from the next available issue.

HINTS FOR AUTHORS
All manuscripts should be sent directly to the editors. By submitting photographs and drawings the sender releases the publishers from claims by third parties. Only works not yet published in Germany or abroad can generally be accepted for publication. The manuscripts must not be offered for publication to other journals simultaneously. In accepting the manuscript the publisher acquires the right to produce royalty-free offprints. The journal and all articles and figures are protected by copyright. Any utilisation beyond the strict limits of the copyright law without permission of the publisher is illegal. This applies particularly to duplications, translations, microfilming and storage and processing in electronic systems.

Springer Automotive Media is a brand of Springer Fachmedien. Springer Fachmedien is part of the specialist publishing group Springer Science+Business Media.

AUTHORS

DIPL.-ING. MARTIN NIJS
is Scientific Assistant at the Institute for Combustion Engines (VKA) at RWTH Aachen University (Germany).

DIPL.-ING. PETER STERNBERG
is Scientific Assistant at the Institute for Combustion Engines (VKA) at RWTH Aachen University (Germany).

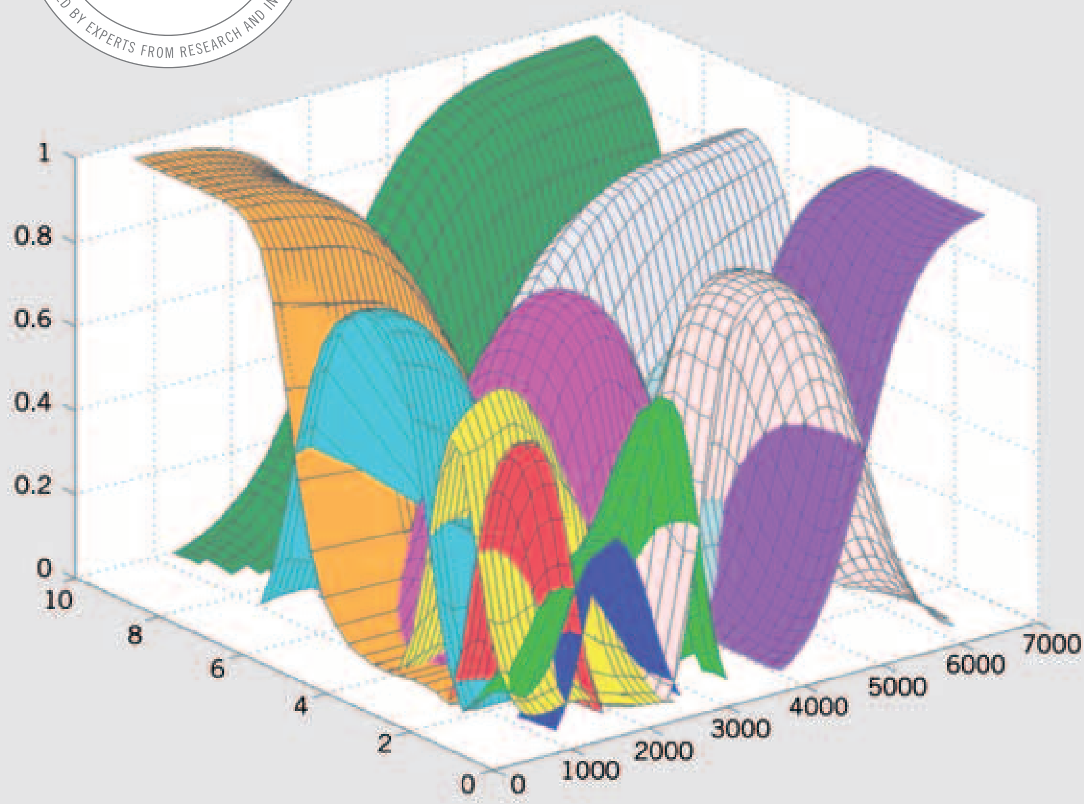
DR.-ING. MICHAEL WITTLER
is Chief Engineer at the Institute for Combustion Engines (VKA) at RWTH Aachen University (Germany).

PROF. DR.-ING. STEFAN PISCHINGER
is Director of the Institute for Combustion Engines (VKA) at RWTH Aachen University (Germany).

AIR PATH MODELS FOR GASOLINE ENGINES WITH EXTENDED VALVE TRAIN VARIABILITY

Two novel engine control models which account for multiple parameters influencing the air path in gasoline engines with extended valve train variabilities have been investigated at the Institute for Combustion Engines at RWTH Aachen University within the scope of FVV (Research Association for Combustion Engines) project 938 "Air path model for variable valve trains". The first model constitutes static as well as learnable physical-neuronal functionalities that are integrated into the engine management software of the FVV engine control unit. The second model concerns a real-time gas exchange algorithm in which high-resolution pressure signals on the intake and exhaust port as well as the cylinder pressure signals were used. Hereby, the respective software was implemented on a high-performance rapid control prototyping unit.

THE SEAL OF APPROVAL FOR SCIENTIFIC ARTICLES IN MTZ.
MTZ
PEER REVIEW
RECEIVED 2010.04.18
REVIEWED 2010.06.20
ACCEPTED 2010.07.10
REVIEWED BY EXPERTS FROM RESEARCH AND INDUSTRY.



1 INTRODUCTION

Today's valve trains in state-of-the-art engines are becoming increasingly complex. For example, such valve trains require fully variable intake and exhaust cam phasing, discrete [1] or continuous valve stroke [2, 3] as well as freely controlled valves [4, 5]. This complexity thereby increases the degrees of freedom resulting in difficult, sophisticated controlling and modelling algorithms in engine management systems. In particular, the precontrol and the air path have to be adapted to these degrees of freedom.

One of the focal components in gasoline engine management systems is the air path model that calculates the mass of fresh air entering the combustion chamber for each combustion cycle. It is hereby essential to exactly calculate the air mass during stationary and especially dynamic operation.

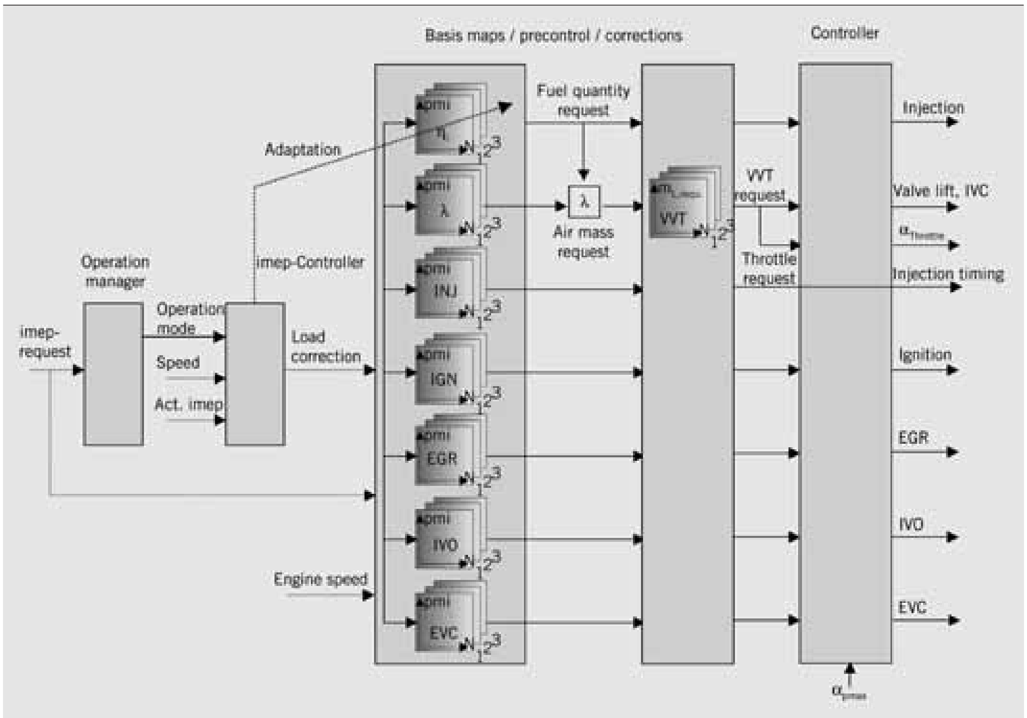
Typical air path models nowadays are based on the parameters of air mass sensors, intake manifold pressure and throttle sensors. Engine managements based on cylinder pressure [6] can use the data of combustion chamber pressure sensors to calculate the gas mass of each individual cylinder [7]. Low pressure indication at the intake and exhaust side as well as the respective temperatures measured at an engine test bench enables a gas exchange analysis in a physical model. This procedure, however, exerts high technical demands on the engine management system (EMS) hardware, especially for real-time calculations.

Besides purely physical models, there are map-based models as well as artificial neuronal networks (ANN), which mimics biological neuronal systems [8]. In this latter system, multiple parameters can be imaged in real time.

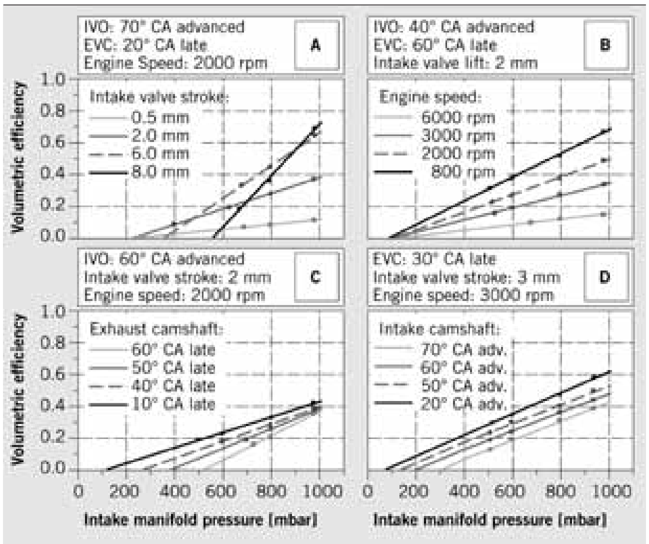
Whereas conventional map-based models are restricted to few parameters, the neuronal networks are advantageous, because they approximate the actual physical process with high precision and use multiple parameters. Besides static neuronal networks, there also online-learnable neuronal network models that can learn directly at the test bench [9].

2 CYLINDER PRESSURE BASED ENGINE CONTROL WITH PHYSICAL-NEURONAL CHARGING MODEL

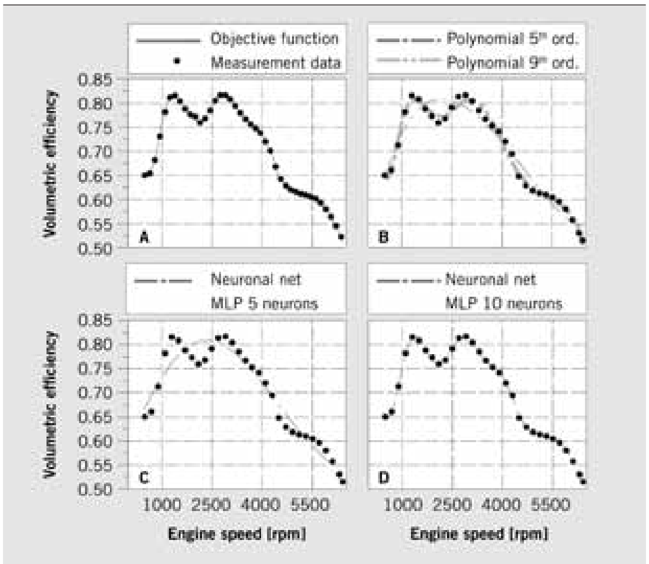
The indicated mean effective pressure (imep) represents the reference input parameter in the basic structure of the engine control model, realized on a rapid control prototyping (RCP) unit (Protronic: RCP system by AFT Atlas Fahrzeugtechnik GmbH) [10]. The operating mode is defined by the operating mode manager. The basic structure ❶ is designed to generate a precontrol consisting of base maps depending on engine speed and the indicated mean effective pressure with dynamic adjustments. The advantage of this structure is that it can be adjusted to the measured indicated mean effective pressure extracted from the cylinder pressure signal. The desired intake air mass from the precontrol dictates the throttle control and the desired valve stroke map. In unthrottled operating mode, the throttle is completely opened or is set to a desired manifold pressure. Depending on throttled or unthrottled operating mode, either the desired valve stroke map is addressed or the stroke is set to a fixed value and the load is thereby controlled by the throttle position.



❶ Overview of the precontrol implemented in the cylinder-pressure based engine control



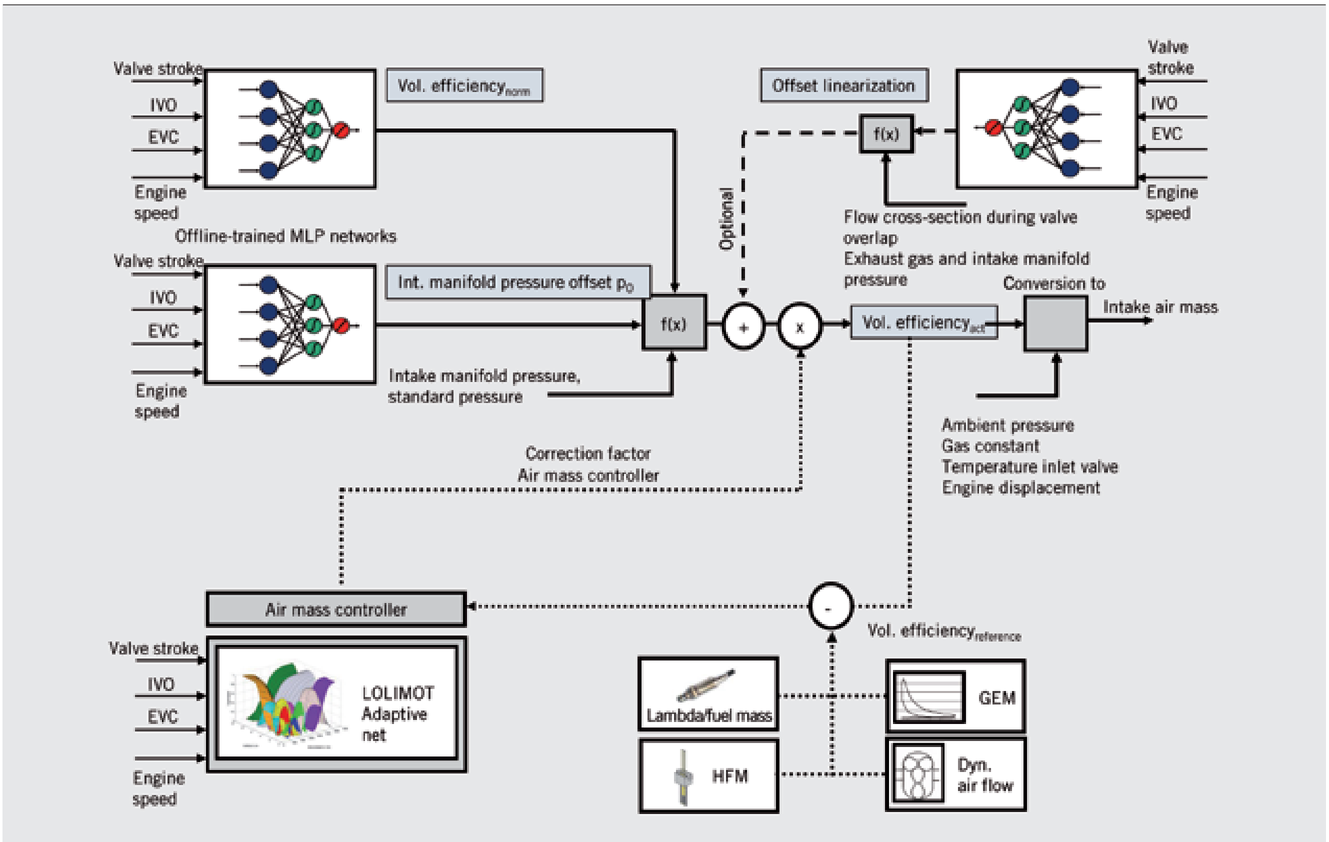
❷ Measured correlation between manifold pressure and volumetric efficiency (VVT engine)



❸ Comparison of different approximation functions

The air path model requires the initial identification of the input parameters that depend on the valve train variability resulting from the valve train configuration. The input values of an engine with intake and exhaust cam phasing and fully variable valve control (VVT) are: the valve stroke, the position of the intake and exhaust camshaft, the manifold pressure and the engine speed.

If the intake and exhaust camshaft timings are maintained constant at a specific engine speed, the relationship between the manifold pressure and volumetric efficiency (engine absorption



❹ Implemented physical-neuronal air path model behavior) on a naturally aspired engine is linear with respect to constant residual gas mass. ❷ (A) illustrates this linearity for an intake stroke variation at constant camshaft timing and engine speed; ② (B) shows an engine speed variation at constant camshaft timing and valve stroke; ② (C) displays a variation of intake camshaft timing at constant engine speed, constant exhaust timing and constant valve stroke. Finally, ② (D) presents a variation of exhaust camshaft timing at constant engine speed, constant intake timing and constant valve stroke.

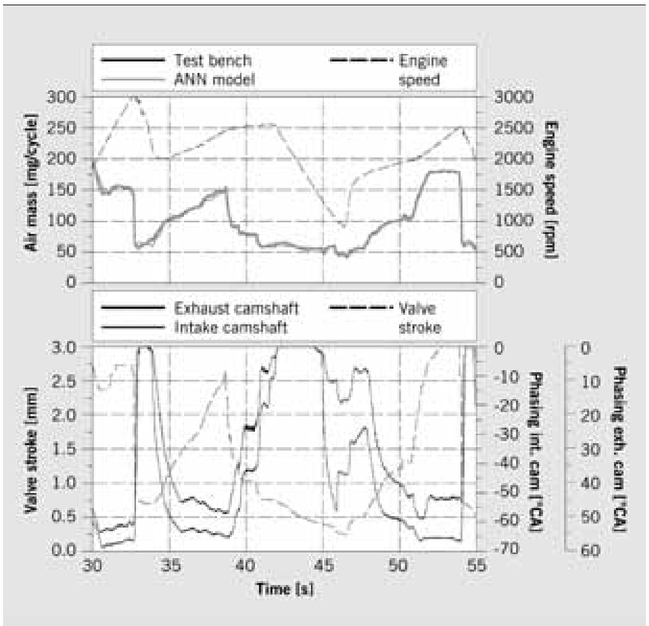
With this linear correlation between manifold pressure and volumetric efficiency, an air path model can be calibrated using the manifold pressure offset p_0 (intersection of the straight line and the x-axis) and the standardized volumetric efficiency $\lambda_{a,norm}$ (defined at 1013 mbar and 293.15 K), whereby both depend on camshaft timing, engine speed and valve stroke. This model is then valid for throttled as well as unthrottled operating mode. The respective coefficients (manifold pressure offset p_0 and standardized volumetric efficiency) are determined by varying the intake manifold pressure for operating points with all the other parameters fixed. Consequently, the engine absorption behavior can be displayed for all degrees of freedom in the air path.

The principal advantage of using neuronal networks is that they are better suited for modelling the multidimensional relationships in the air path for fully variable valve trains than conventional maps or polynomial models. Thus, the neuronal networks yield high accuracy in approximating a physical process on the basis of measuring data and aptly display non-linearities [9].

❸ shows the approximation behavior of the Multilayer Perceptron Neuronal Networks (MLP), which were applied for the modelling of the engine absorption behavior in this research project, compared to polynomial models. ③ (A) shows the highly non-linear relationship between volumetric efficiency and engine speed for a constant combination of valve stroke and camshaft positions derived empirically. ③ (B) illustrates that 5^{th} order polynomial model is unable to fit the measurement data. Even though increasing the polynomial to the ninth order leads to a better general fit, it still displays distinct differences to the target function. By contrast, a MLP neuronal network with ten neurons can achieve a very good fit, ③ (D). It is apparent that the number of neurons has a significant influence on the accuracy of the fit. ③ (C) showing a MLP neuronal network with five neurons in the hidden layer does not fit the target function. Besides the number of neurons, the number of measuring points also has a large influence on the model quality.

The characteristic coefficients "standardized volumetric efficiency $\lambda_{a,norm}$" and "intake manifold pressure offset p_0", which are derived from the linear relationship between manifold pressure and volumetric efficiency, are then stored in two MLP neuronal networks with four input values: valve stroke, intake and exhaust camshaft positions and engine speed for all measured operating points. The applied MLP neuronal networks possess one hidden layer with sigmoid transfer function and a linear transfer function in the output layer.

❹ shows the realized physical-neuronal air path model, which has been implemented in the presented cylinder pressure-based engine control model using the target hardware (AFT Protronic). According to Eq. 1, the actual volumetric efficiency $\lambda_{a,act}$ is calculated with the values "standardized volumetric efficiency" and "manifold pressure offset" subject to the actual intake manifold pressure (pint) and the defined standard pressure p_{stand} (1013 mbar).



❺ Excerpt of a FTP 75 cycle on test bench

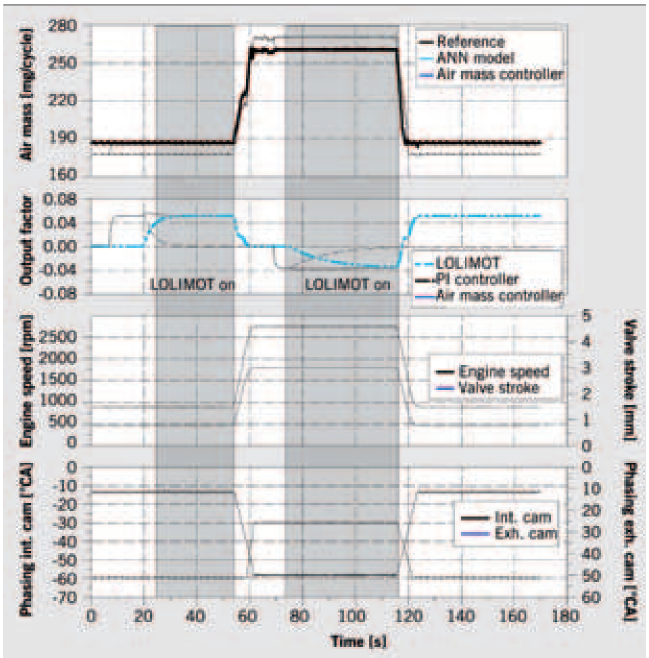
EQ. 1
$$\lambda_{a,act} = \frac{(p_{int} - p_0)}{(p_{stand} - p_0)} \cdot \lambda_{a,norm}$$

In certain cases the linear approach to describe the air charge is not precise enough, especially when the ratio of the integrated flow cross section during valve overlap compared to the integrated intake cross section is large. Therefore a linearization correction is integrated into the air path model. It uses the coefficients derived from the linear approach and saves an offset for the volumetric efficiency ($\Delta\lambda_a$) in a neuronal network. Depending on the ratio of exhaust pressure to intake pressure and the flow cross section during valve overlap a correction for the volumetric efficiency is calculated.

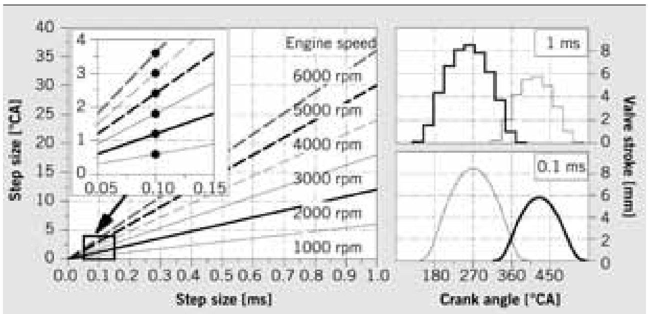
Additionally, an air mass controller, which corrects the air path model, is integrated into the air path model. The correction factor is the difference between the output air mass from the neuronal model and a reference value. This reference value can be obtained either from an air mass flow meter at the test bench, the signal from an hotfilm air mass meter (HFM), an air mass calculated from the measured fuel mass and the lambda sensor signal value or an air mass from a gas exchange model (see chapter 3).

④ shows the extension of the physical-neuronal air path model with the air mass controller. The air mass controller consists of a PI controller, which continuously corrects the model error. The controller is activated, when an operating point is detected as stationary. The integral term is adapted by a learning algorithm based on local linear models (LOLIMOT) [9, 11].

To access the accuracy of the physical-neural air path model a FTP 75 cycle was simulated at the test bench with the air mass controlled turned off. ❺ shows the high accuracy achieved with the developed model for different cam phasing positions, engine



❻ Learning performance of the air mass controller



❼ Demands on control unit hardware for real-time modelling

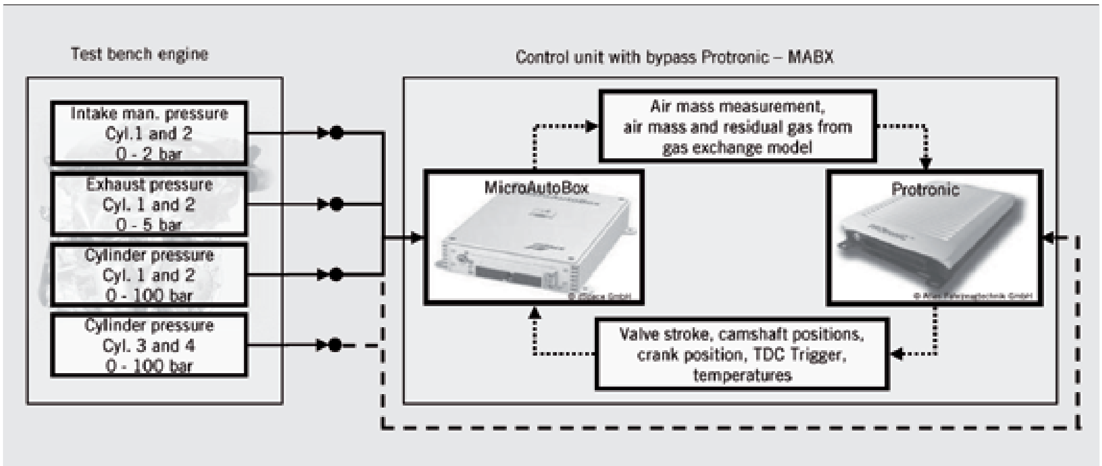
speeds and valve strokes. During dynamic operation a maximum deviation compared to the reference air mass of 7 % was achieved, while during stationary operation the maximum deviation was 4 %.

The learning performance of the air mass controller with the LOLIMOT net in combination with the PI controller is accessed by varying the operating point starting from a set point to an operating point with higher charge and back to the set point. Input parameters for the online adaptation net are: engine speed, intake cam phasing, exhaust cam phasing and valve stroke. ❻ illustrates that, when activating the air mass controller, the air mass corrected by the air mass controller succeeds the reference air mass. When the learning algorithm is turned on, the LOLIMOT net adapts the integral part of the controller, which then consequently turns to zero. Changing operating points back to the set point shows, that the LOLIMOT net returns the factor learned with the PI controller remaining zero.

3 REAL-TIME GAS EXCHANGE MODEL

Implementing a real-time gas exchange model poses high demands on the engine control unit hardware. To compute the algorithms the rapid prototyping control unit "MicroAutoBox" (MABX) by dSpace linked with AFT Protronic was used.

❼ depicts that a maximum step size of 6° CA which was defined at highest engine speed for real-time modelling, results in a sampling time below 0.16 ms. Moreover the picture shows valve stroke profiles defined by a valve stroke grid, independent of engine speed, for two different sampling times (0.1 ms and 1 ms). It is apparent that at a sampling time of 1ms, the valve lift profile is jagged and not a smooth curve. Furthermore the system becomes very sensitive to numerical vibrations. The real-time gas exchange model and the computation power of the "MABX" enable one to set the sampling time of the real time calculation to 0.1 ms. Ac-



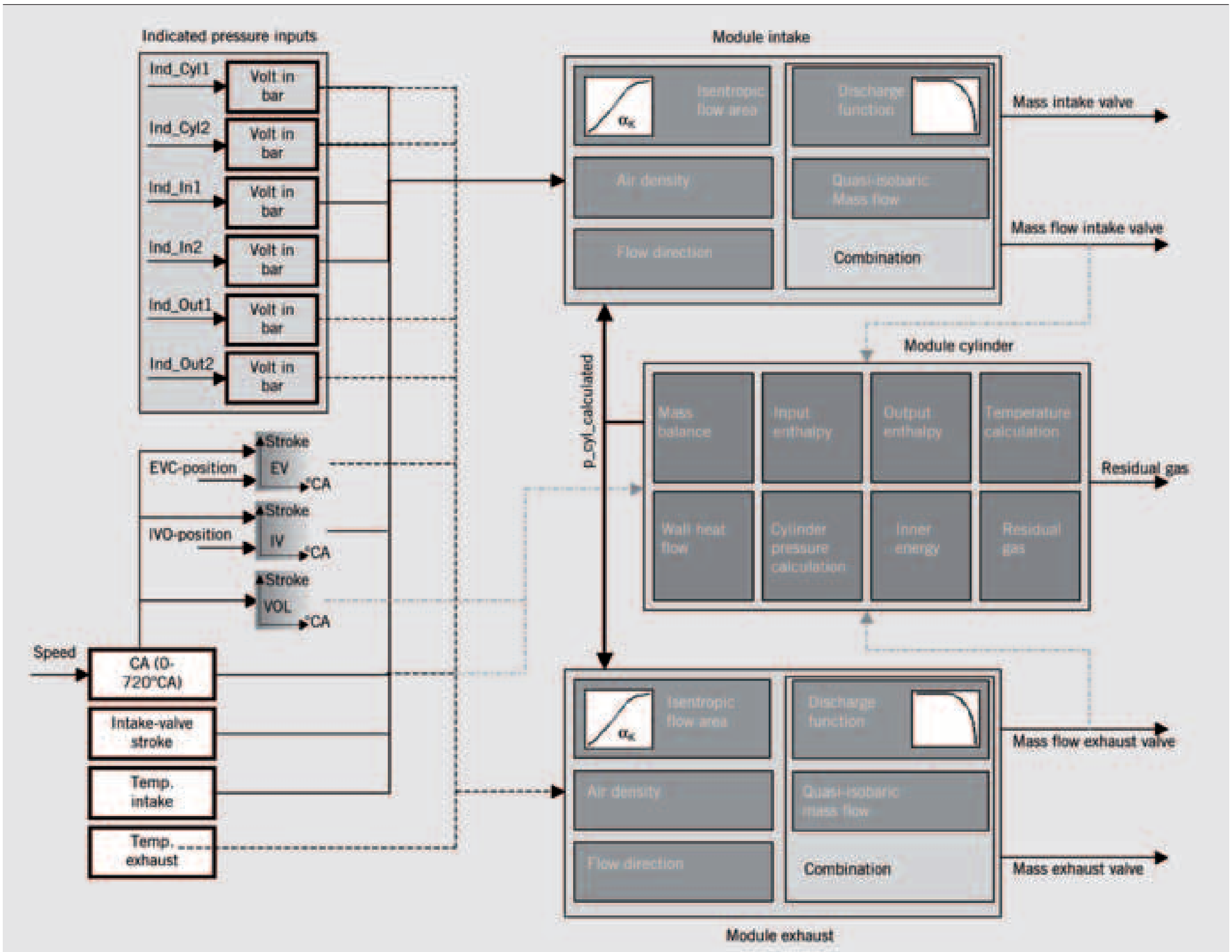
❽ Engine control unit hardware for real-time modelling cordingly, a maximum step size of 3.6° CA at an engine speed of 6000 rpm can be realized.

According to ❽, the four cylinder pressure indication signals from the VVT engine are read into the "Protronic" for imep-controlling. The pressure signals for cylinder 1 and 2 as well as the low pressure signals of the intake and exhaust ports are used as analogue inputs into the "MABX". It also shows the bypass function which enables communication between both control units.

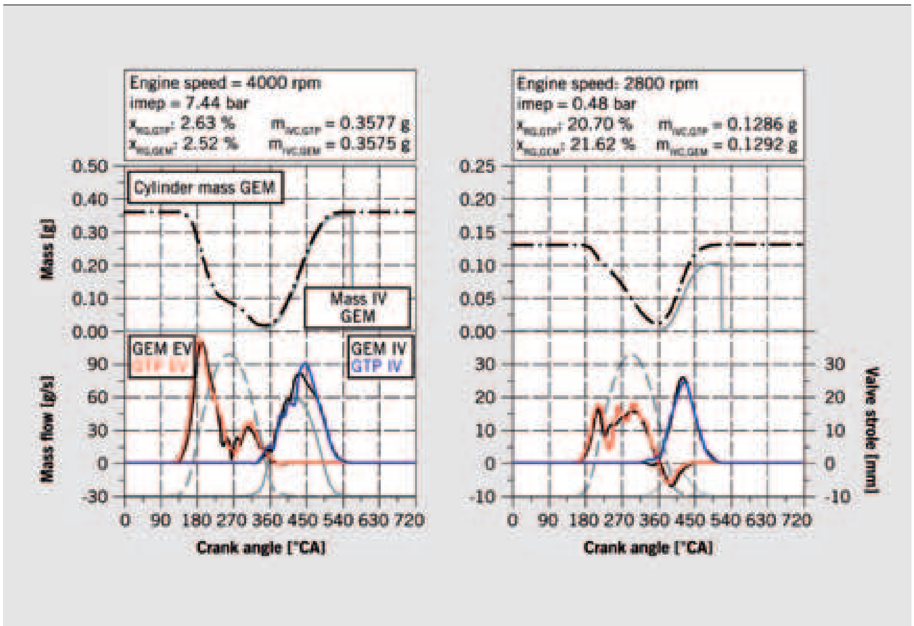
❾ presents an overview of the modules in the gas exchange model and their interactions. Input parameters of the model are the indicated pressures in the intake and exhaust ports as well as the cylinder pressure signal. Furthermore, the valve lift profiles for intake and exhaust have to be stored in a map for VVT engines. For an engine with an electromechanical valve train (EMVT), the valve lift profiles are provided as input signals. Additional values, which are obtained from the test bench or are sent as bypass signals by the engine control unit, are: engine speed, temperatures of the intake and exhaust system as well as valve timing (cam phasing with VVT, Inlet Valve Opening (IVO), Inlet Valve Closing (IVC), Exhaust Valve Opening (EVO) and Exhaust Valve Closing (EVC) with EMVT.

The inlet module contains the discharge coefficients of the intake system. The outlet module is analogous to this. Both modules include a discharge function as an array. Depending on the valve timing, the isentropic flow area is calculated using the valve lift profiles and the discharge coefficients. For each time step, the cylinder pressure in the gas exchange cycle is calculated based on the energy balance. During the high-pressure cycle, the measured cylinder pressure is used to calculate the cylinder temperature applying the ideal gas law. Main input signals for the cylinder module consist of the calculated mass flow rates of the intake and exhaust valves. In its turn, the computed cylinder pressure becomes an input parameter in the intake and exhaust modules. Output values of the gas exchange model are: mass flow rate, integrated mass flow for the valves, the cylinder mass flow curve as well as residual gas fractions. In a subsequent analysis of the model outputs, the parameters volumetric efficiency and charging efficiency are calculated.

❿ shows the values of the residual gas fraction and cylinder mass at IVC as well as the mass flow rates calculated by the real-time gas exchange model (GEM) compared to the results of the 1-D engine simulation tool GT Power (GTP). At operation point 1,



❾ Scheme real-time gas exchange analysis



⑩ Simulation results of the real-time model compared to GT Power

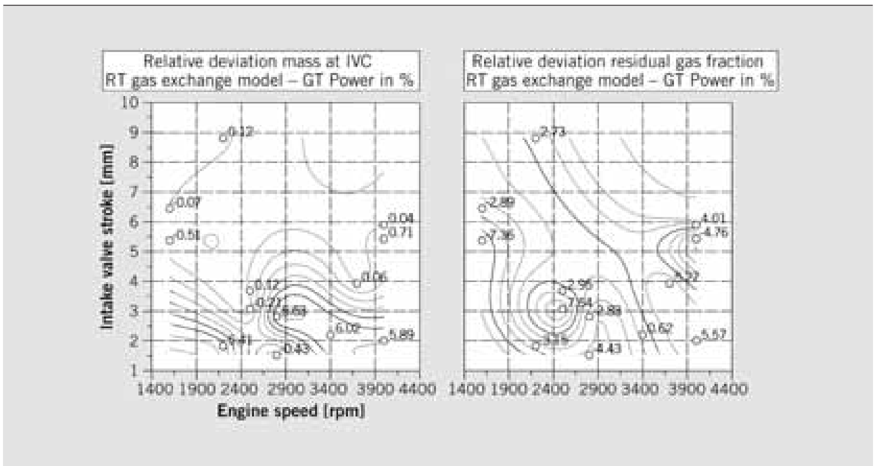
⑩ left, the engine speed is set to 4000 rpm and the valve lift to 5.9 mm. Intake camshaft timing is shifted 55° CA to early IVO, and the outlet camshaft timing was adjusted 20° CA to late EVC. Like to the reference values of the GT Power calculations, the real-time values show a very small divergence for residual gas fraction, cylinder mass and mass flow rate. With regard to the second operation point with 1.5 mm inlet valve lift and full valve overlap, ⑩ right, there is a slightly increased residual gas fraction, whereas the values for cylinder mass at IVC and the mass flow rate fit the reference values and curves well.

⑪ shows the cylinder mass and residual gas fraction as a function of engine speed and intake stroke for the analyzed operating points compared to the reference model. In all, using the real-time gas exchange model based on pressure indication signals produced a mean standard deviation of 3 % for the cylinder mass at IVC and the residual gas fraction for the analyzed operation points.

5 CONCLUSION

Variability in valve trains of gasoline internal combustion engines leads to increasingly complex engine management systems that need to consider multiple system parameters. At the Institute for Combustion Engines (VKA) at RWTH Aachen University, two novel engine control models have been investigated that account for the multiple parameters influencing the air path.

The first model is a physical-neuronal model which computes the cylinder mass for all kinds of different valve train variabilities. This work has shown that neuronal networks are ideal for this task, thereby minimizing the inherent errors of the system. In steady-state operation a maximum deviation of less than 4 % was achieved with the physical-neuronal model compared to reference measurements from the test bench. Additionally, a self-learning algorithm based on local linear neuronal networks, allows self-learning of the whole system during stationary operation. This self-learning algorithm is advantageous,



⑪ Cylinder mass and residual gas fraction as grid for real-time model and GT Power

ABBREVIATIONS

CA	Crank angle
EV	Exhaust valve
EVC	Exhaust valve closes
GEM	Real-time gas exchange model
IGN	Ignition timing
imep	Indicated mean effective pressure
Ind_Cylx	Indicated pressure cylinder x
INJ	Injection timing
IV	Intake valve
IVO	Inlet valve opens
RT	Real time
TDC	Top dead center
VOL	Cylinder volume
$\alpha_{Throttle}$	Throttle angle
α_{pmax}	Peak pressure position
η_i	Indicated efficiency

because is enables fast adjustment of the model to a reference air mass value.

The second model consists of a real-time algorithm capable of running on engine management system hardware, enabling the calculation of the gas exchange in real time. By using the three input parameters intake pressure, exhaust pressure and cylinder pressure, this model is excellent for calculating the charging efficiency and residual gas fractions in combustion engines with extended valve train variabilities.

This proposed real-time gas exchange model can be used to rapidly acquire measurement data and to accordingly calibrate the physical-neuronal model. This innovation will significantly speed up the calibration process of engine management systems, in particular, the air path.

The physical-neuronal air path model as well as the learning function was implemented into the engine management software of the FVV engine control unit.

REFERENCES

[1] Schopp, G.; Klingseis, B.; Angermaier, A.; Assmayr, H.; Schwarz, R.; Kärsch-Kleine, H.: Motorsteuerungsfunktionen zur Ansteuerung von Ottomotoren mit Ventilhubumschaltung. Haus der Technik Tagung „Variable Ventilsteuerung“ Essen, 2007
[2] Schopp Unger, H.; Schwarz, C.; Schneider, J. Koch, K.-F.: Die Valvetronic Erfahrungen aus sieben Jahren Großserie und Ausblick in die Zukunft. In: MTZ Motortechnische Zeitung 07/08, 2008
[3] Harada, J.; Yamada, T.; Watanabe, K.: The new L4 gasoline engines with Valvematic system. 29. Internationales Wiener Motorensymposium 2008
[4] Bernard, L.; Ferrari, A.; Micelli, D.; Perotto, A.; Rinolfi, R.; Vatteneo, F.: Elektrohydraulische Ventilsteuerung mit dem „MultiAir“-Verfahren. In: MTZ Motortechnische Zeitung 12, 2009
[5] Pischinger, S.; Hortmann, K.; Bücker, C.; Lang, O.; Salber, W.; Hahn, J.: Thermodynamical and Mechanical Approach Towards a Variable Valve Train For The Controlled Auto Ignition Combustion Process. SAE 2005-01-0762, 2005
[6] Gotter, A.; Pischinger, S.: Motorsteuerung mit Eingriffsmöglichkeit über ein Applikationssystem für stationär betriebene DI-Benzin-Motoren. Abschlussbericht FVV Vorhaben 843, 2006
[7] Jippa, K.-N.: Onlinefähige, thermodynamikbasierte Ansätze für die Auswertung von Zylinderdruckverläufen. Dissertation an der Universität Stuttgart, 2003
[8] Lichtenthäler, D.: Prozessmodelle mit integrierten neuronalen Netzen zur Echtzeitsimulation und Diagnose von Verbrennungsmotoren. Fortschrittsberichte VDI Reihe 12 Nr. 454, VDI Verlag Düsseldorf, 2001
[9] Nelles, O.: Nonlinear System Identification: Springer-Verlag, Berlin, 2001
[10] Sechtenbeck, C.: Neue Ideen schneller im Fahrzeug mit zielsystemnaher Motorsteuerungsplattform. 14. Aachener Kolloquium Fahrzeug- und Motorentechnik, 2005
[11] Kainz, J.; Beer, J.; Bänfer, O.; Nelles, O.: Lokale Modellnetze – Die zukünftige Modellierungsmethode für Steuergeräte? In: ATZelektronik 06/2008

THANKS

This report is the scientific result of a research project assigned by the Research Association for Combustion Engines (FVV, Frankfurt) and conducted at the Institute for Combustion Engines of RWTH Aachen University under the supervision of Professor Dr.-Ing. Stefan Pischinger. The project was supported by a work group of FVV under the supervision of Dr. P.W. Manz, Volkswagen AG, und K. Grieser, Ford Werke GmbH. We thank the work group and especially Dr. Manz and Mr. Grieser for their considerable help. We also thank the companies AFT (Atlas Fahrzeugtechnik GmbH) for their support during the project and BMW AG for providing the test engine.