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MTZ worldwide 3/2010, as epaper released on 11.02.2010
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ELASTIC MOUNTS with Higher Durability

CAMSHAFT with Roller Bearings

NO_x AFTERTREATMENT for Renault Diesel Engines

ENGINE-TRANSMISSION Interface

LUBRICANT EFFECT on Mixed Friction of Slide Bearings

/// INTERVIEW

Burkhard Göschel
Magna

MORE EFFICIENCY FOR THE COMBUSTION ENGINE

SMALLER AND MORE VARIABLE

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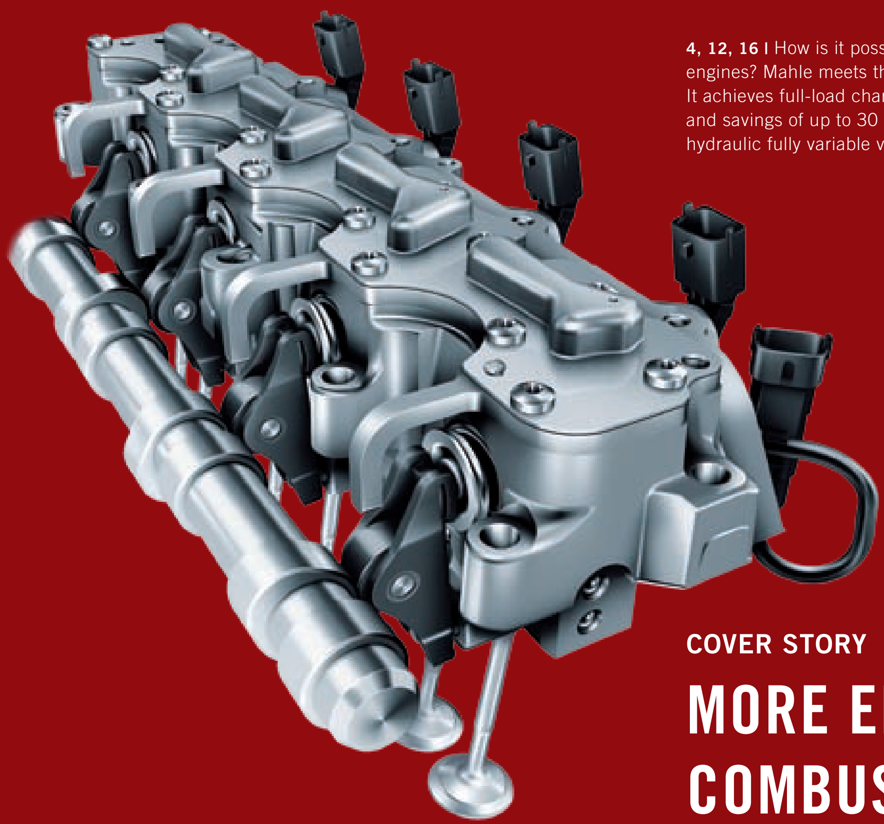

4, 12, 16 I How is it possible to further explore the potential of Downsizing in combustion engines? Mahle meets this challenge with an 1.2-l gasoline engine with three cylinders. It achieves full-load characteristics comparable to an engine with a displacement of 2.4 l and savings of up to 30 %. Schaeffler introduces the series development of the electro-hydraulic fully variable valve train system “UniAir”.

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TECHNOLOGY-NEUTRAL?

Dear Reader,

At our CO_2 conference, Rainer Bomba, State Secretary at the Federal Ministry of Transport, Building and Urban Development, reported on the German government's initiatives to support electromobility. The lively discussion that followed focused on the issue of the extent to which the government should put its weight behind one specific technology.

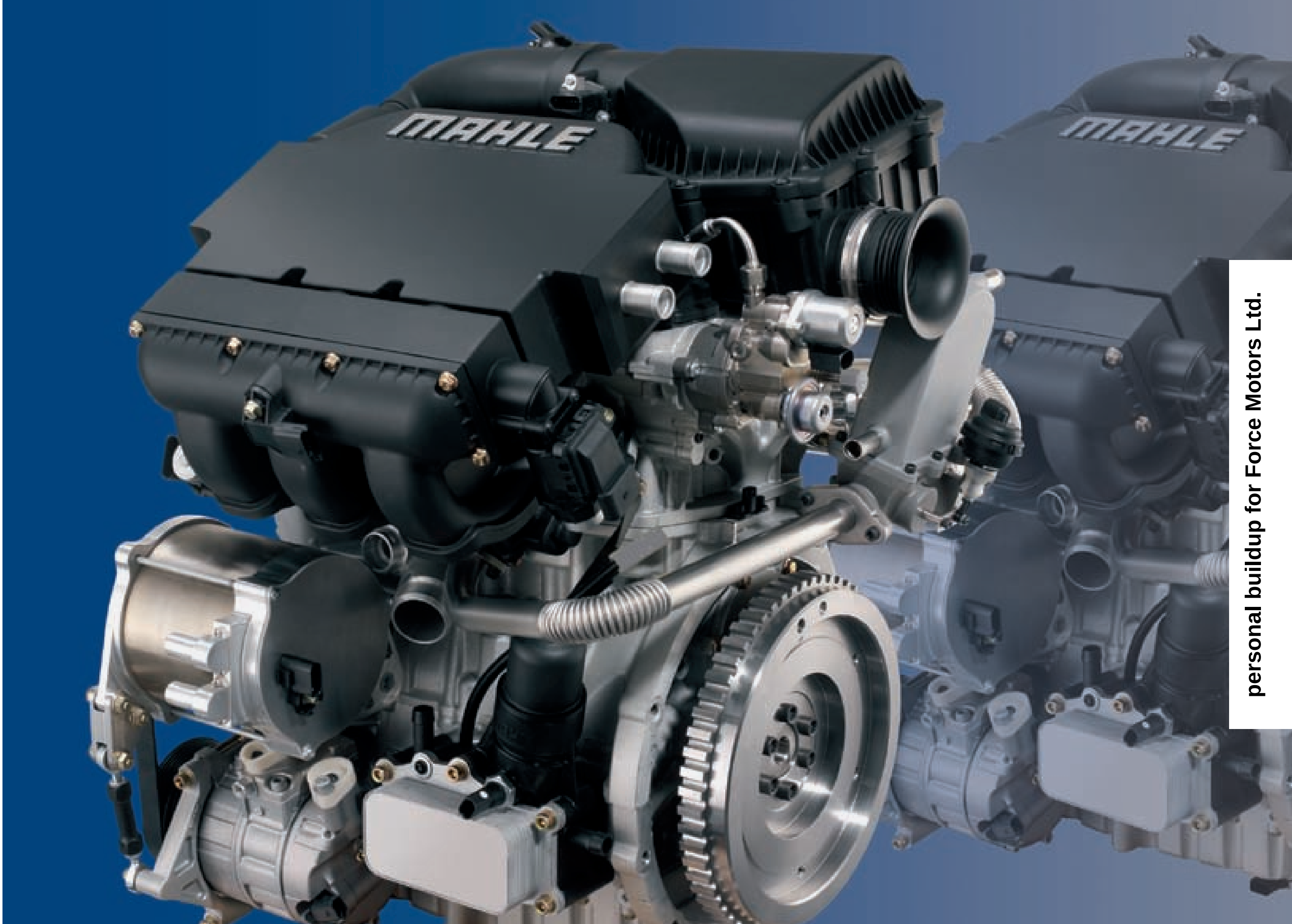
Perhaps I am old-fashioned but I believe that the government has the responsibility to set the right conditions for economic activity. For example, it is obliged to impose fair CO_2 limit values that apply to everyone in order to protect the climate. But it should be left to companies to work out how these limits can be complied with from a technical perspective. The risk of diverting development in the wrong direction by providing misguided support is far too great, also in the field of research. The best ideas will prevail in the end anyway: namely those that are most beneficial to people within the framework of existing legislation.

Bomba said that the move towards electromobility was the key issue of this legislative period. And I am quite certain that he is wrong. I have nothing against electric cars as inner-city vehicles, but it will simply not be feasible to provide individual mobility over great distances with electric vehicles, even if we spend more than the 700 million euros of taxpayers' money currently earmarked for their development. And before we invest billions in establishing a hydrogen infrastructure, we should work at least as intensively on further developing second-generation biofuels.

The government has enough homework to do even if it doesn't see itself as a – potentially vote-winning – coordinator for electromobility. For example, how can we deal with the radioactive legacy of 50 years of heavily supported nuclear energy? Or how can we reach internationally binding agreements on CO_2?

In the end, everyone does what they do best: politicians make laws, researchers generate new ideas and companies develop new products. And we are always there to report and to comment.

JOHANNES WINTERHAGEN, Editor-in-Chief
Munich, 26 January 2010



30 % HIGHER EFFICIENCY WITH 50 % LESS DISPLACEMENT

How can one discover the full potentials of downsizing internal combustion engines? Mahle addressed this issue with a technology demonstrator, a 1.2-l three-cylinder spark-ignition engine, which the company presented to the public three years ago. However, the demonstrator engine was still somewhat young at the time and was unable to fulfil the ambitious targets. But now it can. In this article, Mahle describes how the three-cylinder engine has better full-load characteristics than a comparable engine with a displacement of 2.4 l. It achieves a reduction in fuel consumption and CO_2 emission of more than 30 % in the New European Driving Cycle (NEDC), while maintaining the same power and performance characteristics.

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Maximising fuel economy and thereby minimising CO_2 emissions are among the most important objectives for the future development of engines and vehicles. Various technologies have been developed for gasoline engines with the aim of achieving a significant reduction in fuel consumption. Downsizing, as the combination of reduced displacement and pressure charging is one of the most promising options and is also relatively straight-forward to implement in the short term. Customer acceptance of such a concept relies on achieving good transient response and excellent torque at low engine speeds. Exploring the potential of downsizing at levels in the order of 50 % was the primary motivation for this study. The engine was first displayed at the IAA 2007 and additional information published elsewhere, including [1, 2, 3]. During the following years Mahle has optimized the demonstrator motor and describes now in detail the "proof of concept".

OBJECTIVES

The engine was designed to operate at the extremes of thermodynamic and mechanical performance. To achieve this, careful attention was paid during the design and analysis phase to the following areas

: cylinder head, combustion chamber and GDI injector packaging
: air path and charging system design
: mechanical strength maximisation and friction reduction
: minimising engine weight within the above constraints.

The engine was designed as a three-cylinder in-line configuration with a displacement of V_H = 1.2-l and two-stage turbocharging. The objectives for WOT performance and fuel consumption are summarised in ❶; the engine with the two-stage concept is shown in ❷.

The torque and power targets are aimed at a vehicle application with a kerb weight of m ≈ 1,600 kg. The engine should achieve comparable driving performance to a naturally aspirated engine with a displacement of V_H = 2.4-l, thus demonstrating a 50 % engine downsizing. The engine is designed for $\lambda = 1.0$ operation over the whole engine map and to be capable of achieving EU5 emissions standards.

PERFORMANCE AT FULL LOAD		
Single turbo:		
Torque		
Maximum at n = 2,500 - 3,000 rpm	M_d = 240 Nm	(p_{me} = 25 bar)
Power Output		
Maximum at n = 6,000 rpm	P_{max} = 108 kW	(P_{max}/V_H = 90 kW/l)
Twin turbo:		
Torque		
at n = 1,000 rpm	M_d = 153 Nm	(p_{me} = 16 bar)
Maximum at n = 2,500 - 3,000 rpm	M_d = 286 Nm	(p_{me} = 30 bar)
Power Output		
Maximum at n = 6,500 rpm	P_{max} = 144 kW	(P_{max}/V_H = 120 kW/l)
FUEL CONSUMPTION AT PART LOAD		
Optimum	b_e < 235 g/kWh	
at n = 2000 rpm / p_{me} = 4 bar	b_e < 300 g/kWh	

❶ 1.2-l downsizing engine: performance and fuel consumption objectives

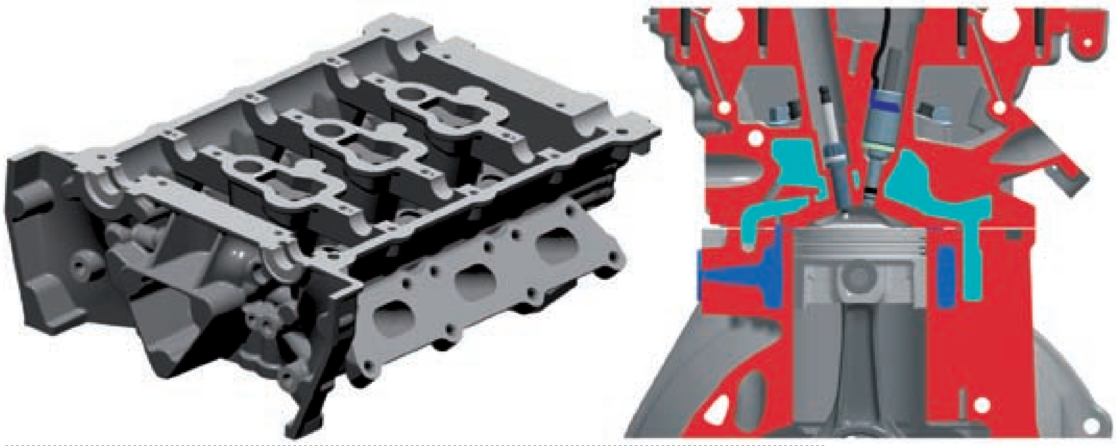
❷ 1.2-l downsizing engine with two-stage charging concept

MAIN DATA, CYLINDER BLOCK AND CRANK TRAIN		
Cylinderblock (material)		A 356
Bore		83.0 mm
Stroke		73.9 mm
Stroke-Bore ratio		0.89
Cylinder displacement		0.400 l
Cylinder number and arrangement		3 in line
Displacement		1.200 l
Bore spacing		91 mm
Conrod length		123 mm
Block height		189.5 mm
Compression ratio		9.75
Firing order		1 – 3 – 2
Crankshaft main journal diameter		48 mm
Crankshaft pin journal diameter		48 mm
CYLINDER HEAD AND VALVE TRAIN		
Cylinder head (material)		A 356
4 valves/cylinder (pentroof) Central GDI (sprayguided) with Piezo injector DOHC architecture with roller finger followers and dual independant cam phasing		
Valve head diameter	Intake / Exhaust	31.4 / 25.5 mm
Valve stem diameter	Intake / Exhaust	6 / 6 mm
Maximum valve lift	Intake / Exhaust	11 / 11 mm
Valve angle	Intake / Exhaust	21.5 / 20 °
OVERALL DIMENSIONS		
Length x Width x Height		438 x 675 x 706 mm
Dry weight (fully dressed)		145 kg

❸ 1.2-l downsizing engine: main design data

CYLINDER HEAD AND COMBUSTION CHAMBER

The main engine design parameters are shown in ❸ and some key features of the design are summarised as follows.The cylinder head is designed with a separate ladder frame for the camshaft bearing, ❹; [1, 2].The combustion chamber is designed around a bore size of 83 mm. High tumble inlet ports are used to generate the charge motion necessary for high BMEP operation. The 4 valve/cylinder layout is arranged with a centrally mounted direct injector for spray-guided operation. The injector, spark plug and valves are positioned to achieve excellent cooling around the injector tip whilst maintaining a compact chamber with low surface area. A multi layer steel gasket with integrated temperature sensor is used. This provides a rapid indication of changes in cylinder head metal temperature and allows for more precise cooling control.



❹ Cylinder head (without ladder frame) and combustion chamber

AIR PATH

The air filter, mass air flow meter and charge cooler have been designed to be engine mounted and demonstrate a high degree of integration [1,2]. Apart from the aluminium charge cooler, all components including the inlet manifold are lightweight plastic, designed to withstand the expected boost pressure levels of more than 2.8 bar. Cooled exhaust gas recirculation (EGR) has previously been demonstrated as a promising technology for reducing the fuel consumption of turbocharged engines at high loads. This is achieved by reducing the need for overfuelling to control exhaust temperature and extending the range of stoichiometric operation. The EGR system on this engine was designed to achieve up to 15 % EGR flow at the highest power conditions. The EGR is introduced to the inlet runners, fed through a barrel type valve. Separate entry points are used for each cylinder for improve transient response. The control valve is also integrated into the intake manifold assembly.

CYLINDER BLOCK AND CRANK CASE

The cylinder block, ❺, uses a closed deck design in aluminium with a separate bedplate for high stiffness. Cylinder bores are parent metal with NIKASIL coating, chosen to achieve the best possible heat transfer. A through-bolting system is used to assemble the head, cylinder block and bedplate which keeps the block in compression, minimising bore distortion and reducing component weights.

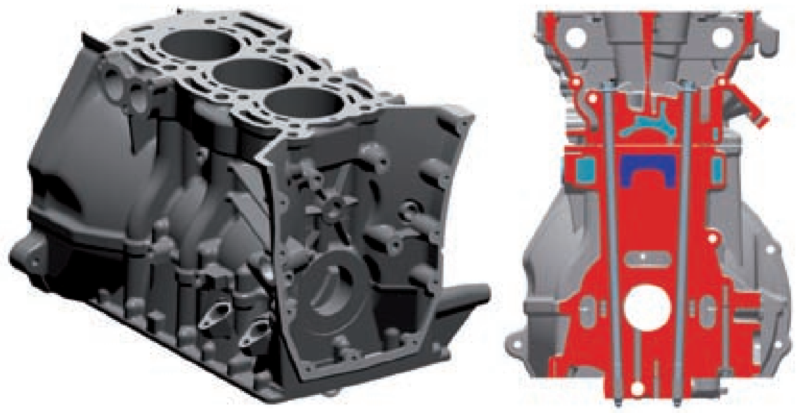
The piston cooling jets are fed via a separate, non-filtered oil supply directly from the oil pump control gallery [1,2]. The bedplate design features integrated baffles avoiding the use of cast-in steel inserts.

CRANK TRAIN, VALVE TRAIN AND BALANCER SYSTEM

The primary aims of the crank train design were to accommodate the expected peak cylinder pressures (140 bar) whilst maintaining low friction and low wear. The crank train features include

- forged aluminium pistons
- a ring pack with two compression rings (one barrel faced ring, one Napier ring) and a 3-piece oil-scraper ring
- DLC coated piston pins
- forged steel conrods
- a steel crankshaft.

❺ 1.2-l downsizing engine: cylinder block and bolting with cylinder head



The DOHC valve train was designed for low overall weight and low reciprocating masses as well as low wear and low friction. The tubular assembled camshafts actuate light weight valves via roller finger followers. The cylinder head and valve train are designed to accommodate a switchable follower design for variable valve lift. The engine balancer system comprises two crank-driven plastic gears carrying the balance weights, designed to reduce weight, package and friction compared to a conventional balancer shaft.

THERMODYNAMICS

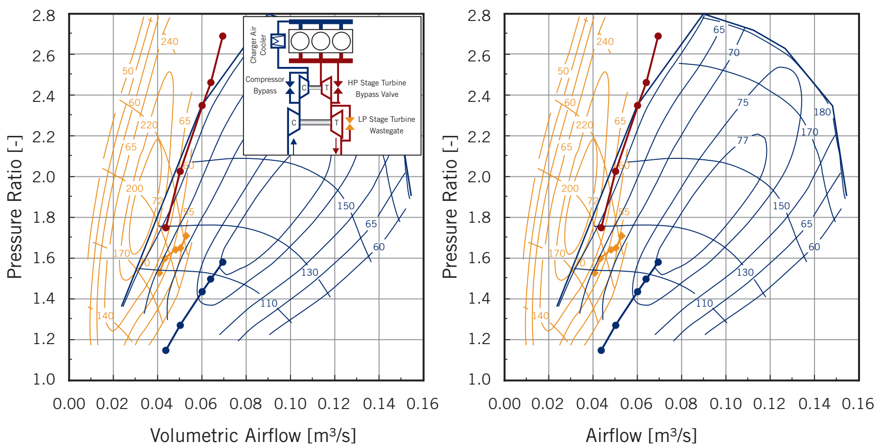
The thermodynamic development testing of the 1.2 downsizing engine has included investigation of the following areas

- optimisation of compression ratio and combustion chamber
- optimisation of ignition timing and cam phasing
- combustion system development
- turbocharger configuration.

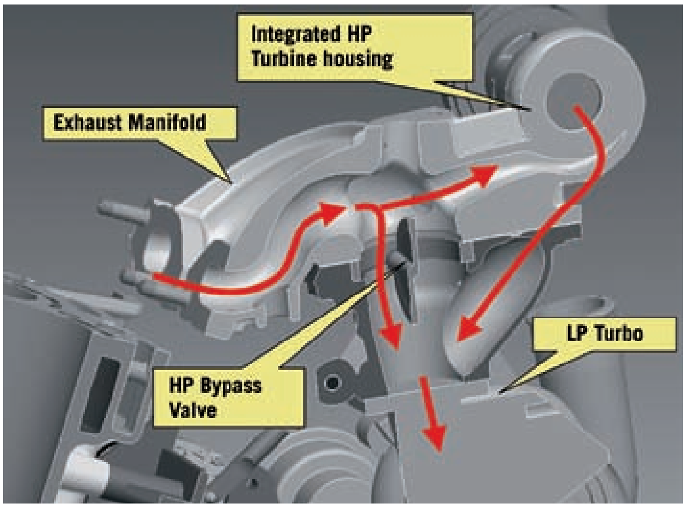
All investigations were carried out with standard gasoline (98 RON). Initially, compression ratio variants ε = 9.25 and 9.75 were investigated and after the first results ε = 9.75 was used for the further development. Initial testing was carried out using piezo injectors in a spray-guided concept, and the optimisations of the usual operating parameters were completed, including

- fuel injection timing
- fuel pressure (at part load between 100 and 200 bar)
- fuel injection strategy (single or double injection per cycle).

Subsequent hardware designs have included multi-hole solenoid injectors



❻ TC01/TC04 compressor maps with characteristics for approx. 20 to 30 bar BMEP at 1,500 rpm (left) and 2,500 rpm (right)



❼ 1.2-l downsizing engine: exhaust manifold with high-pressure compressor bypass valve

and these results will be reported at a later date. The objective for the engine was to operate at $\lambda = 1$ at all speeds and loads and this could be achieved, except around peak power, without the use of EGR [2]. The maximum exhaust gas temperature has been limited in all cases to 1,020 °C. The layout of the turbocharger system is of particular importance due to the ambitious objectives for power and torque, specifically when considering the torque target at n = 1,000/min.

From the outset, the engine was designed with a two-stage turbocharging system. This utilises a small high-pressure turbocharger in series with a larger low-pressure turbocharger. A bypass for the high pressure turbine is provided to allow the work split between the two turbochargers to be varied and for the high pressure unit to be completely bypassed at high power if required [1,2]. Initial investigations with single turbochargers have been carried out to define suitable turbine/compressor configurations. These single turbochargers are characterised as follows:

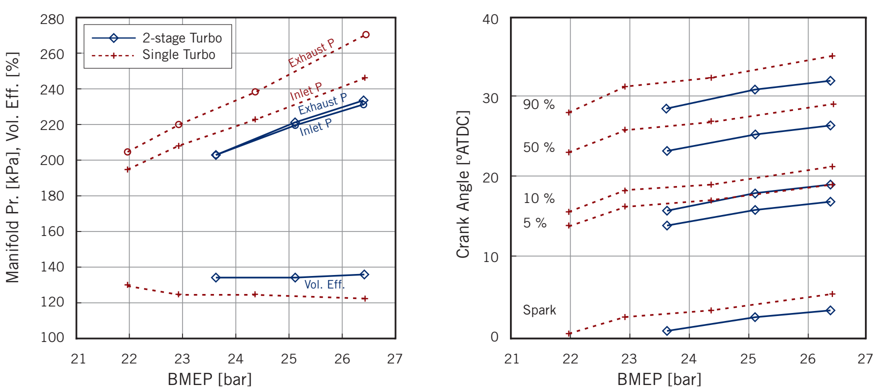
TC01 max. flow 0.06 m^3/s
TC02 max. flow 0.09 m^3/s
TC03 max. flow 0.14 m^3/s
TC04 max. flow 0.16 m^3/s.

After much testing, the TC01/TC04 configuration was selected as the best solution for the two-stage turbocharger system.

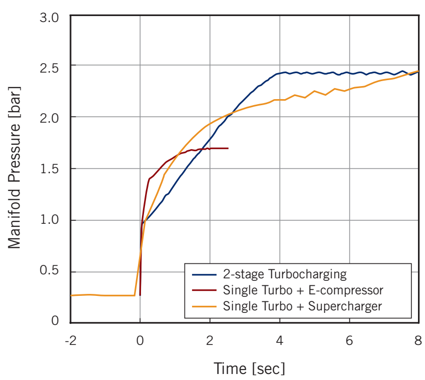
❻ shows the compressor maps for the TC01 and TC04 plotted together. In each case the airflow and pressure ratio for each turbocharger is plotted, as well as an "overall line" which is not related to either compressor map, but included for information. The data at 1,500 rpm (left) illustrates the benefit of the two-stage boost system. Although the TC01 is capable of achieving sufficient boost for target BMEP at this engine speed (pressure ratio of 2.4 for 26 bar BMEP), the required turbocharger speed is very close to the maximum allowed. Substantially reduced turbocharger speed combined with reduced overall backpressure is what makes the two-stage design such an attractive solution.

At 2,500/min (right) the turbochargers combine to achieve a pressure ratio of approx. 2.7 (for 30 bar BMEP). The high pressure compressor is operating very close to the choke line and this is a consequence of running without a larger high pressure bypass for these particular tests. If the turbine work-split could be further biased toward the low-pressure stage, the high-pressure inlet pressure could be raised and this would drive down the high-pressure inlet volumetric flow. Further results with the finalised two-stage design will be available in the near future.

❼ shows the exhaust manifold of the 1.2 l downsizing engine with the integrated high-pressure turbine housing and bypass valve. The design has been refined using thermo-mechanical fatigue analyses, as the system must remain durable in a very challenging environment. ❽ shows BMEP sweeps at 1,500 rpm, comparing single and two-stage boosting. Manifold pressures and volumetric efficiency (left) are shown, as well as combustion phasing information (right). The two-stage boost system provides a useful improvement in apparent volumetric efficiency and the improved combustion phasing at a given BMEP is driven by the reduced pre-turbine pressures.



❽ Pressure in intake and exhaust manifold, volumetric efficiency and combustion phasing with single and two-stage boosting

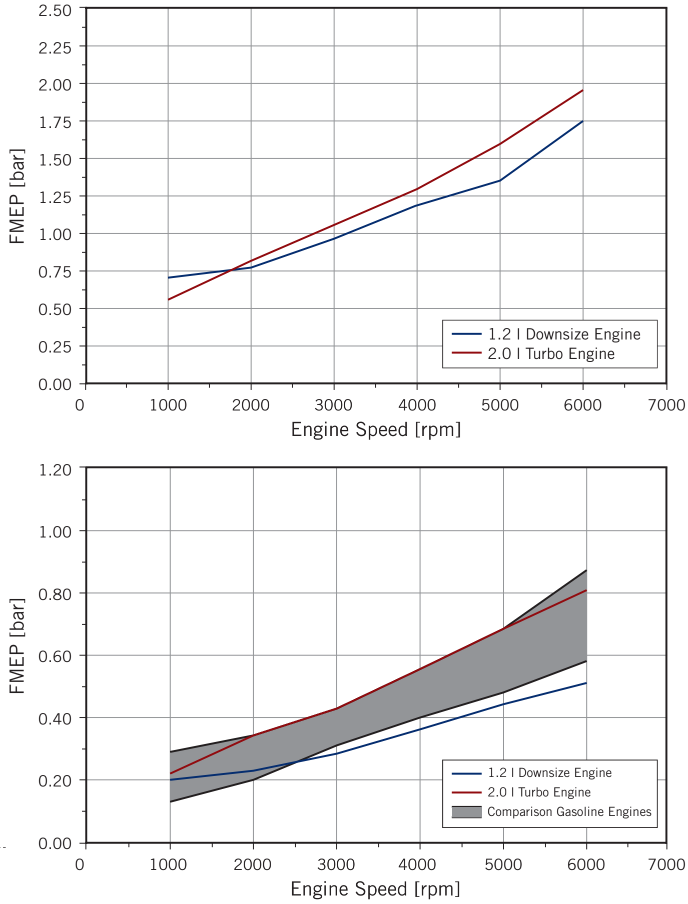


❾ Pressure increase for different boosting concepts for a load step (from 2 bar BMEP, n = 1,250 rpm)

Combustion system development is particularly challenging at low speed and high boost levels. Auto-ignition and misfire must be avoided, as well knock. Large pulsations in the exhaust manifold are driven by late combustion phasing and combine with large valve overlaps to drive unusual gas exchange phenomena at some conditions. Even though considerable cylinder scavenging flows exist, it is clear that some residuals remain and these can have a strong effect on combustion. Turbocharger selection is quite important here, and exhaust cam duration has also been shown to have a significant effect [2].

As mentioned previously, transient behaviour is key for the acceptance of downsizing concepts by the customer. A load step at fixed engine speed is a simple way to assess the transient performance of the boosting system. ❾ shows the results of a load step from 2 bar BMEP to WOT at 1,250 rpm. The performance of the 1.2-l engine with two-stage system shows good potential compared to a single turbo plus mechanical supercharger concept. The best transient performance for our engine was achieved using a single turbo (low pressure TC04) together with an electrical pre-compressor (e-booster).

The development of the 1.2-l downsizing engine is continuing with further optimisation of the two-stage system design and the development of a new single turbocharger concept. This uses our experience from the two-stage testing and will provide very good performance in a more cost effective solution.



❿ Overall and power-cell FMEP

TORQUE AND POWER OUTPUT AT WOT	TARGETS	ACHIEVED
Torque at n = 1,000 rpm	M_d = 153 Nm (BMEP = 16 bar)	153 Nm 16 bar
Max. Torque (n = 3,000 rpm)	M_d = 286 Nm (BMEP = 30 bar)	287 Nm 30 bar
Max. Power (n = 6,000 rpm)	P_e = 144 kW (P_e/V_H = 120 kW/l)	144 kW

⓫ WOT characteristic

ENGINE MECHANICS AND FRICTION

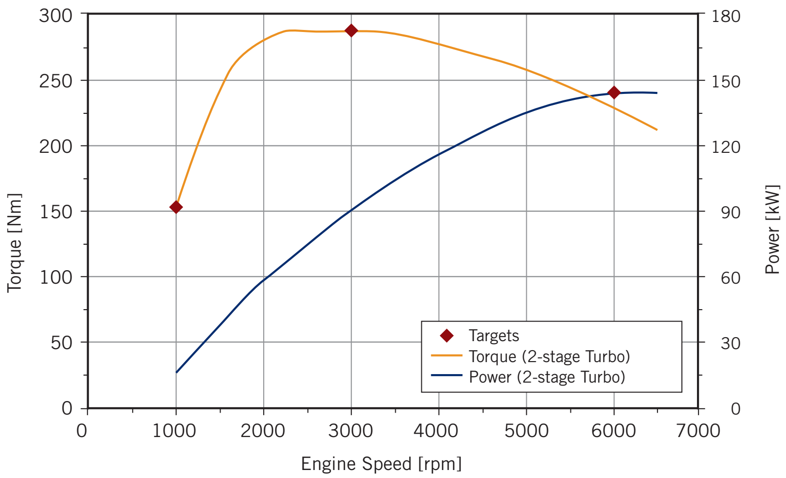
The mechanical components for the 1.2-l downsizing engine were designed to withstand the high mechanical loadings of this concept, but with the additional objective of low engine friction. ❿ compares the complete FMEP of this engine with the FMEP of a turbocharged 2.0 l engine with similar maximum power output. It can be seen that the 1.2-l downsizing engine has a slightly higher FMEP between n = 1,000 and 2,000 rpm but lower FMEP for n > 2,000 rpm. Higher valve-train losses (exhaust spring loads in particular) are the main cause of the small disadvantage at low speed. The good result at medium and high engine speed is due to the very low power-cell friction of the 1.2-l downsizing engine, including the effects of three versus four cylinders, latest technology crankshaft bearings, short stroke design and the very good bore roundness which enables low piston ring tensions. Compared to a turbocharged 2.0-l engine (four-cylinder) with similar maximum power output, the power-cell FMEP is up to 40 % lower. A novel 'Pendulum-slider' oil pump has also been tested on the engine and the effect on parasitic losses investigated [3]. The results show a fuel economy improvement potential of up to 2 %.

RESULTS AND OUTLOOK

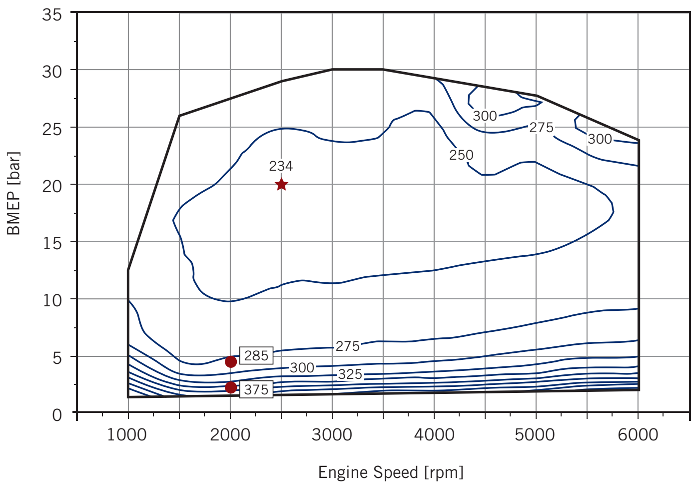
⓫ shows the WOT performance of the 1.2-l downsizing engine. All target values from ① have been achieved, ⓬.The engine brake specific fuel consumption is shown in ⓭. Slightly better values than target have been achieved. This engine map for the brake specific fuel consumption was used for a simulation of the vehicle fuel economy in the NEDC (kerb weight m ≈ 1,600 kg). This results in an improvement of fuel consumption and CO_2 emission of up to 30 %, ⓮.

The development status of the 1.2-l downsizing engine, regarding WOT per-

⓬ All target values



⓭ Brake specific fuel consumption



BRAKE SPECIFIC FUEL CONSUMPTION	TARGETS	ACHIEVED
Optimum	b_e < 235 g/kWh	234 g/kWh
n = 2000 rpm, 4 bar BMEP	b_e < 300 g/kWh	285 g/kWh

⓮ Improvement of fuel consumption and CO_2 emission

formance and fuel economy, demonstrates excellent results for an advanced downsizing concept and provides a good basis for future developments.

These will include the development of a single turbocharger concept and installation of the engine in a vehicle. Pleasingly, the results also show a good conformity with initial predictions and analysis work [1,2], giving us good confidence as we design similar high BMEP engines for other customer applications.

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

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03 I 2010 _ March 2010 _ Volume 71
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Price List No. 53 (10/2009)

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"ALWAYS KEEP THE COMPLETE SYSTEM IN MIND"

The hype surrounding electromobility is leading to technical misjudgements, warns Professor Dr. Burkhard Göschel. In this interview with MTZ, the Chief Technical Officer with responsibility for Development at Magna International discusses important strategic decisions – with a view to sensible improvements in efficiency across the entire portfolio of vehicle powertrains. He is strongly critical of the one-sided research and funding offensive in favour of lithium-ion batteries, which he believes is excluding the alternatives.

Professor Dr.-Ing. Burkhard Göschel has been Chief Technical Officer (CTO) at the Canadian automotive supplier Magna International since September 2007 and has been a Member of the Board of the Swiss capacitor manufacturer Maxwell Technologies since February 2007. From 2000 until reaching the company's age limit in 2006, Göschel, who has a doctorate in mechanical engineering, was Group Board Member Development and Purchasing at BMW AG. Except for the years 1976 and 1977, in which Göschel worked as an engineer at Daimler-Benz AG, the honorary doctor of the TU Munich and honorary professor at the TU Graz can look back on 28 years of continuous service at BMW. The career of the now 65-year-old engineer includes engine development, a period as head of motorcycle development, management of the "Roadster" project and head of "Special Models" and finally head of Complete Vehicle Development.

MTZ _ The developers of internal combustion engines are on the offensive and appear to be stemming the current electromobility tide as champions in the reduction of CO_2. Do you now see a war of attrition developing?

GÖSCHEL _ I am concerned about a perspective that is in some cases limited and insufficiently differentiated – especially because this separation does not exist in the vehicle itself, in particular in the hybrid drive system. Engineers would be well advised to keep the complete system in mind at all times. It may well be the case that one approach is more efficient on the electric side than on the side of the internal combustion engine. The degree of electrification and the characteristics of the systems that can be experienced by the customer are among the few possibilities for differentiation that remain in future powertrains, and especially in conventional engine engineering.

Is the last bastion of the classical engine builder about to fall?

Change has already been initiated with downsizing and in the next few years this will become interesting, dramatic and sometimes painful. Many OEMs will have to apply their engine development and manufacturing activities consistently across different brands, and operating their own transmission production facilities will be out of the question. The internal combustion engine will lose its still strong role as a differentiating feature. It will, if you like, in a figurative sense take on the emotionality of a lithium-ion battery. Standard engines will tend to be three- and four-cylinder in-line engines – with turbochargers, direct injection, fully variable valve timing and balancer shafts. Differentiation will take place in the applications of the basic engines, right through to the design of the complete system, such as hybrid drive versions, which will become established on a wide scale in the future. In the medium term, 1.6 l and 2.0 l engines will fulfil the requirement profiles in premium and luxury vehicles.

Where do you see further potentials for the internal combustion engine?

There is still potential in the combination of valve timing and fuel injection technology with different supercharging processes and new fuels – especially for the petrol engine.

Will natural gas have one last chance?

CNG and ultimately the natural gas hybrid are an excellent means of minimising emissions and CO_2 as efficiently as possible. In the USA, one can see an increased awareness of new fuels, especially CNG, and this is driven by two aspects: the country has very high natural gas reserves and CNG has the image of a clean fuel – in contrast to diesel. An extensive network of filling stations can be set up more quickly than we are experiencing in Germany.

"The combination of natural gas and a hybrid drive system is an excellent solution."

In your view, the transformation in powertrain technology seems to be home-made – not so much driven by competition from the electric car.

I don't know if you can call it home-made. The relationship to the car is in a state of upheaval. It is becoming more rational. At the same time, the development is characterised by an increase in energy efficiency, from the electrification of the powertrain through hybrid drive systems right through to purely electric vehicle. All technical variants will shortly exist side-by-side. A careful consideration of customer benefit and other important issues may help to reduce the number of variants. For example, what will be the customer's experience without the attraction of electric acceleration from a standstill, slow-speed driving or at least parking? Does it make sense to offer a choice between an efficiency function and a boost function or even the option of an electric axle as a traction aid? What is the value of an electric vehicle that doesn't have a reasonable operating range, and one that is even significantly reduced when it is cold outside? Wouldn't it be better to use a range extender system in our climate zone?

The hype that currently surrounds the issue of electromobility may result in an overevaluation or misjudgement of the technical possibilities. What is currently going wrong?

Lithium-ion batteries are currently being seen as the solution to all our problems and as a symbol of technical progress. Unfortunately, the issue has taken on a life of its own on the PR level. There is no doubt that great progress has been achieved. However, decisions by engineers to move in one direction or the other are being determined by public discussion – and serious technical assessments are being ignored. We must ask ourselves whether the lithium-ion battery is always the best storage system or whether double layer capacitors, for example, are not more suitable for small hybrid vehicles. In which cases, for example, is the combination of a battery and capacitors the best means of storing energy? Huge amounts of money and government funding are currently being invested in battery development. Even today, the number of manufacturers seemed far too high for automotive applications. Standardisation has not yet begun.

Lithium-ion batteries are not suitable for hybrid drive systems. Has this awareness still not become established?

Burkhard Göschel talking to MTZ editor Markus Schöttle

The first doubts are becoming increasingly apparent among OEMs. For an electric car, it is currently the most suitable means of storing energy, as slow charging and discharging processes are involved in this case. But its basic principle means that it is not actually suitable as a dynamic energy storage system. I hope that we are not wasting our time.

So are you saying that the nickel metal hydride battery is not yet obsolete?
Without doubt, it is better suited. We should, however, consider using double layer capacitors as a short-term energy storage instead of batteries. For small vehicles, that may be sufficient, or we can use a combination of a battery and capacitors. For an energy requirement of below 100 Wh, the capacitor solution is the cheaper and lighter application option. Electric ranges of beyond 500 m should be left to plug-in hybrids, but we should not forget that we can still expect to achieve significant improvements in the efficiency of conventional drive systems.

Capacitor solutions do not seem to be catching on, however.
The rail industry and commercial vehicle developers take a different view – also in China, by the way. The developers have decided in favour of robust solutions, namely capacitive storage, for example in most urban buses. Here they are concentrating on the defined application profile for hybrid drive systems and are avoiding problems such as temperature sensitivity, life expectancy, wear and maintenance requirements.

Is the electromobility experiment, particularly in Germany, suffering from too much work being duplicated?
There will certainly be a consolidation process in this market. In cell technology in particular, the product and process technology can only be developed by major players. For the time being, the car will not provide a sufficiently large market. In Germany, the major companies in the chemical industry will have to take up the challenge. Perhaps it is only possible through global alliances. Like many other companies, Magna is developing a lot of cell know-how through partnerships, but only in order to gain the necessary understanding for the energy storage systems

According to Burkhard Göschel, Magna will acquire a minority stake in an Asian battery and cell manufacturer

and to get an idea of how the situation might proceed. We deliberately avoid exclusivity in these alliances in order to be open to new developments. In ten years' time, the focus will be on different cell materials, possibly a combination of lithium and air, or storage concepts involving the exchange of a charged electrolyte.

> "We are deliberately avoiding exclusivity in alliances."

What is the contribution of Magna International?
We will acquire a minority stake in an Asian battery and cell manufacturer. The absence of exclusivity will result in the production of a model with which several companies can work together with an established battery manufacturer. This will allow technical standards to mature more quickly and will provide the urgently necessary financial security for all partners.

Which other important challenges need to be addressed in the electric powertrain?
The electric motors, for example, need to be adapted to an electric drive system. As a rule, they are only designed for one operating point and they are too heavy. Their operating map must be much broader for automotive applications.

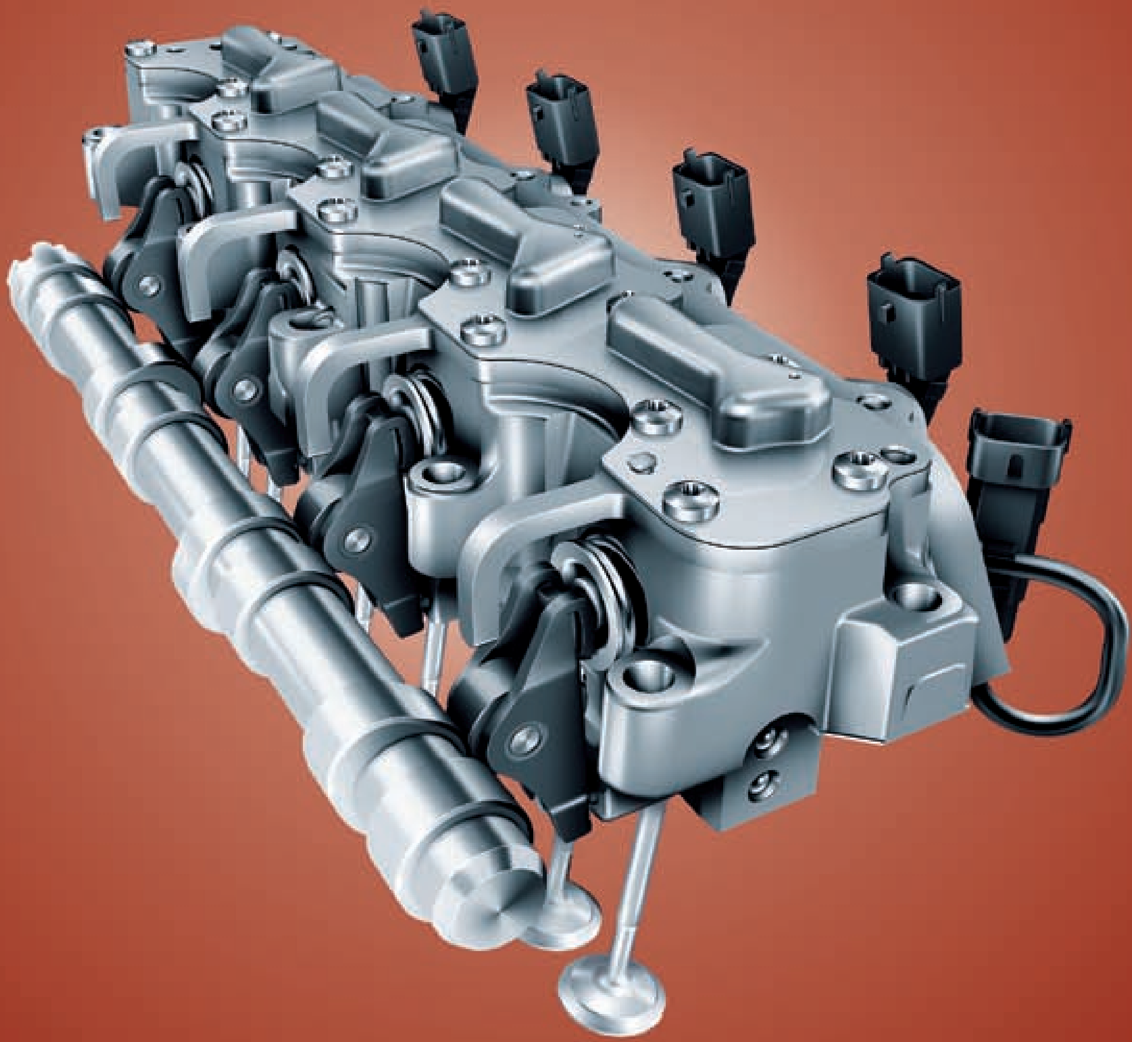
How is Magna positioned in this respect?
Magna has developed its own competence in electric motors, in close cooperation with our own company Bluewave in the USA. We will produce our own electric motors in Italy. I see a further key competence in inverter technology, and I include Magna among the world's best suppliers of this technology. This is based on, among other things, our joint venture with Semikron, which gives Magna access to semiconductor know-how specifically for high currents and voltages.

What does an enthusiastic sports car driver miss when driving an electric car – the sound, perhaps?
When I drive an electric car, I want to enjoy its quietness and a different type of driving experience. This includes playing with such features as adjustable energy recuperation, experiencing almost frictionless "sailing" or attempting to travel as far as possible on a single charge.
The sports car is the classic domain of the hybrid – KERS for the road. In a purely electric car, in which you need 500 kg of batteries to achieve comparable performance, the weight gets in the way of true racing enjoyment.

Professor Göschel, thank you very much for this interview.

INTERVIEW: Markus Schöttle
PHOTOS: Matthias Haslauer

ELECTRO-HYDRAULIC FULLY VARIABLE VALVE TRAIN SYSTEM



Since the end of last year, Fiat has been mass-producing the first fully variable valve train, which is based on electro-hydraulic valve phasing [1]. The Schaeffler Group secured the rights to UniAir technology at an early stage and, as a development partner for the system and its key components, describes the main design steps in this report. Ensuring reliable operation and absolute precision for the system even at very low temperatures and over the entire lifetime – with each solenoid valve carrying out at least 330 million switching operations – was a major challenge.

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FULL VARIABILITY

UniAir, the fully-variable valve train system enables the valve lift of an internal combustion engine to be varied almost without any restriction, limited only by the maximum valve lift curve defined by the cam profile. The system, which was already described in the December 2009 issue, comprises an electro-hydraulic actuator driven by a camshaft with integrated fast switching hydraulic valves, and valve control module software. In contrast to a conventional valve train in which the cam profile is transferred via a rigid element such as a finger follower, UniAir uses a defined volume of oil. By activating the solenoid valves, this oil volume can be controlled in such a way as to specifically change both the engine valve lift curve in terms of the valve opening and closing point and the lift height.

The compact system can be adapted to suit individual engines. As the cam and valve are connected via a hydraulic interface, with the high-pressure chamber acting as a pushrod, the position on the cylinder head can be freely selected.

The system definition for the first application includes three main components, ❶:

: The electro-hydraulic actuators are mounted in the cylinder head at the engine production plant.
: The valve control software (valve control module) is integrated into the engine control system by the engine control unit manufacturer.
: The calibration data set is also integrated into the engine control system.

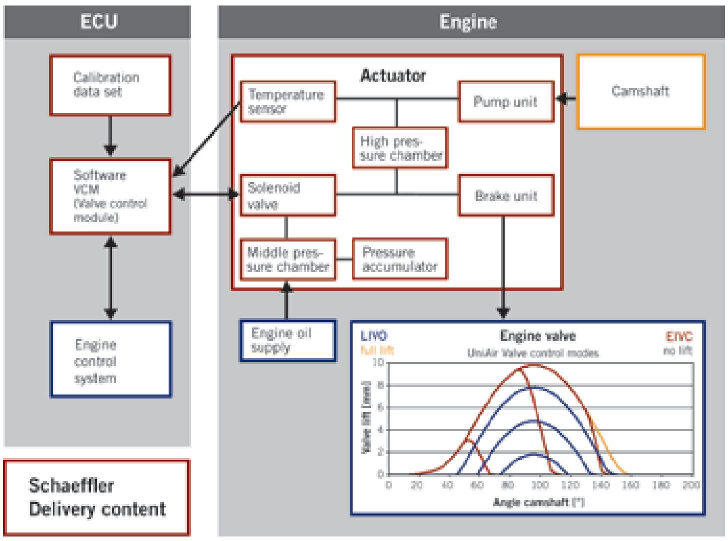
HIGH-PRECISION COMPONENTS

As control elements for the engine valves, the solenoid valves are of central importance for controlling the system. During the design phase of this new solenoid valve, the required switch on and off times, the switching time precision and the durability during the entire lifetime of the system represented special challenges for the developers.

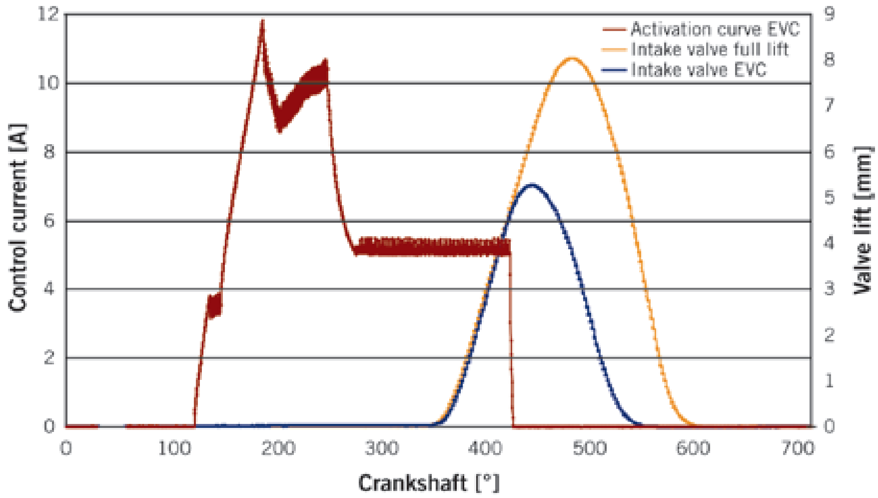
The system architecture with a normally open solenoid valve requires the system to be switched on to generate engine valve lift. Normally, this occurs once per rotation of the camshaft, and during multi-lift operation, it even occurs several times. To ensure refilling of the high-pressure chamber, and therefore to ensure full lift during the next charge-exchange cycle, the solenoid valve is opened for a short time after each cycle. For multi-lift strategies, it must be ensured that, before the solenoid valve is activated a second time, the armature has reached its resting position again after opening for the first time.

❷ shows the activation curve for a solenoid valve and the related engine valve curve. It shows the early valve closing (EVC) mode and the full lift curve as a comparison.

A special activation strategy was developed for the electrical current of the solenoids in order to produce a fast-acting solenoid valve with the lowest possible



❶ Schaeffler components of the UniAir system



❷ Solenoid valve and solenoid valve activation curve

power consumption. The current curve is divided into several sections. Coming from the deactivated solenoid valve, the so-called bias current is applied; this premagnetizes the solenoid valve but does not switch it. In order to ensure rapid and precise closing, an increased peak current is applied at the time of switching. The switching point is determined by the software, depending on the operating condition. After the solenoid valve has been actuated completely, the current is reduced to a holding current in order to save energy. The software then also determines the point in time at which the current is completely switched off, thus opening the solenoid valve again.

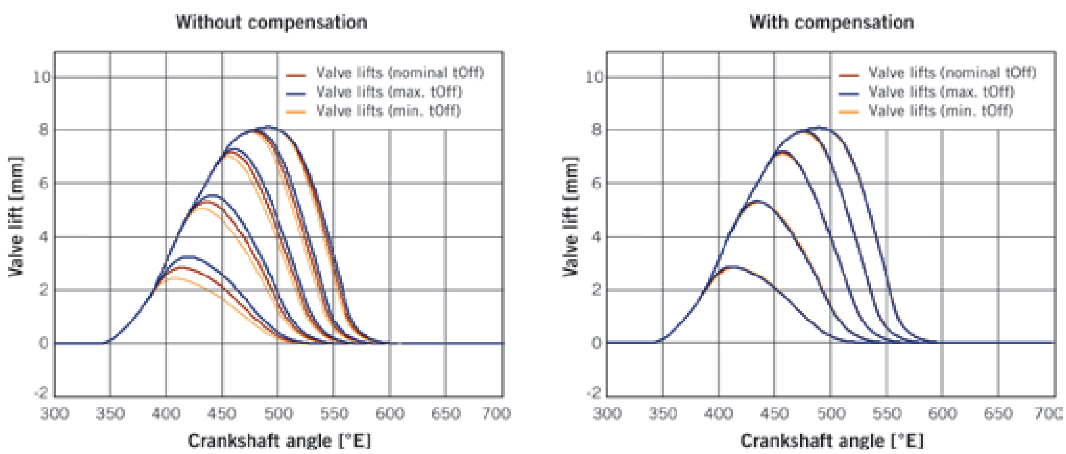
The precision of the opening and closing angles of the engine valves is essential for system function. The switching time precision of the solenoid valve makes a considerable contribution in this regard. During the assembly of the solenoid valves and their subassemblies, various functional values such as flow and switching times are measured on the assembly line. The assemblies are adjusted in such a way that the functional values are within the required range. This means that manufacturing tolerances of the individual parts can be compensated for. Despite tolerance compensation of the individual components, it is still necessary to optimise the precision of the switching times by means of an appropriate compensation function. This compensation is active during the entire lifetime of the product, and therefore also counteracts changes in switching times caused by ageing. This ensures optimum cylinder balancing during the entire lifetime.

The left-hand side of ❸ shows the potential lift curves of individual cylinders without compensation of the off times. The right-hand side shows the curves with active compensation.

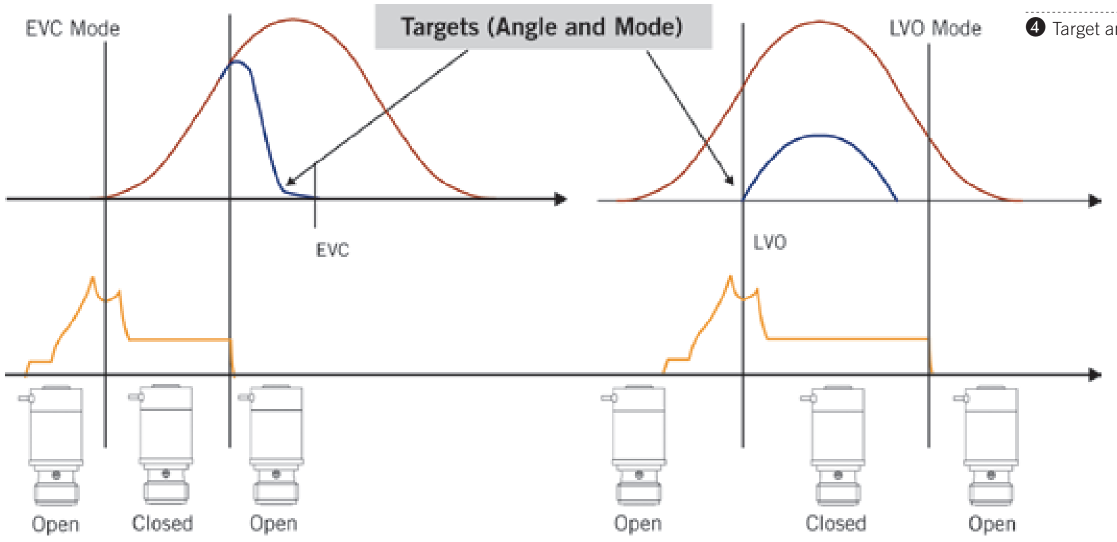
The solenoid valves carry out approximately 330 million switching operations each during the lifetime of the system. This number of switching cycles, at the required precision, poses a significant challenge for the development of the solenoid valve. This new solenoid valve was developed from the concept phase to volume production readiness using the most advanced design and simulation methods in cooperation with Continental Automotive Systems. Both the solenoid valve and the overall system were validated in several function and durability tests on component and system test benches as well as in vehicles.

The solenoid valves are individually controlled by the valve control module software. This implements the requests of the engine control unit regarding valve lift modes with defined opening and closing angles of the engine valves. With this approach, the software considers various factors that influence the behaviour of the system, in order to find the correct actuation point of the solenoid valves in each case, and therefore setting the desired timing of the engine valves, ❹.

In the first step, the switch-on and switch-off times of each solenoid valve must be considered. These are individually monitored using the current curve during each switching process for each cylinder, and then readjusted depending on the operating condition using control maps in the engine control system. The special challenge in this case is the detectability of the current curve over the



❸ Compensation of off times of the solenoid valve for 5000 rpm and 15 °C



4 Target angle and valve lift modes

entire temperature range, and the oil viscosity associated with it. All solenoid valve components must be perfectly matched to each other to secure this function.

But the character of the valve lift curve is also determined by the system architecture and the geometry of the components. This includes the brake unit. This unit is a slave cylinder that converts the hydraulic pressure into the movement of the engine valve via a hydraulic lash adjustment element. Since the engine valve is always closed independently of the cam profile, it is necessary to brake the engine valve hydraulically at the end of its ballistic flight phase. This prevents excessive closing speeds that could otherwise lead to noise and damage to the valves. On the other hand, to achieve fast opening times, the brake is short-circuited by means of a special check valve. Designing all these components in a specific manner ensures that the engine valve is closed in good time under cold conditions (-30 °C) and ensures low closure speeds with hot engine oil.

The valve movements and the brake function in particular are determined not only by geometric and architectural influences, but also by environmental and operating parameters such as engine speed and oil viscosity.

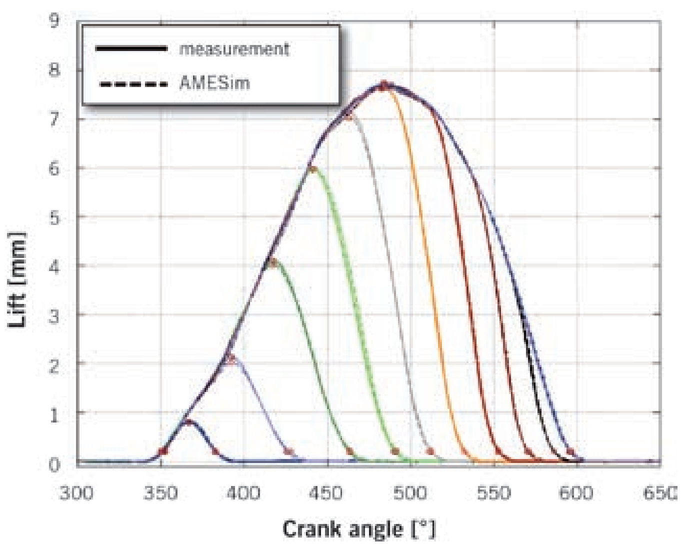
It is therefore necessary to determine the oil viscosity, particularly during cold starts, and the subsequent internal warm-up of the system. In this context, and as the only additional sensor for this system, the temperature sensor is an integral component.

The sensor with an NTC element (negative temperature coefficient) is specially calibrated for use at low temperatures (highest precision at 0 °C), and has a response time of a maximum of 1.4 seconds (τ_{90} in water) – much faster than the temperature measurement already present on the engine for measuring the cooling water and engine oil temperature.

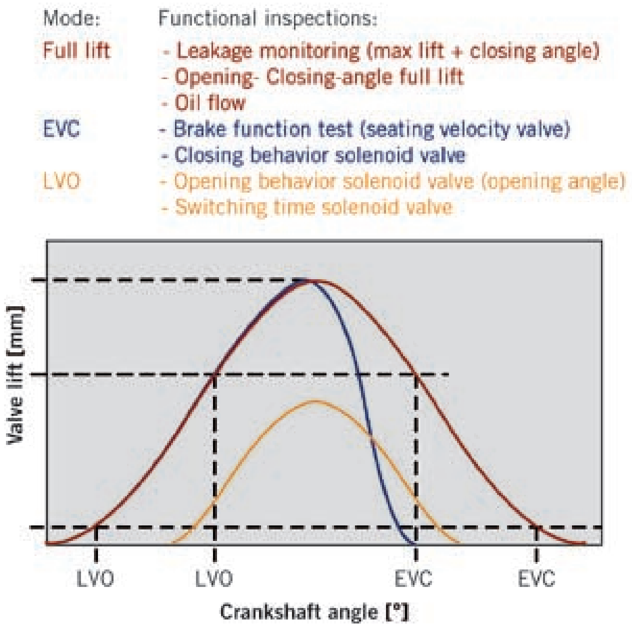
INFLUENCING PARAMETERS OF THE SYSTEM ON THE FUNCTION

The complexity of the system clearly shows that the system function can only be ensured by means of an intelligent interaction between mechanical and mechatronic components, as well as by diagnostic and compensation functions ensured by the software.

During system calibration, actuators and components with known or adjusted geometric and functional characteristics are tested under various boundary conditions (in particular, speed and oil temperature). By developing the calibration data set in this way, the control software can take the influencing factors depending on geometry and architecture into consideration. Various measurement values are also continuously measured during operation, such as the oil temperature, the switching time of the solenoid valves and the engine speed.



5 Valve lift curves – simulated and measured



❻ End-of-line test

DEVELOPMENT AND DESIGN

During the development phase, the system was preliminarily designed using state-of-the-art design and simulation tools. By linking powerful programs such as Matlab/Simulink and AMESim, a comprehensive tool for numerical simulation that facilitates systematic analysis was generated. This tool was used to design the cam profiles and to simulate the valve lift curves of the intake valves, ❺.

The validation of the UniAir system was ensured by means of comprehensive durability, function, robustness and system tests.

Since the influence of the oil viscosity in this electro-hydraulic system is crucial for its function, functional cold start tests were given particular attention. Cold start tests were carried out on complete engine assemblies for temperatures from -30 °C in both fired and motored configurations.

The measurements on the motored complete engine facilitated lift and pressure measurements in the high-pressure chamber during start-up, whereas the starting behaviour and pressure in the high-pressure chamber were measured and analyzed on the fired engine.

While considering borderline samples, the seating velocity, the maximum pressure in the high-pressure chamber, the closing speed and the forces on the lever were investigated, as well as the factors mentioned above.

In order to demonstrate the robustness of the system for unfavourable boundary conditions, extreme tests with oil aeration, oil viscosity, contamination and excessive speeds were carried out.

The function of the UniAir system in conjunction with high viscosity oils was mainly validated by means of cold start tests. During these tests, maintaining the valve closure angle was particularly important, since it is responsible for engine starting behaviour. The influence of oil aeration on the precision of the system was tested over the entire operating control map at high temperatures. The function was also demonstrated at maximum aeration.

The robustness of the system regarding oil contamination was also proven with excessive amounts of contaminant particles.

Function and durability at speeds of up to 7,200 rpm were demonstrated in more than 10,000 operating hours of the endurance test.

MANUFACTURING AND QUALITY ASSURANCE

The function of each assembled actuator is tested at the end of the assembly process on a test bench that, to a great extent, replicates the development test benches, ❻. The test process is automated, and the program is limited to significant operating points (Full Lift, LVO and EVC with certain angles and speeds).

By using the integrated "test cylinder head", the achieved precision of the opening and closing angles and the maximum lift height are monitored.

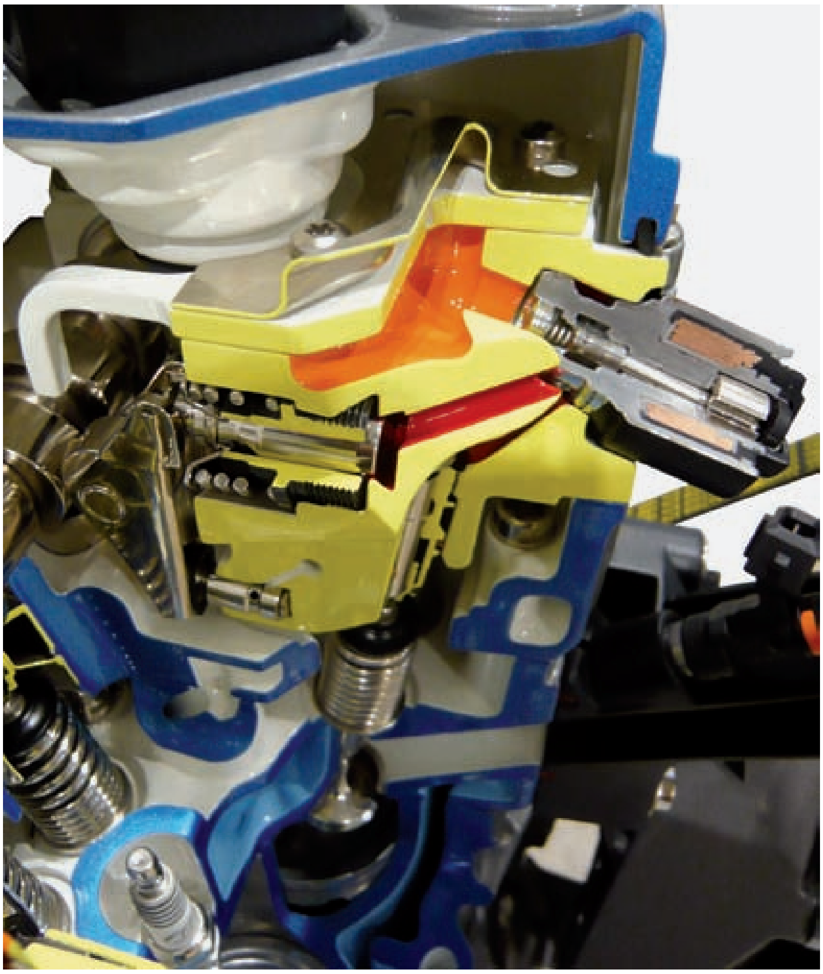
In addition, the function of the brake units and the permitted seating velocity in EVC mode are tested. Furthermore, the switching times of the solenoid valves in LVO mode are tested.

The production measurement data of all steps of the process (including purchased parts) is recorded in a central database, and can be traced at any time – from the finished product back to each single component. This includes, for example, the measured values of the solenoid valves, the oil temperature sensor, the screw-mounting parameters and the press-in forces of the actuator.

SUMMARY AND OUTLOOK

The variability of valve trains is an important component of internal combustion engine technology of the future, and will assist in fulfilling strict current and future legal requirements regarding emissions and fuel consumption, as well as customer requirements.

The fully variable electro-hydraulic valve train system, UniAir, makes an excellent contribution to optimising combustion processes, and expands the potential of current variable valve control mechanisms.



❼ Cross-sectional view of the cylinder head of the Fiat 1.4 l engine with UniAir

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The development of the UniAir system with Fiat means that the Schaeffler Group has succeeded in launching a new valve train system that provides major benefits compared to conventional valve trains, not only for vehicle manufacturers but also for end customers.

The world's first fully variable electro-hydraulic valve control system was launched in the market on September 2009 in the Alfa Romeo MiTo. The engine is a turbocharged 1.4 l petrol engine with an output of 100 kW, ❼. The measured advantages in the first mass production application compared to an engine with a conventional valve train without a camshaft phasing unit are:

- Up to 10 % reduction in CO_2 emissions
- Up to 10 % more power
- Up to 15 % more torque in the lower speed range.

As well as the advantages of increasing performance and reducing consumption, the system offers a wide range of additional benefits. For example, the system is very flexible and can be adapted to meet the requirements of customers and their engines. The UniAir module provides an impressively compact and flexible design and at the same time offers a high level of variability of the valve lift curves. It is operated using conventional engine oil, and the control system can be integrated into the engine control unit.

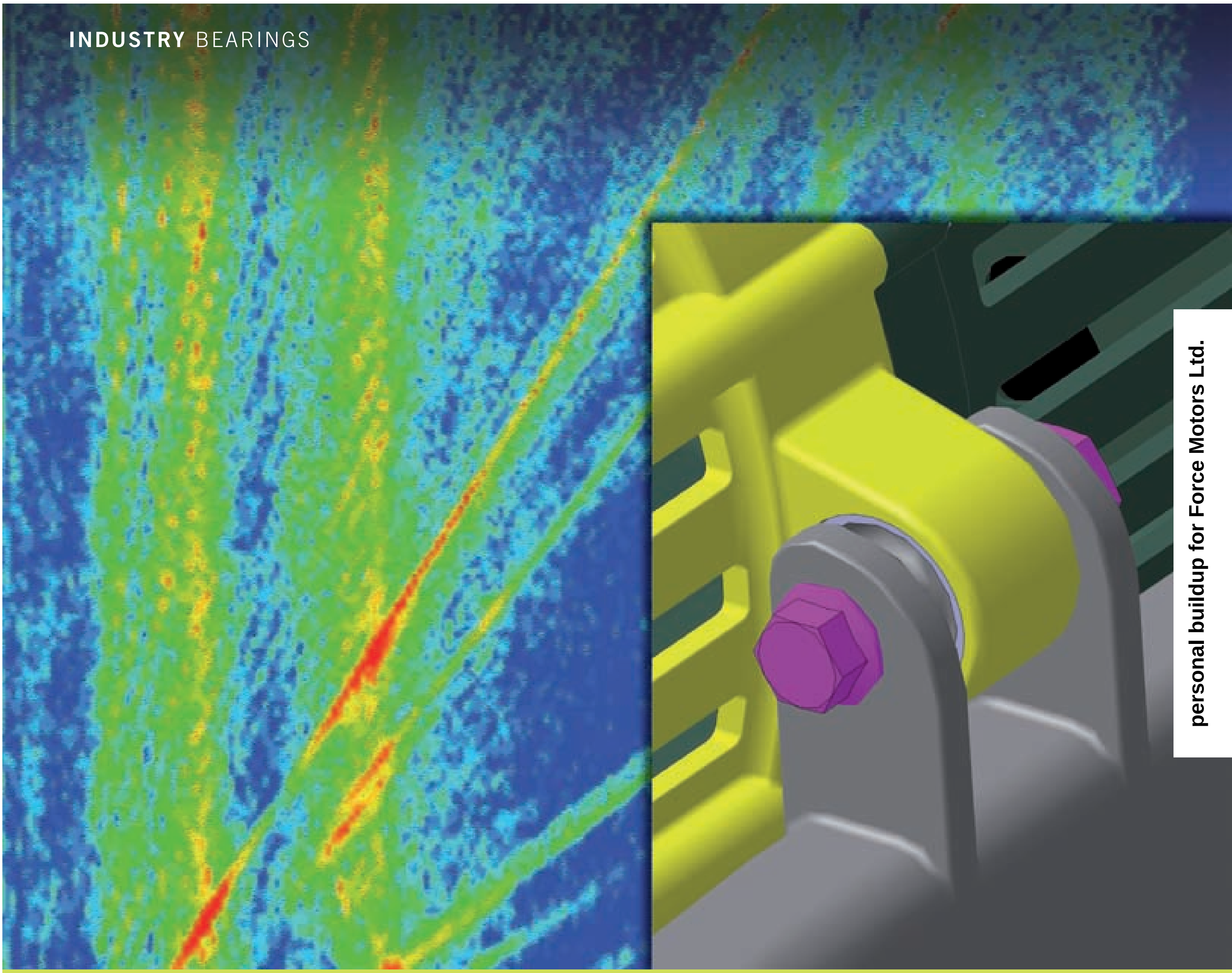
Diesel engines can also benefit from the numerous advantages that the UniAir system has to offer. Initial developments for engines in passenger cars, commercial vehicles and large engines have already been very successful.

In order to meet the requirements of future combustion methods, for example HCCI, in terms of air and exhaust paths, Schaeffler and its partners are already testing the relevant systems in advanced development.

THANKS

Our special thanks go to Piergiacomo Traversa (responsible for Software Function Development), Dr. Razvan Anca (responsible for System Development) and Lothar von Schimonsky (responsible for Testing) for their exceptional support and commitment throughout the Fiat Fire MultiAir project.

We would also like to thank Sarah Kronenberg, Michael Berger, Andreas Kufner and Jürgen Ebert for their valuable contributions to this article.



ELASTIC MOUNTS WITH HIGHER DURABILITY BY VIBRO-ACOUSTIC ANALYSIS

In the development of engine mounts, there is often a conflict of objectives between the lifetime of the mount and the NVH properties of the system being supported. This article by Hofer presents possibilities for achieving noticeable improvements in the lifetime of relatively simple and low-cost engine mounts in spite of difficult boundary conditions with regard to installation space constraints, component stress and acoustic system behaviour.

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Designing the mounting system of engines or power transfer devices in general, several aspects need to be considered simultaneously. If only the loads bearing function or only the acoustic properties of the mounting system are considered, the result will be unsatisfactory. This fact is independent of the type of device under investigation, be it a main engine, an auxiliary engine, accessories or a power transfer device. The mounting systems of an electrically driven axle, transfer cases, etc. in automobiles needs to be designed as well under various considerations, as the mounting system of engines and power transfer devices in water vehicles or similar applications. In this article all these kinds of devices are referred to as the "device" for simplicity.

Typically the device to be mounted is exerted to a torsional load along a principal axis, which is to be supported at the housing side. The available frame structures to support the loads are made of beam profiles in many case, due to space, weight and cost constraints. The load is transferred by the elastic mount, which is to be designed.

An additional problem, which is present in many cases, is an inappropriate ratio of the torsional load and the available design space for the elastic mounting system (size of the mounts as well as size of lever arms) in conjunction with slim inexpensive light weight support structures of low stiffness.

Such conditions often result in the conflict of interests between the durability of the mounts and the NVH properties of the whole mounting system.

This paper demonstrates a way to enhance the durability of relatively simple and inexpensive elastic mounts considerably, although the boundary conditions concerning design space, transferred loads and acoustic system properties are difficult.

PROBLEM AND ANALYSIS

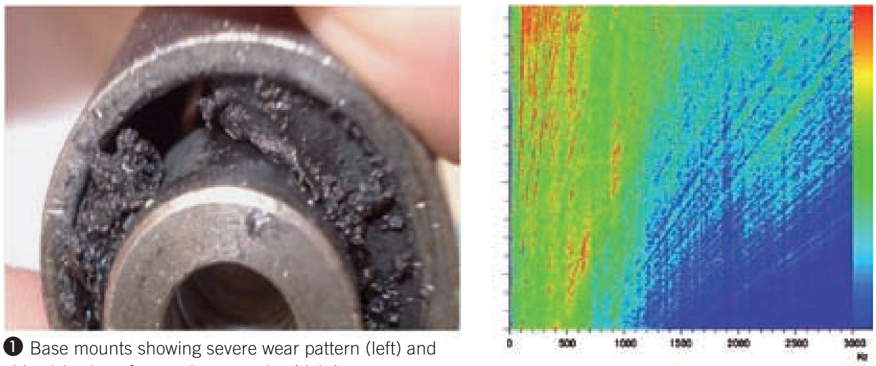
The durability of a mounting system with corresponds to the above described conditions (high load to be supported, short lever arms, minimal design space for the mounts, soft profile support structures) needed to be improved without negative impact on the NVH properties.

In a first approach inexpensive radial-symmetric mounts (base mounts) were used, which showed severe wear after only a short time of operation. The supporting function of the mounts were no longer secured with the observed amount of wear, ❶ (left). The acoustic properties of this system were subjectively rated acceptable. ① (right) shows these acoustic properties objectively.

Several attempts using conventional pocket mounts with and without bumpers, ❷, did not improve the situation considerably due to durability or acoustic reasons. Finally an explicit design of mounts was started for these conditions.

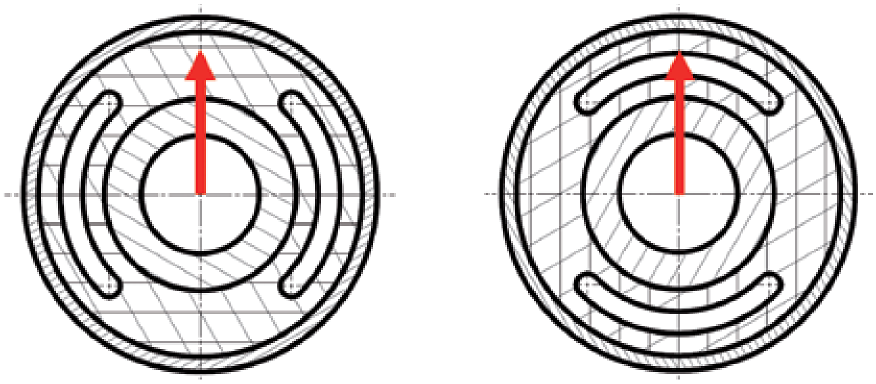
TRANSFER PATHS WITH DEPENDENCY ON DIRECTION

At first the dependency on direction of the transfer of structure borne noise from the device via the mounts to the supporting structure was analyzed. It was obvious that the stronger transfer of structure borne noise was present in the x-direction (radial to the device) were only low loads needed to be supported. On the other hand the transfer of structure borne noise

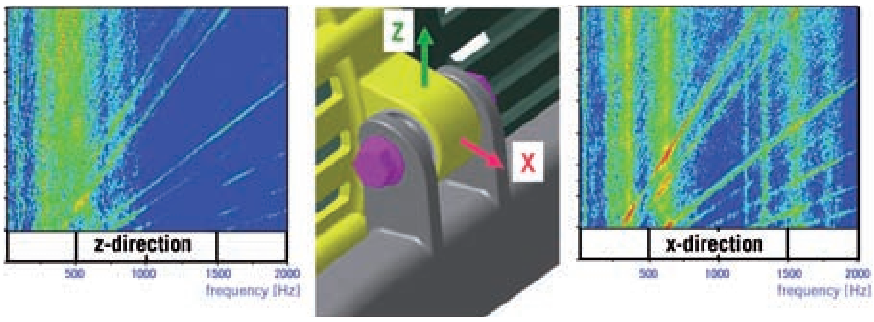


❶ Base mounts showing severe wear pattern (left) and objectivization of acoustic properties (right) was considerably lower in the z-direction (tangential to the device) were the main loads need to be supported, ❸.

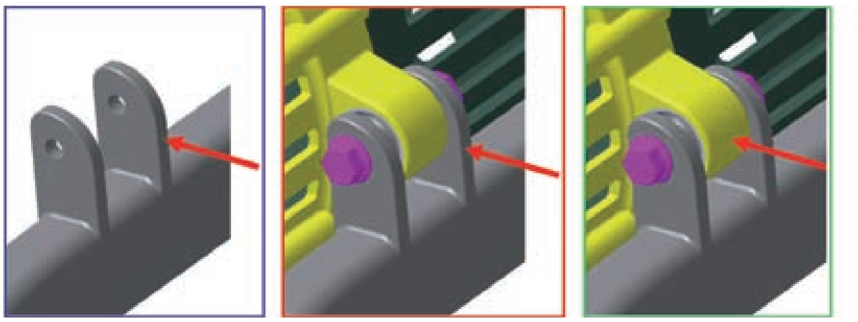
The strong transfer of structure borne noise in the x-direction is caused by two natural frequencies at approximately 400 Hz and 650 Hz, as can be seen in the right part of ③. The main reason for these two natural frequencies will be analyzed in the following.



❷ Conventional pocket mount loaded in web and pocket direction



❸ Dependency on direction of transfer of structure borne noise via base mounts



❹ Impact tests on supporting structure with and without mount and device

INFLUENCE OF THE SUPPORTING STRUCTURE

Impact tests were performed in three different configurations, as shown in ❹.
At first the accelerance of the supporting structure itself was determined ④ (left).
It shows two almost un-damped natural frequencies in the frequency range below 1500 Hz, at 250 Hz and at 650 Hz, ❺, (blue curve). Subsequently the accelerance of the supporting structure was determined with mount and device connected to it, ④ (middle). The natural frequency of the supporting structure itself at 250 Hz was moved to approximately 400 Hz due to the stiffness of the mount that was connected to it, ⑤ (red curve). Finally the transfer of structure borne noise from the device to the support structure was determined, ④ (right). The transfer is dominant exactly at the two natural frequencies of the support structure with connected mount at 400 Hz and 650 Hz, ⑤ (green curve). These two natural frequencies coincide exactly with the two natural frequencies visible in the right part of ③.

In conclusion, the strong transfer of structure borne noise in the x-direction with the two natural frequencies at 400 Hz and at 650 Hz is cause by the supporting structure, ③ (right). The two natural frequencies of the supporting structure itself (slim profile structure) are shifted by the connection with the elastic mount, but they are neither eliminated nor adequately damped.

Now, a mounting system is to be designed that can fulfill the durability requirements and shows acceptable acoustic properties, even under these circumstances.

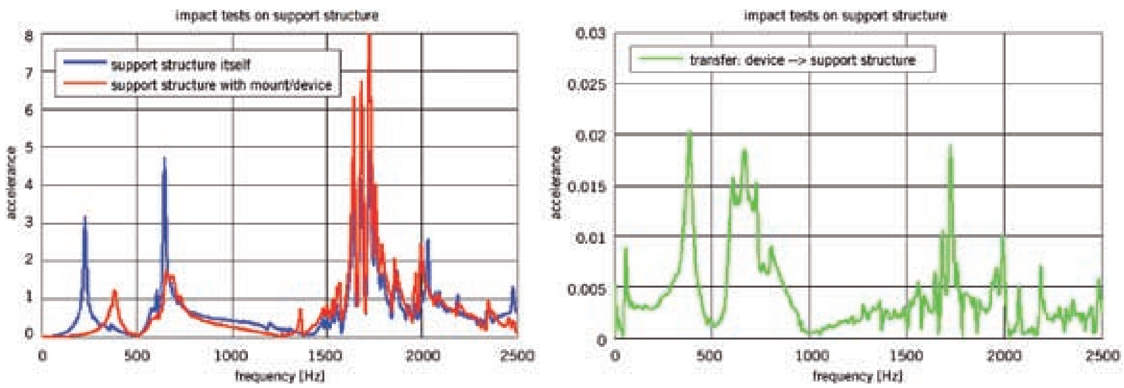
POTENTIAL OF IMPROVEMENT OF THE BASE MOUNTS

The analyses named above have been performed with the radial-symmetric base mounts. Considering the requirements to support loads, a potential for improvement becomes obvious by an asymmetric design of the mounts: in tangential direction (as seen from the device) high loads need to be supported, but the sensitivity to structure borne noise is less. Therefore the mounts can and need to be designed rather stiff in this direction. In the radial direction the sensitivity to structure borne noise is very high, but in this direction the loads to be supported are lower. Therefore the mounts should be designed rather soft in this direction to improve isolation in this direction.

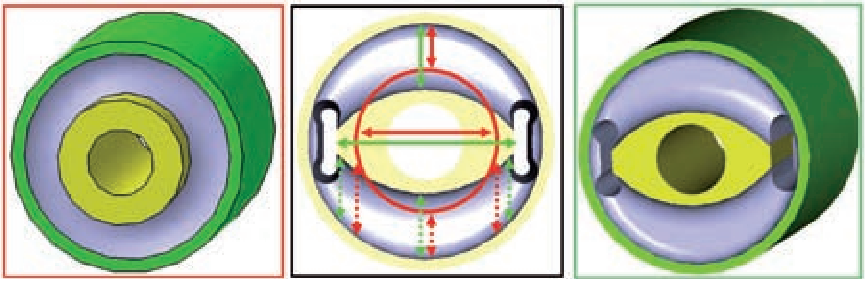
A further measure should be the enlargement of the load bearing area to support the high loads in tangential direction. This would result in a general reduction of the stress level in the rubber material. Finally, the thickness of the isolating rubber material is rather low in the base mounts and should be enlarged for improved isolation.

If the degree of isolation can be improved by geometric means, rubber material of a higher shore hardness can be selected and still the overall acoustic properties are similar to the system using the base mounts.

At the very end, the contours of the core and the shell of the base mounts are considered in detail. ① demonstrates a strongly localized area of wear, which implies a



❺ Results of impact tests on supporting structure with and without mount and device



❻ Implementation of measures for an improved mount concerning durability

localized area of stress concentrations. The contours of the core and the shell are not congruent and thus strongly localized stresses occur under high loads. If the contours are designed congruent, such stress concentrations should be prevented.

HYPOTHESES TO IMPROVE DURABILITY

The goal of the explicit design of a mounting system for the given conditions is an improvement of the durability without implications on the needed design space while preserving the acoustic properties.

Following the analyses named above, five hypotheses are now formulated that will contribute to achieve this goal:

- An asymmetric design with increased stiffness in tangential direction (as seen from the device) and reduced stiffness in radial direction will result in similar acoustic properties of the overall system.
- An enlarged load bearing area in the main loading direction (tangential) will increase the stiffness in this direction, will reduce the stresses in the rubber material and will contribute to an enhanced durability.
- An increased distance of the inner core and the outer shell will reduce the stiffness in this direction and will improve isolation.
- The selection of a higher shore hardness will increase the stiffness in both directions and will improve the durability.
- The design of the load bearing rubber volume with constant cross-section (congruent contours) will reduce stress concentrations in the rubber material and will contribute to an improved durability.

These hypotheses include measures that increase the stiffness as well as decrease the stiffness. It is difficult to say to which extend the measures compensate each other. This will also depend on the degree to which each of the measures is realized or can be realized, respectively. It is as well difficult to foresee, which shore hardness is to be used, to result in an overall system with similar acoustic properties as with the base mounts.

SOLUTIONS AND REALIZATION

❻ shows a design of the mount, with all measures implemented. The core is realized with an oval contour, which yields to an increased load bearing area and simultaneously yields to an increased thickness of the rubber material in the main loading direction. The contour of the core and the shell are designed with identical radii. This results in a rubber volume with constant cross-section. At the sides of the core minimal pockets with bumpers are introduced, which finally gives the desired asymmetric stiffness properties.

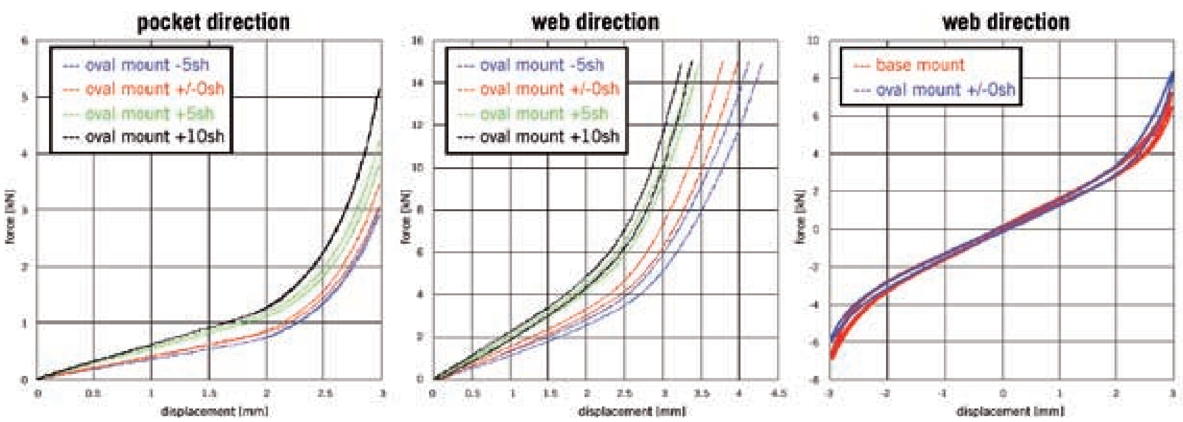
With the chosen implementation, it is expected, that the following measures will compensate each other: enlargement of the load bearing area, increased thickness of the rubber material in the main loading direction and the modification of the cross-section contour. The stiffness in web direction of the new mount should thus be similar to the stiffness of the radial-symmetric mount. It is further expected, that the stiffness in pocket direction of the new mount is considerably lower, as the radial-symmetric mount.

If now the dependency on direction of the transfer paths is considered (sensitive direction for transfer of structure borne noise coincides with the soft pocket direction), it is to be expected, that the new mounts will be acoustically superior to the base mounts, if the same shore hardness is used. Or vice-versa a higher shore hardness can be used to result in a mounting system with similar acoustical properties.

To prove this assumption, the new mount was manufactured in the same shore hardness as the base mount and in -5 sh, +5 sh und +10 sh.

VERIFICATION OF HYPOTHESES

Three kinds of investigations were performed to verify the formulated hypotheses:



❼ Static characteristics of oval-mounts in both directions with base mount for comparison

- elastic properties: static characteristics and dynamic stiffness, in web and pocket direction each
- subjective rating of the acoustics
- durability on test bench investigations.

❼ shows the static characteristics in pocket (left) and web (middle) direction of the oval-mounts as manufactured in the four different shore hardness's. ⑦ (right) shows a comparison of the static characteristic in web direction of the oval-mount versus the base mount, both manufactured in the same shore hardness. The stiffness' in the linear section of ⑦ (right) are identical, proving the hypothesis that the enlargement of the load bearing area, the increased thickness of the rubber material in the main loading direction and the modification of the cross-section contour compensate each other. The oval-mounts with higher shore hardness are stiffer and oval-mount with lower shore hardness is softer.

Further, the hypothesis on the asymmetric stiffness' in pocket and web direction could be proven: all oval-mounts are considerably stiffer in web direction than in pocket direction. These results are similarly true for the dynamic stiffness', as can be seen in ❽.

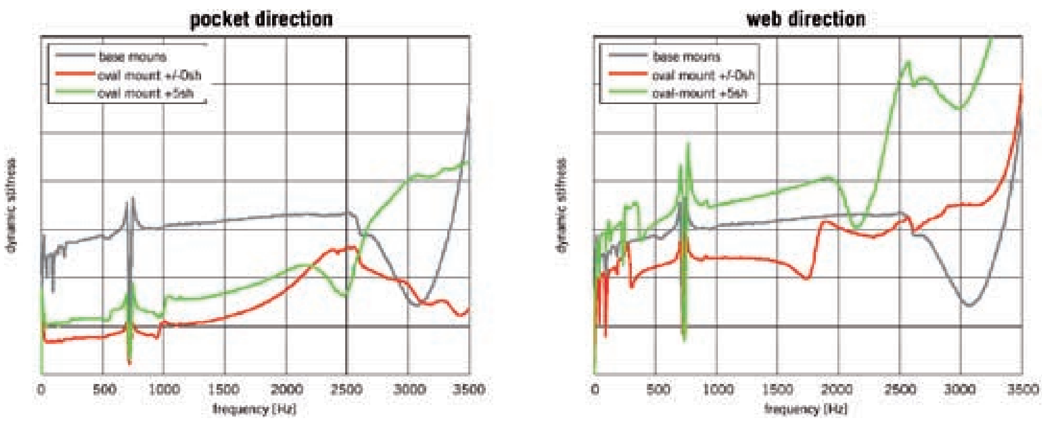
The base mounts and the oval-mounts in all shore hardness's have been mounted in the target application and have been rated subjectively by various test persons under realistic operation conditions. ❾ shows the results of these ratings. In this representation all scores have been normalized by the mean value of the scores for the base mount. The oval-mount with 5 sh increased shore hardness showed similar acoustic properties as the base mount.

Thus a further hypothesis was proven: the asymmetric design of the stiffness' improved the acoustic properties under the given conditions of the dependency on direction of the sensitivity to structure borne noise. Even though the oval-mount with 5 sh increased shore hardness has a higher dynamic stiffness in the load bearing web direction than the base mount, it is acoustically similar because it has a lower dynamic stiffness in the pocket direction which is more sensitive to structure borne noise ⑧.

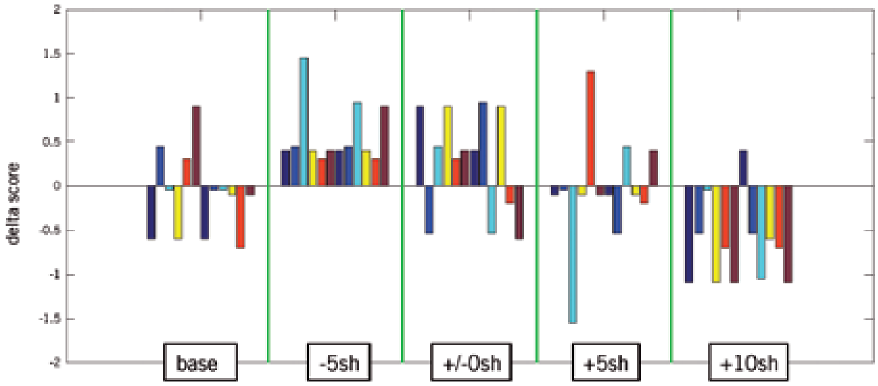
The main goal of this development was to design a mount that is acoustically similar to the base mount with considerably improvements regarding durability. Thus the oval-mount was produced with an increased shore hardness by 5 sh.

These mounts were now compared to the base mount regarding durability, using a uniaxial test setup ("Hydropuls") with cyclic loading in the main loading direction as in normal operation. ❿ (left) shows the results from these investigations. The stiffness of the mounts in the linear area of its characteristic is plotted versus the applied loading cycles. The decrease in stiffness is indicated as a measure of the degree of the progression of fatigue. The decrease of the stiffness is five-fold slower in the oval-mount and thus the fatigue progresses five-fold slower than in the base mount.

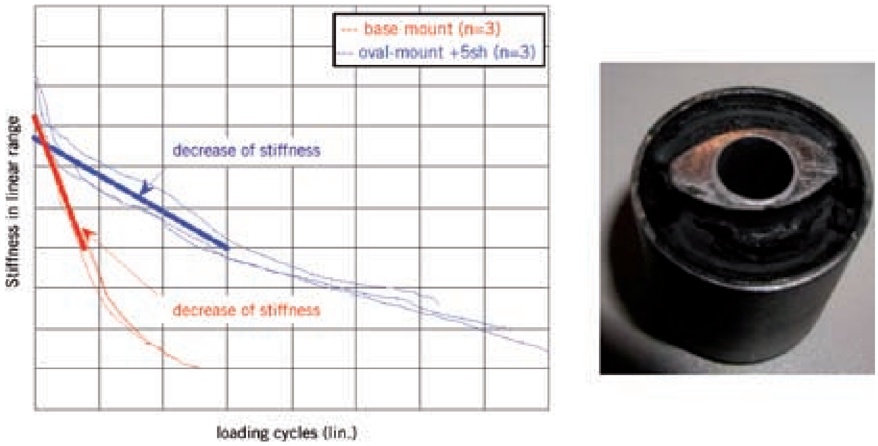
⑩ (right) shows an oval-mount after termination of the whole cyclic loading test. It is well visible that whole rubber volume is destroyed similarly. The localized area of stress concentrations as seen in the base mount could be successfully prevented. Thus the last hypothesis could be proven as well.



❽ Dynamic stiffness of oval-mounts and base mount for comparison



❾ Subjective acoustic ratings of the oval-mounts, in comparison to the base mount



❿ Decrease of stiffness under cyclic loading (left) and wear pattern of oval mount

All the measures introduced to improve the durability resulted in its combination in a strongly increased durability. The five-fold slower decrease of stiffness in the oval-mount corresponds to an expected five-fold increased durability.

SUMMARY

The conflict of interests between the durability and the acoustic properties of elastic mounting systems for engines or power transfer devices is a well known problem. The available design space restricts the possibilities to overcome this conflict in many cases. This paper demonstrates a way to enhance the durability of elastic mounts considerably without negative impacts on the NVH behavior, although difficult boundary conditions concerning design space, transferred loads and acoustic system properties were given. These boundary conditions were too complex for an experimental approach using mounts of different shore hardness's or conventional pocket mounts with bumpers. An explicit development of mounts for these specific conditions was therefore necessary.

A transfer path analysis (TPA) was carried on the radial-symmetric base mounts, which were used as a starting point. Special attention was given to the dependency on direction of the sensitivity to transfer of structure borne noise. The sensitivity to transfer of structure borne noise was found to be especially high perpendicular to the main loading direction were only low loads needed to be supported. The reason for this sensitivity was found to be inherent to the design principle of the supporting structure, which however could not have been changed due to restrictions in the available design space. After a further detailed analysis of the geometry of the base mounts, five hypotheses were formulated that should improve the durability without negative impacts on the acoustic properties:

: asymmetric design of the stiffness' according to the TPA
: increase of the load bearing area in the main loading direction
: increase of the thickness of the rubber material for improved isolation
: increase of the shore hardness by 5 sh
: contouring of the load bearing volume with constant cross-section.

Parts were designed according to these hypotheses and were realized in different shore hardness's. It could be proven that the static and dynamic properties of the new mounts were according to these hypotheses: The increase of the load bearing area in combination with an increased thickness of the rubber material resulted in a slightly reduced stiffness in the load bearing direction when the same shore hardness was used. Using a 10 % higher shore hardness, the stiffness was slightly increased with respect to the base mount. The stiffness was considerably reduced perpendicular to the load bearing direction for all of the shore hardness's used. Several test persons have rated the base mount and the new mounts in all the shore hardness's subjectively for their acoustic properties in a fully blinded test. The new mount with increased shore hardness demonstrated to be acoustically equivalent to the base mount. The stronger transfer of structure borne noise by the stiffer z-direction was compensated by the decreased transfer by the now softer x-direction. The formulated hypotheses concerning the durability were proven by uniaxial cyclic loading tests (main loading direction as in normal operation): the increased load bearing area in the main loading direction, the contouring of the load bearing volume with constant cross-section (homogenous stress distribution) and the increase of the shore hardness leaded to a strongly increased durability. The decrease in mount stiffness – which is a well established evaluation parameter in durability testing of elastomeric mounts – was found to be five-times slower than in the base mounts. This corresponds to an expected five-fold increased durability.

This paper demonstrates that the conflict of interests between acoustics and durability can be overcome with rather inexpensive mounts even under preservation of the given constraints, if a comprehensive vibro-acoustic approach is followed.

CAMSHAFT WITH ROLLER BEARINGS TO REDUCE MECHANICAL LOSSES

Reducing frictional losses in combustion engine and power train applications is imperative to achieve future CO_2 emissions targets. Mahle and Timken have combined their expertise to develop new camshaft technology which could positively contribute to this industry effort. This paper describes the content and the results of this joint work and highlights the potential benefits of the proposed solution.

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In order to reduce the average CO_2 emission of new sold passenger cars, the European Union has decided that car manufacturers will have to pay fines for any gram of CO_2 over the imposed limits. This new regulation will take place in 2012; and at that time, CO_2 emissions will cost between 5 € and 95 € per gram over the limit. Knowing that the 95 € will become the standard for any gram over the limit in 2018 reducing CO_2 will significantly influence the cost of new cars.

This increasing pressure on the automotive industry to achieve better fuel economy and to reduce emissions is pushing market players to continuously innovate to reduce inner engine losses. Frictional losses, which negatively affect the efficiency of several systems (e.g., engines, transmissions, accessories, wheel-ends) in vehicles, can often be reduced by replacing sliding contacts with rolling contacts.

For example, in the last ten years, valve actuators, such as flat tappets have been systematically replaced by rocker arms or finger followers with needle rollers. The next logical step in the reduction of frictional losses in the valvetrain is to reduce the friction of the camshaft support bearings, which are traditionally hydrodynamic bearings. Inserting rolling bearings does not only provide a reduction of the friction, it also reduces the amount of oil required to lubricate the valvetrain, which significantly contributes to the reduction of total fuel consumption.

Mahle and Timken present in this article a camshaft technology named Low Friction Camshaft (LFC), composed of an assembled camshaft with integrated needle roller bearings and pre-validated through simulation and testing,

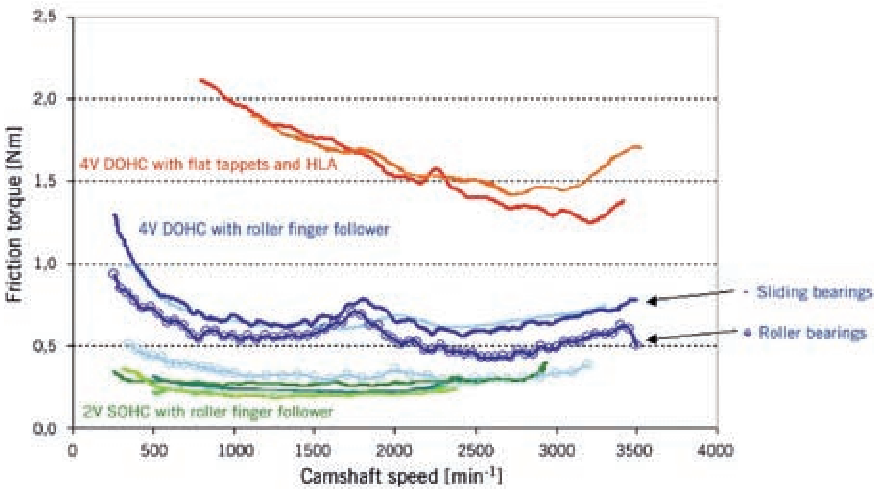
OBJECTIVES AND BACKGROUND OF THE STUDY

Prior to 2007, Mahle and Timken worked separately on various anti-frictionized camshaft projects; however, it eventually became evident that collaboration of efforts would result in the best solution for the market. Indeed, effective integration of needle roller bearings on a camshaft is only possible with a close collaboration between bearing and camshaft specialists. The reduction of interfaces provides Original Equipment Manufacturers (OEMs) products that have minimized technical risks at an optimized system cost.

The objective of the Mahle and Timken engineering teams was to develop an assembled camshaft with integrated roller bearings, meeting the following criteria: cost-effective, mass production compatible, “drop-in” design (compatibility with OEM’s assembly processes and easy adaptation to existing engines) and pre-validated through proof-of-concept activities (e.g., simulation, prototyping, testing). To perform this study, Mahle and Timken have worked on existing, high volume engines in both gasoline and Diesel configurations.

THEORETICAL ASSESSMENT OF TECHNOLOGY BENEFITS

The described market’s goal to reduce emissions and achieve better fuel economy is driving the reduction of inner engine losses. Regarding valvetrain systems, the influence of friction and oil flow are the main focus of the investigation. Both can be optimized with rollerized camshafts.



❶ Comparison of valve train friction of different engines (per cylinder, hot oil)

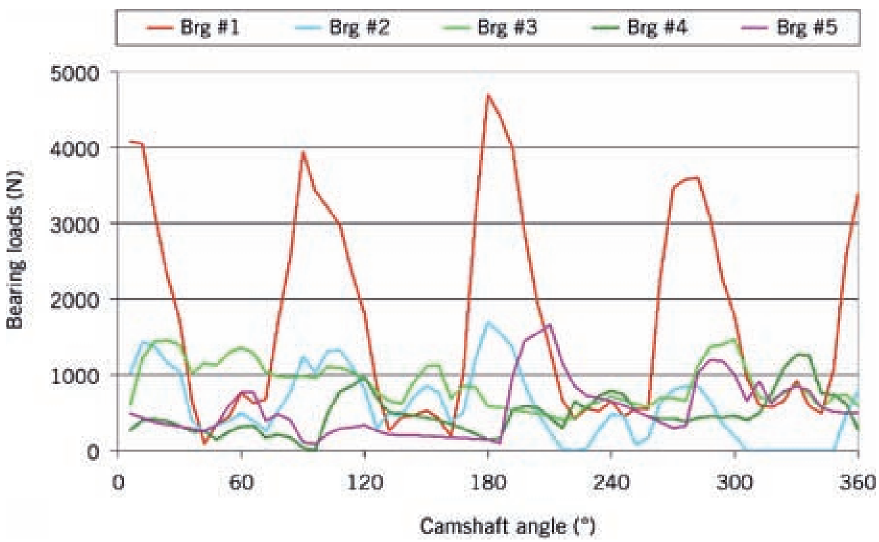
One benefit is to go from a sliding to rolling contact. While rolling contact has been explored for crankshafts, the application of these ideas to camshafts with roller bearings is rather new.

The advantage of roller bearing camshafts compared to standard plain bearing camshafts is the reduction of mechanical losses in the form of friction, especially for cold start behaviour and low engine speeds, ❶.

The other benefit is to reduce the amount of oil that is necessary to lubricate the bearings of a camshaft. Until now, the sliding bearings of the camshaft are lubricated almost directly. This causes engines with a large number of journals to have significant oil flow volume. To ensure sufficient lubrication of the cam followers, additional oil is provided through HLA-sided (HLA: Hydraulic Lash Adjuster) spray holes in the cam follower. Therefore, the oil pump must be designed to support adequate oil flow volume.

The application of the roller bearing camshaft reduces the necessary oil flow, requiring spray oil for lubrication, only. The oil flow volume in the cylinder head can be reduced. Accordingly, the oil pump can be downsized or excess oil can be made available for other systems.

The frictional decreases, as well as the reduction of the oil flow, lead to reduced CO_2 emissions.



❷ LFC bearing loads comparison

APPLICATION STUDY

In order to propose a technical solution that is the best compromise between friction and durability, specific tools have been developed to support the design and the analysis of camshaft bearings.

The calculation takes into account the bearing properties as well as housing stiffness and shaft definition (the model includes system deformation) and therefore, requires some technical and detailed inputs data:

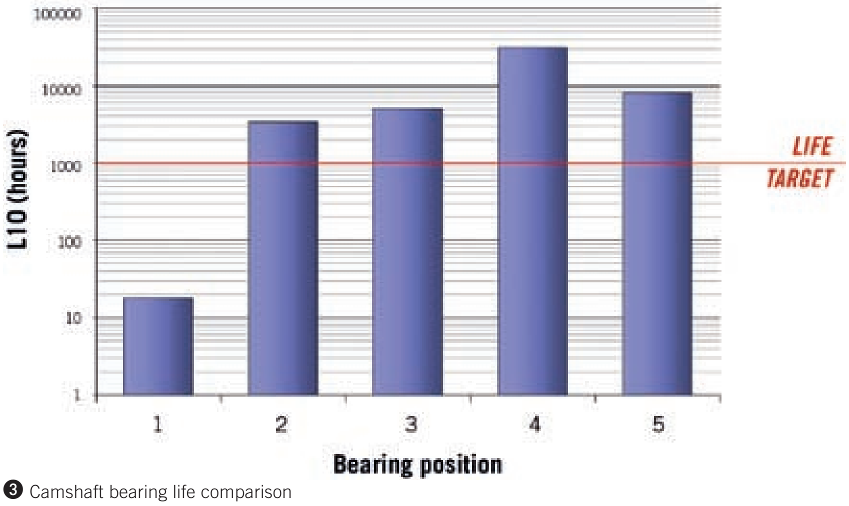
: cylinder head and shaft geometry & material
: operating conditions like speed (duty cycle), lubrication or temperature
: shaft loads (like cam lobes or chain) as a function of the time (separate dynamic valve train analysis can also be performed in parallel).

The low friction camshaft concept is a combination of bearing and camshaft expertise. In order to leverage the design and process capabilities of both technologies, the design analysis system has been split into sub-systems for easy comparison.

Different bearing designs can be considered to support radial load, but needle roller bearing using drawn outer ring and cage have been a preliminary choice. Even if “split or two-pieces” outer ring (Rodliner concept) could be used, it has been decided to take the advantage of the assembled camshaft technology and therefore to use a “single or one-piece” outer ring that offers better bearing properties and reduces customer assembly constraints.

It is also important to highlight that the camshaft is used as the inner race of the bearing and that a plurality of specific outer race and needle profiles can be developed to support to the operating conditions of each bearing positions.

In order to satisfy bearing requirements, the shaft (which is the inner race of the bearing) is made out of high carbon steel. Heat treatment and surface finish specifications are adjusted to the needle bearing needs. Baseline for the different applications is the standard Mahle assembled camshaft. Therefore two design variants were considered. One is the drop-in solution with no changes on the cylinder head which cause to use a solid shaft. On the other side the use of the same tube dimensions with minor changes on the cylinder head.



❸ Camshaft bearing life comparison

Key dimensions of the system were validated through various calculations that take into account most of the operating conditions:
: mounted diametral clearance of the bearing
: retention of the outer ring in the cylinder head
: axial location of components on camshaft and in cylinder head.

These calculations consider operating temperature, materials, thermal expansion coefficient, geometry and tolerances, in order to secure a minimum radial clearance at low temperature (typically -30 °C) a residual press-fit of the bearing in the housing at the maximum operating temperature.

The curves in ❷ show that the bearing 1 handles significantly higher loads than the others. Therefore, the load applied on the camshaft pulley (potentially higher than the loads on cam lobes) is a key input data for the design.

As a direct consequence of the loads, a high deformation of the shaft occurs, inducing various levels of misalignment for each bearing, giving different level of bearing life.

Results shown on ❸ have been obtained by considering the same bearing definition on all the positions. Bearing 1 is clearly the weakest point of the system; therefore, it is recommended that a different bearing be used for position 1.

Alternative designs have been studied in order to achieve the life target at position 1:
: full complement bearing: no cage leading to increased number and length of needles, and therefore bearing dynamic capacity in the same enveloping dimensions
: larger bearing size: increased outer diameter and needle diameter for a higher capacity but with packaging impact
: controlled stress bearing: the introduction of a specific profile on the outer race allows balancing the internal stresses of the bearing between inner (reduced) and outer race (increased), this design solution decreases significantly the sensibility of the bearing to the misalignment and thus improves the bearing life under such conditions, ❹.

❺ shows that an acceptable bearing life could be achieved on position 1, with an appropriate combination of the above design changes.

To evaluate the dynamic behaviour of the camshaft in comparison to a Mahle standard assembled camshaft, FEA calculations of torsion stiffness, bending stiffness and natural frequency were carried out.

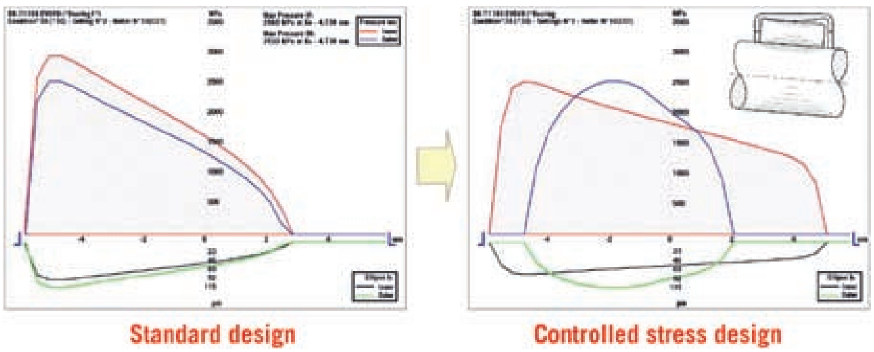
Two different variants were considered (based on general dimensions for camshafts in passenger car engines): a 18 mm solid shaft and a 24 mm tube. The shaft diameter has to be adapted to the according application, with potentially 24 mm diameter and a resulting 30 mm roller bearing for gasoline engines.

Potential oil flow benefits achieved by deleting the bearing oil bores for the LFC camshaft were simulated in advance of the camshaft tests. This simulation was carried out for several oil temperatures and radial bearing clearance values. A significant oil flow rate was found, especially in usual engine working conditions. As the LFC camshaft does not need any oil bores for the bearing lubrication at all, these results show the potential oil flow reduction of the LFC concept.

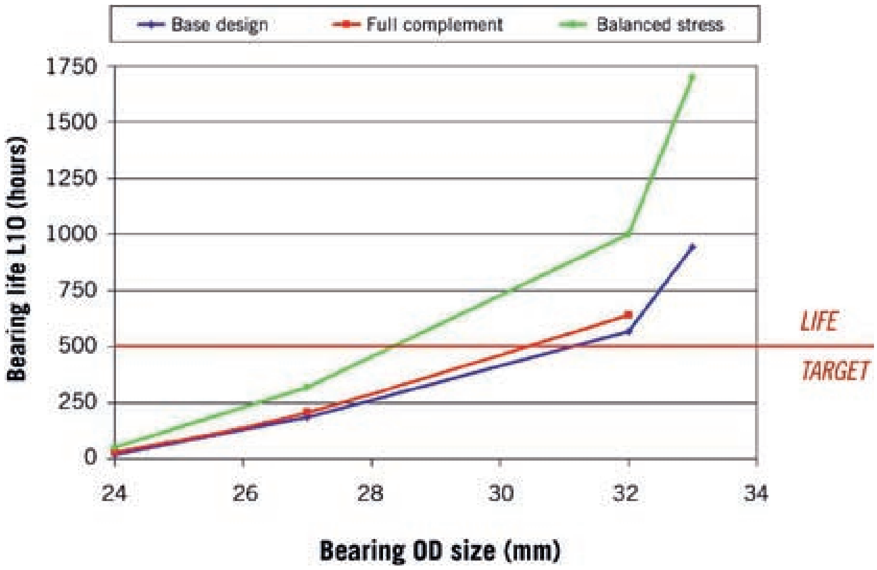
CUSTOMER IMPLANTATION

The LFC can be proposed into two different concepts. The first one is based on a camshaft fitted with bearing caps in the cylinder head as any other camshaft. The second one is a more integrated solution in the cylinder head with closed bearing support.

The "Low Friction Camshaft", ❻, is a Mahle assembled camshaft equipped with several Timken "one-piece" needle roller bearings positioned between two adjacent cam lobes. The use of an "open" outer ring, opposed to a standard drawn cup bearing, combined with a specific cage design (crenels and axial groove) improves bearing lubrication in oil mist condition. This reduces the need for direct



❹ Controlled stress concept



❺ Bearing 1 life improvement

lubrication with potentially contaminated lubricant.

The cage can be symmetrical or asymmetrical in order to fit the design envelope. The outer ring may be machined for additional needs, instead of deep drawn, a process chosen primarily for cost reasons. Machining may be considered for non-standard housing shape and very tight tolerances.

As this concept is a "drop-in" solution (with the advantage of being assembled in the cylinder head like any other camshaft), one important function has been added to the retainer; the cage now ensures the axial positioning of the outer ring in the cylinder head during the engine assembly phase.

In order to avoid any friction issues that may occur in the operating environment, end faces of the cage include wearable bumps. The wear of the cage in these three areas allows an acceptable axial clearance in the system after few camshaft rotations. The new cage design with bearing positioning function is currently patent pending.

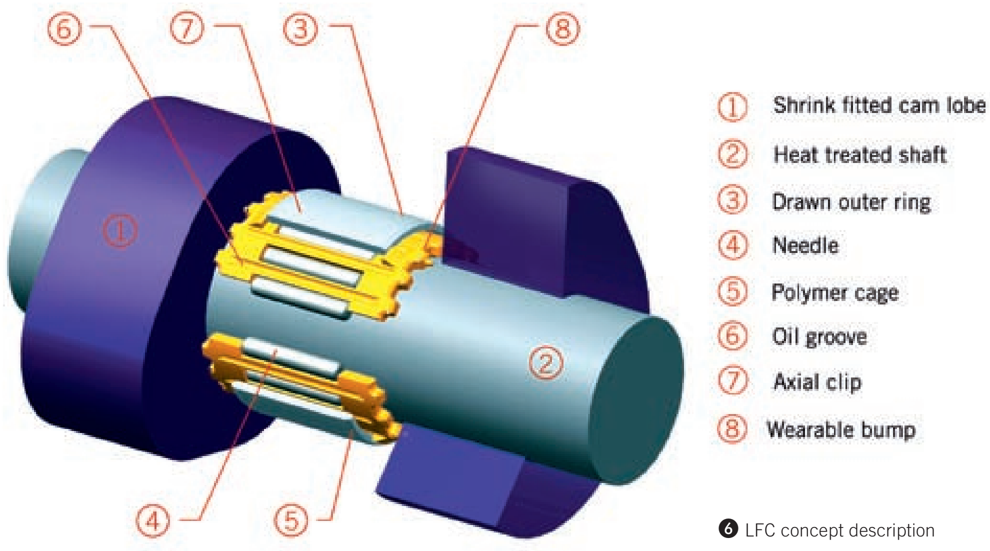
The statistical approach validates that needle rollers are always located in the right area of the outer ring for any conditions of geometry (e.g., chamfers variations, length) and assembly (initial position of the component in the cylinder head).

A next step of system integration for camshafts will be to seat the camshaft system directly on the cylinder head. Therefore the bearing support is part of the camshaft subassembly and will be mounted then on the cylinder head. The bearing support with roller bearing will be added in the assembly process of the camshaft. Benefits of this concept are higher system stiffness, reduces corresponding material input and a simplified handling for mounting process of the camshaft system on the cylinder head at the OEM production line.

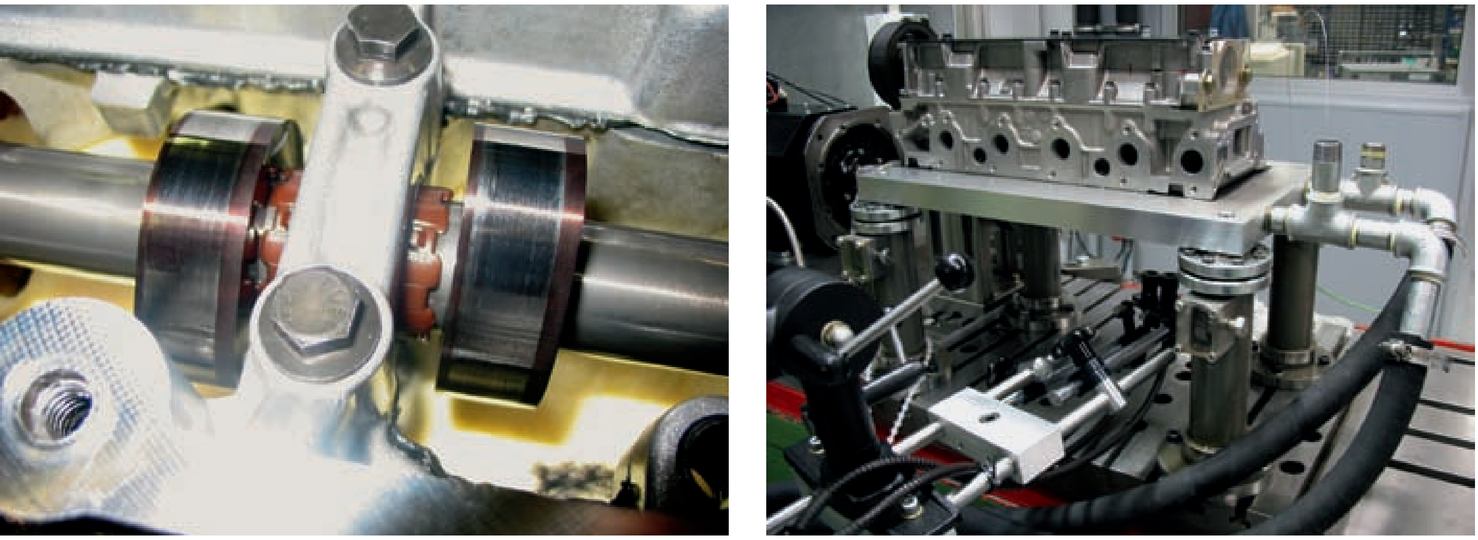
VALIDATION

In order to demonstrate the interest of the solution, a Low Friction Camshaft has been prototyped for adaptation on a four cylinder diesel engine. To validate the LFC concept, a one-to-one comparison was completed versus a standard camshaft. The complete cylinder head was run in a non-fired test, ❼. For both camshaft types, LFC and standard, the oil flow, the friction, the dynamic behaviour and the noise were investigated. A standard cylinder head was used for the tests. The bearing oil bores were closed in combination with the LFC camshaft.

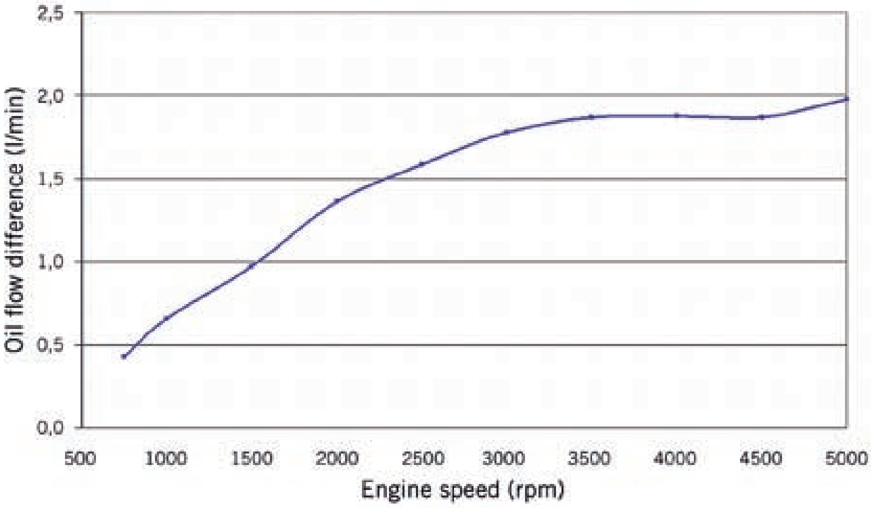
The curve in ❽ shows the difference in oil flow for oil temperature of 110 °C and a radial bearing clearance for the sliding bearings of 45 µm. For a typical clearance of 60 µm the average oil flow is about 5 l/min per camshaft.



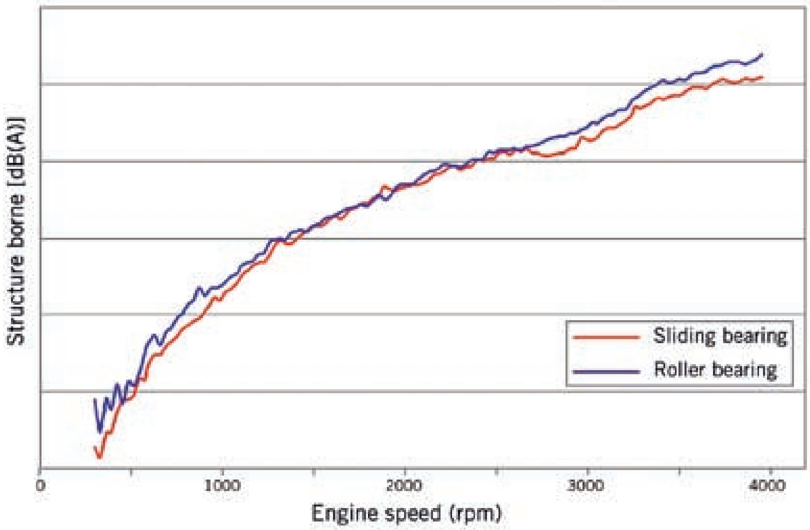
❻ LFC concept description



❼ LFC prototype and test rig



❽ Difference oil flow rate standard camshaft to LFC camshaft



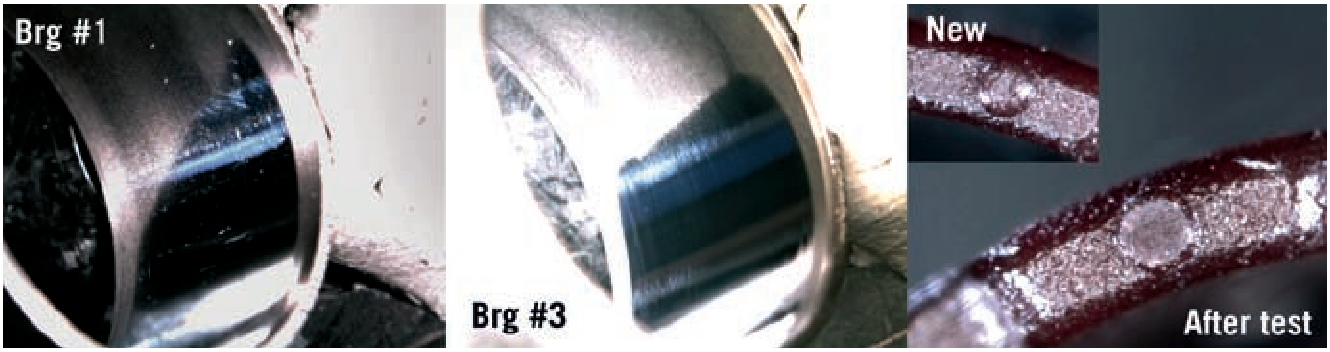
❾ Noise measurement comparison

The friction benefits of a needle bearing compared to a sliding bearing are dependent on oil viscosity. Therefore the best benefits are realized in cold conditions. At 40 °C oil temperature, the LFC camshaft had a decrease of friction up to 80 W and an average of about 20 W compared to the standard camshaft.

As shown in ❾, the roller bearing camshaft has a slightly higher noise level than the plain bearing camshaft. But in subjective impression the roller bearing camshaft sounds similar because of the higher low-frequency noise contribution.

Electrically motored cylinder-head test rigs have been used to validate the durability of the LFC concept. Several durability tests (up to 1000h at full engine speed) have been done with various lubrication conditions at high temperature (clean low viscosity oil and highly contaminated oil with five percent carbon particles) in order to operate in the most severe conditions. Further to these durability tests, and despite their high severity, the behavior of the LFC system is satisfactory and demonstrates that the concept is viable in its current definition.

The camshaft (inner race of the bearing) presents only normal polished areas and the bearing components are also fully functional at the end of the tests. On the bearing outer rings, local polishing can be observed in the upper position (according to the load direction) and reveals also the different level of misalignment of each bearing. Position 1 (close to the belt) is



❿ Outer race and cage after durability test

the most affected by the misalignment, ❿, contrary to the other positions where the polished area is quite rectangular.

In the same way, and as expected, most of the polymer cage have been worn on one side, demonstrating that the wearable bumps play their role to compensate the relative motion of the components in the cylinder head and to maintain an acceptable level of clearance to limit the internal friction. In addition, those tests confirm that a LFC with needle roller bearing works well in an oil mist lubrication condition.

However, this does not validate the performance of bearing 1 on the camshaft, which was identified as the weakest point. A fired engine test must be formally requested to complete the product validation.

SUMMARY AND OUTLOOK

The LFC technology is the combination of Mahle's camshaft and Timken's roller bearing experience. According to customer's demand, different solutions can be developed to integrate a low friction camshaft solution in the cylinder head: If it is requested to minimize the cylinder head modifications and engine assembly process, the LFC and drop-in concept is probably the most appropriate solution. If customer wishes a higher level of integration of the low friction camshaft in the engine head (new engine family development for example), specific bearing support solutions can be designed.

The technology is close to market release and shows benefits in the reduction of inner engine losses. Therefore, this new design helps meeting CO_2 targets of modern combustion engines.

According to the results of this study, the friction losses for a small four-cylinder engine can be reduced per camshaft up to 80 W (cold oil), about 20 W in average. The oil flow of the oil pump can be reduced up to 5 l/min (hot oil) which is equivalent to about 45 W.

The potential impact on CO_2 emissions for a four-cylinder engine is estimated to be at least 0.5 % to 1 %, with a potential of 1.5 % for V6 DOHC engines. However these values need to be confirmed through fired engine and vehicle testing.

Regarding the penalties for exceeding the CO_2-limits as from 2012 the LFC-technique is an appropriate instrument to decrease the CO_2 emissions with minimum investment. Even for the first step of 1 g/km CO_2 emission the on-cost for LFC can amortize the penalty.

Cost studies have been made for two different shaft sizes and for production volumes between 50,000 and one million pieces per year. According to these studies, the cost-to-benefit ratio (in € per percentage of CO_2) appears to be extremely competitive, versus other technologies such as turbo charging, variable valve lift and variable valve timing.

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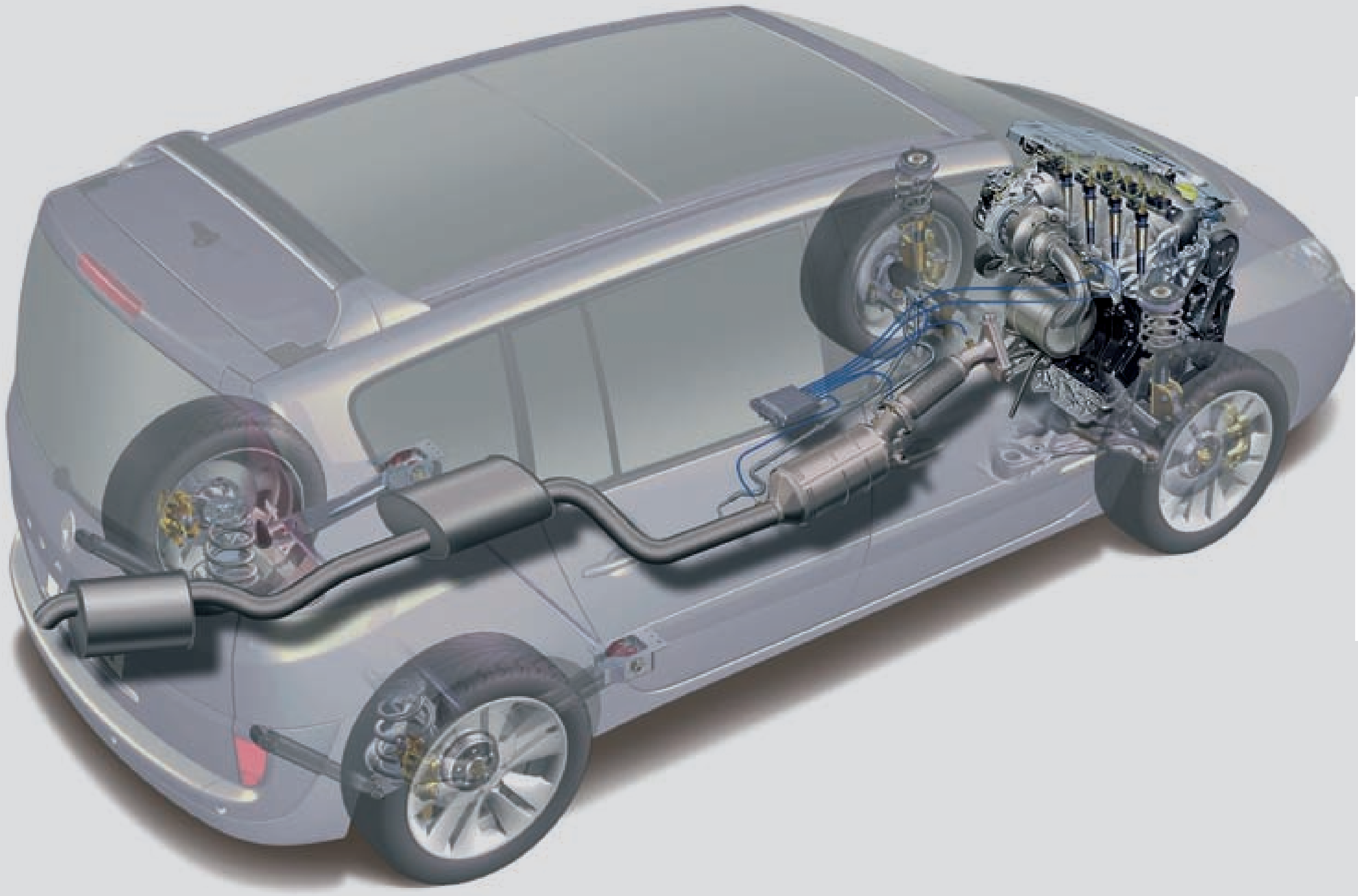

DEVELOPMENT OF NO_x AFTERTREATMENT FOR RENAULT DIESEL ENGINES

The achievement of future emission standards for diesel engines requires the use of NO_x aftertreatment systems, in addition to engine internal improvements. The NO_x trap system, developed by Renault, enables heavy vehicles like minivans to reach future emission standards and maintain the diesel CO_2 advantage. Equipped with the NO_x trap, the Espace dCi 175 complies with Euro 5 and looks even further ahead with a technology that is ready for Euro 6.

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SYSTEM OPTIMIZATION

Since several years, tremendous efforts have been put into research and development by the automotive industry, in order to improve diesel engines characteristics, while maintaining or even improving the well-known CO_2 advantage of compression ignition engines. The challenge is now mainly on diesel engine exhaust emissions reduction in view of future emission legislations. Engine measures, like combustion and injection system improvement, EGR system or turbocharger optimization are definitely a necessary step. However, the achievement of future emission standards does require the use of exhaust gas after-treatment systems, in addition to engine internal improvements. The Diesel Particulate Filter is now widely applied to diesel engines. Nitrogen oxide (NO_x) emissions reduction by exhaust gas aftertreatment, and the complete system optimization still remain the most challenging issues.

The NO_x storage catalysts (NSC) applied to diesel engines have been launched in series production only very recently under different names [1-4]. Although NO_x adsorber application for diesel engines emission control is very challenging, Renault have decided to develop the complete NO_x trap system and apply it to a diesel engine, in order to anticipate the Euro 5 emission standard and be ready for Euro 6. This paper summarizes the main features of the system, the results achieved with respect to exhaust emissions and customer requirements, and finally points out the advantages of this technology.

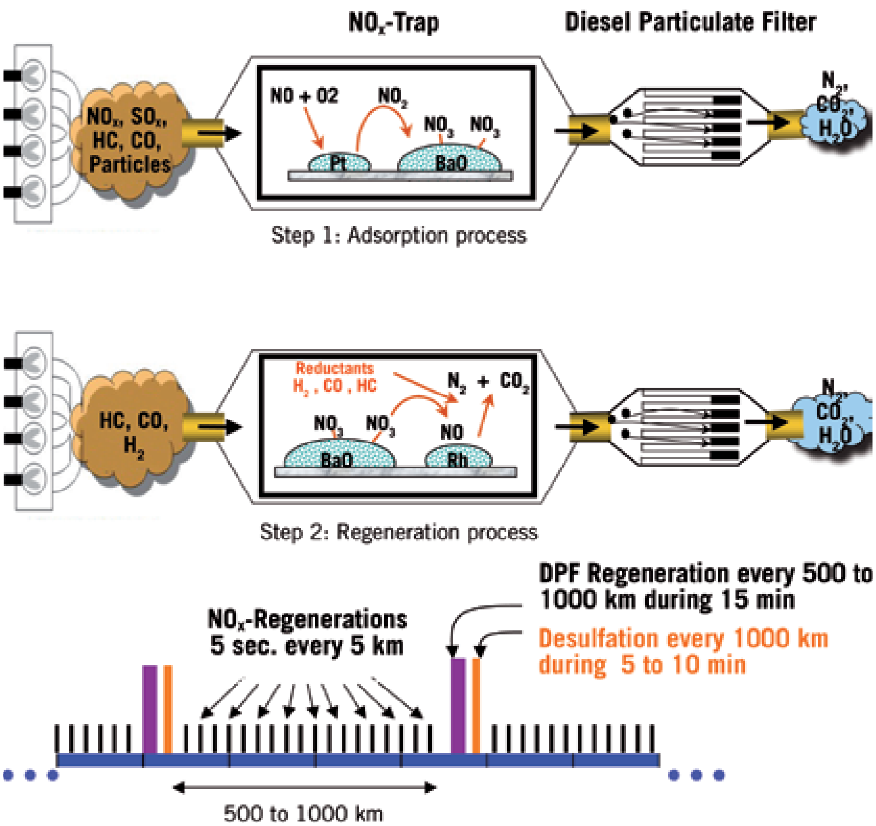
NO_x TRAP: OPERATION

The NO_x trap material has three basic functions:

: It continuously treats hydrocarbons (HC) and carbon monoxide (CO).
: It stores nitrogen oxides.
: It transforms nitrogen oxides into harmless componets.

In order to reduce NO_x emissions, the NO_x trap system works according to a two step process, ❶.

During step one, the nitrogen oxide emissions coming out from the engine are getting stored onto the NO_x trap. The platinum converts nitric oxide (NO) into



❶ NO_x trap principle and operation

nitrogen dioxide (NO_2). The barium, which is oxidized into barium oxide, traps and holds NO_2 as part of an aqueous barium nitrate solution ($Ba(NO_3)_2$).

During step two, the engine runs rich, thus producing high amounts of reducing components able to react with the NO_x that have been stored during step one. The NO_x emissions are then converted into harmless components like nitrogen (N_2), while reducing components are oxidized into water (H_2O) and carbon dioxide (CO_2). To ensure a high level of NO_x reduction, the NO_x trap has then to be managed alternatively between step one and step two. The duration of step one significantly exceeds the duration of step two, ①. The storage phase lasts during about 10 minutes, respectively 10 km, and the NO_x regeneration ($DeNO_x$) event is a five-second process that cannot be noticed in vehicle.

SULPHUR POISONING

Sulphur poisoning has a big influence on the NO_x trap activity. The mechanism for sulphur poisoning is assumed to be similar to the mechanism of NO_x adsorption. Sulphur dioxide (SO_2) emitted by the engine is oxidized to sulphur trioxide (SO_3) on platinum site and then reacts with wash-coat components to form sulphate species on the trapping material. Desorbing these sulphate species is much more difficult than desorbing nitrate species and is only possible if high temperature levels are coupled with rich mixture conditions. The sulphur impact is considered as a limitation in NO_x trap operation, since it has a great impact on NO_x storage capacity. Taking into account the sulphur coming from a 10 ppm sulphur fuel and from the low SAPS engine oil consumption, 2 g of sulphur are stored in the catalyst after 2000 km.

Desulphation events occur every 1000 km and are generally phased with diesel particulate filter (DPF) regenerations to get the benefit of high temperature, ①. Rich condition coupled with high temperature levels of around 600 °C are required to ensure desorption of sulphur species. Sulphur can be desorbed as SO_2 but also as H_2S and COS. It is possible to enhance SO_2 formation during the $DeSO_x$ process by applying a good management of air-fuel ratio and temperature. To protect the catalyst from a too high level of temperature, it is also necessary to periodically switch back to lean condition, as described below in the paragraph "NO_x trap control and management strategy".

ENGINE AND VEHICLE DATA

The NO_x trap system has been developed and applied to the Espace equipped with a 2.0-dCi-175-engine. This application is the high power version of the 2.0-dCi-engine, which complies in its previous configuration with underfloor DPF to Euro 4 emission standards. The base engine for the NO_x trap system development is the improved version of the 2.0-dCi-engine without NO_x aftertreatment, originally developed for the Laguna III. ❷ summarises the engine and vehicle characteristics.

To achieve further NO_x reduction, the exhaust gas aftertreatment system has been modified compared to the Euro 4 technical definition. In addition, in order to control the complete system and to give relevant information to the engine electronic control unit, a significant number of sensors have been added. ❸ presents the exhaust line layout in the vehicle.

Vehicle certification requires to take into account a margin in order to compensate for engine drift and dispersion, and some NO_x penalty that are induced by active regeneration events.

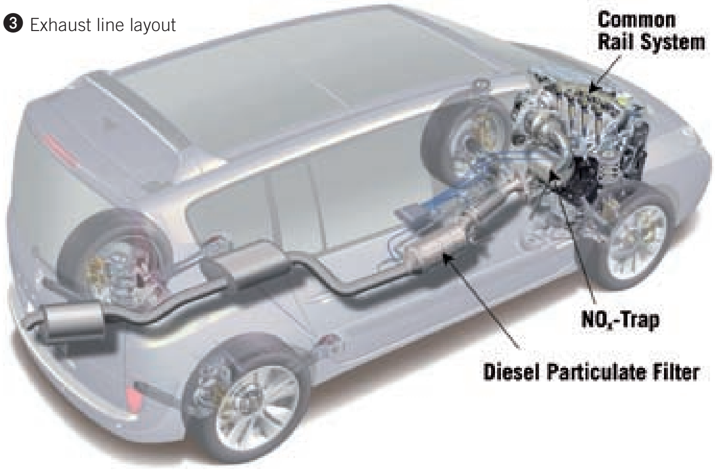
Thermal ageing has a very big impact on NO_x trap efficiency. The NO_x storage capacity is drastically reduced with ageing temperatures higher than 750 °C. Therefore the catalyst ageing must be taken into account to define the engine tuning objective. Tests are realized with aged material so as to satisfy legislation until 160,000 km.

Since sulphur is always stored in the NO_x trap and considering that sulphur has a great impact on NO_x storage capacity, the mass of stored sulphur has to be controlled. Specific tests are done to evaluate the right sulphur level for $DeSO_x$ events and to ensure that the stored sulphur does not exceed the threshold to satisfy legislation requirements. The mass of sulphur that has to be considered to cope with legislation is 3 g. It is thus necessary to adjust the mass of sulphur below this limit and usually, the stored sulphur mass is regulated around 2 g.

CYLINDER CONFIGURATION	In line 4
DISPLACEMENT	2.0 l
BORE	84 mm
STROKE	90 mm
NUMBER OF VALVES	16
COMPRESSION RATIO	16:1
FUEL INJECTION SYSTEM	Common rail system, Max. Pressure 1600 bar
INJECTOR FLOW	450 cm^3/100bar/30sec, 6 holes
EGR	Water-cooled, electrically controlled valve, external by-pass
TURBOCHARGER	VNT
MAX POWER MAX TORQUE	127 kW at 3750 rpm 360 Nm at 1750 rpm

VEHICLE TYPE	Espace 
TRANSMISSION	Manual, 6 gears
SCX	0,913
INERTIA	1930 kg (Espace), 2040 kg (Grand Espace)
TIRES	225/60 R 16

❷ Engine and vehicle characteristics



❸ Exhaust line layout

NO$_x$ TRAP CONTROL AND MANAGEMENT STRATEGY

As mentioned above, the NO_x trap is a periodic aftertreatment system requiring lambda 1 ($\lambda \geq 1$) combustion modes to achieve catalyst purges. The basic requirements for $\lambda \geq 1$ are:

: no customer perception (drivability, noise)
: minimum HC, CO, CO_2 emission and oil dilution penalty
: the consideration of engine mechanical specifications.

To cope with these difficulties it is needed:

: to manage both air mass flow and EGR rate independently
: to use a specific multi-injection diagram (timing and quantities)
: to regulate the first two values to take into account scattering and drift of the system
: to manage transitions between lean and rich mode
: to use a torque structure to take into account these new combustion modes.

Thus a new air system software has been designed, based on a model based air system, in order to be able to control independently air mass flow and EGR rate and their trajectories during transition. The turbocharger regulation has been adapted with specific tuning parameters for lambda 1 mode. Additionally, new injection combustion modes have been added to cope with DeNO$_x$ lambda 1 and desulphation (DeSO$_x$) lambda 1 requirements.

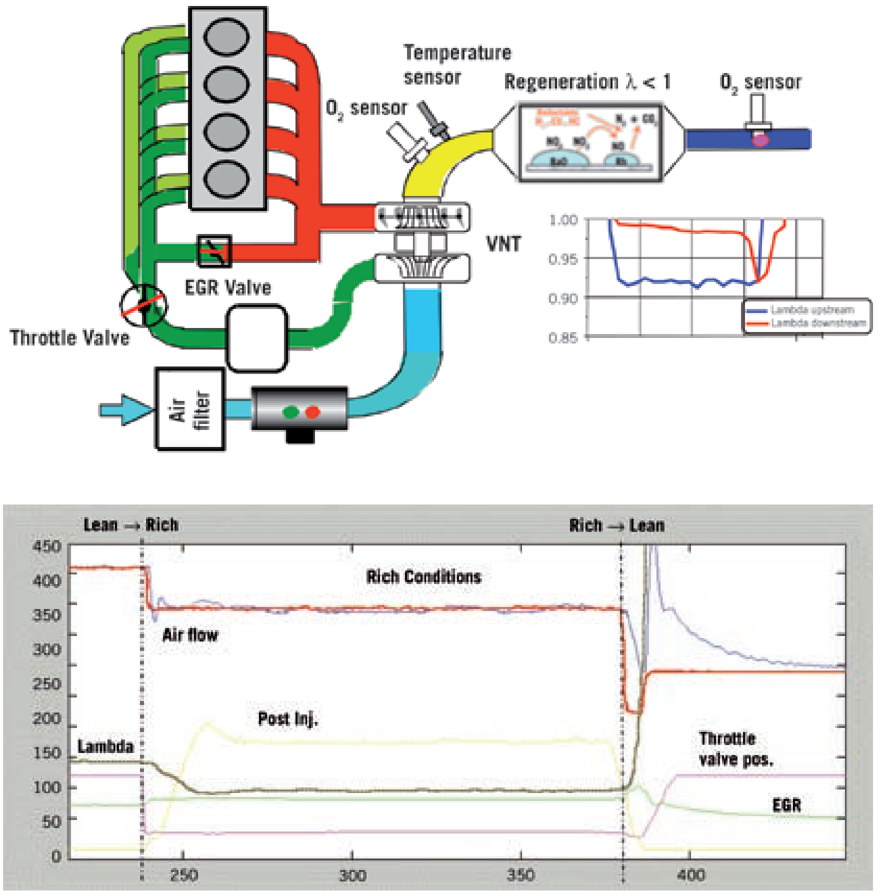
A Specific lambda 1 torque structure has been implemented to take into account new dynamic torques losses due to throttling and post-injection torque efficiency. Furthermore a new post-injection quantity regulation dedicated to lambda 1 mode has been introduced based on the H-Infinity concept and upstream NSC Universal Exhaust Gas Oxygen (UEGO) sensor.

A new transition structure has also been developed to achieve the complete task in less than 1 s. The main difficulty is to take into account the differences of response time between the injection system and the air system.

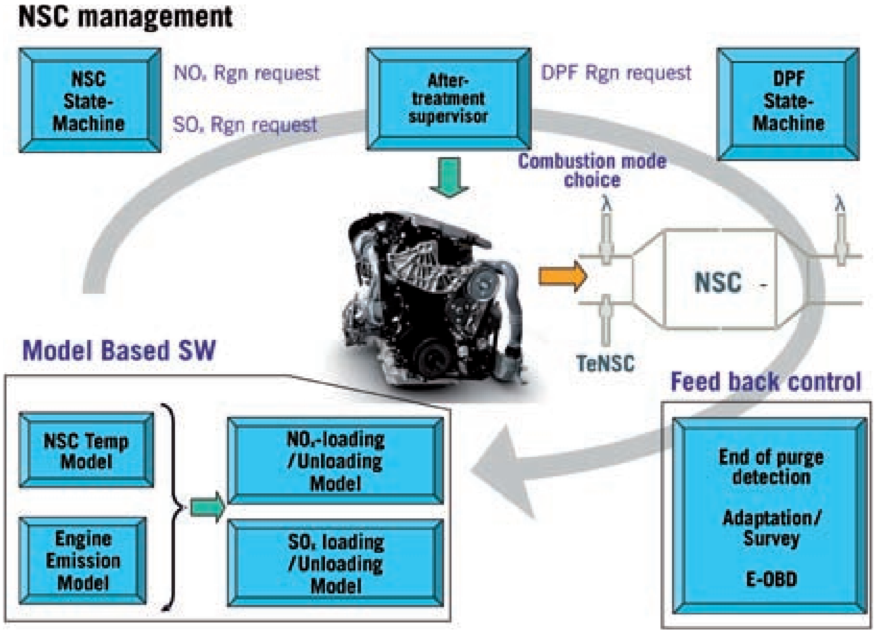
The behaviour of the system during transition from normal injection mode to Lambda 1 mode is controlled by decreasing the air mass and increasing the injection quantity, ❹. Throttle is actuated to reduce fresh air mass flow, while at the same time the EGR valve position is adapted to maintain the right EGR rate. Additionally the position of turbocharger nozzle is adapted. When the air system transition has begun, injection transition is started to compensate for torque losses, until lambda 1 is reached. Lambda 1 is then regulated.

AFTERTREATMENT SOFTWARE

The development of the NO_x trap management strategy has been driven by two



❹ Schematic of engine control variations during transition from normal to lambda 1 mode, and behaviour of main control values



❺ Overview of software architecture

main issues, the time when the engine control unit decides to order a NO_x regeneration process and the time when the NO_x regeneration is stopped. A too early or too late NO_x regeneration process would respectively lead to either extra fuel penalty, or NO_x emissions exceeding legislation. A too long regeneration step would generate extra fuel penalty and lead to unconverted reductant emissions. Therefore the software architecture, ❺, has three essential functionalities:

- a software for NO_x emission and NO_x storage estimation
- a block for rich spike ordering
- a protocol for accurate detection of end of purge.

To estimate the mass of NO_x that has been stored onto the NO_x trap, two models are running in parallel. The first one is an engine-out NO_x emission prediction model, which gives an instantaneous estimation of the NO_x flow rate exhausting from the engine. The second one is a NO_x storage capacity estimation counter, that takes into account the mass of NO_x already stored, the output of model 1, the output of the NSC temperature model, the lambda value etc. The combination of models 1 and 2 allows the system to give an accurate estimation of the mass of nitrogen oxides stored on the NO_x trap, and the amount of NO_x emissions that have got trough the aftertreatment system, ❻.

A rich spike is ordered when the estimated mass of NO_x exceeds a threshold that has been previously calibrated in the engine control unit. This limit is also modulated depending on surrounding conditions like for instance exhaust gas temperature, engine speed and load, engine coolant and ambient air temperature.

Two routes can be applied for end of purge management. The first one is a closed loop control based on signal processing of the downstream UEGO sensor. The second one is based on a NO_x unloading model depending on driving conditions. Depending on its state, the system switches between both.

The management of the desulphation process is based on the same principle than $DeNO_x$, a model based software estimating the sulphur contained onto the NO_x trap. Depending on this value and on driving conditions like for example the vehicle speed, engine rpm, NSC temperature and DPF state, a dedicated state-machine decides to start a $DeSO_x$. Therefore lean-rich toggling is managed to avoid thermal ageing of the trap, and to interrupt the $DeSO_x$ event when estimated efficiency is enough, or when driving conditions do not allow to maintain the desulphation.

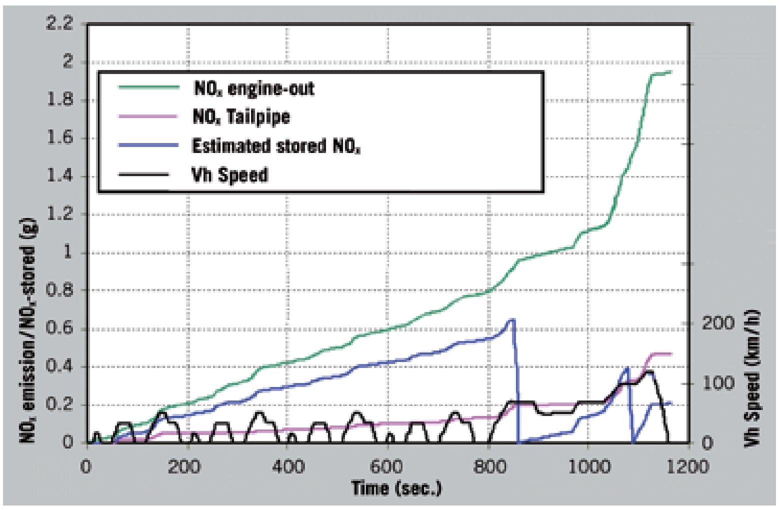
The NSC On board diagnostics module monitors the NO_x, HC and CO efficiencies during NSC purges, while diagnostic is based on the signal processing of both UEGO sensors.

Referring to the Euro-4-DPF-engine, the Euro-5-NSC-DPF-engine requires specific components to manage the NSC, ❼:

- an upstream NSC UEGO sensor
- a downstream NSC UEGO sensor
- an upstream NSC temperature sensor
- an intake manifold pressure and temperature sensor
- a Euro 4 Engine Control Unit modified to comply with these new needs.

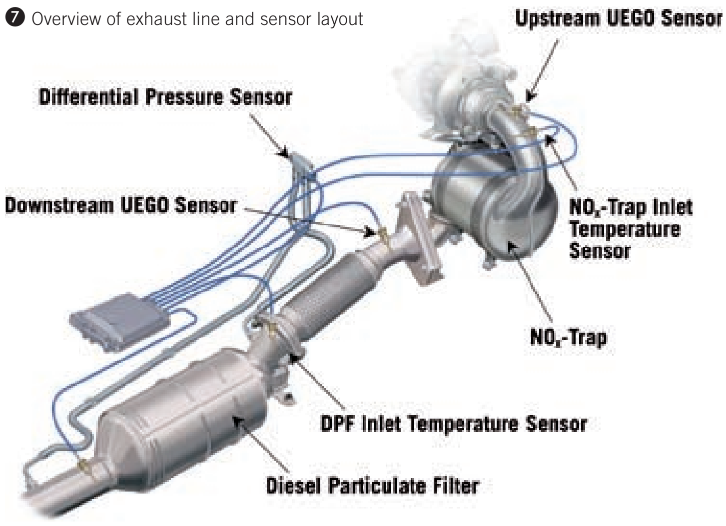
RESULTS

The results on the New European Driving Cycle (NEDC) are shown in ❽. Engine-out NO_x emissions are maintained at a low



❻ Example of NO_x trap management on New European Driving Cycle

❼ Overview of exhaust line and sensor layout



level below 0,25 g/km. The NO_x trap efficiency can be estimated between 50 and 80 %, depending on catalyst ageing and on the mass of sulphur in the trap.

The main challenge of the industrial development was to maintain all customer relevant characteristics at the same level as the Euro-4-DPF-version of the Espace 2.0-dCi. During NO_x regenerations, as fuel injected quantity is increased compared to corresponding lean conditions, and as engine efficiency is decreased (throttling), an increase of fuel consumption is induced. The longer the rich phase duration will be, the higher the increase of fuel consumption, while the duration of the rich phase is mainly driven by the mass of NO_x that has to be reduced. Considering the amount of NO_x reduction on the Espace 2.0-dCi application – roughly 0.1 g/km on NEDC cycle – an average value of 2 % could then be expected. The optimization of the engine combustion and EGR system, together with the complete tuning optimization has led to the result that no fuel penalty was observed between the NO_x trap Euro 5 version with respect to the original Euro 4 production version.

Noise control is also a key parameter in the management of a NO_x trap system. The lambda 1 calibration methodology takes into account the noise constraint by minimizing two parameters, the combustion noise difference between lean and rich modes as well as the combustion noise gradient during the transition between lean and rich modes.

At 35 km/h vehicle speed, the vehicle driver is able to hear a change in the engine noise when the combustion noise difference between lean and rich conditions becomes higher than 1 dB(A). When the vehicle speed increases, the rolling and aerodynamic noise levels become higher than the combustion noise. It is, for example, possible to accept a 2 dB(A) combustion noise difference at a vehicle speed of 60 km/h. These combustion noise constraints were taken into account for the engine tuning. Thus it was possible to obtain in all driving conditions an inaudible NO_x trap regeneration.

The big change in the combustion process has also to be managed with respect to vehicle drivability. The torque response of the engine in rich condition has to be the same as in lean condition. For that purpose a specific torque control strategy is applied.

To obtain an in cylinder rich combustion, the cylinder pressures are decreased and a post injection is applied. These two conditions enhance the impingement of the fuel spray on the cylinder walls and the mixing with the oil film. The mixture then bypasses the piston and enters the oil pan, decreasing the lubrication quality of the oil. Late injection combined with post injection is also used to regenerate the DPF, which leads to increased oil dilution. It is thus necessary to consider this constraint early in the development. After complete tuning, and by a fine optimization of the system, the service interval could be maintained at the same level as the Euro-4-version, therefore without any impact on customer regarding maintenance cost.

CONCLUSION

Thanks to its particulate filter and NO_x trap the Renault Espace 2.0 dCi complies with the Euro 5 requirement and is equipped with a technology that high-end vehicles particularly will use to meet Euro 6. Although this project has been very challenging, it has been shown that the NO_x trap system combines several advantages:

- vehicle passengers do not notice it – no impact on driving pleasure, noise, service intervals or fuel consumption, thanks to a careful and complete optimization of the system
- it is designed to equip all dCi engines in the Renault range as it replaces the current diesel oxidation catalyst
- it requires no additive systems and is entirely self-sufficient.

The NO_x trap management requires complex engine control strategies and the system cost also depends on precious metal prices. A reduction of costs for the system will be a challenge and will be addressed by the optimization of catalyst material and by the reduction of precious metal content.

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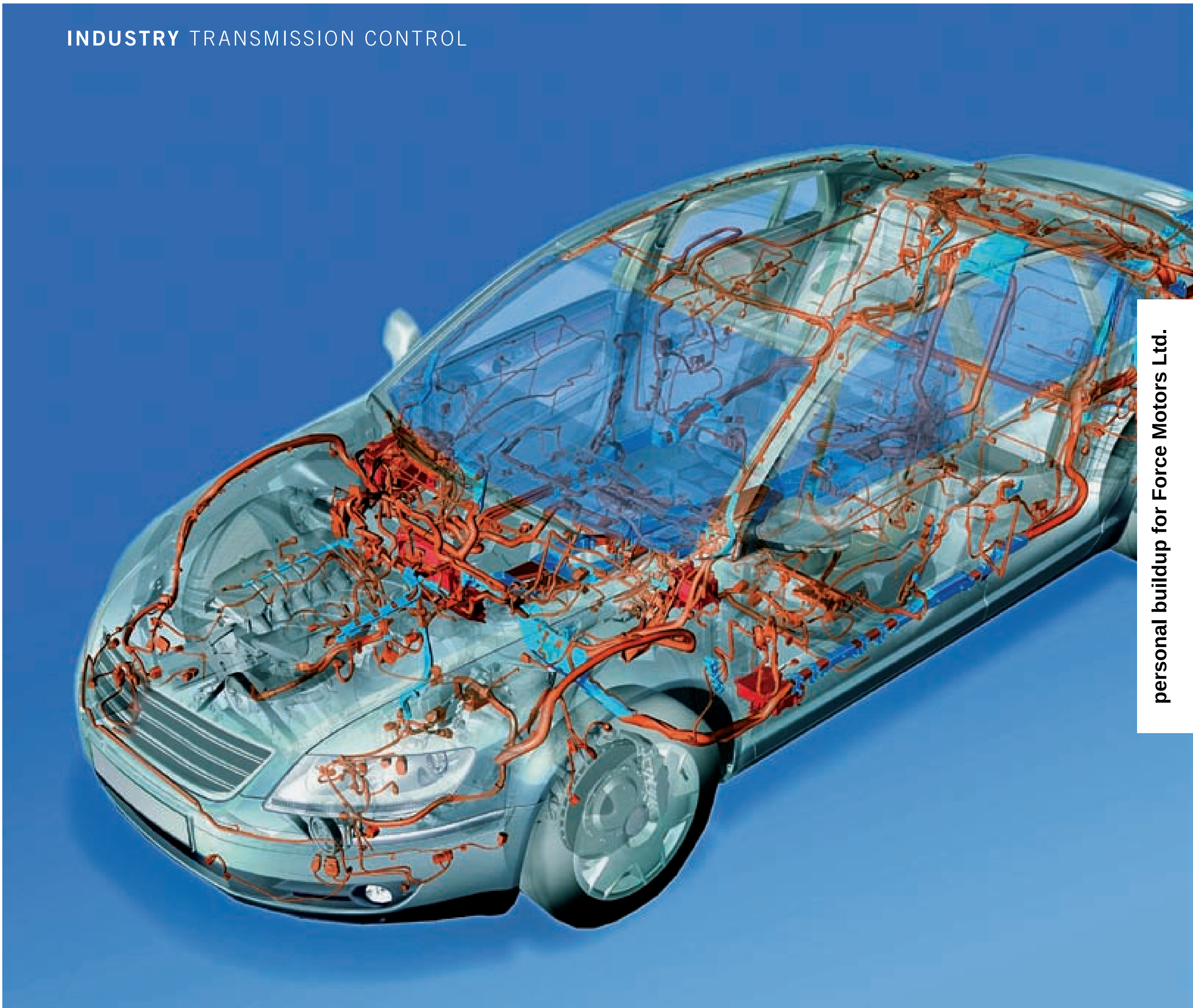
	NO_x (g/km)	HC (g/km)	CO (g/km)	PM (g/km)
ENGINE-OUT	0,24	0,47	2,25	0,05
EXHAUST	0,124 (aged NO_x trap) 0,080 (fresh NO_x trap)	0,048	0,226	0,005

❽ Emission results on New European Driving Cycle



ENGINE-TRANSMISSION INTERFACE FOR MORE COMPARABILITY AND TRANSPARENCY

The rising demands on efficiency, emissions and comfort are giving rise to complex network structures in vehicles. The challenge lies in minimizing future development effort, in minimizing possible failures as well as in the ability to test and re-use individual components. Here, IAV GmbH presents a proposal for creating a standardized, easy-to-configure communication layer that brings comparability and transparency to different systems without jeopardizing the expertise of the particular car manufacturers concerned.

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INTEGRATING HYBRID FUNCTIONS INCREASES COMPLEXITY

In 2005 [1], discussion turned to using the transmission management system as the ideal site for the hybrid master controller. The hybrid master brings together all hybrid-specific functions, such as energy management, the source of propulsion selected as well as the actuation of components like interrupting clutches and adapted transmissions.

A key function of the hybrid master is to coordinate the powertrain when changing over between systems (for example engine start-stop, drive torque vectoring). These system changeovers have a substantial influence on ride comfort. This is where integrating hybrid functions in a vehicle increases the complexity of system architecture, making the interfaces more difficult to design.

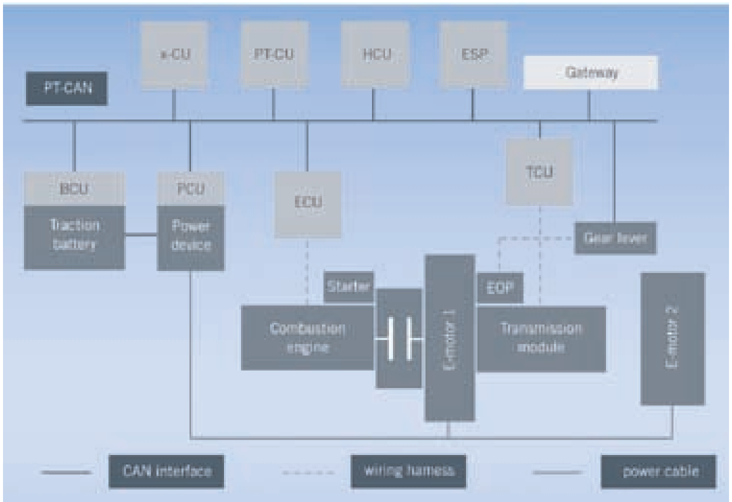
Based on IAV's hybrid experience, a proposed concept was put forward for a standardized communication layer. Using a communication layer of this type in the form of a torque interface in relation to the powertrain permits savings and simplifies development work for all of the parties involved.

It would make it easier to integrate new development concepts from Tier-1 or Tier-2 suppliers into OEM projects. This could revitalize parts of the automotive development market, also producing opportunities for small development companies with innovative ideas and solutions.

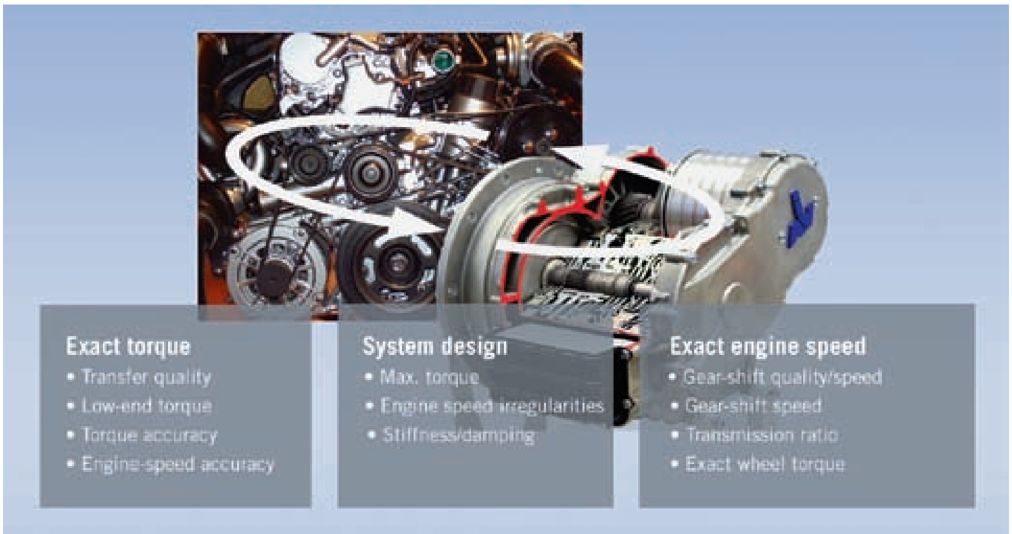
OVERVIEW OF NETWORK COMPLEXITY IN TODAY'S VEHICLES

Within the last 50 years, many control units have been integrated into the vehicle for ride comfort, safety and torque vectoring. The limits of CAN communication have meanwhile been reached, with further communication systems, such as Flexray, LIN, MOST, TT-CAN, having found their way into the vehicle.

In terms of communication, extremely high demands are placed on the powertrain CAN with its time and safety-critical functions. This is where all information on torque and power output come together.



❶ Overview of the communication structure in a hybrid powertrain



❷ Communication relationships in the powertrain

❶ shows a general communication structure of the type found today in a hybrid vehicle. In addition to the familiar control units for engine (ECU), transmission (TCU) and drive stability program (ESP), hybrid powertrains (PT-CU) also need to integrate the power electronics (PCU), battery control unit (BCU) and hybrid controller.

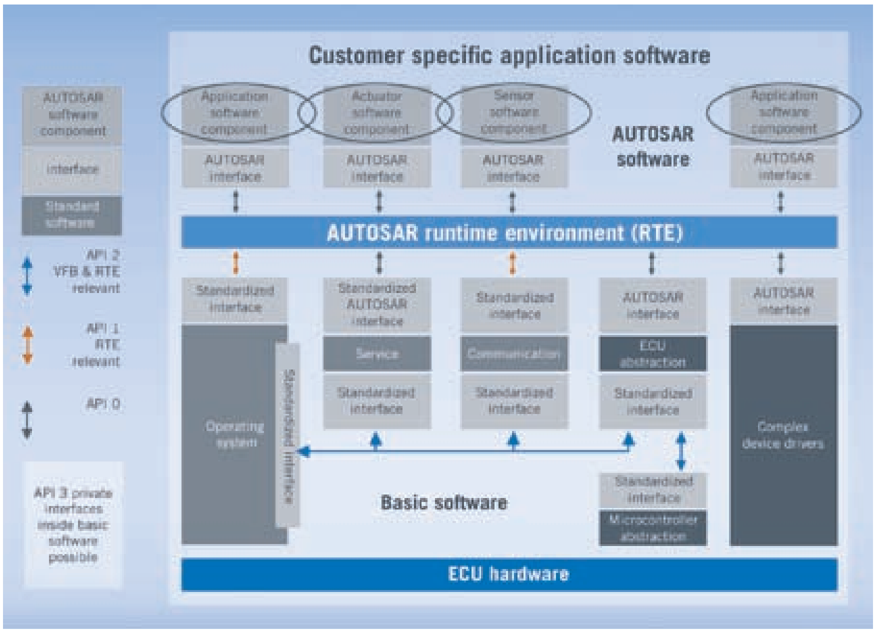
The driving forces behind a growing complexity are the ever-greater demands being made on emissions and fuel consumption, higher safety standards, consumer demands for greater comfort and, with computing capacity increasing, the ability to integrate additional functions. This also comes with numerous drawbacks: The main one being the high level of development input brought about when new systems or functions are integrated into existing systems. In many cases, the tests required for validating a system only take place at module level and, for cost reasons, often neglect the interface problems. This demands complex combined testing systems. Porting, reusing or substituting components/functions is only possible with major adaptation work. Defining a standardized communication architecture with predefined interfaces is one way of overcoming these drawbacks.

COMMUNICATION IN THE POWERTRAIN

Based on IAV's experience with various car manufacturers and suppliers, communication relationships in the powertrain can be summarized using the example of engine and transmission as presented in ❷. The aspects common to all communication concepts are defined on the basis of these relationships, with each manufacturer implementing them in its own specific way. Here, signal quality and signal timing are defined within the scope of the CAN protocol in relation to vehicle structure and powertrain type.

Essentially, communication is limited to information being exchanged on the current state of a component, the prediction of possible future states as well as the transfer of system-relevant events. In terms of the engine-transmission interface, this means communication of torques, engine speeds and transmission ratios with current values and bandwidth within the prediction. In hybrid powertrains, these are additionally joined by the following functions:

: energy management for balancing the requirements of all sub-systems (for example battery management, air-conditioning system, engine warm-up function)



❸ Proposed architecture in compliance with Autosar [2]

: torque vectoring between combustion engine(s) and electric motor(s) at times of acceleration and recuperation
: torque-vectoring function for controlling stability in all-wheel drive structures assisted by electric motor.

A standardized interface structure must be configured on the basis of these requirements. In doing so, the challenge is to convince all OEM and component suppliers to use such a standard.

A number of approaches in this context are already being discussed. For instance, there are recognized statutory requirements (standardized OBD communication protocol) or the activities of the Autosar partnership [2] that many companies already belong to today. ❸ shows an overview of the Autosar architecture.

CONCEPT FOR A STANDARDIZED TORQUE STRUCTURE

The components in the powertrain can be divided for a standardized torque structure into the following classes, also refer to ❹:

a) torque sources (TS)
b) consumption components (CC)
c) torque-conversion components (TCC)
d) coordination components (CC).

Depending on the class the particular component is allocated to, it must use a specific information structure. Applying indices, multiple-application components are easily configured and integrated on a modular basis.

Using the example of the engine control unit (ECU) as being representative of torque sources, it can be said that information is made available, such as engine speed and torque as well as their physical limits. In addition to the current values, it is also necessary to provide the drive master with information relating to the next computation stage (fast torque) and prediction time base (slow torque).

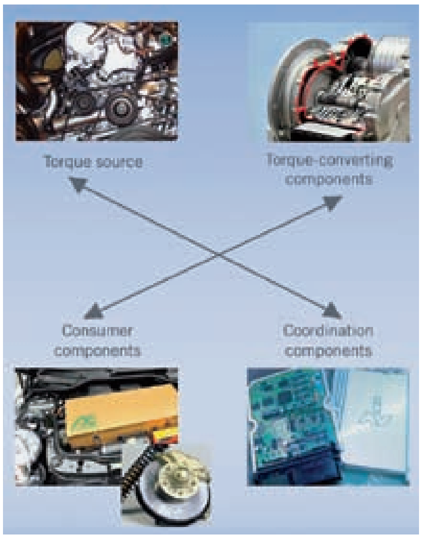
In contrast, the dynamics are often not critical in relation to electric motors. These, however, are associated with additional relationships, such as maximum permissible currents, that need to be communicated alongside the standard information, like engine speed and torque.

In the same way as the sources of torque, the power consumers must be bound to the same communication protocol. They too need information on the state of the torque source and, in turn, provide information on their own state. Compared with the energy sources, the difference in this case is that torque is always in the negative range.

By way of example, mention must be given to the generator whose load is regulated in relation to battery charge state. In hybrid powertrains, it is necessary to ensure that the electric motors produce positive torque (propulsion) and negative torque (recuperation).

The following information must be provided by the power consumers:
: current engine torque request
: maximum and minimum engine torque request
: current torque loss (absolute value)
: Maximum and minimum torque loss (in relation to working point)
: prediction of torque loss in relation to the working points requested.

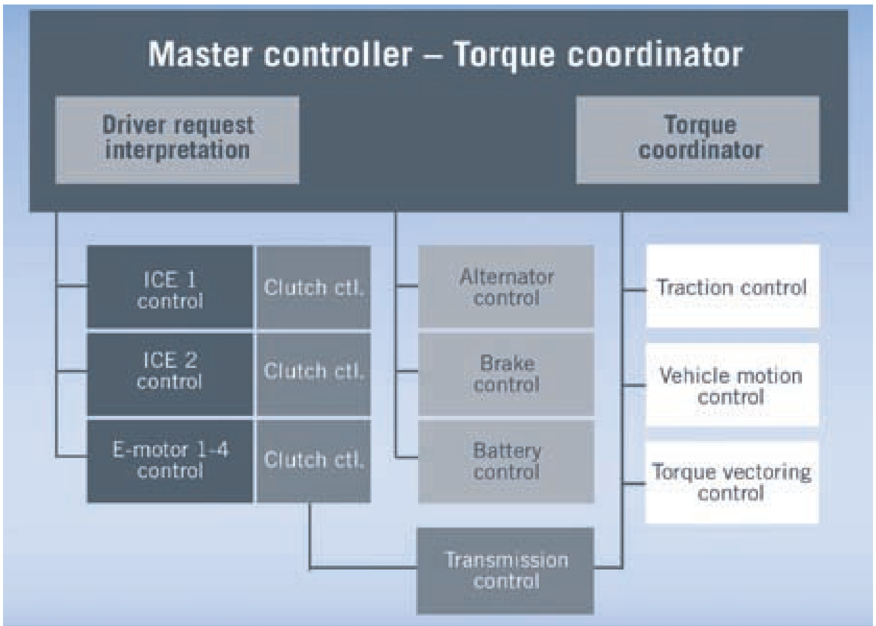
It is characteristic of torque converters to distribute propulsion power in the way requested by the driver, convert it and pass it on to the driving wheels. In addition to this, a torque-converting component must – if equipped with switching elements – communicate its state to the other components. The current drive power losses belonging to the relevant conversion stage must also be communicated to the coordinator. The information necessary for an optimum operating point



❹ Classes of components in the powertrain

(for example fuel consumption) is then transmitted via the standardized communication protocol to the coordination unit responsible.

In addition to the contents of the power-consumer communication protocols, the following converter information must also be mentioned:
: state of clutches and brakes
: state of any hydraulic converter
: gear-shift state (gear-shift system active)
: current gear (current ratio).



❺ Concept for a standardized torque structure in the powertrain

The last component is the master controller. This is where all torque-relevant information, such as current and predicted values, comes together. The driver request is interpreted here as requested torque. Based on all available information, the coordinator decides which source delivers which torque at which road speed. The torque coordinator is responsible for varying the way in which torque is split between the various sources (boosting for electric motor, steady-state torque for combustion engine). This is also where the energy balance, including battery charge-state monitoring, is generated, with it being necessary to avoid critical states. The master controller is the main module for controlling the other components. This module is typically developed by the OEM and integrated in an available powertrain control unit (ECU, TCU or HCU).

Proceeding from the described physical structure, the functional interconnection of components is configured on the basis of the given physical conditions.

: wheel-torque based interface for coordinating vehicle acceleration
: motor-torque interface for torque-generating components and power consumers
: interface for coordinating the electrical energy
: engine-speed and road-speed interface for the torque-converting components.

Timing, time base, name convention and units must be defined to satisfy a standardized communication layer and utilize the benefits to the full extent. ❺ shows a concept for torque structure.

SUMMARY AND OUTLOOK

With the complexity of vehicle systems growing all the time, development costs are exploding. In future, this effect must be kerbed. One possible way of doing this it to standardize various parts of the development process, such as communication structure in the manner described here by IAV, as a basis for permitting substitution, ease of verification and comparison. This will make it possible to improve the quality of the systems that are developed and also bring down the costs of development.

As a result, and provided it conforms to the standard interface, each component developed can be tested while still at the manufacturer. This will ensure that a component can be ported to other systems and also make it easy for the non-specialists to verify its suitability. As a result, manufacturers will be able to concentrate fully on developing the component without having to provide resources for delivering the necessary communication structure.

Components could be developed offline and then – meeting the standard – be offered on the market. One example in this context would be the master controlling function which could then be integrated or ported with ease and on a flexible basis. Under the boundary conditions described here, there is consequently no reason why the hybrid master should not be sited in the transmission control unit.

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Therefore, since the No. 4 issues of 2008, ATZ and MTZ have the status of refereed publications. The German association "WKM Wissenschaftliche Gesellschaft für Kraftfahrzeug- und Motorentechnik" supports the editors in the introduction and execution of the peer review process. The WKM has also applied to the German Research Foundation (DFG) for the magazines to be included in the "Impact Factor" (IF) list.

In the ATZ|MTZ Peer Review Process, once the 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.

LUBRICANT EFFECT ON MIXED FRICTION OF SLIDE BEARINGS

To improve the effectiveness of the lubricant effect on slide bearing mixed friction, experimental investigations with regard to tribology under mixed friction conditions were carried out within the framework of the FVV project No. 872 at the Institute of Machine Elements and Construction Technology (IMK) of Kassel University. The results investigated at con-rod bearings obtained by means of special testing technologies served to validate simulation calculations. In consideration of lubricant-specific parameters like shear- and pressure-behaviour it is possible to achieve very best simulation results in the mixed friction area.

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PEER REVIEW
RECEIVED 2009-10-09
REVIEWED 2009-11-12
ACCEPTED 2009-12-04
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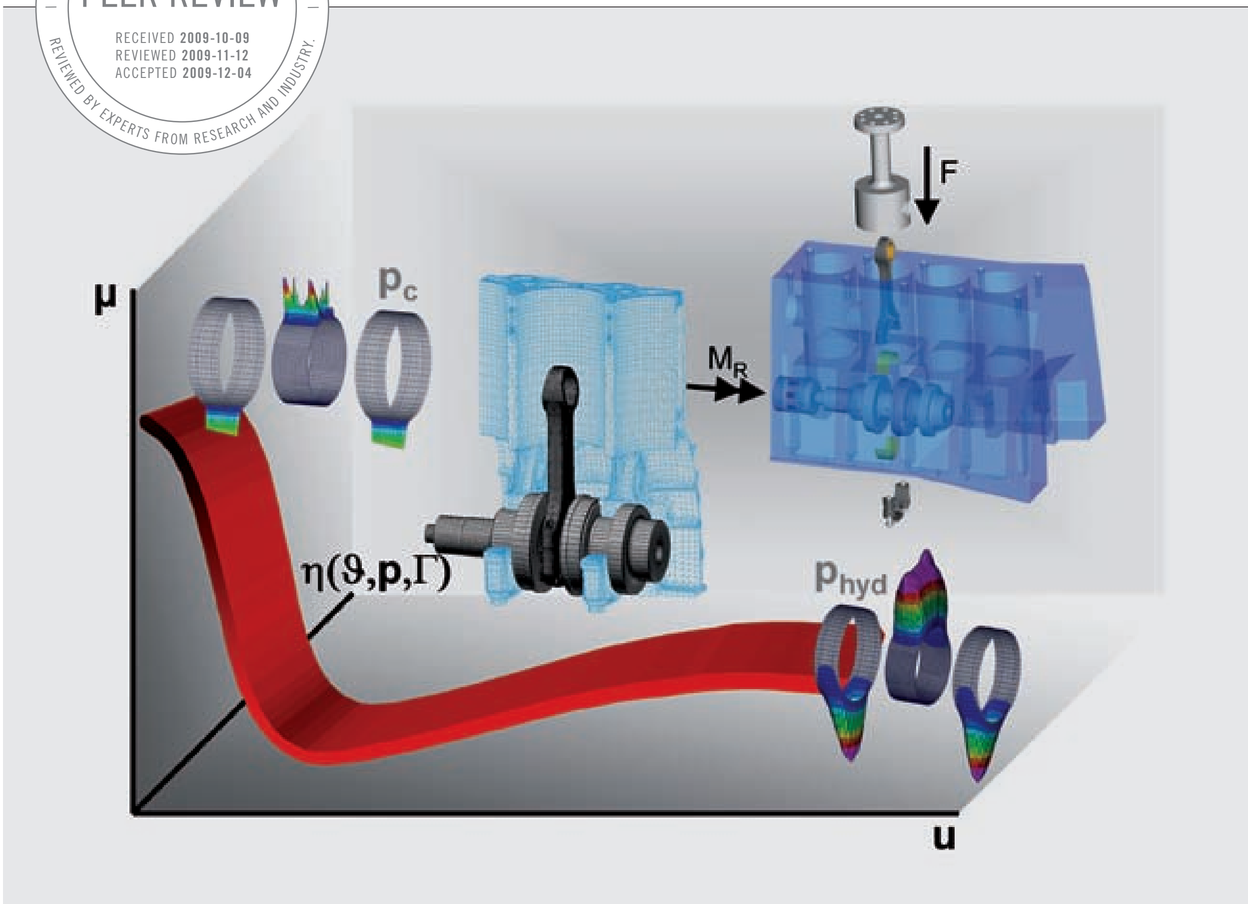

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1 Introduction
2 Testing Technology
3 Simulation Technology
4 Test Conditions
5 Results
6 Mixed Friction Load Limit

1 INTRODUCTION

The development of combustion engines has been determined for several years by the effort to minimize fuel consumption. Smaller turbocharged engines working with direct injection are being built more and more. This downsizing in connection with further strategies to reduce consumption, as for example stop/start automatics, creates an extreme increase of tribological loads of the components in the engine.

With the help of special testing technologies the Institute of Machine Elements and Construction Technology (IMK) of University Kassel investigated the effect of lubricant additives on mixed friction at con-rod bearings and main bearings in this research project. Through the use of lubricant specific values the results of simulation technology could be optimized.

2 TESTING TECHNOLOGY

An important part of the work was the development of a high-dynamic component test bed for engine slide bearings, ❶, with which the friction behaviour of con-rod bearings and main bearings can be measured integrally under engine-like load amplitudes. As component bearer for this test bed the original crank housing of a four-cylinder diesel engine is used. The double mounted uncranked testing shaft is driven by a rotation-regulated electric motor. The angle-triggered load application takes place via a high-dynamic servohydraulic linear cylinder on an original con-rod. The application of the high dynamic load amplitudes is achieved using elaborate valve technology.

The control of the test bed is carried out with the help of a newly developed closed-loop control system (Zyklo-Regler = cyclo regulator). This makes it possible for the first time to investigate con-rod bearings under real engine conditions in a rotation speed area between 20 rpm and 8000 rpm on a components test bed.

The recording of the integral friction momentum of con-rod and main bearings is done using a strain gauge (DMS) operated measuring cam in the powertrain. To investigate the effect of the oil on wear behaviour in the case of mixed friction in the con-rod bearing, a con-rod bearing casing was developed that makes the operation with two separate oil circulations for main and con-rod bearings possible, ①, bottom. This is the case for both the physical parameters of temperature, pressure and, volume flow and for the condition of the oils (fresh oil, old oil, contaminated oils). To detect mixed friction states a two-channel contact tension measuring system was used, by mean of which a separate evaluation for con-rod and main bearings is possible.

For the quantitative assessment of the contact tension signal, the contact tension parameter C* was introduced [1], that specifies the ratio of mixed friction and hydrodynamic lubricant proportion during a working cycle. By definition a hydrodynamic lubrication state is present with $K_i = 1$ (contact tension > 5 mV) and mixed friction and/or solid body contact with $K_i = 0$ (contact tension < 5 mV). The C* value is ascertained in the area of the load, it delivers the mixed friction intensity via a working cycle in the form of a value, ❷. It is ascertained according to the following Eq. 1 [1]:

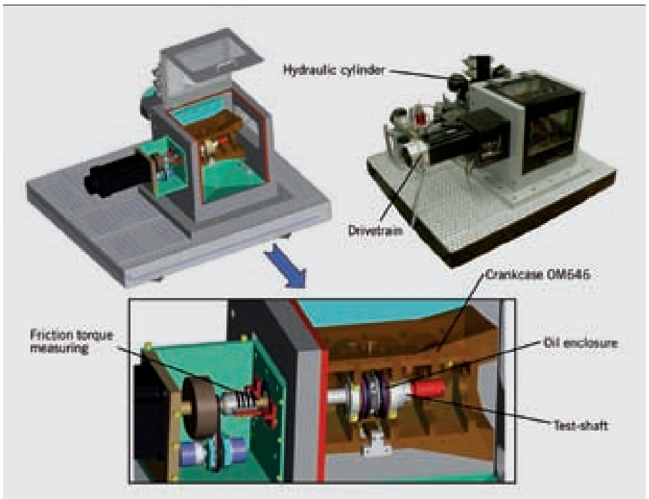
EQ. 1
$$C^* = \frac{\sum_{i=1}^{n} K_i}{n}$$

3 SIMULATION TECHNOLOGY

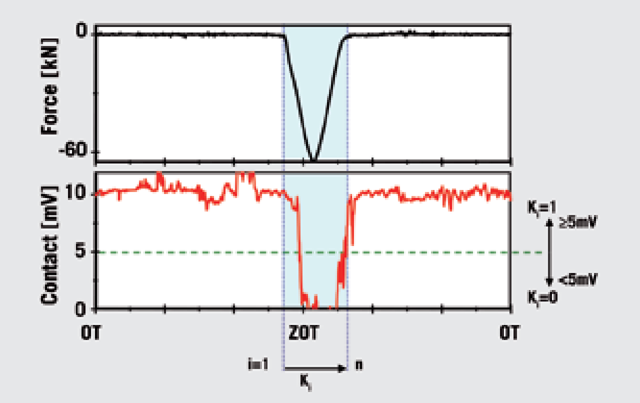
The EHD simulation calculations (elasto-hydro dynamic theory) were carried out with the program system First. This simulation tool is used to determine movement behaviour of elastic, rotating components in engine technology. To calculate the bearings reactions a special slide bearing module (Tower) was used. This module Tower and the program system First were developed at IMK in Kassel and IST Ingenieurgesellschaft für Strukturanalyse und Tribologie mbH in Aachen.

The calculations were made using an extended Reynolds differential formulae solver (DGL) to take the surface roughnesses into consideration. The recording of the effect of microstructural properties of the surfaces on the hydrodynamic support pressure behaviour and on the solid body support parts is taken into consideration in the calculation by means of contact pressure models and flow sensors. In the areas of pure hydrodynamic (high rotations speeds and low loads) and/or with large lubricant gaps surface roughness has no significant effect on the pressure build-up in the slide bearing. The lower the gap heights, the greater the effect of the surface microgeometry on the pressure build-up, in the case of solid body contacts especially on the friction momentum and contact pressure.

The calculation for the friction force was developed by Vogelpohl. It is a combination of Newton's law of shear stress and Coulomb's law of friction. By integrating local contact pressures p_c over



❶ Structure of the components test bed



❷ Ascertaining the C* value by means of the contact tension signal

area parts involved in the contact A_c and then multiplying with a solid friction value μ_c one gets the solid body friction force F_{Reib}, ❸. From the measured microstructure of the surface the characteristic diagram of contact pressure has been calculated and is now used for the determination of the friction moment as a function of the gap height. The hydrodynamic proportion of the friction moment is determined via the shearing strain in the lubricant, the proportion of the solid bodies through the use of the characteristic diagrams dependent on the contact area and the height of the lubrication gap.

The simulation technology also takes into account the viscosity under strain (by temperature, pressure and shearing). With a so-called VT-diagram the viscosity in the absence of strains is determined. Viscosity as a function of pressure is determined using a calculation following Roelands equation in Eq. 2 [4]:

EQ. 2

$$\eta = \eta_0 \cdot e^{\left[\frac{\alpha \cdot kp}{z}\left(-1 + \left(1 + \frac{p}{kp}\right)\right)^z\right]}$$

Factor k_p is a constant pressure of approximately 1980 bar. Factor z is the pressure viscosity index and η_0 the viscosity under ambient pressure. The loss of viscosity due to shearing is determined using a calculation by Carreau presented in Eq. 3:

EQ. 3

$$\eta = \eta_0 \left(1 + (B \cdot \Gamma)^2\right)^{(C-1)/2}$$

The values B and C are Carreau's coefficients and Γ is the shear rate. Carreau's coefficient C indicates the gradient of the log η line. Coefficient B is the reciprocal value of shear strain at the transition from Newtonian behaviour to shear thinning. In order to take the loss of viscosity into account that results from shear strain the knowledge of so-called HTHS values is used in such a way that with variation of parameters lubricant specific coefficients are used whose course intersects with the relevant HTHS value. ❹ shows six different courses, three belonging to lubricants V0, V3.1 und V3.2. and three belonging to lubricant V1.1. Carreau's pair of coefficients B and C are abbreviated as Ci (i = 1, 2, 3) and C_{11}i (type V1.1).

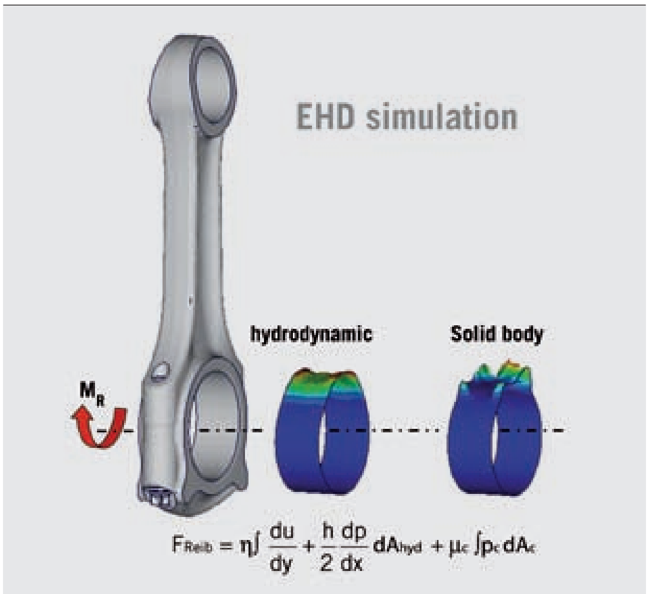
4 TEST CONDITIONS

For experiments defined lubricants and materials were used. These and the test program "Friction Behaviour at Lower Mixed Friction Stribeck" are explained in the following more in detail.

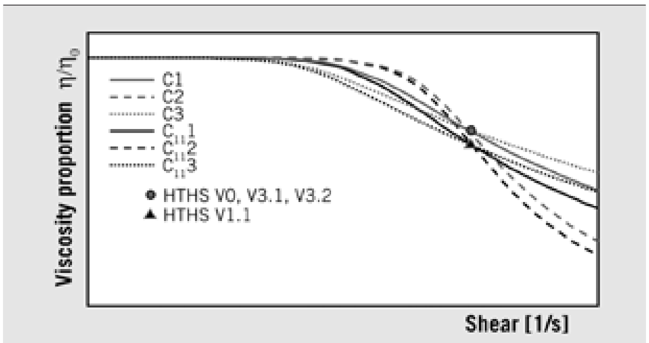
4.1 LUBRICANTS AND MATERIALS

The basis of the different lubricant variations is a reference oil with known chemical composition (reference oil type V0). Its formulation was chosen in such a way that it is possible to vary different individual properties of the oil, without changing the base oils. Staring with the reference oil the following variations, ❺, were investigated:

- Type 1.1: With this type the HTHS value was lowered to 2.6. The base oil was exchanged to yield a viscosity class of a SAE20 oil. The ratio of the base oil classes groups III and VI was kept constant. The viscosity index (VI) was also kept to an identical value of 161 by reduction of the VI improver.
- Type 3.1: The basis of this type is also the reference oil. The friction modifier (FM) of the reference oil was replaced by an ash-free oil whose mode of operation is on the physical level.



❸ Calculation of friction parts in simulation technology



❹ Viscosity loss as a result of shear strain

: Type 3.2: This type differs from the reference oil equally only with the changed friction modifier (on a molybdenum basis). The basis of the mode of operation here is on the chemical level.

The lubricants were provided by Fuchs Europe Schmierstoffe GmbH.

With respect to the materials choice a normal material combination in the automobile industry was chosen. The combination of a steel shaft (42CrMo4 nitrated) with a sputter bearing (Kolbenschmidt KS S30S) was used.

4.2 TEST PROGRAM "FRICTION BEHAVIOUR AT LOWER MIXED FRICTION STRIBECK"

In order to ascertain the lubricant effect on deep mixed friction various rotation speeds are driven at constant oil inlet temperature and constant load amplitude. To create lubricant characteristic friction momentum runs the total friction momentum of basic and con-rod bearings of the individual rotation speed levels are recorded. The rotation speed band stretches from a maximum of 1500 rpm to a minimum at 20 rpm. In the upper rotation speed areas the individual driven rotation speeds are nowhere near each other. With falling rotation speed – especially in the area of deep mixed friction – a close gradation of the rotation steps is chosen.

The evaluation and transfer to a lubricant-characteristic in the form of measuring data are shown in ❻. The left side shows the single measurements of the load applied (top diagram) and that of the friction momentum (bottom diagram). This is true for a working cycle at 100 rpm of six consecutive test series. For a better overview and comparability of the individual measurements the measuring curves were phase-shifted against each other. Selected representative friction momentum maximums of the relevant rotation speed step of a test series are superimposed on the rotation speed (Stribeck curve), ⑥, right. Of the six test series carried out in ⑥ only the last three are used for the evaluation, because with these measurements the friction momentum is at a constant level as a result of the completion of the inlet process. As the criterion for consideration of a measurement a deviation in a maximum of three consecutive measurements of $\Delta M_{r,max} < 0{,}3$ Nm is applied. To show the lubricant characteristic Stribeck curve the arithmetic average value of the three representative measurements is used. The diagram on the right in #6 shows the averaged friction momentum curves with the permissible scatter bands superimposed on the rotation speed.

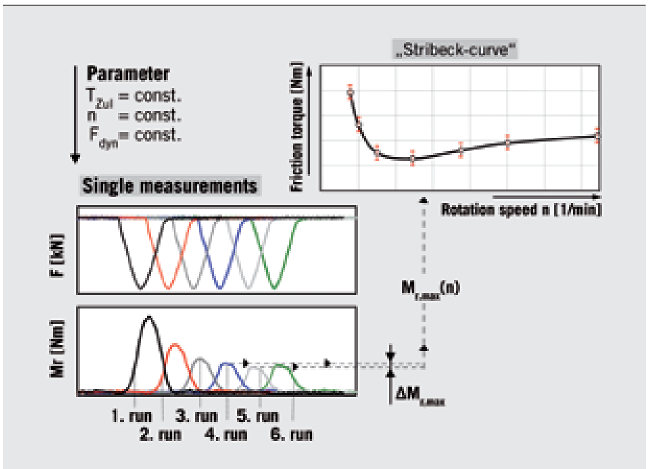
5 RESULTS

❼ shows a scatter band of the measurement results for the lubricants investigated and the corresponding simulation results with different parameters. The results are shown for the combination of the sputter bearing on the steel shaft at 100 °C oil supply temperature.

The calculations for the reference oil, ⑦, diagram top, were carried out successfully with three parameter variations. In the area of 200 to 100 rpm the simulation result is almost congruent with the parameter variations C1 and $\mu_c = 0.05$ (black curve) on the measurement. With decreasing rotation speed there is not enough friction with these parameters. In this area a more exact correlation can be achieved using the other parameter variations (grey

NAME	REFERENCE	HTHS TYPE	FM TYPES	
TYPE	V0	V1.1	V3.1	V3.2
BASE OIL	Group III+IV (a)	Group III+IV (b)	Group III+IV (a)	Group III+IV (a)
FM			FM I (phys.)	FM II (chem – Mo)
VII		Ref. VII reduced	Ref.	Ref.
ν_{40} [MM²/S]	70.6	39.8	70.7	72.1
ν_{100} [MM²/S]	11.7	7.8	11.8	12.2
VI [-]	161	161	161	166
HTHS [MPA*S]	3.5	2.6	3.5	3.5

❺ Rheological data of the test oils



❻ Testing results with single measurements and Stribeck curve

line, parameter C1 and $\mu_c = 0.06$ and dotted black line parameter C3 und $\mu_c = 0.05$).

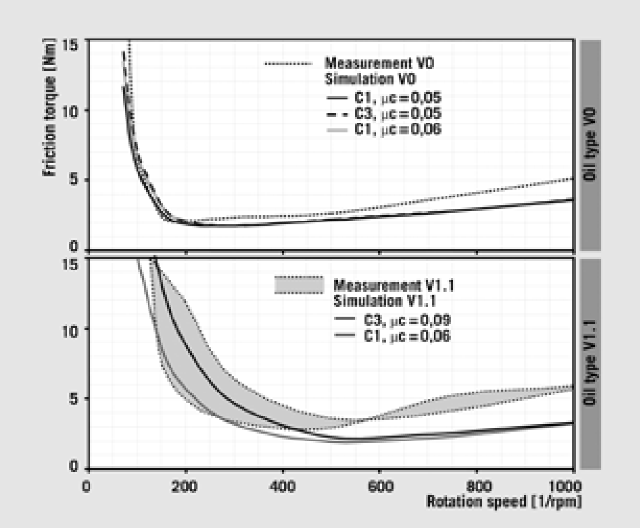
The test scatter band of the type V1.1 can best be shown in the mixed friction area by varying the solid body friction value µc and altered Carreau coefficients in the simulation (diagram bottom).

❽ shows the results of the combination steel/sputter for the types V3.1 and V3.2. The type V3.1, ⑧, diagram top, can be shown with a solid body friction value in the area of 0.03 to 0.05. A variation of the Carreau coefficients (C1 and C2) produces an earlier friction momentum rising with C1 because of lower shearing in this rotation speed area. Related to the minimal lubricant gap height the shearing for this type is 0.8×10^6 1/s at 100 rpm and falls to 0.6×10^6 1/s at 60 rpm.

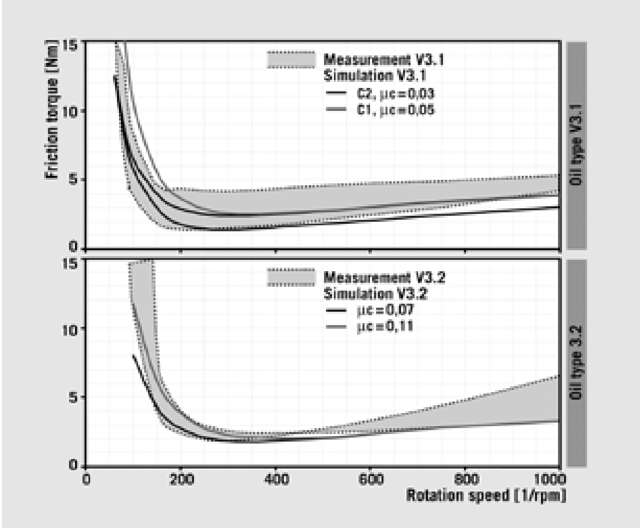
The diagram bottom in ⑧ shows the results of the type V3.2. In the area of mixed friction the measurement can easily be covered by solid body friction values of 0.07 to 0.11. In the deep mixed friction the gradient of the calculated friction momentum is not enough to cover the scatter band of the measurement on both sides.

6 MIXED FRICTION LOAD LIMIT

To compare the various lubricant types with regard to wear protection ability the concept of the mixed friction load limit is intro-



❼ Results of the combination steel/sputter for the types V0 and V1.1



❽ Results of the combination steel/sputter for the types V3.1 and V3.2

duced. This contains the mixed friction load (computational solid body contact pressure) and the mixed friction intensity (measurement technical C* value). Here, the following thesis is offered:

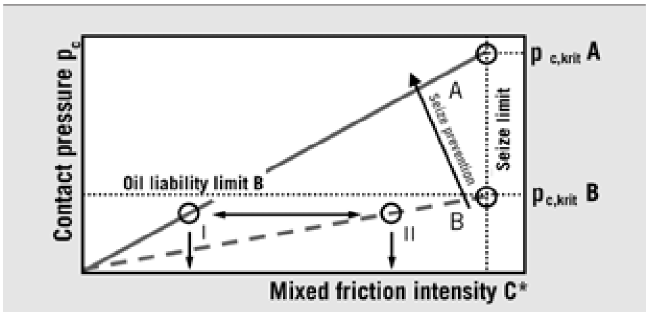
: With increasing contact pressure the mechanical load rises on the bearing material, which because of the relative movement generally leads to increased wear.
: With decreasing C* value the mixed friction proportion rises, which generally also leads to increased wear.
: Higher contact pressure at simultaneously high C* value means:
  a) less wear, because reduced to zero mixed friction proportions can produce no or reduced wear
  b) A protective effect of of the lubricant (by, for example, the chemical effect of the EP/AW additives on the surface), because a high contact pressure and a high C* value generally exclude each other
: Accordingly a high contact pressure at simultaneously high C* value means an optimum of wear protection provided by a lubricant.

This consideration, in view of the word choice "high contact pressure" and "high C* value", is relative.

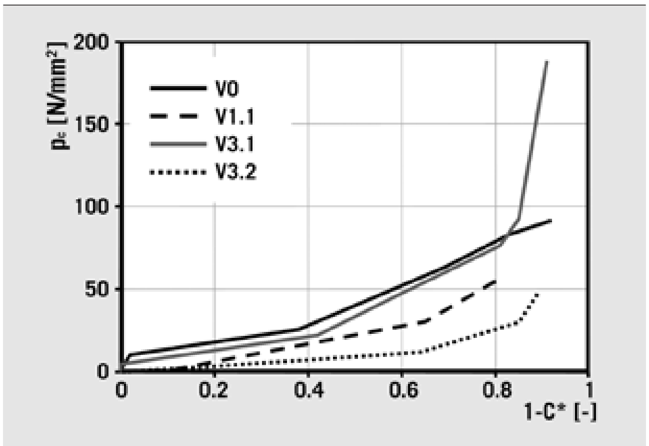
With an exact adjustment of the calculated friction momentum runs with the measurement results the mixed friction load limits of the lubricants can be used in the form presented subsequently for the choice or evaluation of lubricants. The adjustment ensures that the calculation model with the chosen parameters reproduces the measurement as exactly as possible, with the result that a direct comparison of the simulation results (solid body contact pressure) is permissible with the measurement results (mixed friction intensities).

For the sake of representation the contact pressures calculated are superimposed on the measured mixed friction intensities. This representation has the advantage that it is detached from the rotation speeds and thus transferable to arbitrary rotation speeds. ❾ is intended to clarify this thesis. The flatter a curve progression the higher the mixed friction proportions at low load. If the load is raised, this means an increase of mixed friction proportions, which in turn increase the probability of wear.

⑨ shows the curve progression of two different lubricants (A und B). Each lubricant has a lubricant load limit ($p_{c,crit}$) and a wear limit. The lubricant load limit is reached when a critical C* value is not achieved and as a result a critical contact pressure (wear criterion) achieved. In the case of this critical C* value the bearing



❾ Thesis of the mixed friction load limit



❿ Mixed friction load limits of the combination steel/sputter at 100 °C ambient temperature is operated in the mixed friction without interruption during the load phase. This means that the surfaces are at no time protected by a lubricant film. Continuing operation in this area leads to the immediate bearing feed. This behaviour could be observed in some Stribeck tests when the limit friction momentum (15 Nm) was exceeded. In the test bearing wear was reduced by limiting the drive momentum (60 Nm).

The intersection of these two limits (wear limit and lubricant load limit) is defined as mixed friction load limit. It specifies the maximum contact pressure load of a material/lubricant ($p_{c,crit}$) at the wear limit. The wear limit is determined by the last rotation speed level set by exceeding a momentum limit (15 Nm).

The comparison of the two lubricants A and B (points I and II) at a certain contact pressure shows clearly different mixed friction intensities. While lubricant A is still in moderate mixed friction, lubricant B is just under the wear limit in deep mixed friction. ❿ shows the evaluation of the results in the form of mixed friction load limits.

As can be seen in the results of the Stribeck tests the lubricant type O in both material combinations shows very good friction behaviour. In the case of the combination steel/sputter the type V3.1 can show lower fiction momentum at still lower rotation speeds and thus higher mixed friction, ⑩, which can be seen at the higher mixed friction load limits. The poorer friction behaviour of the types V1.1 and V3.2 is also clearly recognizable in this diagram.

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THANKS

The results of mixed friction conditions on slide bearings for crank drives of combustion engines arose at the research project No. 872 of the Research Association for Combustion Engines (FVV) at the University of Kassel. The authors thank the chairman of the workshop „Schmierstoffeinfluss auf Gleitlagermischreibung“, Dr. Hans-Jürgen Füßer, Daimler AG, for his encouragement. Also sincere thanks to the Fuchs Europe Schmierstoffe GmbH for the preparation of the lubricants and to the KS Gleitlager GmbH (Kolbenschmidt Pierburg AG) for the preparation of the slide bearings.