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WORLDWIDE

03 March 2011 | Volume 72

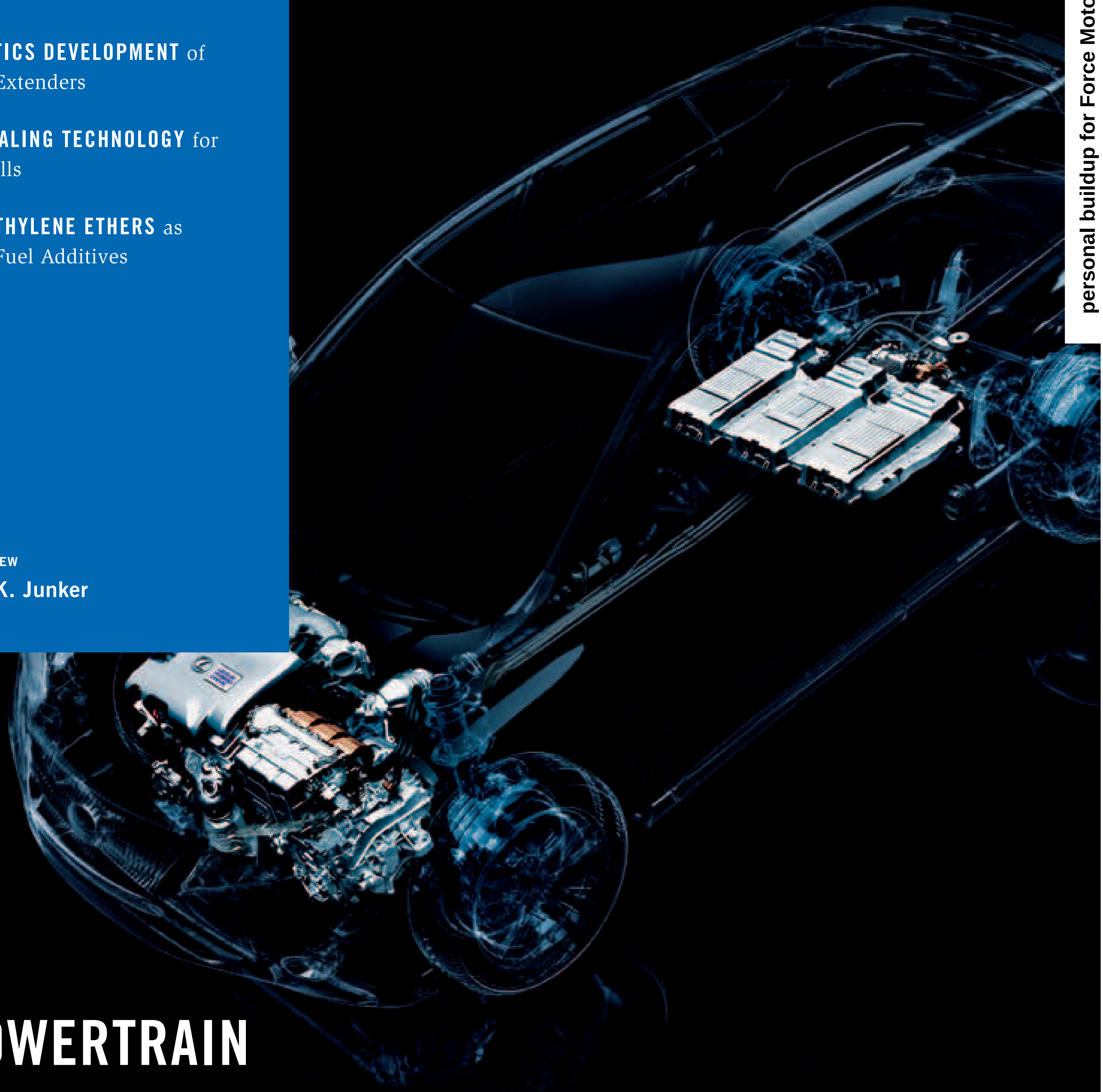
ACOUSTICS DEVELOPMENT of Range Extenders

LEM SEALING TECHNOLOGY for Fuel Cells

OXYMETHYLENE ETHERS as Diesel Fuel Additives

/// INTERVIEW

Heinz K. Junker
Mahle



POWERTRAIN ELECTRIFICATION

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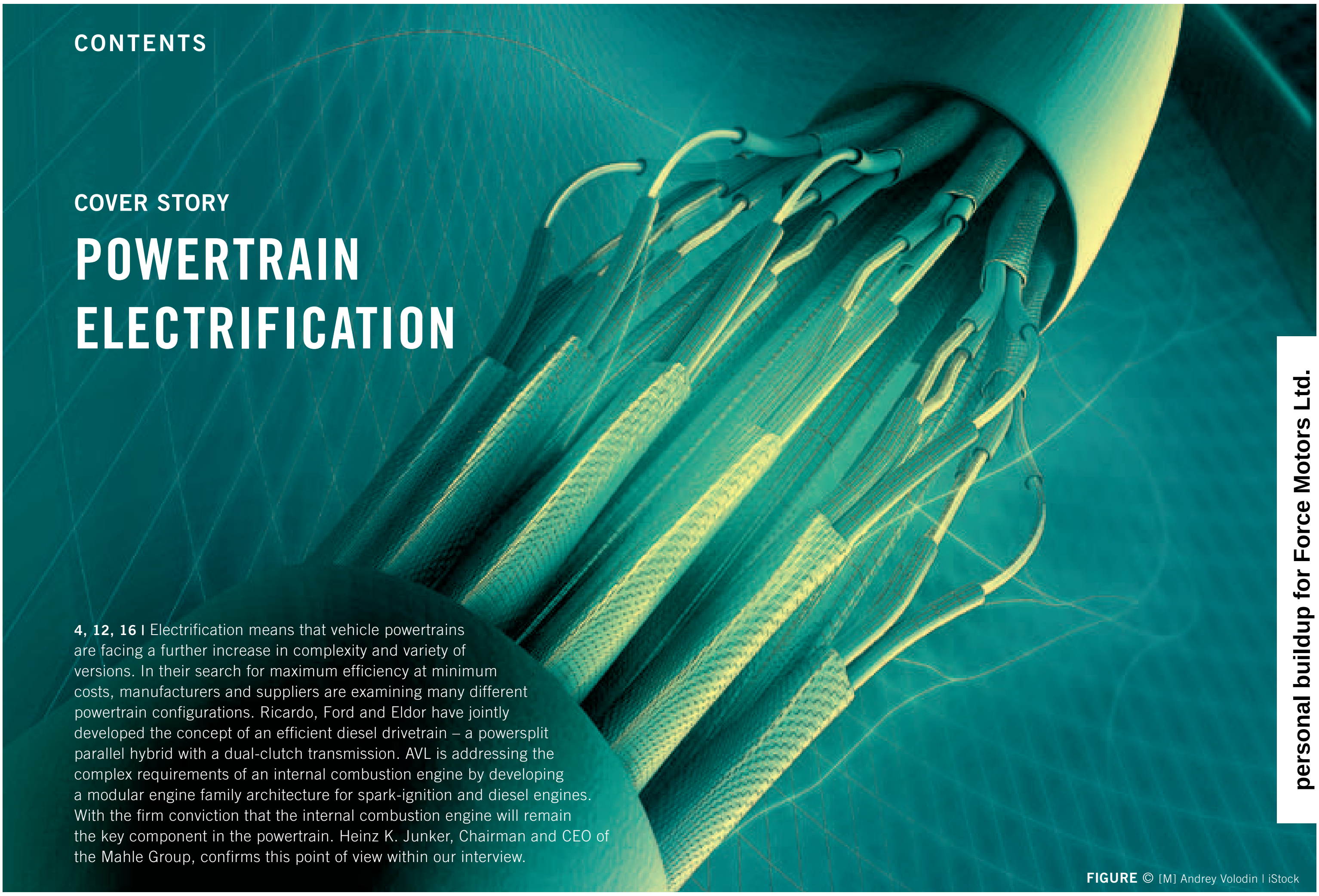

COVER STORY

POWERTRAIN ELECTRIFICATION

4, 12, 16 I Electrification means that vehicle powertrains are facing a further increase in complexity and variety of versions. In their search for maximum efficiency at minimum costs, manufacturers and suppliers are examining many different powertrain configurations. Ricardo, Ford and Eldor have jointly developed the concept of an efficient diesel drivetrain – a powersplit parallel hybrid with a dual-clutch transmission. AVL is addressing the complex requirements of an internal combustion engine by developing a modular engine family architecture for spark-ignition and diesel engines. With the firm conviction that the internal combustion engine will remain the key component in the powertrain. Heinz K. Junker, Chairman and CEO of the Mahle Group, confirms this point of view within our interview.

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COVER STORY

INDUSTRY

RESEARCH

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COVER FIGURE © Lexus

PRODUCTIVE TRANSFORMATION

Dear Reader,

What do you spontaneously associate with electrification? Only a few years ago, most drivers would probably have said the hybrid, which was becoming an increasingly familiar concept particularly due to the Toyota Prius. Today, electrification has become more of a vague term, as manufacturers and suppliers pursue different approaches under the same heading. Their repertoire extends from a cautious start/stop system right through to fully electric powertrains. The advantage of electrification is its great potential to improve fuel economy and to reduce emissions, at least locally. And all those numerous variables on the way to an efficient system may be either a blessing or a curse. Increasing complexity and costs must be seen as a challenge. Intensive research, creativity, competition and promising approaches are clearly the positive aspects.

The ongoing transformation in powertrain development is driven by an awareness that we are currently using up finite resources and need more environmentally friendly drive systems. Electrification – among other things – may possibly offer a solution in the long term. Politics is also a driving force, but in a different way. It already sets a course by effectively raising public awareness of electric powertrains. And then there are the driving cycles. They may well tip the scales, resulting in electric cars becoming established due to CO_2 subsidies. That would be counter-productive, especially in this highly innovative phase in which the efficiency of the powertrain is being improved in so many different ways. It is also being used to find out which alternative drive systems make sense in the long term. The fact that electrified powertrains have great potential is also shown in the cover story of this issue.

RUBEN DANISCH, Vice Editor-in-Chief
Wiesbaden, 23 January 2011

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DIESEL HYBRID POWERTRAIN FOR PASSENGER AND LIGHT COMMERCIAL VEHICLES

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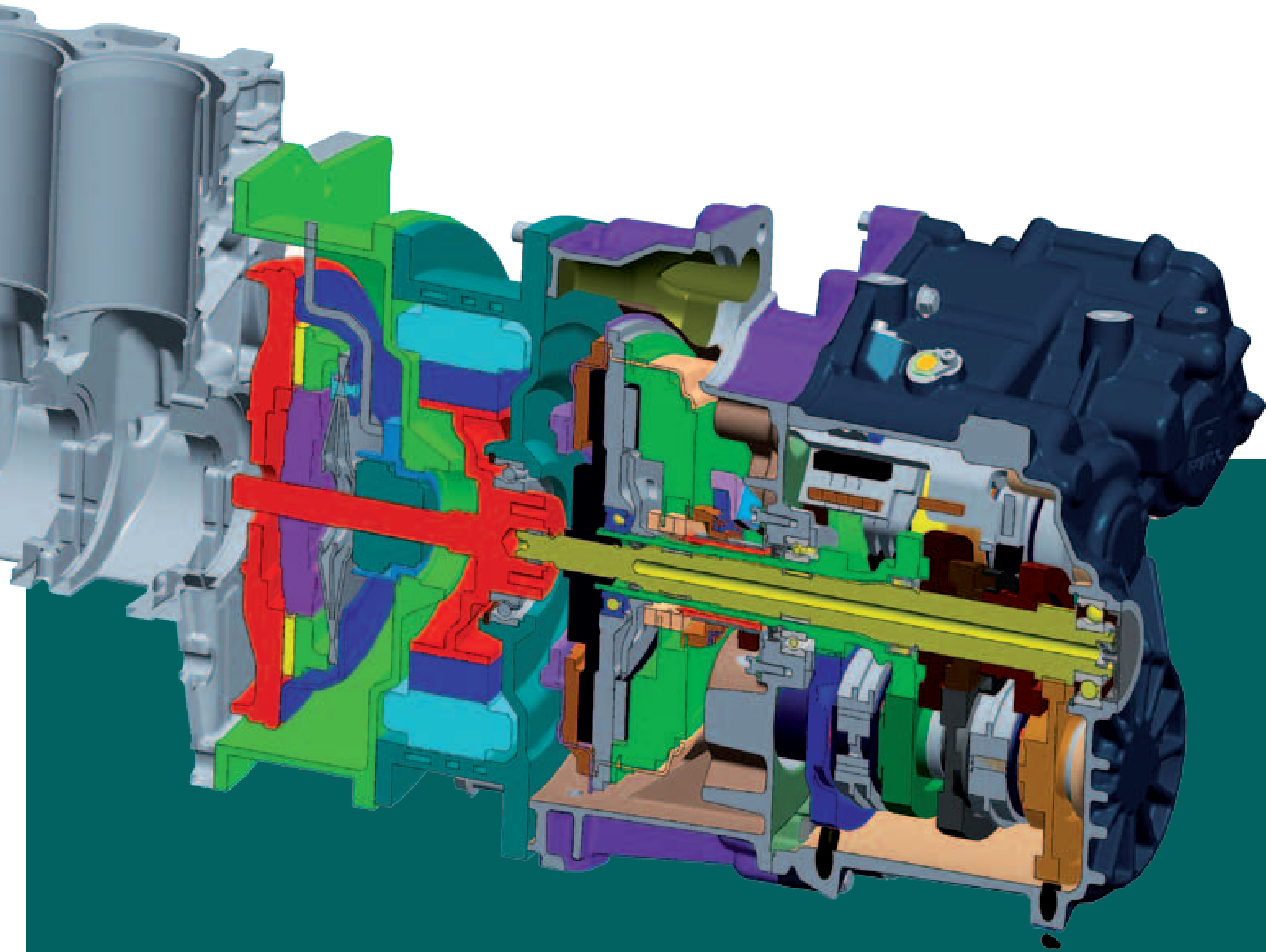

As part of the European funded FP6 project Hi-CEPS, Ricardo, Ford and Eldor have collaborated to develop an efficient diesel powertrain concept suitable for a family of vehicles on a C-segment platform, including a light commercial vehicle variant. The powertrain architecture incorporates a downsized diesel engine mated to an electromechanically actuated dual clutch transmission developed by Ricardo.

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THE PROJECT

Fuel-efficient and cost-effective powertrain solutions are a vital requirement for the transition to lower-carbon solutions. The challenge of integrating the key technologies to achieve a fuel-efficient diesel hybrid powertrain is further increased by the need to facilitate a modular powertrain strategy. This project has explored the concept using full vehicle simulation to determine the optimum powertrain layout to meet the needs of the target vehicle. Realisation in hardware has been achieved and a full testbed optimisation programme is about to start. This article shows the path from concept through to hardware and concludes on what the research partners expect to achieve in the final results.

VEHICLE REQUIREMENTS

In principle, it is generally expected that the behaviour of a hybrid vehicle should be transparent to the final customer. This means that the customer can expect predictable behaviour of this vehicle which is in line with the behaviour of existing conventional vehicles and independent of the operating mode of the vehicle. Therefore, the starting point for this project was to define the requirements for the hybrid vehicle which meet this over-arching requirement.

The target application was a C-segment small commercial vehicle for city delivery and distribution, a Ford Transit Connect. Since this particular vehicle is based on a platform shared with the Ford Focus and Ford C-Max, the benefits from hybridisation could be applied across a wider range of vehicles. This opportunity will inevitably lead to reduced system on-cost as application volumes increase. The issue of system cost and therefore product on-cost is not explicitly dealt with in this article; however another sub-project in the Hi-CEPS project is studying the benefits of this which will be reported separately. The operating modes considered were as follows:

- stop/start
- pure electric vehicle mode (EV)
- propulsion boost function
- regenerative braking
- internal combustion engine drive (ICE).

The key vehicle requirements are summarised as follows:

- pure EV mode possible up to 55 km/h
- acceleration 0 to 100 km/h < 17.5 s at gross vehicle mass (GVM)
- vehicle top speed 165 km/h
- e-motor nominal torque > 70 Nm
- e-motor mechanical power > 16 kW for ≥10 s
- launch/pull away in pure EV mode possible, 0 to 54 km/h in 17.6 s at GVM
- support engine cranking in cold conditions whilst powering the vehicle.

ARCHITECTURE DEFINITION

The powertrain architecture was initially defined as a diesel series-parallel split hybrid with a dual clutch transmission, comprising of the four cylinder 1.6 l Ford Duratorque engine, the Ricardo eDCT, a seven speed dual clutch transmission with electromagnetic actuation, the Eldor AC permanent magnet electric machines as traction motor(s), an Eldor AC permanent magnet belt-driven starter/generator (BSG), Eldor dual inverters and Ford Escape Ni-MH standard production Batteries.

The inclusion of a BSG in the architecture is intended to satisfy the requirements fast engine start while in electric-only mode to respond to increased load demand, additional generation capacity to charge the high voltage traction battery and the possibility for limited series hybrid mode (primarily for development purposes).

The disadvantages of the BSG include added drag on the engine (increased Friction Mean Effective Pressure, FMEP) and increased cost and complexity. Since the cold-cranking torque of the diesel engine increases significantly below 0 °C this, combined with reduced battery performance at sub-zero ambient conditions, meaning that it is possible that the BSG may not be able to start the engine. This work will therefore explore the feasibility of deleting the BSG from the architecture.

The use of a dual clutch transmission in a hybrid powertrain architecture presents a number of opportunities. Acceleration performance and driveability are superior compared to standard manual transmissions whilst retaining comparable efficiency. The objective of the initial work was to explore how the Ricardo eDCT could be incorporated into the hybrid powertrain to achieve even greater benefit, such as positioning the traction motors on each of the transmis-

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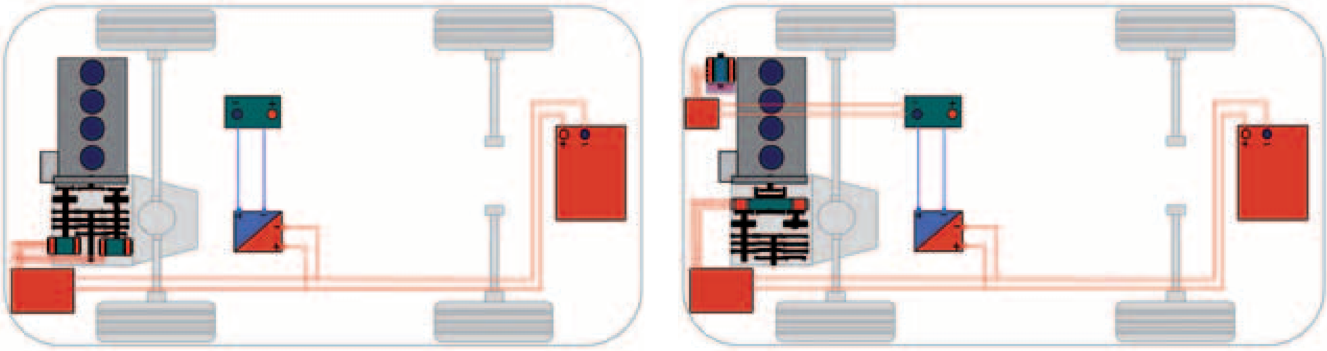


❶ Vehicle and powertrain architecture – dual traction motors mounted in eDCT (left); vehicle and powertrain architecture – single traction motor between engine and transmission (right)

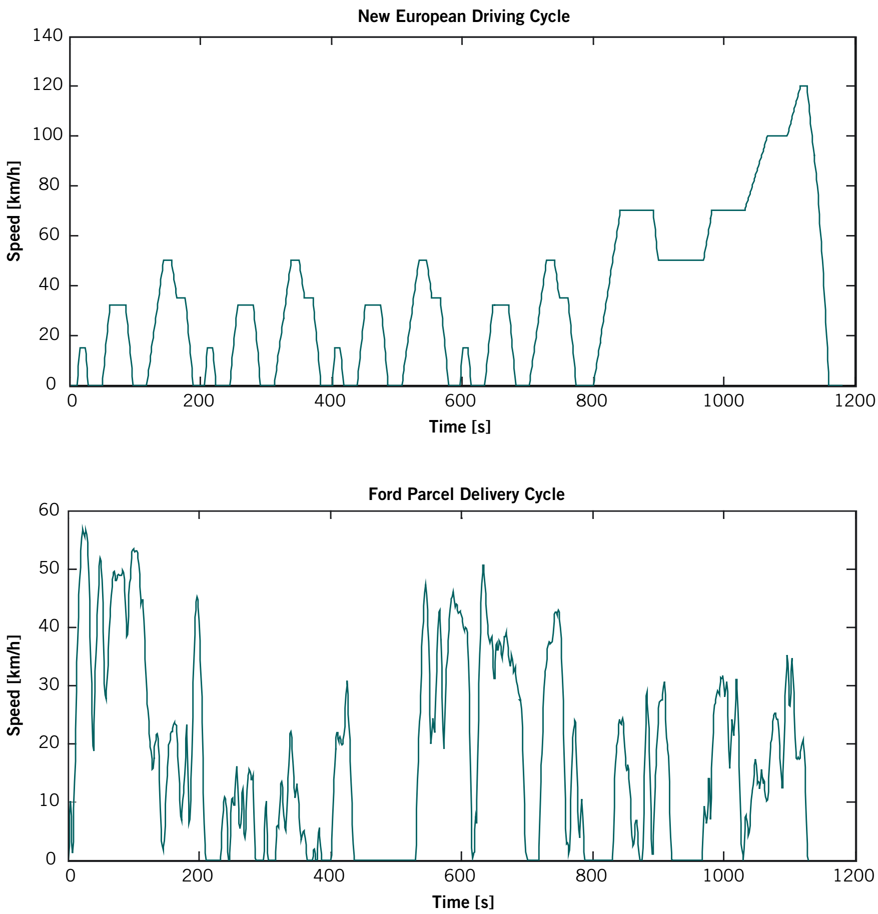
sion shafts (i.e. one on the even gear set, the other on the odd gear set), as shown in ❶ (left). The benefits of this, compared with the traction motor positioned between engine and transmission, ① right, needed to be understood and considered against other factors such as package, complexity, cost and control overhead.

These two concepts, amongst others, were explored using Ricardo V-Sim, a simulation tool developed in-house based on Matlab-Simulink to carry-out such investigations on different hybrid architectures. The value of hybrid architecture depends on a vehicle usage pattern specified by a family of possible drive cycles. For this analysis, the New European Driving Cycle (NEDC) and the Ford Parcel Delivery Cycle (FPDC), ❷, were used to explore the architecture options.

The V-Sim model was used to estimate the driveline power requirements over each of the two drive cycles. It was found that for the target application the power requirements are mostly within the ± 20 kW range where transitions outside this range are very limited, apart from the highway section of the NEDC, ❸. A total electric-only driveline power of 20 to 25 kW, with a typical permanent magnet AC motor power and torque characteristic was therefore selected for the ongoing analysis.

The V-Sim model was then configured into the range of powertrain architectures being considered; a selection of these is shown in ❹. Dynamic Programming techniques were applied to each V-Sim model and drive cycle instance to ensure a optimal strategy was being considered in the comparisons between architectures. It is beyond the scope of this article to provide details of each optimisation; however, the outline control strategy was based on the following key principles:

: Electric-only mode is used without engine support as long as the battery state of charge (SOC) exceeds a certain level or torque demanded is achievable from the electric machine only.
: In hybrid mode, generation from fuel is performed when engine is on with sufficient speed and battery SOC is sufficiently low. The engine has a minimum "on" time of 15 s. By this rapid on/off transits of the engine with the corresponding negative effect on driveability are avoided. Also, there are possible durability concerns owed to repeated stop/start events behind this strategy. Positive results achieved



❷ New European Driving Cycle (NEDC) (above); Ford Parcel Delivery Cycle (FPDC) (below)

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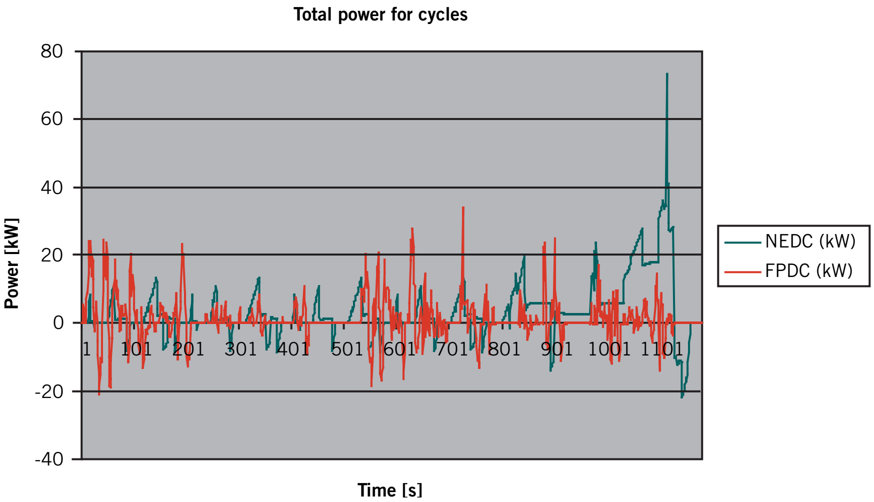


❸ Driveline power requirements for each drive cycle

These results clearly show that an architecture configured with the traction motors integrated into the eDCT tended towards a more efficient solution.

ARCHITECTURE CONFIGURATION SELECTION

A full appraisal was carried out through qualitative and quantitative analysis of the various architecture configurations. This appraisal included driveline efficiency comparisons (reported above), package appraisals using 3D CAD concept modelling, system drivability/response appraisal through assessment of transmission control requirements and experienced subjective opinion and cost analysis through high level Bill of Materials and subsystem cost estimates.

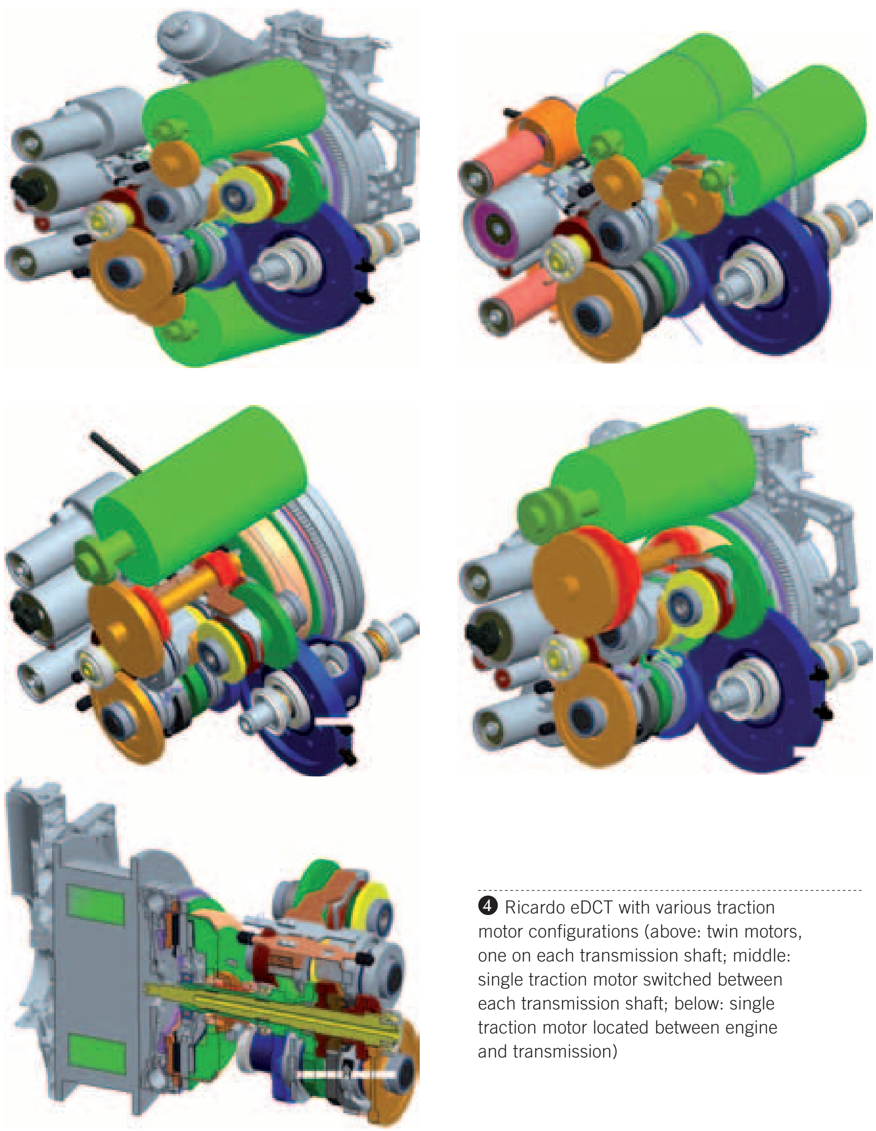
through control system development on other Ricardo hybrid programmes also suggested this "on" time.

The electric machine is used to boost performance based on accelerator pedal position. The magnitude of regenerative braking is a function of brake pedal position, vehicle speed and SOC. Primary gear selection is based on accelerator pedal position and vehicle speed. Secondary gear selection (for eDCT dual motor solution) is either one gear up or down from primary.

This approach has allowed relative rather than absolute comparisons to be made between the different architecture configurations being considered. This was sufficient to allow key decisions to be made prior to concept selection for more detailed analyses.

The V-Sim models, with Dynamic Programming methodologies, were used to compare overall driveline efficiency over the NEDC and FPDC drive cycles. By way of comparison, results achieved on NEDC for a selection of the configurations are provided below:

- single traction motor located between engine and transmission: baseline
- single traction motor located between engine and transmission with BSG: 1 % improvement
- twin motors, one on each transmission shaft: 4.4 % improvement
- single traction motor switched between each transmission shaft: 2 % improvement.



❹ Ricardo eDCT with various traction motor configurations (above: twin motors, one on each transmission shaft; middle: single traction motor switched between each transmission shaft; below: single traction motor located between engine and transmission)

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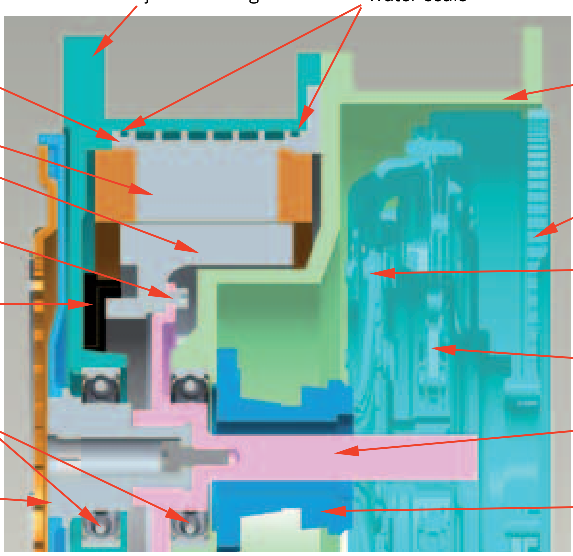

❺ Hybrid module

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The key conclusion from this stage of the study was that the single traction motor located between engine and transmission gave the greatest overall benefit for the target vehicle application for the following reasons: The eDCT could be common with non-hybrid powertrain variants, reducing cost and complexity. A single electric traction motor should be cheaper than a twin motor arrangement. The full driveability benefits of the eDCT could be achieved. The package feasibility could be improved with an axially compact traction motor.

SYSTEM DEVELOPMENT

With the eDCT design essentially unaffected, the work then focussed on the design of a hybrid module consisting of engine clutch, electric traction motor and coupling to eDCT dual clutch pack.

❺ provides a cross section of the assembly with key components identified, at concept stage. Key attributes of the hybrid module design shown are as follows:

: two part water jacket/housing (manufacture and supply benefits)
: split rotor shaft (assembly benefits)
: bolted rotor armature core – shaft assembly (manufacture and supply benefits)
: asymmetric rotor core (package benefits)
: twin bearing support for Rotor
: single mass flywheel and 220 mm dry clutch for engine.

Analysis of the critical dimensions showed that 107 mm of axial length is required to house the electrical machine (against a 90 mm width stator) and a further 92 mm for the integration of the engine clutch giving a total length of 199 mm. The provisioning of significant volume around the clutch slave cylinder (CSC) shows that future optimisation could allow the nesting of the engine clutch within the rotor inner diameter, reducing the overall length in to the region of 120 mm. This has not been pursued due to a number of uncertainties with the approach that will only be understood during testing of the hardware (engine inertia limitations and required engine clutch operational duty).

The diameter of the system is constrained by the centre distances of the mating transmission units input and differential axis. This constraint is the overriding determinant of the stator and rotor widths when meeting the required motor performance characteristics.

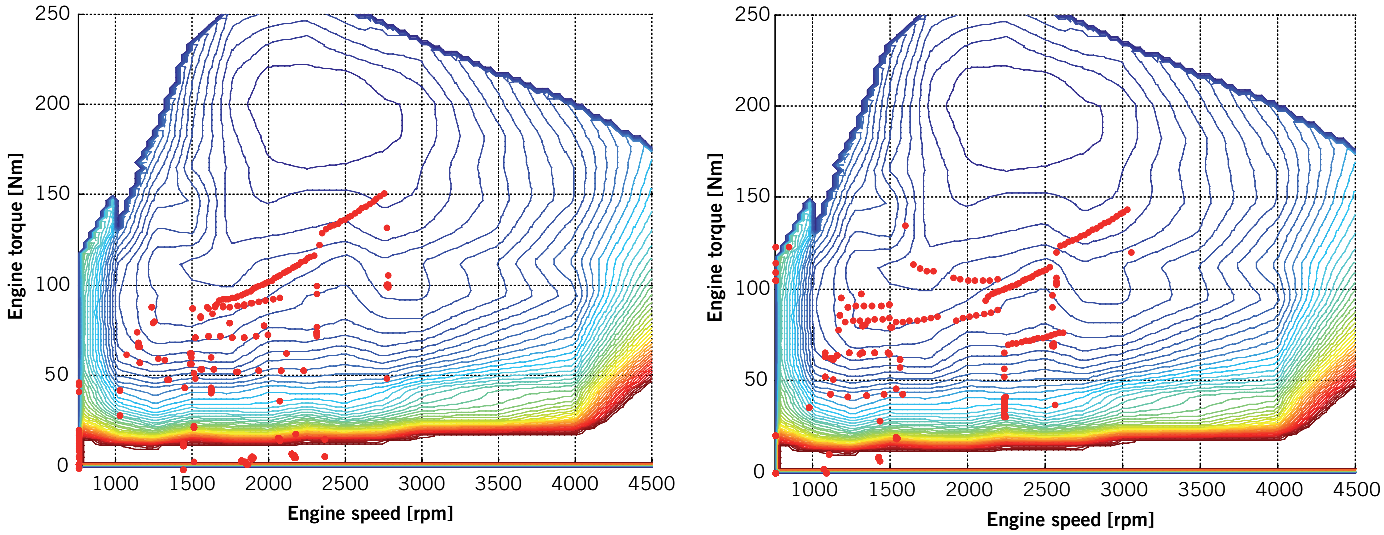
The total mass of the hybrid module was estimated to be 46.3 kg. This is a dry weight, excluding any coolant and does not account for the battery, inverter or controller systems. It is envisaged that future development of the engine clutch, as already described, could lead to further weight savings for the clutch disk, cover and single mass flywheel elements.

The integration of the electrical machine has been achieved through a two-part water jacket and bolt-on rotor armature core. This approach allowed Eldor to carryout the design of the electrical machine parts independently of Ricardo's overall packaging and integration work without compromise to the final assembly.

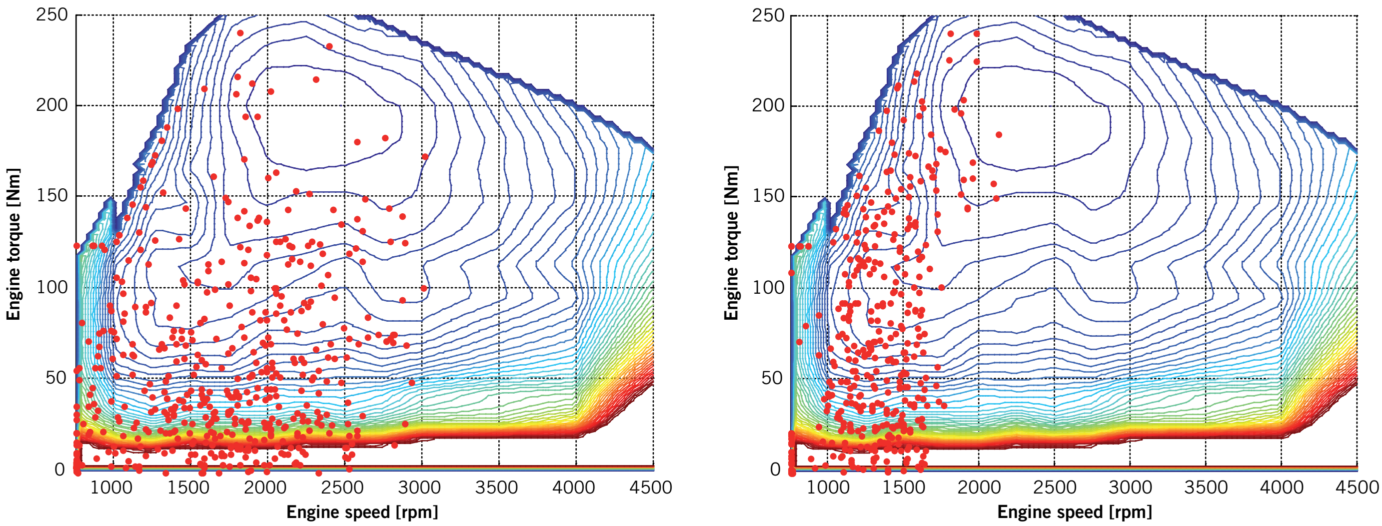
The key to the integration of the rotor armature core into the powertrain is the rotor shaft assembly. This assembly provides the power flow for both the electrical machine and engine to the eDCT. A two-piece design is necessary for assembly purposes since two deep groove ball bearings were deemed necessary to adequately support the shaft. The downstream part includes a number of welded subassemblies; a splined disk for interfacing with the dual dry clutch (shown in orange), a carrier disk (shown in blue) and a hub (shown in grey). The upstream part provisions a radial register for accurate location of the motor armature core with necessary fasteners, a spline interface for the engine clutch and a pilot bearing interfacing with the engine crank palm.

SIMULATION RESULTS

The V-Sim model was updated to represent the designed systems and design-intent electric traction motor. Analyses were carried out over the NEDC and FPDC drive cycles, with negligible change in battery SOC over the drive cycles. The results obtained over the NEDC are detailed below



❻ Brake Specific Fuel Consumption (BSFC) maps showing the operating points over the NEDC (left: target vehicle with five speed manual transmission; right: target vehicle with hybrid module and Ricardo eDCT)



❼ Brake Specific Fuel Consumption (BSFC) maps showing the operating points over the FPDC (left: target vehicle with five speed manual transmission; right: target vehicle with hybrid module and Ricardo eDCT)

For the target vehicle with five speed manual transmission, a fuel consumption of 5.2 l/100 km was calculated, which equals 139 g/km of CO_2 emission. For the target vehicle with Ricardo eDCT the fuel consumption was 5.1 l/100 km translating into 135 g/km of CO_2 emission. For the target vehicle with hybrid module and Ricardo eDCT fuel consumption dropped to 4.0 l/100 km equalling 107 g/km of CO_2 emission.

The corresponding Brake Specific Fuel Consumption (BSFC) maps showing the operating points over the NEDC are shown in ❻: the simulation results are summarised by plotting the engine operating points over the BSFC map. This shows how the different builds optimise the operation of the engine. The results obtained over the FPDC are detailed below:

For the target vehicle with five speed manual transmission, a fuel consumption of 8.7 l/100 km was calculated which equals 230 g/km of CO_2 emission. For the target vehicle with Ricardo eDCT, the fuel consumption was 8.0 l/100 km which is equivalent to 211 g/km of CO_2 emission. For the target vehicle with hybrid module and Ricardo eDCT the specific fuel consumption was as low as 5.7 l/100 km which equals 151 g/km of CO_2 emission. The corresponding Brake Specific Fuel Consumption (BSFC) maps showing the operating points over the NEDC are shown in ❼.

The BSFC plots show how the engine operation points have been moved as the eDCT and hybrid functions are introduced. These show the engine operating at more efficient points and then for the hybrid functions, the low loads are reduced. It is anticipated, however, that operating a Diesel engine at higher loads will lead to increased NO_x emissions. The authors firmly recognise this issue but

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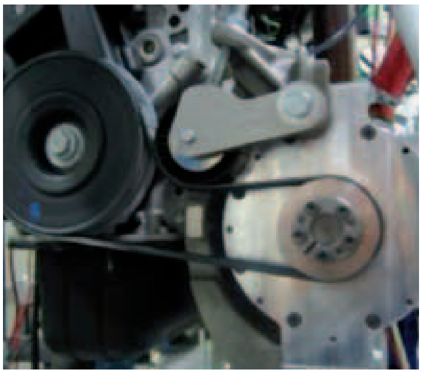
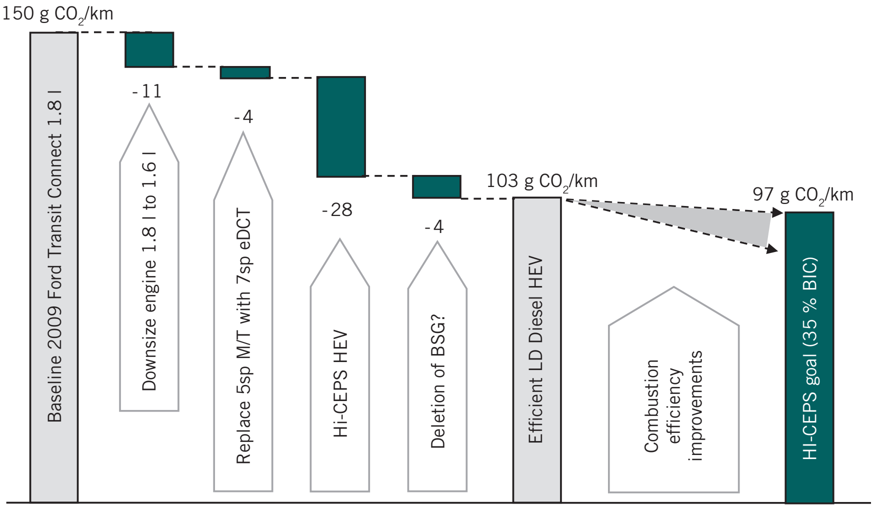


❽ BSG installed on the diesel engine (left), complete hybrid powertrain installed on Ricardo test bed (right)



❾ CO_2 walk showing how the HICEPS overall goal will be achieved

note that since this is being addressed in a separate Hi-CEPS project workpackage it will not be dealt with in this article.

HARDWARE REALISATION & DEVELOPMENT

Ricardo, Ford and Eldor have worked closely to procure all of the key components of the diesel hybrid electric powertrain. ❽ shows BSG mounted on the diesel engine and the complete hybrid powertrain installed in the Ricardo engine testing facility ahead of the full test programme. The realisation of the complete hybrid powertrain has therefore been completed and full transient testing in a proprietary Ricardo test facility is currently underway, although results are yet to be yielded.

CONCLUSIONS

The work performed to date demonstrates that a diesel hybrid powertrain incorporating a Ricardo eDCT in a light commercial vehicle will yield strong improvements in fuel economy whilst retaining maximum commonality with conventional powertrain variants.

The HICEPS project has targeted an overall 35 % improvement over the NEDC drive cycle. Whilst the analyses reported here show a result slightly short of this, the project team foresee further improvements, including the removal of the BSG to reduce engine drag losses (as well as weight and cost); this will require an alternative engine starting protocol and ancillary loads to be served by the traction motor.

A CO_2 “walk” showing an estimate of how the target could be realized is illustrated in ❾.

A very important conclusion is that without further refinements, the same vehicle and powertrain could achieve around 40 % improvement in fuel economy under urban delivery conditions. This, combined with the cost effective Ricardo eDCT, makes the business case more attractive for the target customer.

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"CLOSE TO THE OPTIMUM"

Professor Heinz K. Junker, Chairman and CEO of the Mahle Group, heads one of the biggest German component suppliers and is a fierce defender of the future of the piston engine. So is he the ideal partner for an interview on the subject of electrification? We think so.

Prof. Dr.-Ing. Heinz K. Junker has been Chairman and CEO of Mahle GmbH since 1996. Heinz Junker was born in 1949 and studied engineering in Aachen, where he also completed his doctorate. Over the past one and a half decades, he has turned Mahle into an internationally leading supplier of engine components and technology through numerous acquisitions and co-operations. The takeover of Behr announced in 2010 will expand the portfolio of the company, which was formerly purely a manufacturer of pistons, to include thermal management. Anyone who talks about engine technology with Junker, who is always ready to go into detail, would never notice that he spent the first two decades of his professional career primarily in the field of chassis technology.

MTZ _ Professor Junker, it came as a bit of a surprise when you accepted our invitation to give an interview on the subject of electrification.

JUNKER _ I'm sometimes criticised for saying that the internal combustion engine will still be around in 20 years' time. But I also have to address the issue of electrification.

Do you sometimes have doubts that this statement will not be able to withstand the huge political pressure in the end?

Even politics cannot change physical laws. But if things were allowed to develop on their own, the world would perhaps be a different place than one where there is massive political influence.

It would, of course, be rational to set certain CO_2 limits and then let manufacturers and the market find their own way of achieving them.

I think that that is how it will work in the long term. We should really leave it to the actors on the market to find a way to meet CO_2 targets. One should not prescribe which proportions electric or fuel cell vehicles have to achieve.

The driving cycles for vehicle approvals favour electric vehicles, and therefore a certain amount of technology control is already taking place.

Let us wait and see how the cycles develop. Can we really live with our EU cycle, which is far from realistic, until 2020? The only cycle that is close to reality with regard to CO_2 emission today is one of the different US cycles.

It may be questionable whether the electric car will become a reality – but electrification has begun and is already changing the conventional internal combustion engine.

That depends very much on how great the degree of electrification is. When one hybridises a powertrain, its first all makes life easier for the internal combustion engine. It no longer has to pull as much weight as today. But that is still a normal internal combustion engine that we know today. The situation is quite different when we consider a range extender.

Will the trend towards greater variability and therefore greater complexity in the internal combustion engine continue?

I believe it will. Although we have to distinguish whether the internal combustion engine is part of a hybrid powertrain or is only fitted with a start/stop system. Certain variabilities that one can integrate into an internal combustion engine are obsolete if one can operate it in a full hybrid. They are then simply no longer effective because the dynamics can be better controlled by the electric motor components. In such cases, it is important to keep an eye on the cost-benefit relationship.

What would a CO_2-optimised internal combustion engine for a mid-size car then look like?

With our downsizing concept engine, we are already close to the optimum. Its power and torque figures are at the level of a present-day turbocharged two-litre spark-ignition engine, which itself corresponds to a naturally aspirated engine with a displacement of between 2.5 and 3 litres. We have further development steps at the planning stage in order to demonstrate how we

> "Any concept that does not fit into an existing architecture will have a hard time."

can achieve less than 100 grammes of CO_2 in a mid-size vehicle weighing up to 1500 kilogrammes in 2020 without making a full hybrid. Today, we are still at 135 grammes, but the potential has not yet been fully exploited.

Will this engine then have two-stage exhaust gas turbocharging?

We have already built various versions, including ones with twin supercharging. However, we favour single turbocharging because we believe among other things that we can reach the targets more cost-effectively with other measures.

One of these measures might be even more variability in the valve train – there are already a lot of new approaches in this area. Which ones are you pursuing?

Every system developed by a supplier that does not fit into an existing cylinder head will have a hard time. For that reason, we favour the "cam-in-cam" concept, which can be integrated into any existing cylinder head architecture. It then comes down to the question of whether you fit one or two phase adjusters on the outside. Furthermore, such a system can be used in either a spark-ignition or diesel engine, either on the intake or exhaust side.

When the limits are achieved in 2020, has the potential then been fully exploited?

A further reduction in CO_2 will probably no longer be possible without a full hybrid. Vehicles heavier than 1500 kilogrammes will probably even have no alternative than to use a plug-in solution, allowing them to drive certain greater distances fully electrically. The exciting question from my point of view is whether it will be sufficient in 2020 to achieve the limits with fleet fuel consumption. Or whether there will be absolute upper limits for heavy vehicles.

What will solutions for much smaller vehicles then look like?

I see three scenarios in the long term. Fully electric driving for pure city vehicles. Or the use of an extremely fuel-efficient and low-cost internal combustion engine with a start/stop system. Or the combination of a battery-powered drive system and a range extender module, which I would include as an option for – let's say – 2000 Euros. That will apply to vehicles weighing less than 1000 kilogrammes, the majority of which are also very cost-sensitive.

How big will the market segment for range extender vehicles become?

I don't think that we can expect a volume market here.

As such engines – especially in small production volumes – are not being produced by any manufacturer today, is that a future business field for Mahle?

That is difficult to predict. We are currently developing such an engine, which we will probably present at the IAA for the first time. If there is customer demand, we will be happy to talk about it. We would certainly be in a position to produce some thousands of engines relatively quickly.

For the range extender, you favour a conventional piston concept and not a Wankel rotary engine.

The Wankel engine does, of course, have advantages in NVH and in packaging. Perhaps both solutions will exist on the market.

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Mahle's CEO Junker still sees potential in mechanical systems

Having more electricity on board is certainly an opportunity for the auxiliary units.
We will certainly also take advantage of this opportunity. One must not forget, however, that mechanical solutions directly driven by the crankshaft also offer further potential. For example, our controlled pendulum-slider oil pump, which is now going into production in large numbers. Depending on the vehicle and engine, it can cut CO_2 by between two and three percent. It is clear that in full hybrid vehicles you need an electric air-conditioning compressor in order to ensure passenger comfort. Even in vehicles with a start/stop system, thermal management will become an issue, at least in summer temperatures. But here too, there are also simpler technologies, such as a storage evaporator.

You are now also working together with Behr in this sector. How great are the synergies with Mahle's core business?
We already have joint development projects, for example for integrated intercoolers that are integrated into the intake system. The next topic is EGR, which is now an essential part of every diesel engine. EGR is now also being introduced into spark-ignition engines, simply because we increasingly want to eliminate full load enrichment. One can also observe that Japanese manufacturers are using EGR to dethrottle the engine. And in most cases, a systemic EGR approach also requires a cooler. In today's designs, we are not yet at the optimum, neither in terms of function nor with regard to costs. What is required is more integration.

You plan to have acquired a majority stake in Behr by 2013. Is that still the case?
From today's perspective, yes. It is our plan, after the acquisition of a majority stake, to run Behr as a division of Mahle, even if Behr of course legally remains an independent company. That means that we have two years to prepare integration. The key objective is to harmonise processes and structures and already to make use of initial synergy potentials

"We should leave it to the actors on the market to find a way meet CO_2 targets."

What is the reason for the slow, step-by-step approach?
We wanted to have the freedom to be able to invest in other issues than only in thermal management if necessary. What is more, we want to try to finance everything from free cash flow. As we know, other forms of financing can lead to unpleasant problems.

Finally, let us talk briefly about pistons. When will we see the first steel piston for passenger cars from Mahle?
An exciting question. It will certainly come for the diesel engine, and for spark-ignition engines it is under discussion. I would not like to give a firm schedule at the moment.

Will it take longer than three years?
I don't think so.

Thank you very much for this interview.

INTERVIEW: Johannes Winterhagen
PHOTOS: KD Busch

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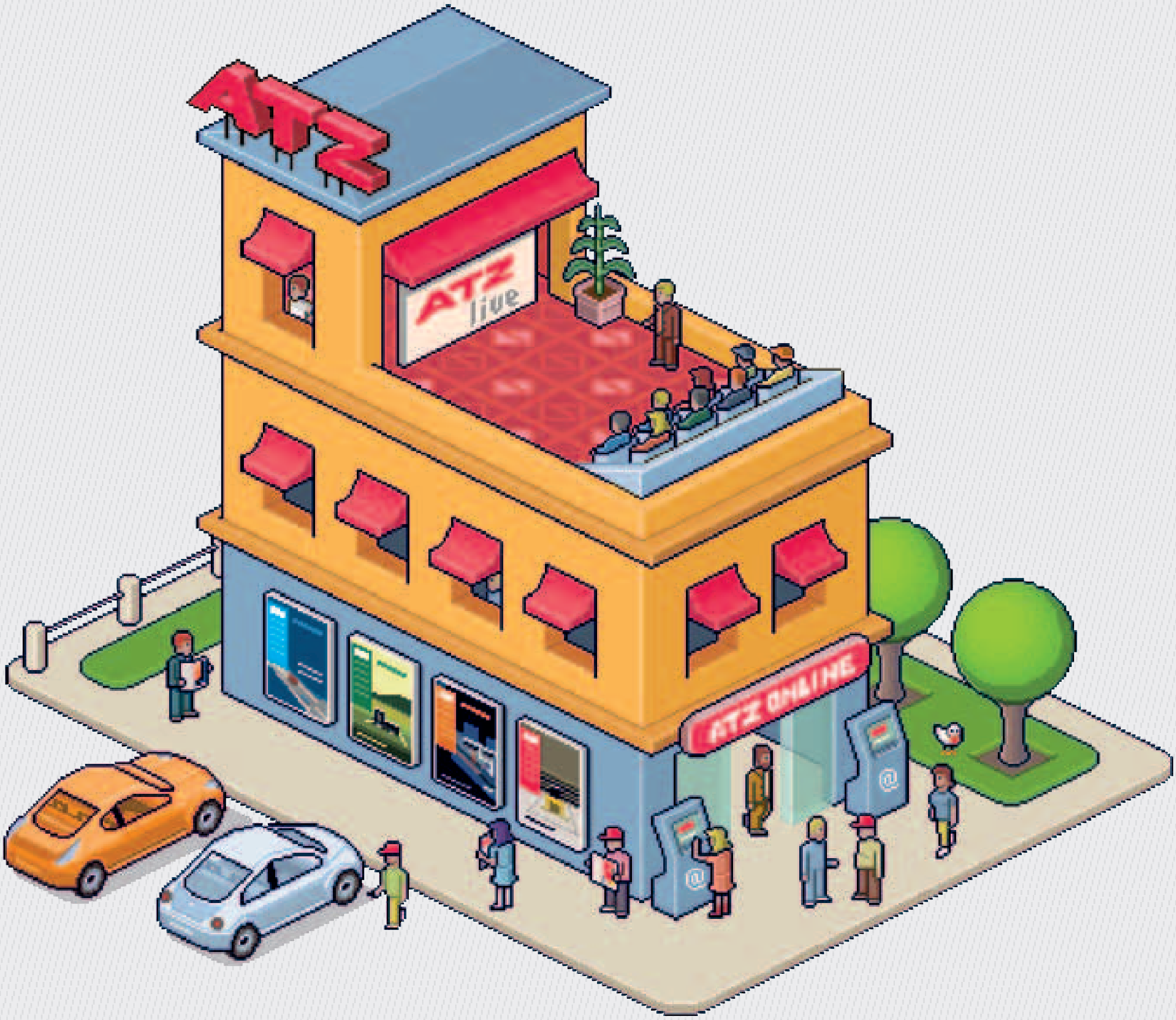
ATZ. A VARIETY IN AUTOMOTIVE KNOWLEDGE.



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THE INTERNAL COMBUSTION ENGINE AS KEY COMPONENT

The trend to powertrain electrification leads to a huge increase in the variety and complexity of new powertrain systems. The internal combustion engine – which will remain the key powertrain component for the foreseeable future – faces new demands including reduction of production costs and the flexibility to produce different engine variants in changing volume splits without compromising engine efficiency. AVL proposes a modular engine architecture in answer to these contradictory requirements.

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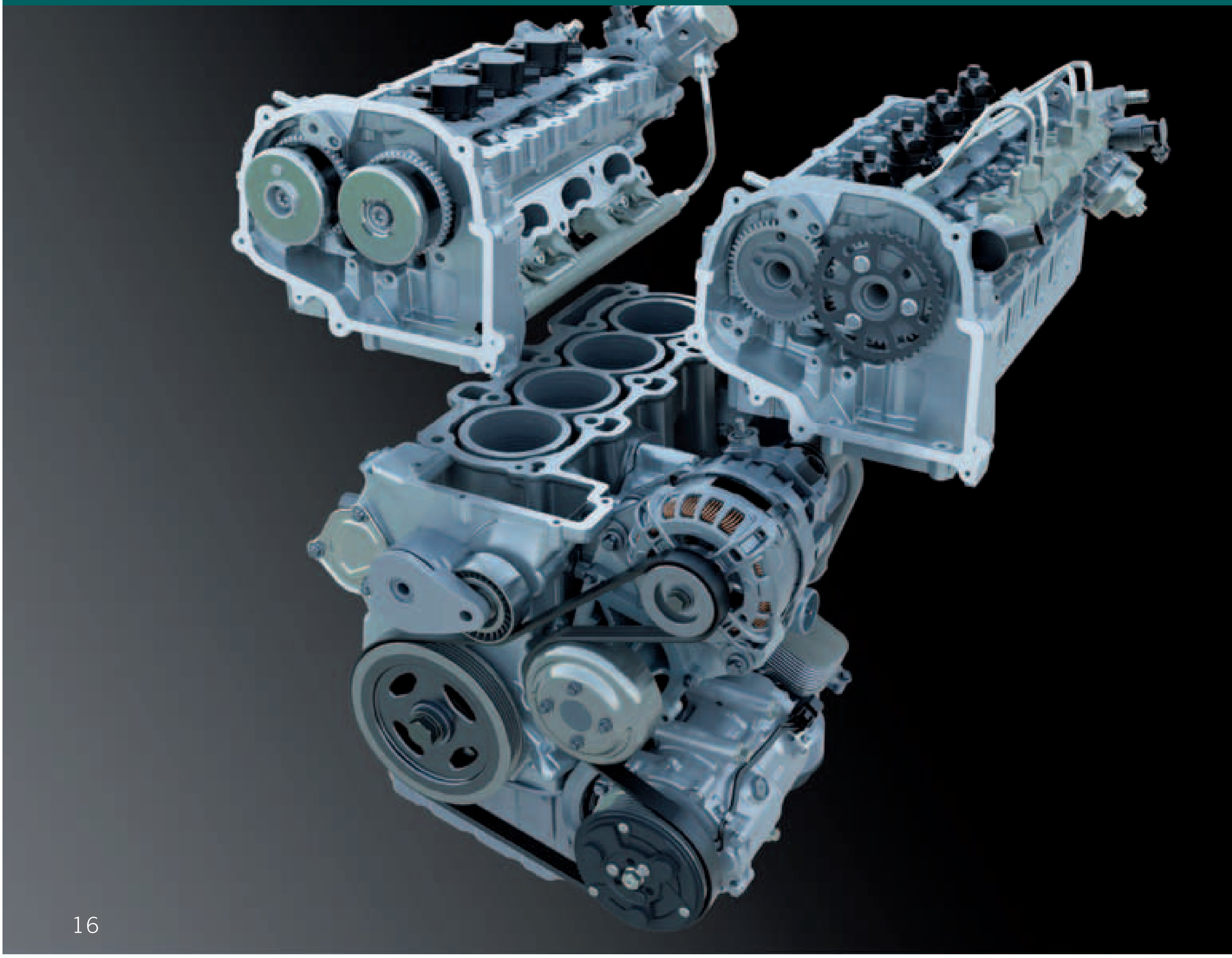

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GASOLINE AND DIESEL ENGINES WILL KEEP THEIR PLACE

Future worldwide CO_2 scenarios with stringent fleet average fuel consumption targets as well as increasing customer demand for fuel efficient vehicles are the most relevant drivers in the development of future powertrain concepts. Electrification will play an ever increasing role in the fulfillment of these targets; however the internal combustion engine will remain a decisive component of integrated powertrain systems with the degree of electrification adapted to the specific application.

The continually increasing number of vehicle and powertrain variants with different degrees of hybridization leads to highly diversified requirements on the combustion engine. Gasoline and diesel engine have strengths in different application and usage profiles and both will have their place in future powertrain line-ups. In both cases "downsizing" is a promising route and drives the trend to turbocharging and direct injection for all engine configurations. The consequently increased mechanical and thermal loads on the base engine structure have to be given special attention when defining a new engine family in order to avoid an excessive friction penalty.

At the same time it is necessary to reduce the cost of the base ICE, in order to balance the increased technology costs of the other powertrain system elements. This places high demands on the engine manufacturing strategy, which has to produce multiple variants in unpredictable volume splits. A modular engine architecture offers a possible solution.

OVERVIEW OF FUTURE POWERTRAIN LAYOUTS

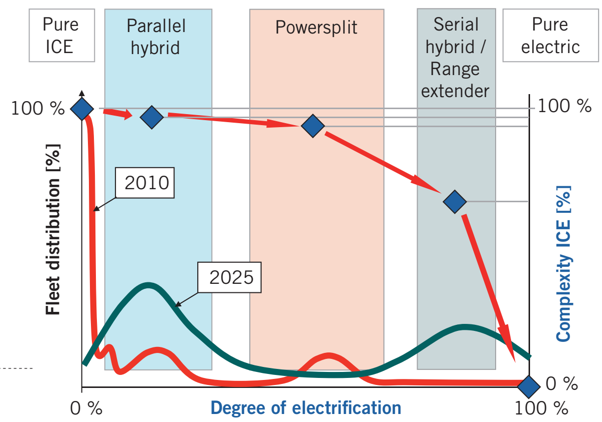
Various predictions have been made for different development scenarios of powertrain concepts in the next 15 years [1, 2, 3, 4]. Although there remains high uncertainty in these predictions due to the fast pace of technology development on several fronts and unknown influence of future legislation and taxation incentives or penalties, a continued strong future for the ICE is a common theme. ❶ shows one scenario for the volume split in 2025 in comparison with today's status.

In this model the ICE as sole prime mover will be reduced to a niche product – although still comparable to the total volume of pure electric vehicles. The volume products will be either parallel hybrids (ranging from micro- to mild-hybid according to application) or serial hybrids.

PARALLEL HYBRID

In these configurations, the ICE is the prime mover and energy source for the electrical consumers on boards. The engine – since it is mechanically coupled with the wheels of the vehicle – has to cover the full range of engine speed and load operation, including idling and transients.

Here the emphasis will be on a downsizing approach to shift ICE operation to higher loads and therefore better efficiency; reduction of mechanical friction will also be an important factor in improved overall efficiency. Since CO_2 reduction is the main driver for these developments, we can expect that Diesel engines will also have a strong role to play.



❶ Impact of electrification on the ICE

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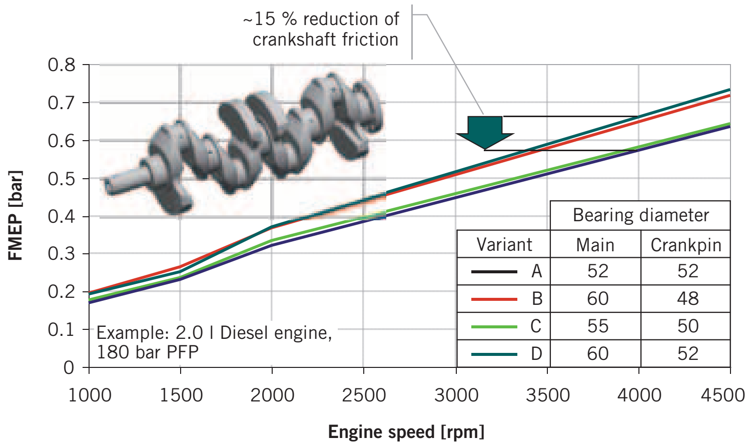


❷ Crankshaft friction related to bearing diameters

SERIAL HYBRID

Here the vehicle is electrically driven and the ICE engine provides power to generate electricity, either exclusively or in support of a plug-in charging system. Because the ICE is no longer mechanically coupled with the driving wheels, it becomes possible to restrict the range of ICE operation to certain preferred speed/load conditions and to eliminate transient or idling conditions. Therefore the ICE can be simplified to some extent. Especially for range extender applications – where the engine is designed for occasional use only – unconventional concepts can be considered [5].

STANDARD ARCHITECTURE OF AN ENGINE FAMILY

Engines are typically designed in families which share main dimensions and architectural features (e.g. block material, bore pitch), with variation of certain parameters (e.g. bore, stroke, no. of cylinders) in order to cover a wide range of outputs and applications. This approach offers synergies in both manufacturing (common parts, common manufacturing facilities) and the engine development process (common cylinder unit). The disadvantage of this approach is that the architecture is set by the most complex, highest loaded variant, which potentially leads to penalties in weight, friction and cost for the entry level versions – although these make up the bulk of the production volumes.

For example, the specification of the crankshaft bearing diameters strongly influences the crankshaft friction and thus engine friction, and is driven by bearing unit loads, oil film thickness and crankshaft stiffness, ❷. The layout depends strongly on the maximum loads due to gas pressure and inertia [6, 7, 8]. A compromise here, as is typical in an engine family, leads to a measurable increase of CO_2 emissions.

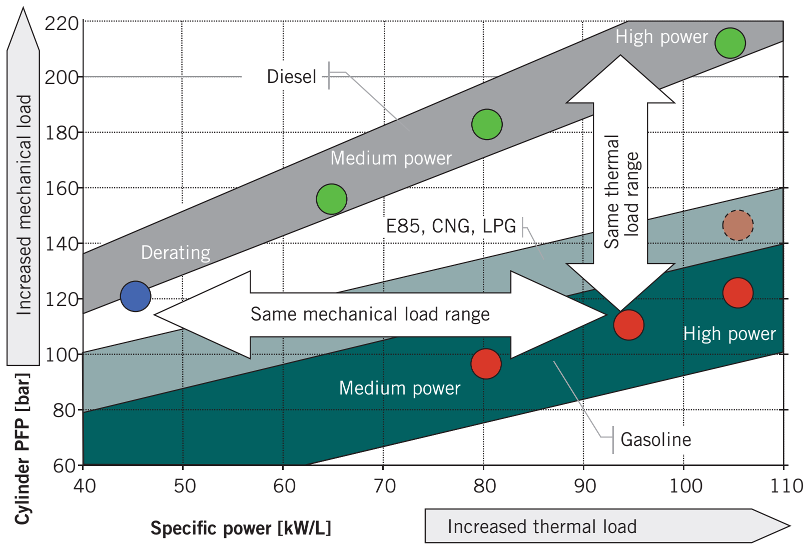
MODULAR ENGINE ARCHITECTURE: A NEW CHALLENGE

The trend to downsizing leads to increased thermal and mechanical loads on the engine components, ❸. The increased application of turbocharging and direct injection on the gasoline engines to increase specific output leads to a gradual convergence of gasoline and diesel engines in terms of cylinder pressure (increasing for the gasoline engines well above 100 bar) and inertia loading (due to reduction of rated speed of the downspeeded gasoline engines) [9].

For Diesel engines the highest power ratings are feasible with cylinder pressures around 200 bar, in combination with increased exhaust temperatures and injection pressure. On the other hand, there is an opportunity to cover mass market power requirements with reduced peak pressure around 120 bar (de-rating approach [10]).

For TC DI combustion systems, the ideal bore-stroke ratio lies between 1.05 and 1.15 for both combustion types. Therefore common bore and stroke for diesel and gasoline applications is a reality. Bore pitch – for inline engines determined by the need for cooling between the cylinders for the increased power densities – can also be commonised. Therefore the basic conditions for a unified engine family are provided.

In order to achieve future stringent targets for weight and friction, the penalties typical of the standard engine family approach have to be avoided. This can be achieved by building in a certain minimum degree of flexibility to the manufacturing facility. The main dimensions and performance variants of such an engine family are shown in ❹ for the example of a four-cylinder engine of 400 cm^3 per cylinder.



❸ Peak firing pressure requirement of gasoline and diesel engines

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		DIESEL CR-TCI				GASOLINE GDI-TCI		
CHARGING SYSTEM		WG-TC	VGT	Two-stage series-sequential	Two-stage Series intercooled	WG-TC	WG-TC + E-charger	Two-stage series-sequential
MAXIMUM POWER	kW	74	104	128	166	128	151	166
	@rpm	3700	4000	4200	4400	4500	5200	5000
MAXIMUM TORQUE	Nm	239	272	317	380	272	304	380
	@rpm	1500	1700	1500	1500	1500	1800	1500
SPECIFIC POWER	kW/l	46	65	80	105	80	95	105
PFP	bar	120	150	180	210	95	110	125
BORE	mm	77						
STROKE	mm	85.5						
BLOCK HEIGHT	mm	215						
CONROD LENGTH	mm	131				142		
COMPRESSION HEIGHT	mm	41				30		
MAIN JOURNAL Ø	mm	50				45		50
CRANKPIN Ø	mm	45	50		53	45		

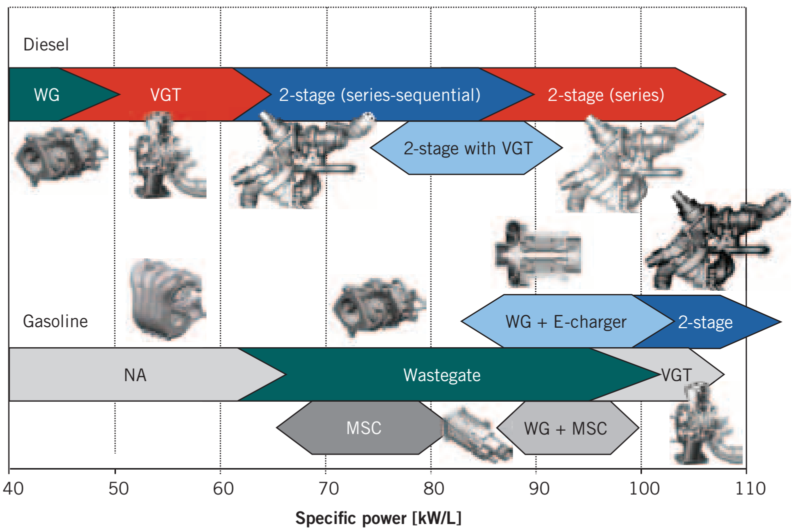
❹ Main dimensions and performance variants of a modular engine family (400 cm³ per cylinder, four cylinder)

VARIANT DIFFERENTIATION WITH THE CHARGING SYSTEM

❺ shows the current range of charging systems applied, or under development, for conventional powertrains. These systems provide a wide range of specific power and torque, covering a wide range of vehicle applications without the need for additional swept volume variants and therefore a rationalized base engine production.

By adding electrical boosting in the low engine speed range the matching of the exhaust gas turbocharger systems can be focused on the high speed area. This extends the potential application of each charging technology towards higher specific power, leading to a more cost effective overall package.

The electrical boosting may be in the form of torque input directly at the vehicle axle or at the ICE crankshaft (BSG or ISG), or indirectly via electrically driven charger air compressor. The indirect approach has the advantage of multiplying the effect of the supplied electrical boost energy.



❺ Modular engine family architecture – charging systems

MODULAR COMPONENT ARCHITECTURE – BOTTOM END

❻ shows an example of an engine family with gasoline and diesel derivates based on a cost-effective family architecture approach with common cylinder unit. The complete range of power ratings is clustered into two ratings each (medium and high power) for the diesel and for the gasoline variants, giving a total of four base engine configurations. Flexibility to machine main- and crankpin-bearing diameters in two or three steps allows a friction optimized layout for each configuration. The increased piston height of the diesel engine can be accommodated within a common block height by adjusting the conrod length. The conrods differ also in length as well as big end diameter and shaft cross-section.

For all but the high power diesel variant, a cost-effective yet low-weight aluminum HPDC cylinder block [11] can be applied. The HP Diesel with significantly higher cylinder pressure will require a further reinforcement by use of inserts or change to a higher strength alloy.

MODULAR COMPONENT ARCHITECTURE – TOP END

The cylinder heads for gasoline and diesel applications are in detail unique due the different requirements of the com-

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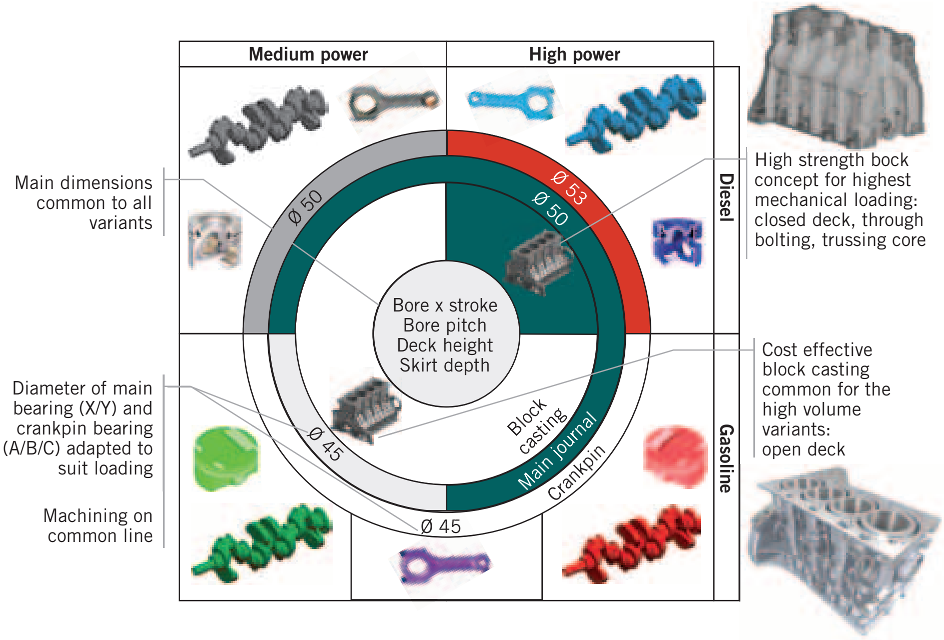


❻ Modular engine family architecture – bottom end

bustion systems on intake ports and combustion chamber geometry. The valve angle is below 5° for the Diesel and in the region of 30° for gasoline, leading to different camshaft locations. Nevertheless, it is possible to commonise flange locations for the inlet and exhaust manifolds. A common modular architecture with compact base cylinder head and separate die-cast cam carrier module allows commonisation of key machining e.g. cylinder head height, ❼.

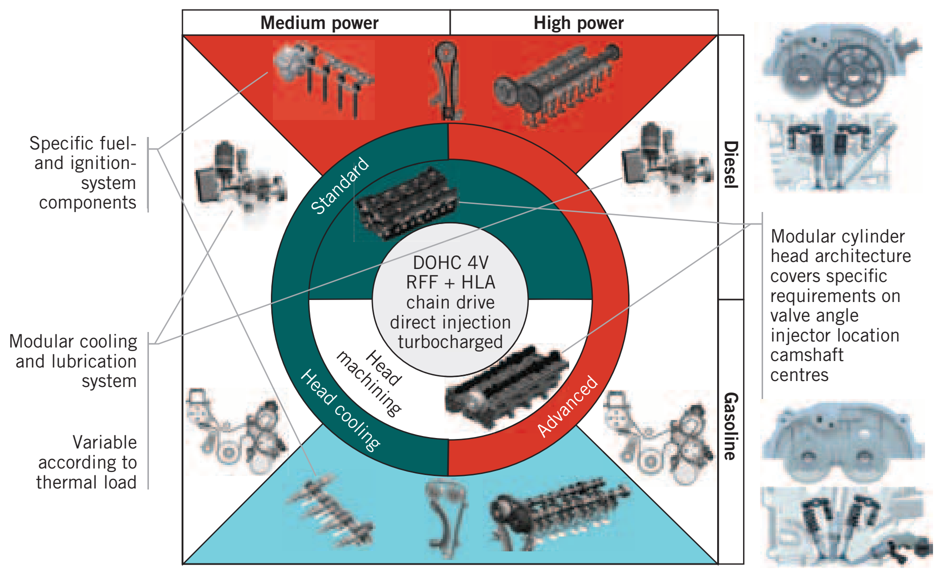
Chain timing drives have unique layouts but can be based on a common approach for assembly and sealing, allowing for assembly on a common line.

Other peripheral components such as oil filter/cooler module – in this example integrated with the auxiliary bracket – oil pan and covers can be common to all variants subject to vehicle packaging.

With this modular approach the different engine variants can be produced in parallel with a minimum number of unique components while avoiding the main penalties of a standard approach.

EXTENSION OF THE ENGINE FAMILY BY DIFFERENT NUMBERS OF CYLINDERS

A further extension of the engine family with a three-cylinder covers lower power requirements of smaller vehicles. Three- or even two-cylinder engine variants are also of interest for use in serial hybrid or range extender applications due to the lower power requirements and reduced package space for the ICE as only a small part of the overall powertrain package.



❼ Modular engine family architecture – top end

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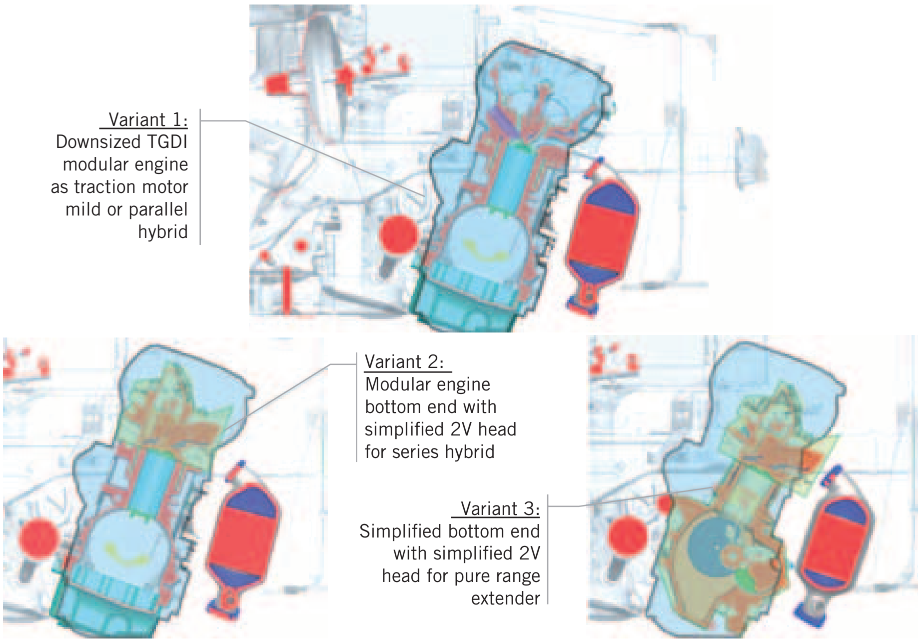

❽ Modular engine family compared to dedicated range extender

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For these applications, some simplification of the engine technology is possible due to the reduced engine operation range. For example, the gasoline engine may be simplified by change to port fuel injection, natural aspiration and fixed valve timing. This can be accomplished within the existing engine family architecture, thus maximizing the utilization of the manufacturing investment.

On the other hand, depending on the production volumes, it may be appropriate to design a more heavily customized engine taking full advantage of the opportunities for simplification. For example, for a pure range extender application a 2-V-cylinder head provides satisfactory results while resulting in a more compact overall package and lower weight [12]. These options are compared in ❽.

CONCLUSION

The internal combustion engine remains a key component of the electrified powertrain. Its design will be strongly influenced by that context. Scenarios from pure range extender for single-point operation, to a modular engine family covering the widest possible scope of applications can be imagined. The optimum solution for each vehicle manufacturer will be different depending on production volumes and vehicle product range.

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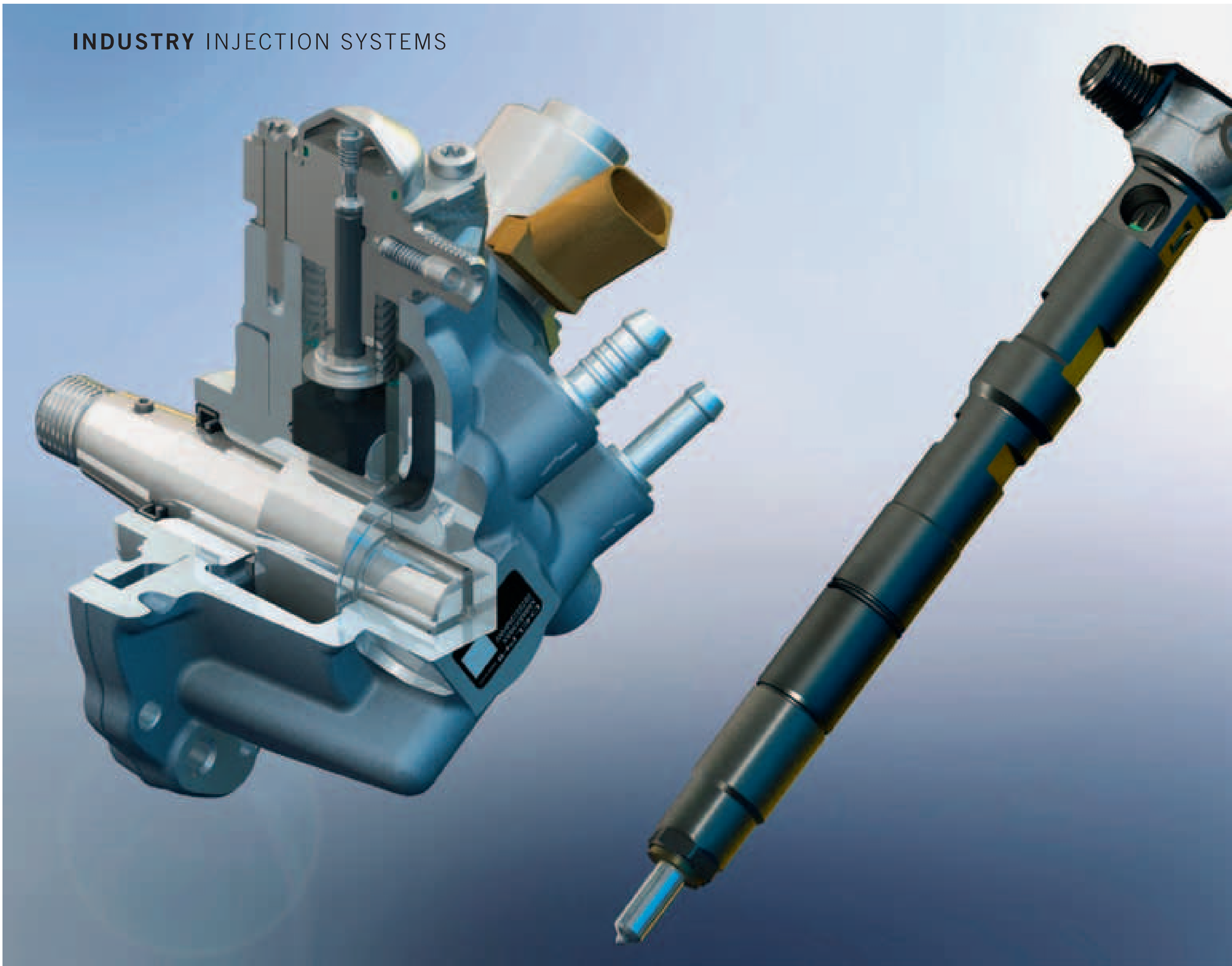

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COMMON RAIL SYSTEM FROM DELPHI WITH SOLENOID VALVES AND SINGLE PLUNGER PUMP

In order to achieve a further reduction in the emissions and fuel consumption of diesel engines, Delphi has developed a new common rail fuel injection system with solenoid valves. Its single plunger injector pump is driven at engine speed and can develop a pressure of up to 2000 bar. The engine management system offers extensive control strategies for engine and exhaust aftertreatment functions. A 1800 bar version of the fuel injection system has been used in series production since 2010.

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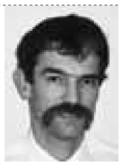

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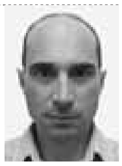

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THREE SYSTEMS FOR DIFFERENT APPLICATIONS

In the area of diesel common rail systems Delphi differentiate between three different product families, ❶: the high technology FIE path via direct acting piezo injectors; the standard high value fast servo solenoid system; and the UPCRS FIE path, via an unit pump based common rail for small diesel engines with overall swept volumes below 1.0 l for sub compact vehicles.

The direct piezo common rail system has been published many times [1], which is why in the following text is focussed on the Multec CR system with solenoid type injector and the new DFP6 pump. The solenoid injector offers a very fast injector needle lift, which supports minimum injection quantities and a stable injection slope. As an alternative to DFP6, Multec offers the option of using DFP3 type pumps with two or three plungers for larger engine applications.

A 2000 bar application of the solenoid system has been in mass production since 2008 [2, 3], and a further upgrade to 2200 bar is currently within the development phase. For Euro 5 compact ultra low CO_2 cars, the optimised Multec system was recently launched on a three cylinder engine [4]. This system features the DFI1.5 injector with 1800 bar injection pressure and multiple injection capability, combined with a novel DFP6 single plunger high efficiency, high speed, downsized common rail pump.

For future engine applications with lower power density and emissions levels up to Euro 4, the Unit Pump Common Rail System (UPCRS) is offered which is capable of 1600 bar injection pressure and multiple injections.

KEY OBJECTIVES FOR THE HIGH PRESSURE PUMP

The key objectives for the design and development of the DFP6 pump are summarised in ❷. They include: reduced mass and size of packaging; increased efficiency; 2000 bar rail pressure; and a speed capability of 6000 rpm. Operating the high pressure pump at engine speed, i.e. a 1:1 drive ratio, allows for a reduced package size and provides the required high pressure fuel delivery capacity with just one pumping plunger. A cast aluminium housing and front plate both help to reduce the pumps' weight, whilst giving maximum flexibility to reduce the package size and increase machining efficiency. To achieve the improvements in efficiency, internal losses were reduced by using the single plunger concept, the new DFP3 based hydraulic head with forged on high pressure outlet, and a compact drivetrain with reduced friction losses. To achieve 2000 bar the shoe design evolved from the well established DFP1 design into a very simple and lightweight component carrying the roller. A twin lobe cam profile was also used, which allows for synchronisation of injection to pumping with four-cylinder applications. 2000 bar rail pressure is foreseen as the current design limit and will be offered with the next pump applications.

The final design borrows heavily from both previous CR pump families, DFP1 and DFP3, whilst incorporating innovation in all areas. All this leads to the best in class pump mass of only 2.4 kg.

LOW AND HIGH PRESSURE CIRCUIT

The cam box serves as a large internal volume in which the pressurised fuel inlet

❶ Three key product families of Delphi common rail systems: direct acting piezo system (left), solenoid system (middle) and the unit pump based common rail (right)

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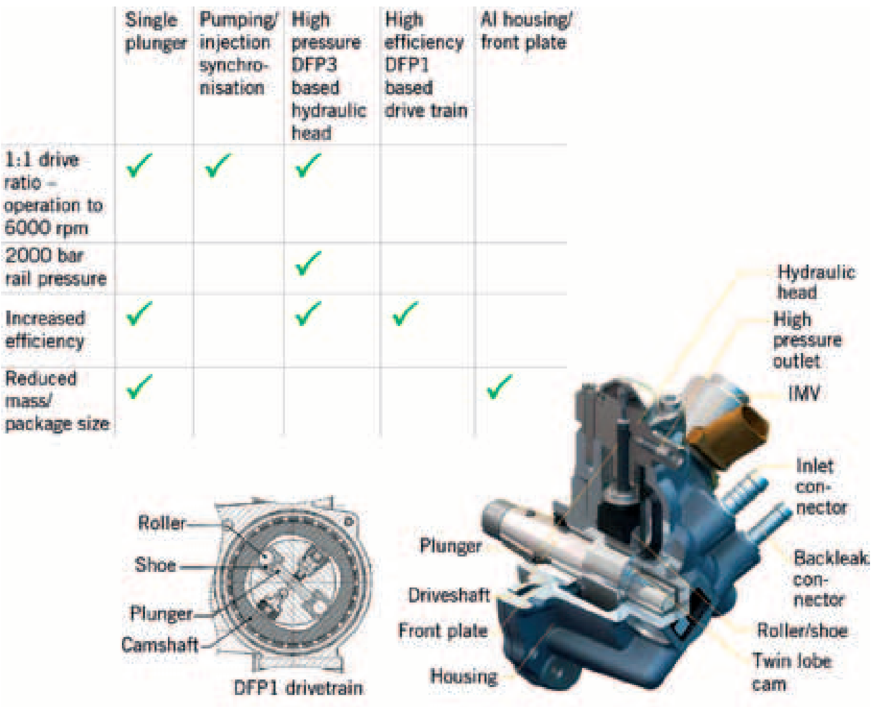


	Single plunger	Pumping/ injection synchro-nisation	High pressure DFP3 based hydraulic head	High efficiency DFP1 based drive train	Al housing/ front plate
1:1 drive ratio – operation to 6000 rpm	✓	✓	✓		
2000 bar rail pressure			✓		
Increased efficiency	✓		✓	✓	
Reduced mass/ package size	✓				✓

❷ Key drivers and objectives for single plunger pump design and development

is directly fed to avoid expensive deep hole drillings.

Two bearings support the pump's cam shaft, both of which are subject to a throughflow of fuel. By ensuring the back of both the front and rear bearings are connected to the fuel return line and are therefore close to atmospheric pressure, a pressure difference is created across the journals providing a quasi force-flow arrangement. This ensures that fresh fuel is constantly delivered to the bearings, thus reducing the operating temperature of the bearings and the pump cam box fluid. This architecture is a distinct advantage for low cranking speeds at engine start and for vehicle stop/start control systems, which in turn offers a reduction in the fuel consumption and carbon footprint as there is a flow across the journals before the pump starts to turn.

The fuel is transported via the inlet metering valve and the inlet valve into the pump's compression chamber.

With the DFP3 design, the inlet valve was contained in a small inlet valve housing separate from the pump hydraulic head. A patent for the new integrated inlet valve concept of DFP6 has been filed. The inlet valve is no longer incorporated in a separate housing; instead the valve stem is directly assembled into the head via the plunger drilling as shown in ❸ (right).

This leads to various advantages:

: avoidance of the expensive and difficult to machine metallic seal between inlet valve housing and head; this also removes any potential high pressure leak paths to the external environment

: a reduction in the large assembly loads required to seal the inlet valve against rail pressure

: more than a 50 % reduction of the dead volume, which increases the efficiency

: a reduction in the hydraulic resistance of the valve due to the size of the fuel annulus on top of the hydraulic head

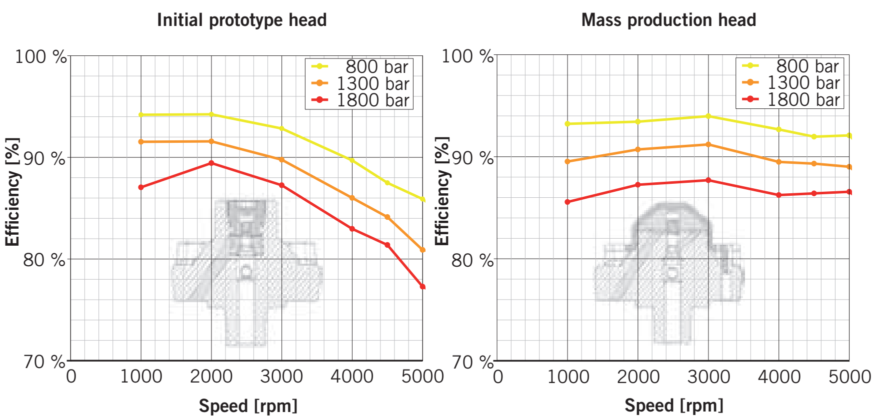
: an increase of 185 % for the inlet flow area due to removal of the separate cartridge.

The last two items generate advantages in efficiency specifically at high speed. An 8 % gain in output flow at 5000 rpm compared to the previous generation valve has been achieved.

③ compares the volumetric efficiency performance for the single plunger pump by interchanging the hydraulic head assembly from an early head with the old inlet valve design to the new integrated valve design without any inlet valve housing on one identical test pump.

Whilst the old valve design is very efficient in its original applications with pump speeds below 3500 rpm, the performance at speeds typical for 1:1 drive ratios is less impressive for rail pressures between 800 and 1800 bar. The new patented integrated inlet valve impressively shows its advantages at speeds beyond 3500 rpm: DFP6 sets the benchmark with the highest efficiency in the market.

The outlet valve is integrated into the high pressure outlet, which is forged onto the head. This leads to reduced components stress levels, and further avoidance of all potential high pressure leak paths to the environment. Metallic knife edge seal surfaces, which are difficult to control in production and which challenge the material stress capabilities have been removed by integrating both valves. Hence, an elimination of approximately 100 MPa of stress in the hydraulic head has been achieved.



❸ Volumetric efficiency effect of single plunger pump inlet valve design – lines of constant rail pressure; assembled inlet valve (left) and integrated inlet valve (right)

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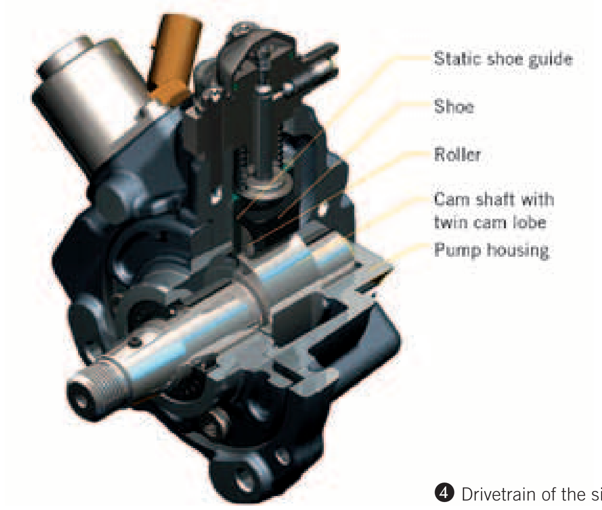


❹ Drivetrain of the single plunger pump

In addition the valves are less stressed during assembly leading to a reduction in seat distortion and leakage. The new single plunger pump is the first high pressure diesel pump which strictly avoids any potential high pressure leak path to the external environment.

DRIVETRAIN

The DFP6 drivetrain evolved from the DFP1 design, ❹. Significant developments have been made to ensure robustness at today's higher rail pressures of 2000 bar. As with DFP1, a roller/shoe assembly rides on the cam. The roller diameter has grown from 9.5 mm in DFP1 to 12 mm for DFP6 to meet the new pressure demands and to reduce the maximum Hertz stress.

A static shoe guide has been selected from various design concepts, and is pressed into the pump housing to guide the roller/shoe assembly movement. This patented solution prevents lateral rotation of the roller/shoe assembly as it passes over the cam top dead centre. This is a large supporting factor for product robustness, with the particular benefit of this concept being a reduction in reciprocating mass.

Specifically for the shoe guide design, conflicting parameters between component stress levels, parts' function, machining and assembly have had to be considered: e.g. internal machining after the shoe guide insertion into the housing to ensure perpendicularity, planarity and surface finish and to remove any assembly distortion at the final stage.

A plunger return spring is used to avoid the roller lifting off the cam profile at high speeds, i.e. to prevent so-called ski-jumping. Rolled journal bearings support the drive shaft. Different from standard solutions based on a polytetrafluoroethylene (PTFE) coating, a solution based on polyether ether ketone (PEEK) is used. PEEK offers an increased robustness at mixed friction conditions specifically after engine start when the vehicle is operated in stop/start control mode.

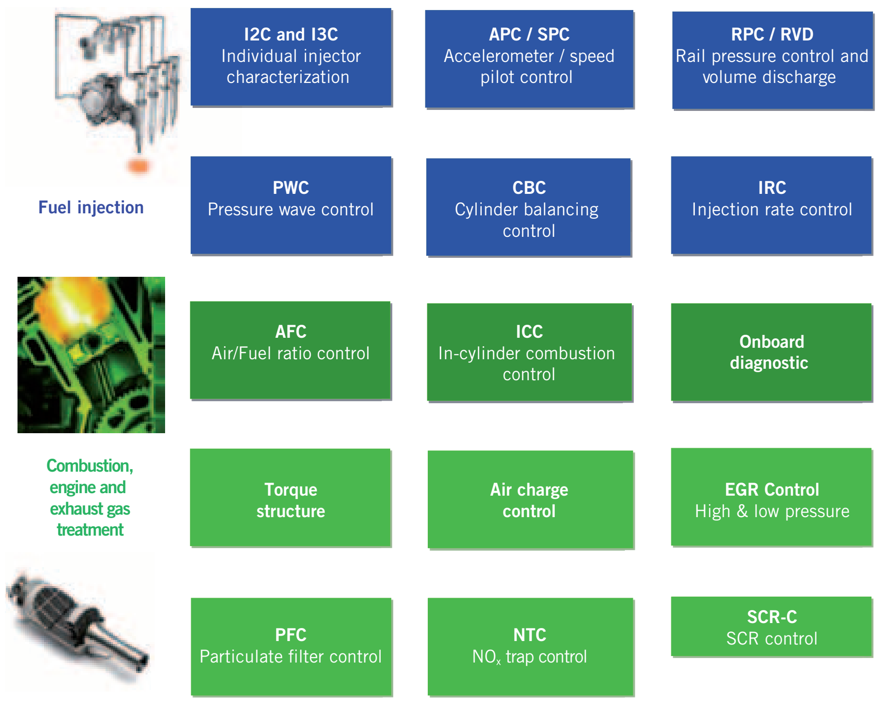
ELECTRONIC CONTROL UNIT

The DCM3 electronic control unit (ECU) family has been developed for Euro 4 and Euro 5 applications. Together with the DFI1.5 injector's battery voltage and low drive energy requirements, it offers low thermal losses and can be packaged within a small and light envelope. A 200 MHz microprocessor is available. On a three-cylinder engine, total injection flexibility is offered by providing a three injector drive bank architecture. The ECU supports additional features to improve CO_2 emissions such as vehicle stop/start control functionality, smart generator control and thermal management capabilities.

KEY CONTROL STRATEGIES

The key control strategies, ❺, focus on the fuel injection control, presented with a blue background and on the combustion engine and exhaust gas aftertreatment control, marked up in green. Only a few of the features displayed within the picture will be explained herewith.

It is well known that end of line individual injector characterization has been deployed on common rail systems in serial mass production. The most recent form of this is the I3C. Another specific



❺ FIE system's ECU ensures fuel injection control (blue) and combustion engine and exhaust gas aftertreatment control (green)

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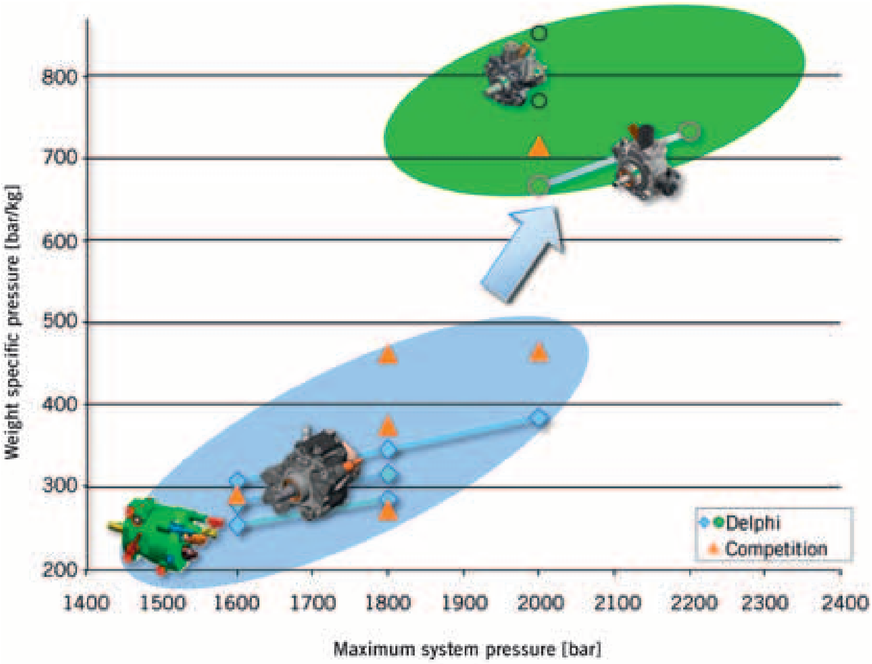

6 Rail pressure to weight ratio of common rail pumps

patented fuelling control strategy, APC, is being applied to precisely control smallest fuel injection quantities via continuous learning of injector behaviour on the vehicle. The patented PWC is ensuring consistent injection quantities in the case of non synchronised pumping events, like the use of a 1:1 drive ratio two pumping strokes single plunger high pressure pump fitted to a three-cylinder engine. The strategy takes into account the instantaneous pressure at the time of every single injection event and adapts the injection duration in order to deliver the required fuel quantity. Consequently a common pump drivetrain architecture can be maintained with the four-cylinder variant of the same engine family.

In the lower portion of 5 it is shown that a torque based engine control is used together with model based air and EGR control. The introduction of up-to-date modern exhaust aftertreatment systems has led to the implementation of multiple engine operation modes. The mode control module manages the engine mode prioritisation and selection, the transition between modes and the sequence of action on engine variables. This allows the engine operation to be controlled to maintain a seamless mode transition, without change in e.g. torque or noise being noticeable to the driver.

FUEL CONSUMPTION REDUCTION VIA MASS REDUCTION

6 shows the weight specific pressure over the system's pressure for several high pressure pumps. Of course the reduction in mass of any individual component directly supports a reduction in CO_2: Delphi's DFP1 and DFP3 pump generations (shown as images) are positioned similar to their competitors (orange coloured triangles) in this chart. However, from DFP3 to DFP6 pump generation the weight specific pressure has been more than doubled, in which the single plunger pump now represents the benchmark.

For Euro 3 systems the HP pump contributed 50 % of the FIE system's weight; however on a modern Euro 5 engine even with a three-cylinder engine the contribution is now little more than 20 %. The entire new Multec common rail FIE system has reduced its weight on Euro 5 applications to only 60 % of its former weight on Euro 3.

CONCLUSIONS

The new Multec diesel common rail fuel system was developed to achieve ultra low CO_2 emissions. This system is mainly based on the newly developed DFP6-type high pressure pump family and the fast solenoid DFI1.5 injector in combination with the 200 MHz microprocessor DCM3.7 electronic control unit. The single plunger DFP6 pump generation is a class leader. It operates at engine speed and compared to previous pump generations various enhancements have been implemented in all areas of the pump design. A new efficient ECU has been developed in conjunction with sophisticated control strategies that exploit the best capabilities of the Multec system. The complete system, with 1800 bar pressure rating has now been released on a five seater vehicle, equipped with a three-cylinder diesel engine, which has the world's lowest CO_2 emissions. This new system will be further developed for larger engine displacements including 2.0 l applications.

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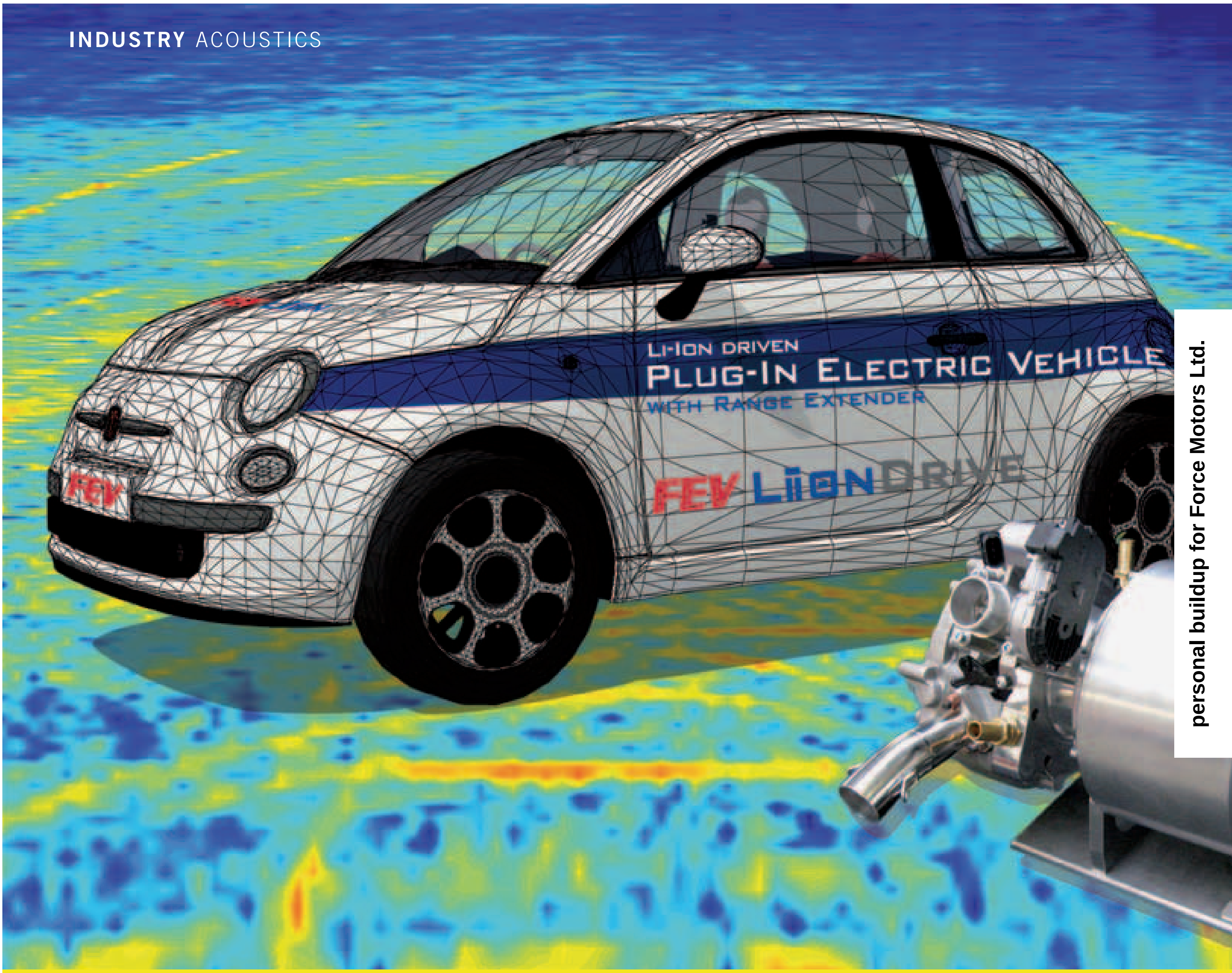

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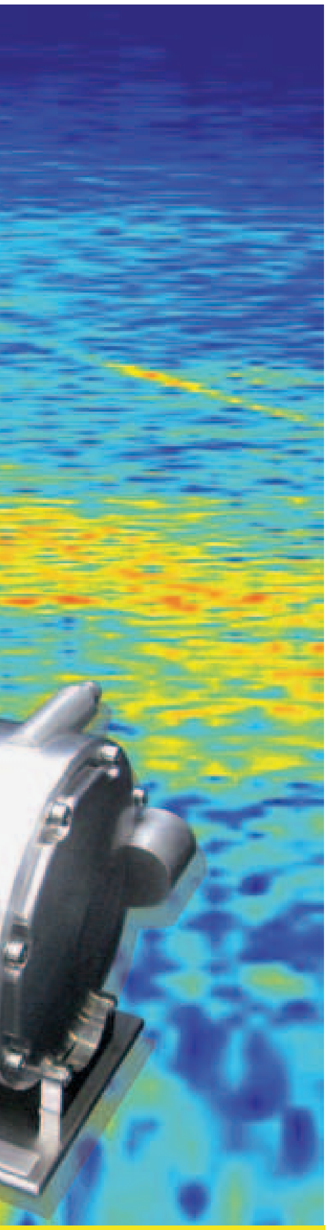
ACOUSTICS DEVELOPMENT OF RANGE EXTENDERS FOR ELECTRIC VEHICLES

Acoustics is one of the greatest challenges in the development and application of range extenders for electric vehicles. FEV Motorentechnik GmbH has developed a special operating strategy for range extenders that can achieve a significant improvement in their NVH performance.



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PLUG-IN HYBRIDS PLAY AN IMPORTANT ROLE

The very intense debate about CO_2 has sparked an interest in the electrification of the drivetrain. The vision of electricity produced from renewable energy sources that can fulfill our demand for mobility with almost no effect on the environment for the extended future is fascinating. This situation is characterized by the willingness of energy suppliers to "act" as new players on the mobility playing field.

A dynamic phase of these changes has already begun. On the one hand, actual vehicles must be provided and on the other hand, we need the infrastructure that is required to use these vehicles. As far as vehicles are concerned, so-called plug-in hybrids, which have a battery that can be charged through the electricity grid and can use both a purely electrical as well as a combustion engine to run, play an important role in addition to purely electric vehicles. The question about marketability and customer acceptance of the different range extender drive concepts when it comes to noise, vibration, and harshness (NVH) characteristics is fascinating and has not yet been answered in practice. The subject of this article is the acoustic integration of a range extender. Vehicles developed by FEV Motorentechnik GmbH were used as a basis for the discussion [1,2].

DEFINITION OF A RANGE EXTENDER

The most common question associated with battery-powered electric vehicle is: "How far can I get with the vehicle?" This question describes to the point the subconscious fear of potential buyers of being stranded one day with their vehicle with an empty battery. With electric vehicles, people constantly have to make choices: "warm feet, cool head, or getting there?" This is due to the fact that the energy that must be provided for the vehicle's climate control also comes from the battery and, as a result, limits the range, which is not great to begin with, even further. From today's point of view, the best answer to this problem is a range extender with a combustion engine, which permits a range that is virtually independent from the battery. This is expected to have an impact on the end customers' acceptance of electric vehicles.

Battery-powered electric vehicles with their currently limited usage profile seem to be ideal for use in urban areas. This application range calls for a range extender engine that helps expand the use of urban vehicles. The range extender engine increases the range of the FEV Liiondrive for instance from 80 km with purely electric drive to a total range of 300 km with one full tank of fuel. The result is a special range of requirements on the engine with the following characteristics:

: small
: light-weight (~ 50 kg)
: low-cost (~ 1000 €)
: limited output range (20 kW to 35 kW), maximum performance is reached with purely electric operation
: excellent NVH characteristics
: can be offered as an option.

Standard range extenders are suitable for battery-powered electric urban vehicles. However, in the larger classes of vehicles, full hybrids with more battery power (plug-in hybrids) are preferable. This class of vehicles can then also be used as urban vehicles

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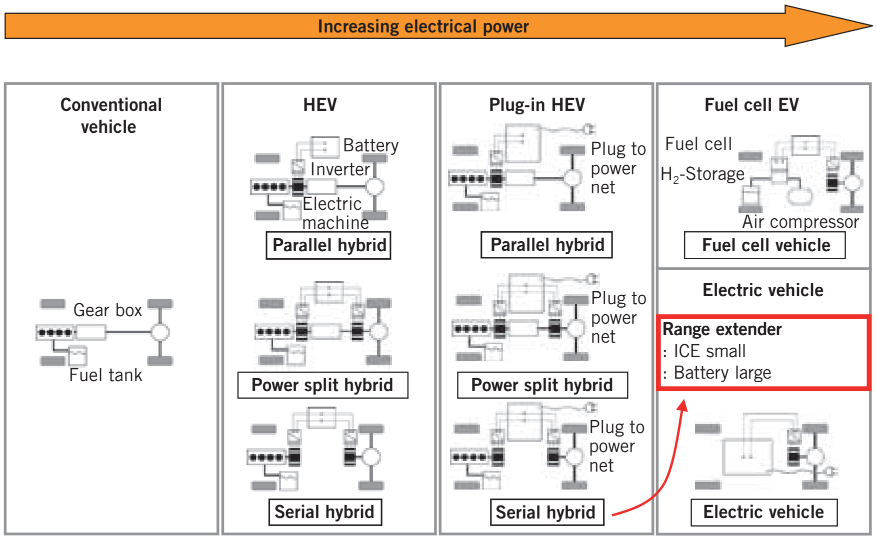
with a purely electric drive that can be charged from the power grid, ❶.



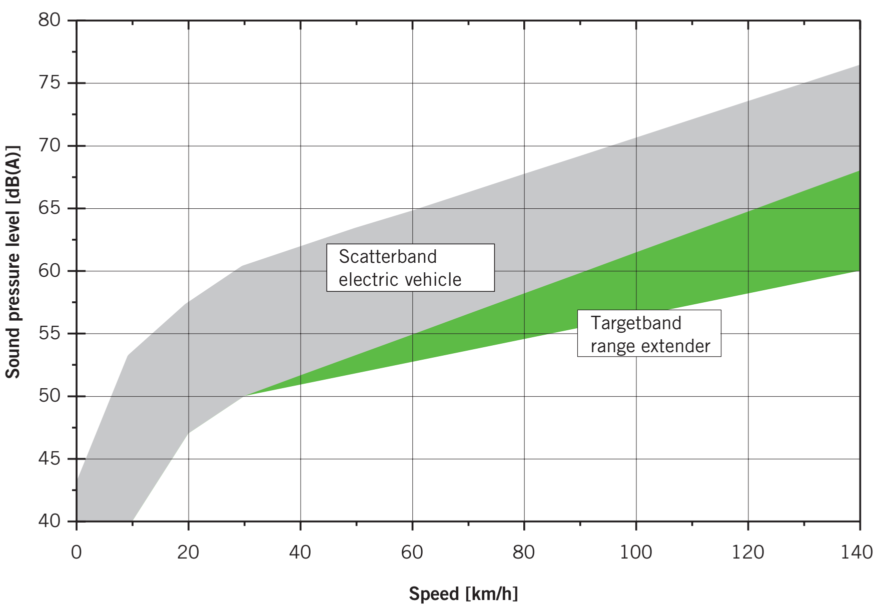
❶ Hybrid drivetrains

NVH TARGETS

Electric vehicles have different NVH characteristics than vehicles with combustion engines. The drive-related noise levels are greatly reduced and have higher frequency content. The road-induced noise that increases with the driving speed therefore remains unchanged as does the wind noise that is typically prevalent starting at approximately 100 km/h, ❷. This results in a tangible increase in the comfort of electric vehicles when driving at lower speeds in combination with an increased awareness of external sounds. In vehicles with an additional combustion engine, this positive experience can end abruptly once the combustion engine starts up. An excerpt of NVH targets defined for a range extender engine from the customer's perspective shows an apparent contradiction to the requirements specified earlier:

: "no noticeable, additional vibrations"
: "idles like a normal gasoline engine"
: "I think of it like a refrigerator; it automatically switches on and off and hums quietly"
: "should not sputter like a single-cylinder motorcycle"
: "should not howl like a scooter".

Implementing these statements in the form of technical targets means that 20 to 35 kW should be provided without noticeable vibrations and at an interior sound level of approximately 40 dB(A). In order to find an acceptable engine keeping NVH aspects in mind, other factors must be considered in the planning phase. A revving engine sounds unnatural in combination with a standing or slowly moving vehicle ("vehicle is ready to pounce"). On the other hand, the individual cylinder events can be clearly heard at low rotational speeds in engines with a low number of cylinders ("motorcycle sound"). Above all, there is the question to what extent the engine noise in the vehicle can be reduced using secondary measures, since the above aspects will then have an increasingly smaller impact. The result of masking the road and wind noise shares – without considering special spectral masking effects – is a speed-dependent target range for the sound pressure level of a range extender in the passenger compartment, ②.



❷ Target noise values in the vehicle interior (FEV scatterband)

ENGINE DESIGNS

In addition to reciprocating piston engines with one to three cylinders, rotary engines can also be considered for use as range extender engines. In the long term, fuel cells seem to have the greatest potential to serve as a quiet source of power. A conceptual evaluation of whether these kinds of engines are suitable is performed considering the criteria costs, manufacturability, weight, installation space, consumption, emissions, and NVH. As a combustion engine, the single-disk rotary (Wankel) engine has advantages in terms of package, weight, and NVH compared to smaller reciprocating piston engines. In particular the single-cylinder engines, when used without any additional measures, must be considered to be critical in terms of NVH. ❸ shows the package situation of different range extender modules.

Through the FEV-VINS transfer path analysis, we know the general composition of drivetrain-excited interior noise and vibration in vehicles. We make a distinc-

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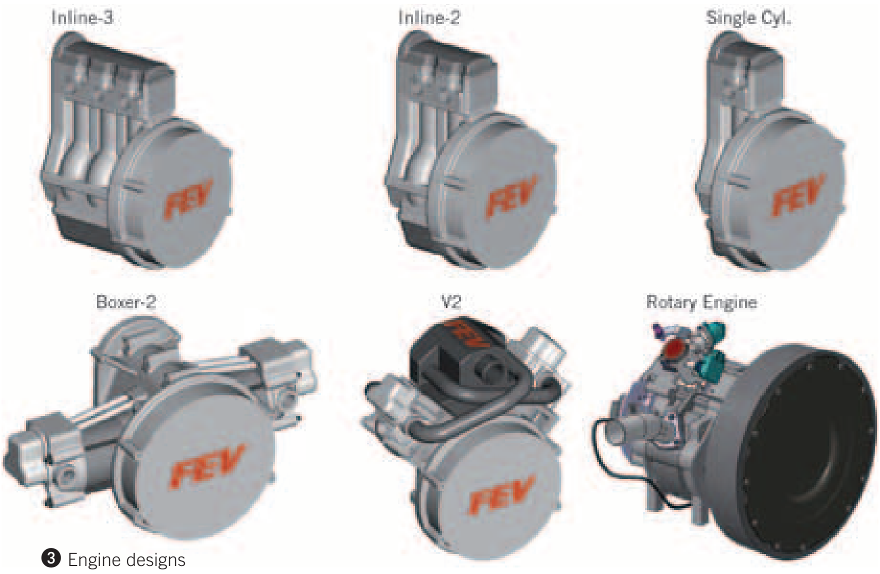

❸ Engine designs

tion here between structure-borne and air-borne shares.

STRUCTURE-BORNE SOUND

The vibrations of the engine are audible in the interior as structure-borne sound – they often dominate the interior noise in the frequency range up to 1 kHz – and at the same time lead to vibrations of the steering wheel and the seats. However, the whole body vibrations of the engine are irrelevant for the outside noise. We basically distinguish between gas force and inertia force-borne vibrations, ❹. Depending on the load and type of design, gas force-borne vibrations prevail at speeds of up to approximately 3000 rpm. Then, at higher rotational speeds, the inertia-borne vibrations become important due to their quadratic dependency on the rotational speed. This is why it is important to know as much as possible about the operating strategy when designing the engine.

As a response to the cyclic gas forces in the combustion engine, a dynamic roll moment forms about the crankshaft axis. Although increasing the flywheel mass – possible here thanks to the electrical generator with the most rigid connection possible – reduces the rotational speed irregularity, it will not reduce the rolling motion of the entire engine. Even the single-disk rotary engine, which is known for its low vibrations, produces considerable rolling motions around the crankshaft axis of the 1st engine order. Engines with multiple cylinders continue to have advantages here. FEV is working on finding innovative solutions in this area in order to minimize the gas force reaction motions of the entire range extender assembly. A special feature of a range extender assembly is the external, static freedom of torque, since the combustion engine torque is compensated directly by the inverse electric generator torque.

High operating speeds are required especially for a compact range extender assembly with high power density. It is recommended here to use engines with low, free inertia forces; ideally, the rotary piston engine even is inertia force-free. With additional technical effort, free inertial forces and torques can be compensated using typical balancing systems. They are shown in gray in ④.

Attachment points with sufficient initial dynamic body stiffness – e.g. 10^7 N/mm – must be provided for the mounting of the entire range extender assembly. For this purpose, additional body reinforcement is required depending on the installation position in the vehicle. The mounting concept of choice seems to be a three-point suspension with two support mounts in the neutral rotational axis of the engine and a soft anti-roll bar link, which – in the absence of static torque – merely has to reduce the engine motion due to road vibrations and vehicle acceleration.

AIR-BORNE SOUND

Typical sources of air-borne sound from the range extender unit are direct radiated engine noise, intake noise, and exhaust noise. They determine the outside noise and also contribute to interior noise. All previous investigations have shown that the noise levels of the alternator, e.g., due to electromagnetic field excitation, are negligible compared to combustion engine noise.

Direct engine noise is made up of mechanical and combustion-generated noise levels. In addition to optimizing the structure of the engine, an acoustic capsule can provide a good means for considerably reducing the engine noise. The task of the charge exchange devices is, in addi-

	I1	I2			V2 (90°)	B2	I3	Rotary
Pin offset		360°	180°	90°	360°	180°	120°	
Firing interval	even 720°	even 360°	uneven 180°-540°	uneven 270°-450°	uneven 270°-450°	even 360°	even 240°	even 360°
Mass balance system		Bal. shaft 1st order		Bal. shaft 1st order	Counter weights		Bal. shaft 1st order	Counter weights
Mass force 1st order	1 F_{01} (osc. v.)	[2 F_{01} (osc. v.)]	0	[1,4 F_{01} (osc. v.)]	[F_{01} (rot. +)]	0	0	0
Mass force 2nd order	1 F_{02} (osc. v.)	2 F_{02} (osc. v.)	2 F_{02} (osc. v.)	0	1,4 F_{02} (osc.)	0	0	0
Mass moment 1st order	0	0	a F_{01} (osc. h.)	0,7 a F_{01} (osc. h.)	0,5 b F_{01} (rotating)	b F_{01} (osc. v.)	[1,7 a F_{01} (osc. h.)]	0
Mass moment 2nd order	0	0	0	a F_{02} (osc. h.)	0,7 b F_{02} (osc. h.)	b F_{02} (osc. v.)	1,7 a F_{02}	0
Torque fluctuation order	Harm. 0,5th	Harm. 1st	Inharm. 0,5th; 1,5th; 2nd; 2,5th	Inharm. 0,5th; 1st; 1,5th; 2,5th	Inharm. 0,5th; 1st; 1,5th; 2,5th	Harm. 1st	Harm. 1,5th	Harm. 1st

a = bore pitch F_{01} = mass force amplitude of 1 cylinder, 1st order
b = bank offset F_{02} = mass force amplitude of 1 cylinder, 2nd order

❹ Gas and inertia forces

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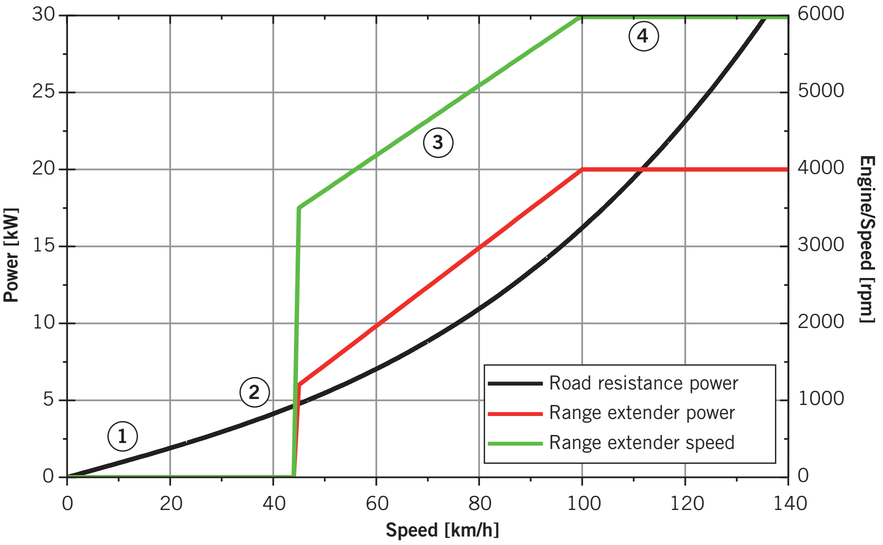

❺ Range extender operating strategy of the FEV Liiondrive

tion to filtering the air and cleaning the exhaust, to lower the intake and exhaust noise. This presents a significant challenge, particularly on the exhaust side, which has up to 20 dB higher excitation levels than the intake side. To manage the high and low-frequency pulsation excitation caused by single and two-cylinder engines, which the ear perceives as individual events at low speeds, large volumes and cross-section changes are required with a correspondingly large amount of installation space. The slot-controlled single-rotary engine can produce extreme exhaust impacts, thus exciting the silencer structure to produce impulsive shell noise.

OPERATING STRATEGY

The operating strategy provides – similarly as with hybrid vehicles – an additional degree of freedom for optimizing NVH. Current investigations are therefore dealing with the application of a function and noise-optimized operation strategy to range extenders. In addition to providing power, the range extender can also take over thermal tasks for charging the battery and controlling the climate of the interior compartment.

First applications consist of using the range extender, which is activated when the charge level of the battery was low, at stationary operating points. It is soon becoming clear that, when the vehicle is stationary and at low driving speeds below 30 km/h, operating the combustion engine should be avoided if possible from a NVH point of view, since the noise it emits is not sufficiently masked by the driving noise, ②.

At the moment, FEV assumes that the driver can use an operating mode switch to preselect the following range extender modes: automatic, electric, and operating range. More recent operating strategies have shown clear progress here under NVH aspects. Since range extenders can only provide power with an optimum degree of efficiency, if the generated energy is directly transmitted to the engine and not stored as battery-powered electric energy, the operating characteristics are currently coupled to the torque regulator via the accelerator pedal. The following operating strategy is used here, ❺:

1. no range extender operation unless absolutely necessary (thermal management, battery state of charge) below 30 km/h
2. hysteresis range for start and stop between 30 and 50 km/h
3. linear velocity-dependency with increase from the lower to the upper operating speed between 50 and 100 km/h
4. operation at the upper operating speed (maximum power in the range masked by other noises) from 100 km/h.

This strategy features the highest degree of masking due to the driving noises. The dynamic impression of a speed-dependent operating rpm is additionally supported by coupling the throttle valve position to the accelerator pedal. By closing the throttle valve in deceleration mode, we not only achieve a positive load relationship of the range extender noise, but can also prevent inadvertent electric output peaks caused by adding the recuperation power of the drive motor and the range extender unit.

CONCLUSION

Integrating a combustion engine-operated range extender into an electric vehicle represents a great challenge when it comes to noise, vibration, and harshness. The tough requirements in terms of costs, weight, performance, package, consumption, and emissions seem to be incompatible with the desire for supportive operation. After two years of development, we can see very promising results that can ease the conflict between these objectives. In addition to designing a range extender unit that is optimized with regard to function using innovative solutions, the operating strategy that must be selected provides a degree of freedom that can be used systematically to influence the characteristic of the preferred NVH behavior.

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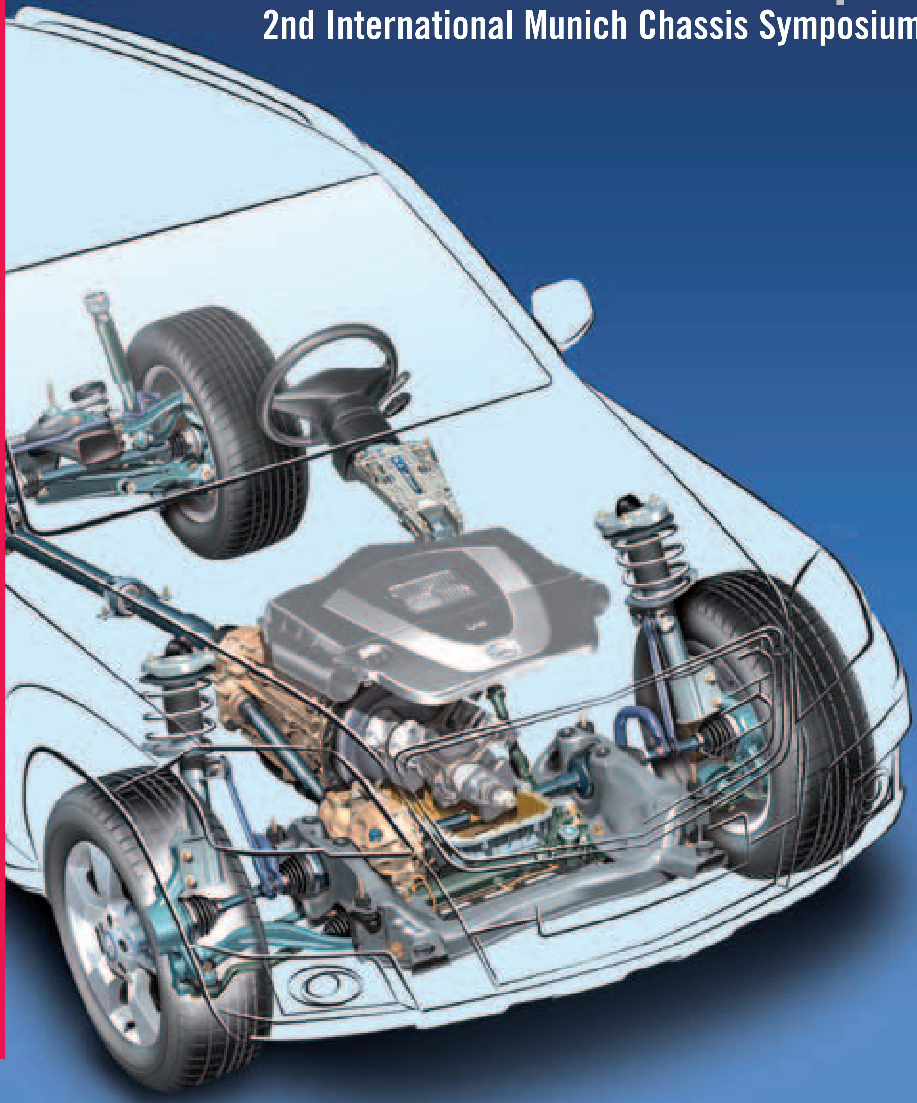

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OXYMETHYLENE ETHERS AS DIESEL FUEL ADDITIVES OF THE FUTURE

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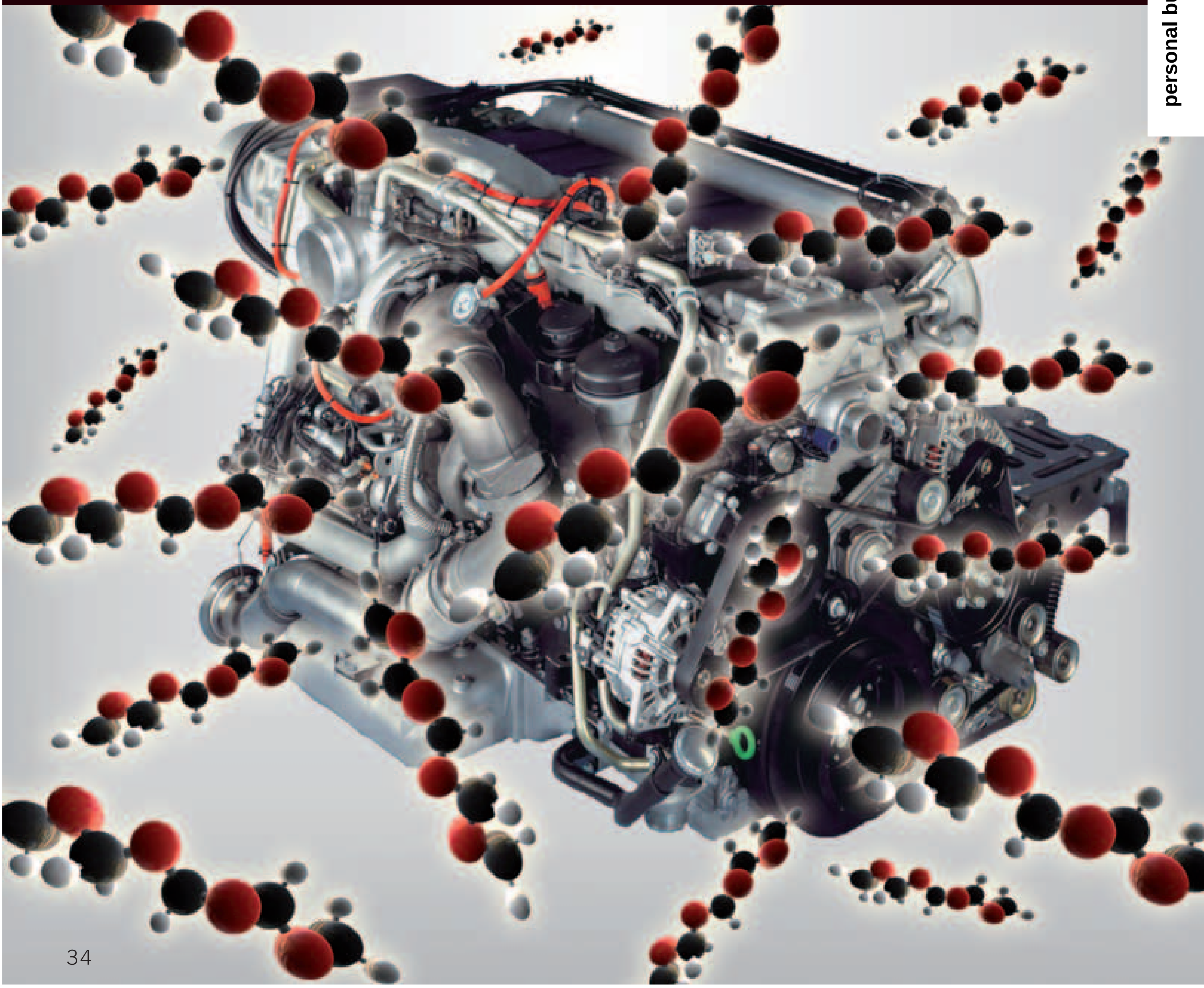
The limited reserves of easily accessible fossil raw materials and the climate-related effect of their use are driving the synthetic production of fuels as non-fossil energy sources. The economical boundary conditions make the realisation of ecological requirements such as the use of residual biomass and the recycling of CO_2 considerably more difficult. MAN Truck & Bus and Emissionskonzepte Motoren analysed synthetic fuels of the type oxymethylene ether (OME) as diesel fuel additives, which are relatively easy to synthesize from methanol.



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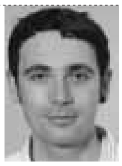

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MOTIVATION

European climate protection legislation requires second generation fuel components to be introduced in the near future [1]. This makes the search for alternatives to Fame (Fatty Acid Methyl Ester) and HVO (Hydrogenated Vegetable Oil) for use as bio-components for diesel fuel a top priority. Synthetic fuels made from residual biomass can make a significant contribution to the production of fuel additives via the intermediate product methanol [2]. CO_2 can be recycled into methanol by reaction with electrolysis hydrogen [3]. Methanol is a petrol and not suitable as a diesel fuel component [4, 5], but can be dehydrated to form dimethyl ether (DME) with a high cetane number. DME burns in the diesel engine without producing soot and offers the basis for extensive NO_x reduction inside the engine. However, as a liquefied gas in pressure tanks, DME has considerable disadvantages compared with conventional fuels in the established fuel logistics process [6]. That is why the search is on for high-molecular and therefore liquid ethers that can be produced from methanol. The simplest representatives of these compounds are the easily accessible oxymethylene ethers (OME) [7]. Analogously to DME, OME was selected as the short form for the oligomeric polyoxymethylene dimethyl ether, $CH_3O(-CH_2O-)_nCH_3$. These ethers have a chain structure made up of oxymethylene units ($-CH_2O-$), the number of which (n) determines the molecular size and properties, ❶.

Methanol derivatives of the type OME, which are similar in terms of their physical properties to diesel fuel, have numerous advantages over methanol:

: high self-ignition properties (see cetane numbers, ①)
: miscible with diesel fuel in any desired concentration
: good material compatibility
: no toxicity.

The simplest OME representative, monooxymethylene dimethyl ether OME (n = 1), which has already been thoroughly investigated as a diesel additive for reducing emissions and is better known as methylal [8], is too volatile for use as an admixture for diesel fuel. Therefore the high-boiling 1:1 mixtures of tri and tetraoxymethylene dimethyl ether (OME n = 3,4), which are being investigated with regard to their emission-reducing properties as part of this work, are suggested as additives [7]. Dioxymethylene dimethyl ether (OEM n = 2) assumes an intermediate position with a boiling point of 105 °C. It is expected that such volatile diesel fuel components increase the proportion of a premixed combustion. OME (n = 2), which was only available in small sample quantities, was investigated using a single-cylinder engine.

TEST ENGINES

The test series were conducted on two different engines, an externally turbocharged single-cylinder research engine D2061LX

	UNITS	DK B7(EN590)	DME	OME N=1	OME N=2	OME N=3	OME N=4
FORMULA		$CH_{1.83}\,O_{0.01}$	C_2H_6O	$C_3H_8O_2$	$C_4H_{10}O_3$	$C_5H_{12}O_4$	$C_6H_{14}O_5$
MELTING POINT	°C		-141	-105	-70	-43	-10
BOILING POINT	°C	180-390	-25	42	105	156	201
DENSITY, LIQUID AT 15 °C	kg/m³	820-845	668	867	961	1021	1059
KINEMATIC VISCOSITY AT 40 °C	mm²/s	2-4.5	<0.1		0.64	1.05	1.75
CETAN NUMBER		>51	55	50	63	70	90
O-CONTENT (%)	m-%	1.2	34.7	42.1	45.3	47.1	48.2
VOLUMETRIC CALORIFIC VALUE H_u AT 15 °C	MJ/l	35-36	18	20*		19*	19*

❶ Physical characteristics of diesel fuel, DME and OME (n=1-4) *Calculated values

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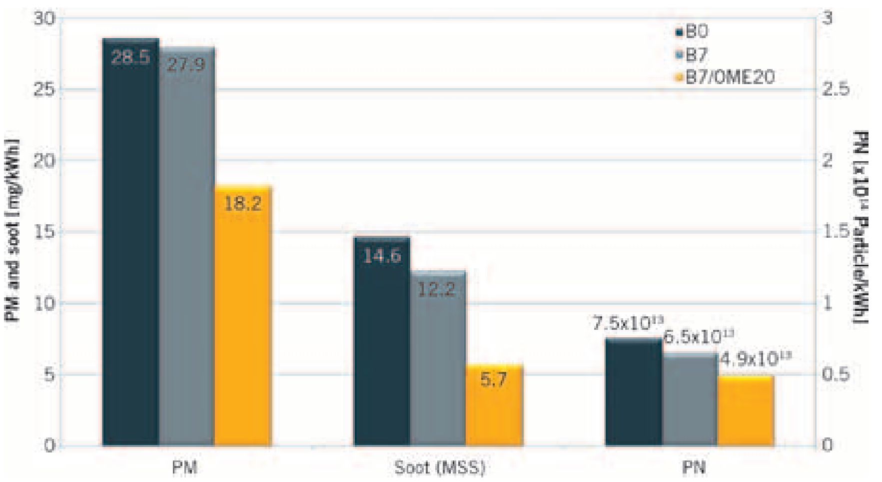
	UNITS	LIMITS DIN EN 590:2010		B0	B7	B7/OME20
		min.	max.			
DENSITY AT 15 °C	kg/m³	820	845	838	836	876
KINEMATIC VISCOSITY AT 40 °C	mm²/s	2	4.5	2.42	2.52	1.73
CLOUDPOINT	°C				-8	-6
SULPHUR CONTENT	mg/kg	-	10	7	5.3	3
LOWER CALORIFIC VALUE AT 15 °C*	MJ/l			36.0	35.5	33.3
CETAN NUMBER		51	-		52.5	55
FAME CONTENT	V-%	-	7	0	6.3	4.8
WATER CONTENT	mg/kg	-	200		59	98
OXYGEN CONTENT	m-%			0	1	12
CARBON CONTENT	m-%			86.4	85.6	75.5
MONO-AROMATICS	m-%			24	20.1	14.6
DI-AROMATICS	m-%			1	1.6	1.2
TRI+-AROMATICS	m-%			0.2	0.1	0.1
POLY-AROMATICS	m-%			1.2	1.6	1.3
CO_2 AQUIVALENT*	kg/kg			3.17	3.14	2.77

❷ Fuel analysis *Calculated values



❸ Raw particulate emissions in the ESC for the six-cylinder engine

(capacity: 1.75 l, power output: 55 kW at 1800 rpm) and an MAN six-cylinder engine D2676LOH24 (capacity: 12.4 l, power output: 353 kW at 1900 rpm). With its two-stage turbocharging, common rail high-pressure injection with max. 1800 bar, λ-regulated EGR and an oxidation catalytist, this engine complies with the Euro V emission limits. As such, it can be considered a basic engine for the Euro VI era [9]. The oxidation catalyst almost completely eliminates the organic-soluble components of the particulate emissions, thus enabling compliance with the Euro V particulate limits. However, for the investigations described here, only the raw emissions without the oxidation catalyst were considered.

MEASUREMENT EQUIPMENT

Gaseous emissions were measured at both test benches with a multi-component exhaust analyser (MEXA 9000 for CO, CO_2, NO_x, HC) and an Ansyco FTIR for other gaseous components. For the particulate emissions, an AVL MSS 483 (Micro Soot Sensor) was used for soot measurement (elemental carbon, EC) at both test benches.

On the six-cylinder engine, a Nova Microtrol 4 (partial flow dilution tunnel for gravimetric PM determination), and an AVL APC 489 (advanced particle counter for particle number) were additionally used. The sampling was carried out on both engines in the raw exhaust. For fuel consumption, a Rheonik (RMH 03) was used on the six-cylinder engine and an AVL 733 fuel balance on the single-cylinder engine. Cylinder pressure indication was carried out at both test benches using an AVL Indimodul 621.

FUELS AND BLENDS

An overview of the fuels and fuel blends used is provided in ❷. A comparison of the properties of the B7/OME20 fuel (B7 with a fraction of 20 Vol. % OME, n = 3,4) with the requirements of the standard EN 590:2010 shows that the two largely correspond. Only the viscosity is slightly lower and the density higher. OME was provided by BASF (Ludwigshafen) and blended by ASG (Neusäß). The lower calorific value of the blend is 33,3 MJ/l, 6 % lower as B7. All fuels were in barrels and fed into the test bench's fuel system using fuel pumps.

REDUCING EMISSIONS USING OXYMETHYLENE ETHER BLENDS

On the six-cylinder engine, fuel consumption remains almost unchanged in the comparison between pure mineral diesel (B0) and diesel with a Fame content of 7 % (B7). The nitrogen oxide emissions are slightly reduced – by 5 %. With B7/OME20, the NO_x emissions are decreased by 25 %. This effect is due to the lambda-controlled EGR of the six-cylinder engine. A gravimetric additional consumption of 13 % was measured. The calorific value-based fuel consumption, however, remains the same. At 19 MJ/l, the calorific value of OME (n = 3,4) is 45 to 50 % lower than that of diesel due to its oxygen content of 47 %, but higher than that of methanol (15 MJ/l) and DME (18 MJ/l). The CO_2 emissions for the two test cycles

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are calculated from the fuel composition and consumption. In comparison to B7, the CO_2 emissions with B7/OME20 are virtually constant with a slight increase of 1.2 % in ESC and 0.5 % in ETC.

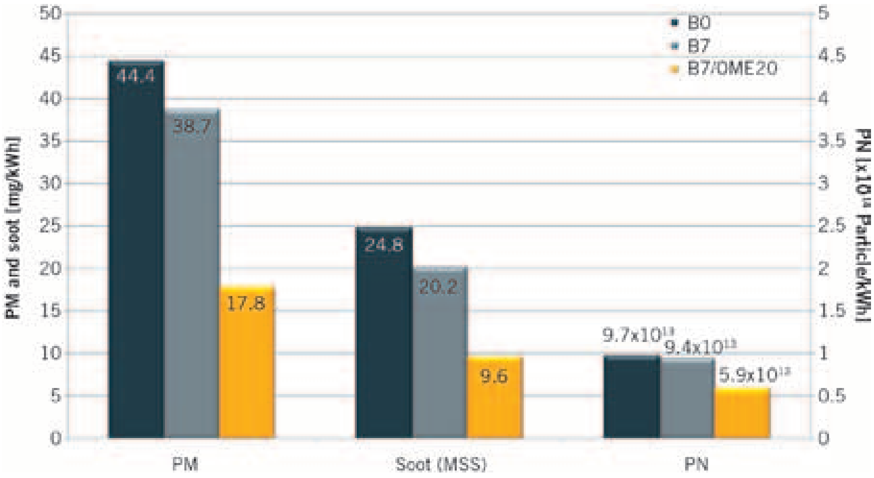
The particulate emissions of the six-cylinder engine determined in the ESC (European Stationary Cycle) for the three fuels are summarized in ❸. As expected, the most marked feature is the soot-reducing effect of the OME additive. Comparing B7 with B7/OME20, soot emissions (MSS) are decreased by 60 %. The gravimetric particulate mass (PM) is reduced by 40 % and the particle number by 25 %.

❹ shows the particulate emissions in the ETC (European Transient Cycle) for the different fuels. The gravimetric PM emissions and the soot emissions are diminished by 50 %. The particle number is reduced by 40 %. Worth noting is the significant particle reduction in the ETC compared with the stationary test. The soot-reducing effect of the B7/OME20 detected in the overall engine performance map is illustrated in ❺. The percentage improvement in the main working range of a commercial vehicle engine is up to 90 %. None of the results shown so far are NO_x neutral. At individual points with the same EGR controller position, a more significant soot reduction was detected.

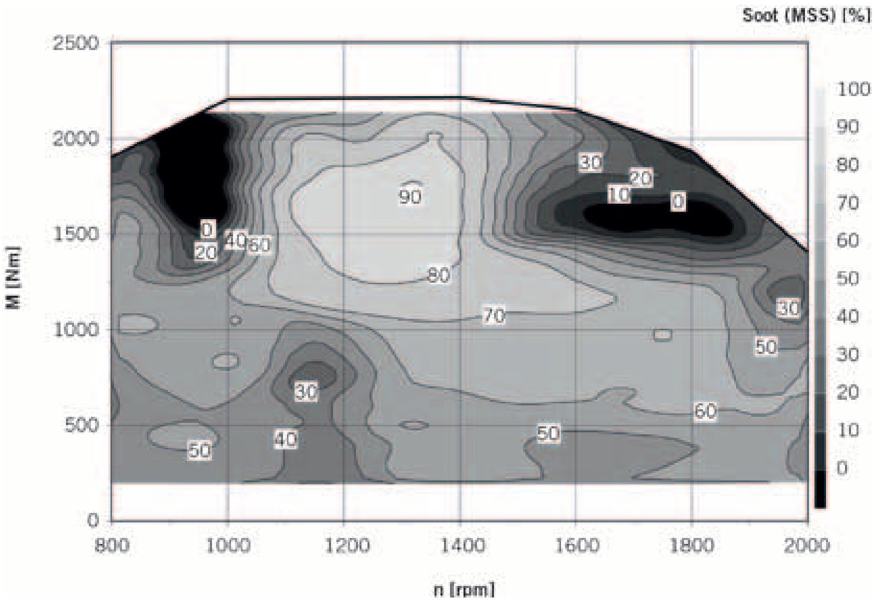
The tests on the D2061 single-cylinder were carried out at four selected operating points, ❻ and ❼, with varied engine speed and load. In the series of tests on the single-cylinder, B0/OME10 with 10 Vol. % OME (n = 2) was added to the additive-free diesel (B0). The basic measurement was performed with B0 at all operating points with 20 % EGR rate and a rail pressure of 1800 bar in the common cail system. The increase in NO_x emissions during operation with B0/OME10 was offset by adjusting the EGR rate to achieve an identical NO_x level to the basic measurement. The NO_x emissions vary between 8 g/kWh (OP1), 2.5 g/kWh (OP2 and OP3) and 2 g/kWh (OP4).

With B0/OME10, a reduction of 30 to 40 % in soot emissions can already be achieved in the selected operating points. At low load and high engine speed, the soot-reducing effect on the single-cylinder is less marked than at the usual points, ⑥ and ⑦.

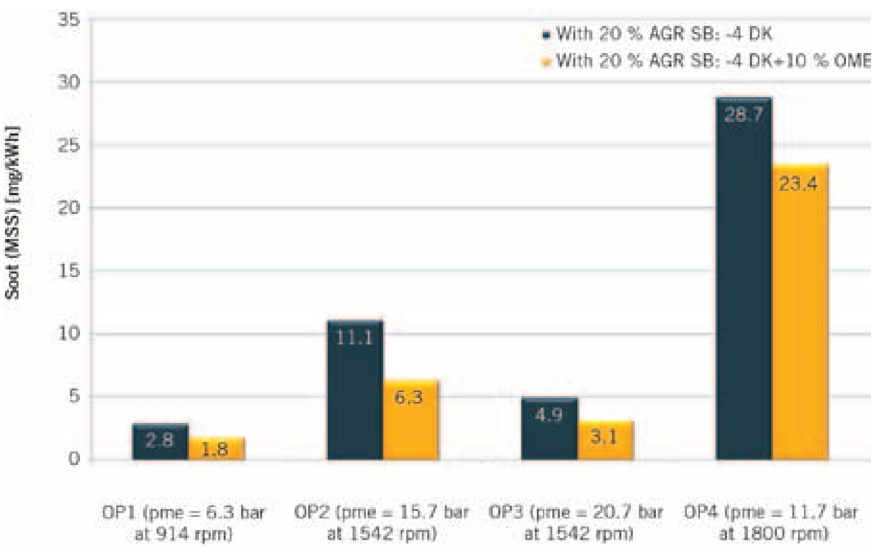
A look at the calorific value-based fuel consumption of the mixture does not pres-



❹ Raw particulate emissions in the ETC for the six-cylinder engine



❺ Partial-load performance map of soot reduction using B7/OME20



❻ Single-cylinder soot emissions using BO/OME 10

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sure any additional consumption. The high pressure indication shows no discernible difference in combustion characteristics either on the six-cylinder engine or the single-cylinder.

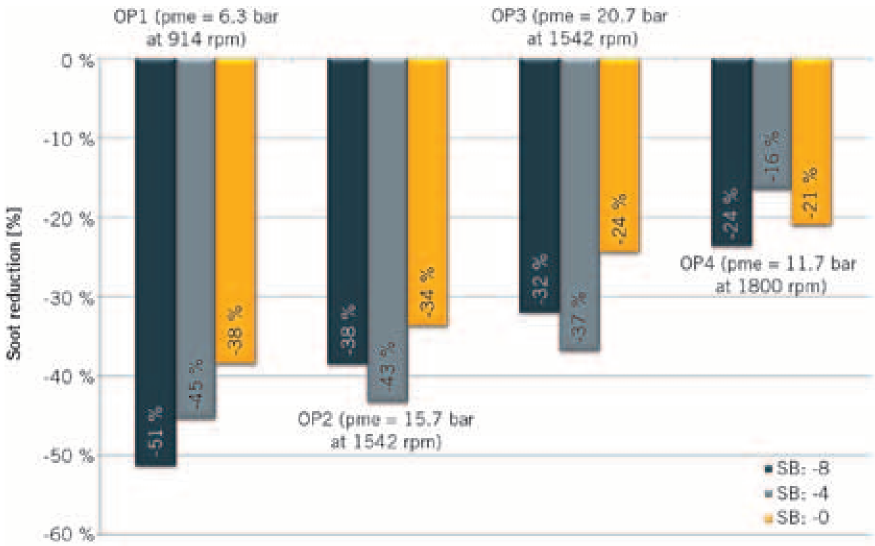
OUTLOOK

There are already a large number of scientific findings relating to the soot-reducing potential of the compounds containing oxygen (particularly of ethers, esters and carbonates) in the diesel engine combustion. Information about the influence of oxygen content and molecular structure (which are often linked to increases in NO_x emissions) on the soot-reducing effect is often contradictory [10]. The efficiency of the engine combustion of OME can be higher than that of diesel fuel due to the mole number increasing effect similar to that of methanol [11]. In the search for optimal oxygenate additives taking account of emission reduction and production costs, the three-year research project "Ereka" [12] was initiated with the institute for internal combustion engines at the Technical University of Munich (Professor Wachtmeister) as the research center and other industrial partners.

SUMMARY

Limited fossil fuels resources and the climate change associated with the CO_2 problem demand a shift from conventional towards more renewable fuels. We propose as a blending conponent for diesel fuels the OMEs. They can be easily produced via syngas/methanol from waste biomass or from recycling of CO_2 by H_2. The benefits of the OMEs are not only their physical properties, being similar to conventional diesel fuels, but also their significant reduction of soot emissions and additionally the increasing EGR compatibility for internal NO_x reduction and their relatively low production costs compared to those of hydrocarbons.

Use of the fuel blend B7/OME20 shows a reduction in emissions in the ESC/ETC cycles with PM (-40/-50 %), soot (-60/-50 %) and particulate number (-25/-40 %). Particularly notable is the marked reduction in particles of over 70 % in a broad performance map area. Even just a small proportion of OME in B0/OME10 led to a decrease of over 40 % in particle emissions in the single-cylinder engine. OME blended diesel fuels have the potential of significantly reducing local emissions as well as global CO_2 emissions.



7 Single-cylinder soot emissions – percentage change (BO/OME 10)

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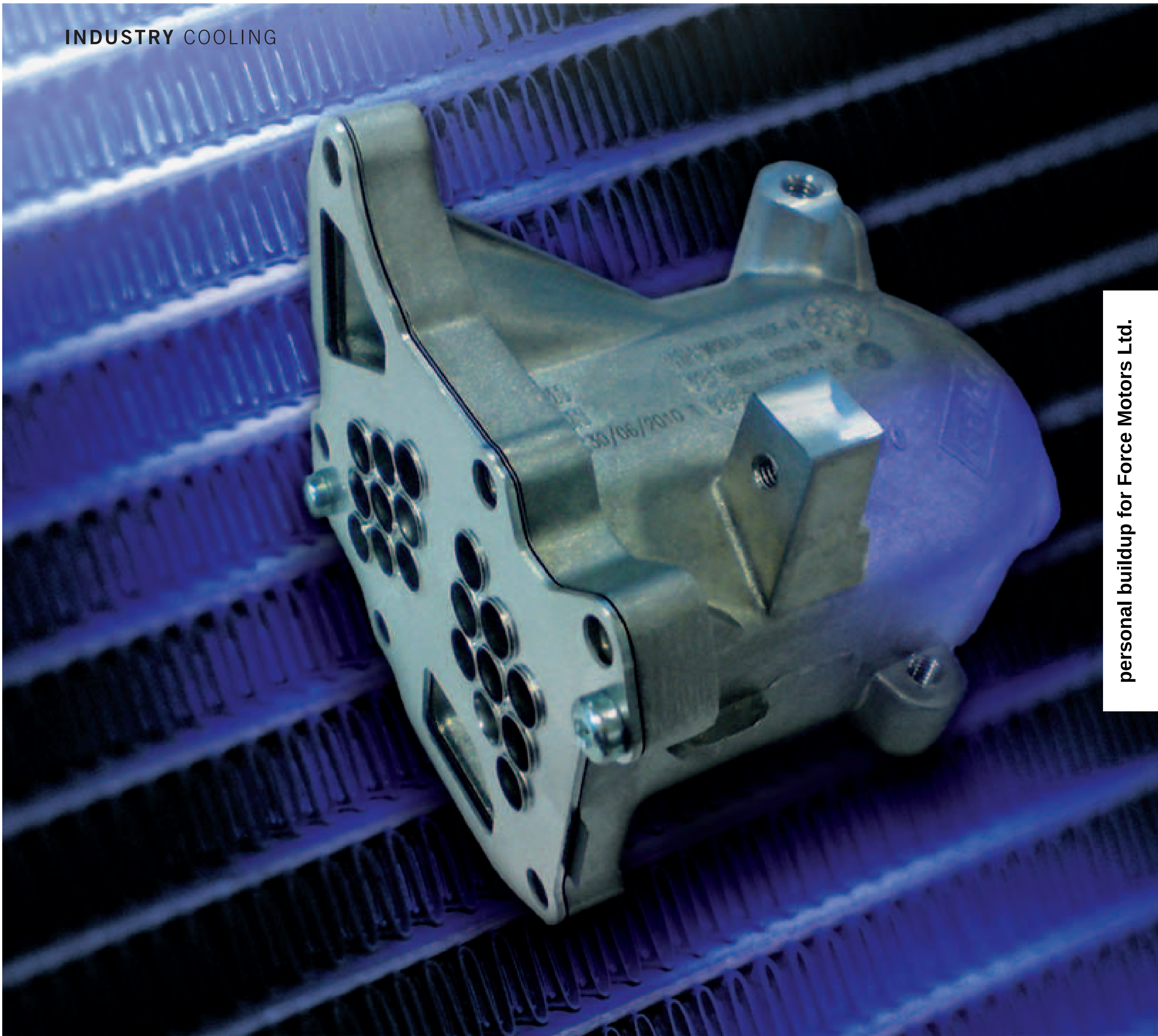

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TECHNICAL ENHANCEMENTS IN POWERTRAIN COOLING

Increasingly stringent emission standards and the requirement to improve fuel economy are major drivers of innovation in the powertrain cooling sector. Visteon solves the increasingly complex and ever more demanding powertrain cooling requirements with advanced cooling technologies to improve thermal management and the robustness of heat exchangers.

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REQUIREMENTS

Vehicle manufacturers are continuing to explore engine downsizing using turbocharging and new engine combustion technology which use higher amounts of exhaust gas recirculation and intercooling to save fuel and improve emissions. These trends are driving the need for a new generation of powertrain cooling products and new approaches to thermal management of the vehicle's powertrain system.

In addition, pedestrian impact legislation, low speed vehicle crash resilience and weight reduction are changing the shape of the front of the vehicle, reducing the space available to package the powertrain cooling systems and components. This creates a need for more efficient and compact heat exchanger solutions that can fit in the available space and still achieve the necessary heat rejection performance.

For automotive suppliers, this presents unique integration challenges. Visteon is addressing these industry trends with a portfolio of technologies that are designed to improve thermal efficiency and achieve the same or better performance from a more compact solution.

Global exhaust emission standards continue to be tightened as concerns regarding greenhouse gases and global warming require the vehicles to emit less exhaust gases. In established markets, at the forefront of emission reduction, this is driving further development of new engine technologies, while the emerging markets, which lag behind on the emission reduction timeline, are adopting the older technology solutions.

The high cost of fuel continues to be the main factor driving consumer demand for higher performing engines with higher combustion efficiency and better fuel economy. In response, vehicle manufacturers are increasingly using boosted engine technology and efficient engine combustion which in turn leads to an increased demand for intercoolers and exhaust recirculation technologies. These more efficient engines also present a new challenge, ensuring that the cooling system can provide sufficient heat to provide cabin comfort, ❶.

Focusing its powertrain cooling developments into three core areas, Visteon's strategy is to develop heat exchanger product solutions for key components within the engine cooling system, ❷:

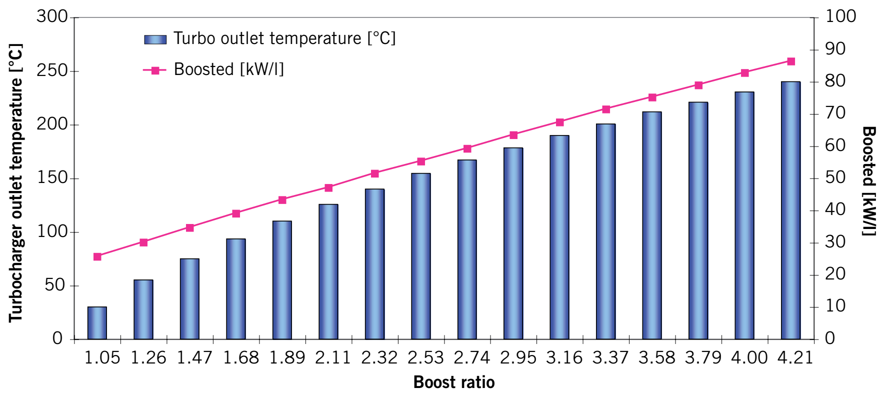
: intercooler solutions
: radiator technologies
: exhaust thermal management solutions.

INTERCOOLER SOLUTIONS

As vehicle manufacturers continue to downsize engines to improve fuel consumption, air intake boost pressures have recently risen by 30 % on gasoline engines and over 100 % on diesel – with modern diesel engines boosting to over 2.5 bar (gauge). This requires significant modification of the intercooler design to withstand the higher pressure cycling and heat rejection. Visteon has developed a new range of tube and fin geometries that enable the company's solutions to achieve higher heat rejection and withstand the higher pressures.

WATER-COOLED CHARGE AIR COOLER

When a higher volume of air is being compressed, two challenges must be addressed. Firstly, as the higher volume can accentuate turbocharger lag and provoke vehicle drivability issues, vehicle manufacturers need a solution that helps minimize the com-



❶ Example of the impact of turbocharging boost ratio on outlet temperature

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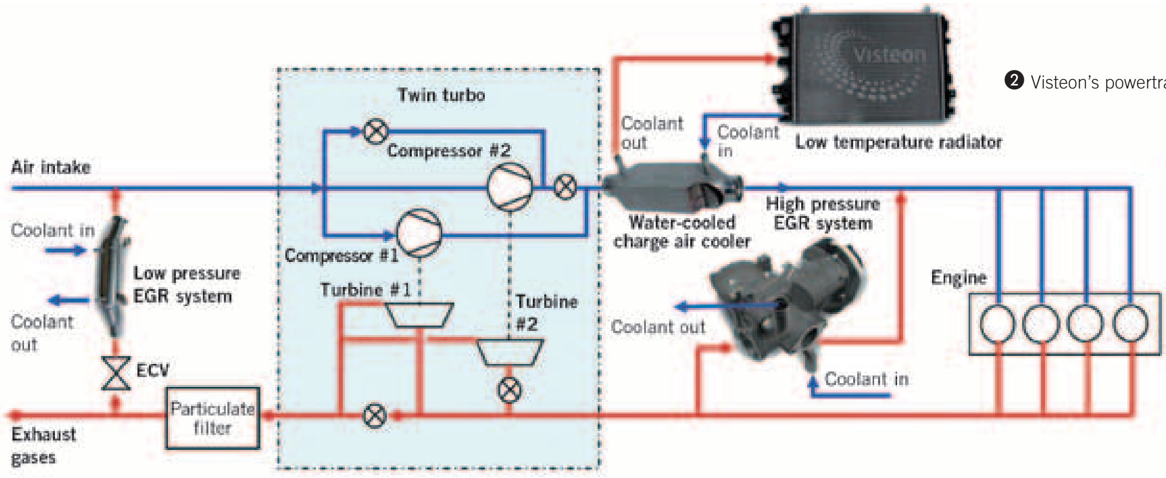
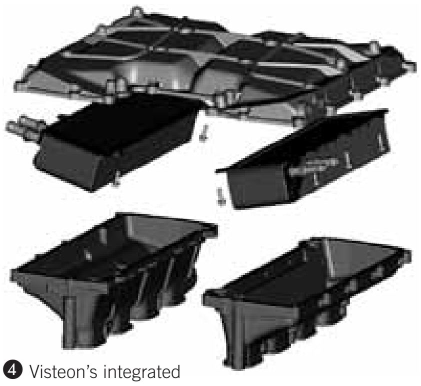


❷ Visteon's powertrain cooling portfolio

❸ Standalone water-cooled intercooler



❹ Visteon's integrated intercooler in the Jaguar manifold

pressed air intake volume. The second issue is the heat rejection capacity in applications where it is not feasible to integrate or package a large enough air-to-air intercooler.

Visteon's water-cooled intercooler technologies have been developed to provide a flexible solution and a viable alternative to an air-cooled heat exchanger, ❸.

The construction of Visteon's water-cooled intercoolers have been designed so they can be used as both a standalone unit, or alternatively, integrated into the engine intake manifold. Visteon can achieve a smaller package volume when compared with traditional water-cooled intercoolers by using high efficiency fins coupled with enhanced coolant passage turbulation to maximize the effectiveness of the products.

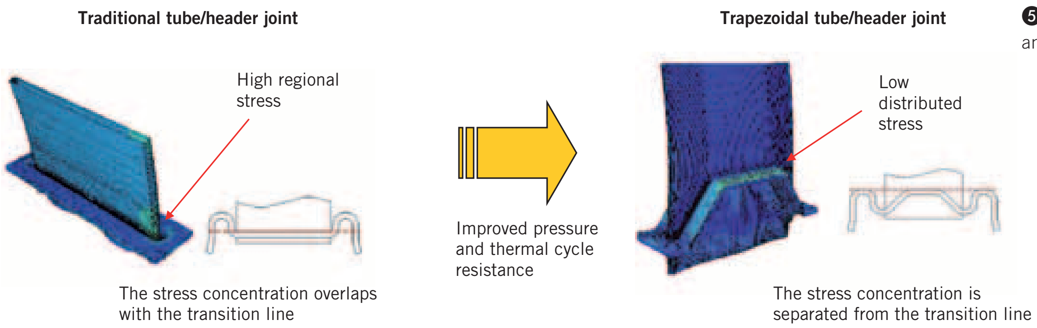
An example of Visteon's water-cooled intercooler technology can be found on the 5-l-supercharged engines used across the full range of Jaguar and Land Rover vehicles, ❹. This technology enabled Jaguar Land Rover to achieve market leading engine performance within the predetermined package space.

ENGINE-COOLING HEAT EXCHANGER EFFICIENCY

Today's engines are not only more efficient but are also more powerful. When used at maximum capacity, the heat rejection requirements have increased. However, the amount of space available under the bonnet for heat exchangers is significantly constrained due to pedestrian impact legislation, low speed vehicle crash resilience and weight reduction targets. As such, heat exchangers that can reject more heat from a smaller package volume are required.

To achieve higher efficiency and higher heat rejection capacity, Visteon has developed a new range of tube fin solutions which form the building blocks of an effective cooling system. This includes two patented technologies which place Visteon in a uniquely competitive position.

The first technology is known as a "trapezoidal header". The joint between the header and the tube is generally accepted as the area of weakness for both pressure and thermal cycle tests. Illustrated in ❺, a normal header is flat and the highest stress regions are concentrated around the nose of the tube. The trapezoidal form moves the highest level of stress away from the tube nose and distributes it more evenly across the tube-to-header joint, significantly reducing the maximum stresses experienced in this area. The result is a more robust joint which will extend the service life of the part.



❺ Comparison of traditional and trapezoidal tube/header joint

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❻ Example fin with strong fin feature

❼ State of the art EGR cooler for Euro V emission standard: U-bend EGR cooler concept

The second technology is “strong fin”, ❻. This addresses pressure cycle resistance, providing more support to the tube by reducing deflection in the tube itself and, as a result, reducing the stress. The extra strength is obtained by additional form in the fin. Adding form to a flat surface significantly increases its ability to resist compressive forces – the principle which forms the basis of the concept.

The new range of tube and fin solutions, optimised for performance, enables Visteon to install a thinner radiator than was possible previously. Reduced core thickness not only reduces component weight and package volume but can also lead to a lower airside pressure drop. This allows for the use of a lower powered cooling fan and therefore less electrical load. All of these are key factors in helping reduce overall vehicle weight, improving fuel economy and reducing emissions.

An example of this radiator downsizing is the 10 mm radiator core recently launched in the Ford Fiesta. The radiator replaces the significantly deeper 13 mm core found on the previous model, reducing package space and component weight. The radiator has the smallest core depth currently in production on a passenger car and can be up to 25 % lighter than a competitor’s equivalent product.

LOW PRESSURE EXHAUST GAS RECIRCULATION

Growth in diesel and lean burn gas engines is driving growth in demand for exhaust gas recirculation (EGR) coolers. In order to increase the combustion efficiency and reduce exhaust emissions, vehicle manufacturers are increasing the use of EGR across the engine operating cycle.

To meet this diversified demand, Visteon has developed both high pressure and low pressure EGR solutions. High pressure EGR coolers can be supplied in I-flow and U-flow versions to meet engine package requirements. All EGR cooler designs can be combined with a bypass function, including a robust exhaust bypass valve (EBV).

Visteon’s U-flow EGR cooler is an innovative solution, this U-bend EGR cooler enables a much higher capacity heat exchanger to be packaged and offers efficiency improvements up to 20 %, ❼. Visteon U-bend EGR coolers incorporate different Visteon patented concepts, including unique U-shaped corrugated heat elements which enable the use of alternative, low cost materials for the cooler housing. A vibration damping system, as well as manufacturing assembly technologies, are also part of the protected elements.

LOW PRESSURE EGR

To meet future emission targets, vehicle manufactures are increasingly turning towards more complex EGR systems using a combination of low pressure and high pressure EGR.

In low pressure EGR applications, the exhaust gas is taken post particulate filter, the ‘cleaned’ exhaust gas is then fed back into the inlet of the turbo charger. The low temperature EGR has a higher density which enables a higher concentration of EGR to be used to tackle the more stringent future emission requirements by improved mixing with the fresh charge. When low pressure EGR’s are used, Visteon has developed charge air coolers which incorporate an anti-corrosion treatment to significantly reduce deterioration of the aluminum tubes by the exhaust gas.

❽ Exhaust heat recovery system

EXHAUST HEAT RECOVERY SYSTEMS (EHRS)

In colder climates, more efficient engines are not rejecting enough heat to the coolant to maintain a comfortable cabin environment. Conventional technologies use positive temp coefficient (PTC) electric heaters which consume electric and engine power. As an alternative solution, Visteon’s EHRS system reclaims waste exhaust heat and passes this heat back to the engine, transmission and cabin, ❽. By accelerating the warm-up process, the EHRS system can be used to speed up cabin heating and/or engine warm-up. Faster engine warm-up results in the engine achieving optimal temperature quicker, with faster catalyst light off and takes the engine out of the inefficient warm up calibration earlier. Quicker warm up also improves fuel consumption by bringing the lubricating oils up to operating temperature quicker, reducing frictional losses in the engine/transmission.

CONCLUSION

Visteon’s strategy is to continue to invest and deliver high efficiency intercooler solutions, heat exchanger technologies and exhaust thermal management solutions that will enable to meet future fuel economy, performance and emission targets.

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LEM SEALING TECHNOLOGY FOR FUEL CELLS

The patented Liquid Elastomer Molding (LEM) technology from Federal-Mogul makes it possible to manufacture extremely flat, light seals that are simultaneously pressure and media resistant. LEM seals offer significant, design-based advantages for sealing fuel cells. Benefits are, amongst others, efficient sealing capability, minimal seal compressed thickness, and serviceability of the stack.

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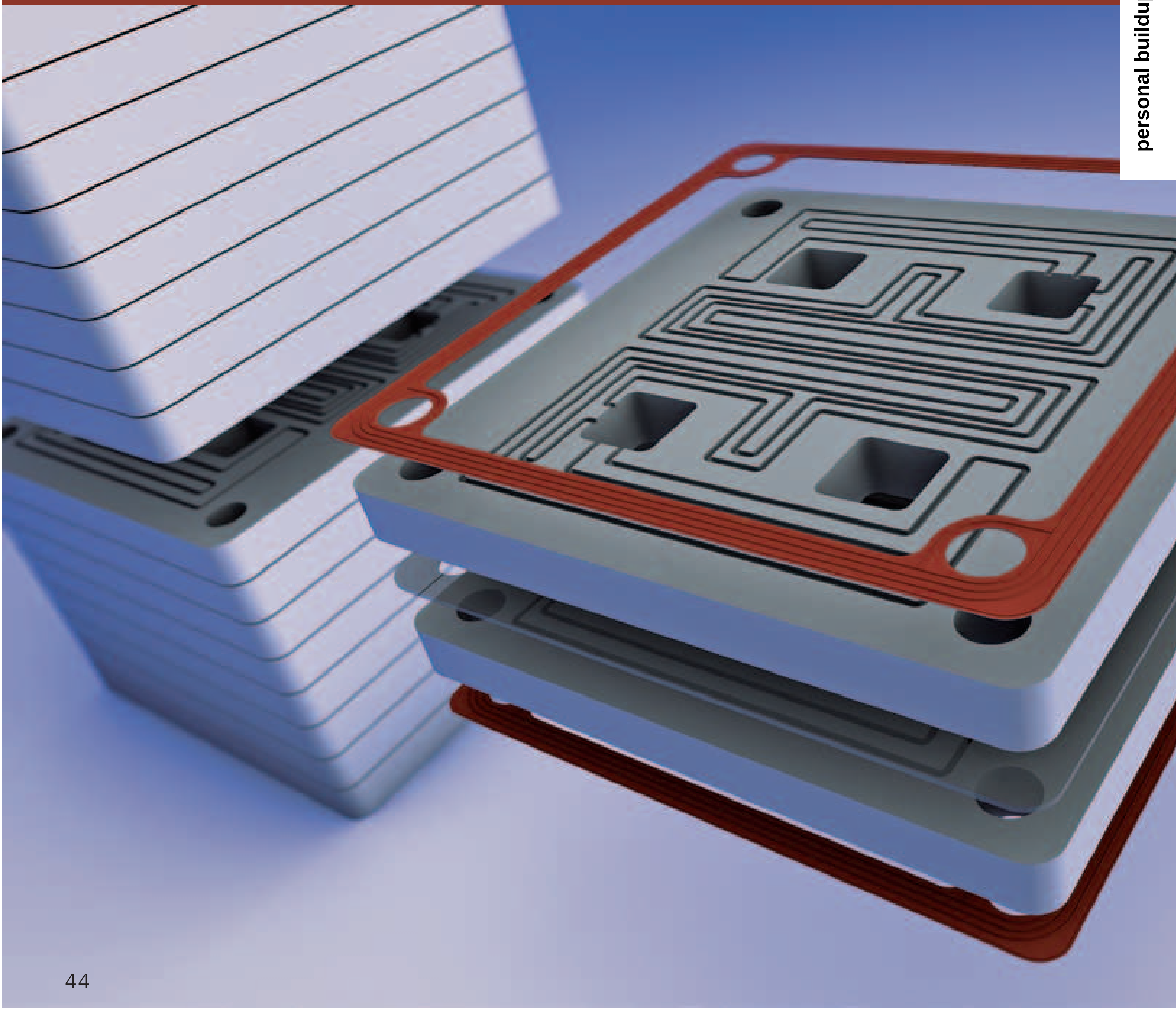

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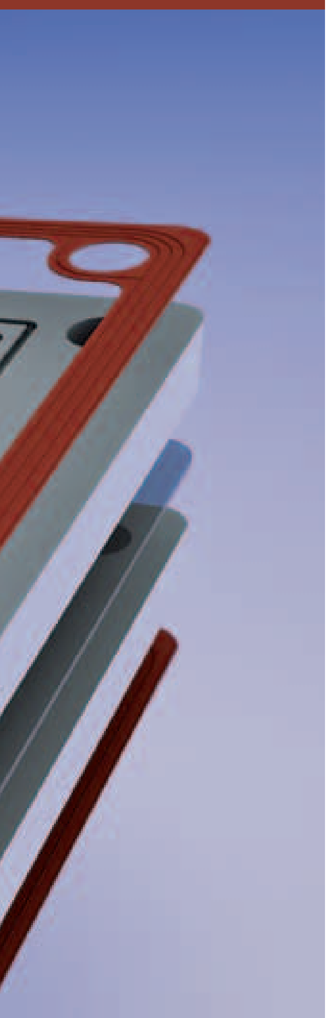

POTENTIAL OF LEM SEALINGS

New drive systems like the fuel cell and traditional ones like the combustion engine are in some respects faced with the same challenges: Both should evince the highest possible energy density. For that reason, they demand compact aggregates with low weight and high efficiency. Lightweight construction plays a major role. However, with increased energy density comes increased stress on the individual components through pressure and temperature, which can make lightweight construction a challenge.

For that reason, Federal Mogul has been using LEM seals for sealing oil filters, oil coolers, oil pans, valve hoods, water pumps and vacuum pumps for a good ten years now. The construction and operational characteristics of these seals have made them into an economical solution to the problem.

With a compressed thickness in the range of just 0.3 mm and their low weight LEM seals make a very low sealing clearance possible, while the sealing joint simultaneously promotes high rigidity. For all that, LEM seals achieve a tight seal with high media resistance at low clamping force. To seal fuel cells today, solutions are often chosen that are based on liquid systems, such as wet-built sealing materials (Formed-in-Place-Gaskets, FIPG). A severe disadvantage of FIPG is that it can be nearly impossible to take a stack apart, since the sealing material, as a rule, is tightly affixed to the sealed component. In contrast, LEM seals are ready-to-install components. A stack with LEM seals can therefore be opened again if this proves necessary for maintenance or repairs with the seal being easily exchangeable.

CONSTRUCTION AND DESIGN OPTIONS

The core of a LEM seal is formed by a solid substrate with a height of around 0.2 mm. After a suitable surface pre-treatment, a pattern of liquid silicone is placed on the substrate using a procedure related to screen printing. In a press mold with 3D-geometry, this network of silicone is distributed in a defined pattern on the substrate surface under pressure and heat. This allows the elastomer to cross-link within a few seconds, thanks to a special cross-linking system, which makes for short cycle times. A final stamping process for the seal contour provides clean, precise edges on all sides. During the press forming of the silicone, two functional, supplemental sealing elements are created: On the one hand the process creates a flat silicone layer with a height of 50 µm to 100 µm, which later provides the micro-seal. On the other hand, one to three line sealing elements with a defined cross section are created around every media channel in the vulcanizing form, assuring pressure-resistant sealing, ❶. The geometry of these line sealing elements can be aligned three-dimensionally to a maximum linear pressure to the flange. Depending on the application, the height of the sealing element can be between 0.05 mm and 0.4 mm.

With the patented process, there are other geometric liberties available for the arrangement of the sealing elements. Depending on requirements, the silicone can be partially applied to the substrate. The three-dimensional sealing elements can receive a variable height over the span between two screw points. With this topography, it is also possible to achieve even linear pressure on the sealing element with an upward bend on the flange between the points where bolting force is applied. Additionally, silicone

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❶ Cross-section of seal with flat 50 µm silicone layer, three line sealing elements as well as an edge sealing element

❷ Example of a complex LEM seal for dividing several media from each other and sealing them against the environment

seals can be created on the seal's edges, in order to seal off T-joints and high pressure areas. An additional degree of freedom is found in the substrate material itself. Here one can in principle use a broad spectrum of materials.

MATERIAL PROPERTIES AND OPERATIONAL BEHAVIOUR

Using the patented manufacturing process, complex geometries can be generated on the substrate, sealing different media off from each other as well as from the environment, ❷. Even in geometries with several media channels, the substrate provides a comparatively high seal rigidity. This does not just have advantages in operations, since, though LEM seals are relatively light components, the substrate affects a high dimensional stability and makes both assembly and operation as a whole easier. Likewise, thanks to the rigidity of the substrate in combination with the micro-geometry of the sealing elements, it is possible to create seals for flange widths of only 4 mm.

In machine-processed flange surfaces, the required roughness factor for the seal lies at an R_z of ≤25 µm, under ideal conditions of up to 30 µm. Cast surfaces may evince a standard R_z of up to 30 µm. With that, LEM seals are no different from other seals with respect to the requirements on the component surface, and instead tend to be less demanding.

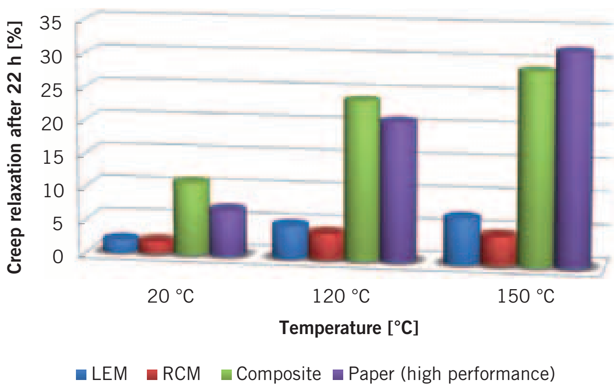
Due to the very flat seal geometry, the effective contact surface for the silicone sealant to the surrounding media is extremely small. For that reason, LEM seals have been validated and have proven themselves in use in automobiles for the media oil, coolant, air and water.

With respect to its potential use as a sealing technology for fuel cells, the small seal clearance is advantageous in reducing permeation in gaseous media. The latter is an advantage for use in fuel cells, because the cells must be protected from contamination even before application, while impermissibly high hydrogen outgassing is unacceptable simply for safety reasons. LEM technology brings optimal properties for both of these requirements.

LEM seals cover a temperature range from -40 °C to 180 °C and are suitable for use at a medium pressure of up to 30 bar. Compared to many other sealants, silicone has the unique property of retaining its sealing characteristics and flexibility to a great extent even at low temperatures. This also speaks for use in demanding mobile applications.

Tests have also shown that LEM sealing elements provide an extraordinarily good dynamic recovery under variable pressure and under effects of high temperatures, ❸. The good creep resistance means that the sealing properties are guaranteed for a long time. With respect to the partially high level of operating temperatures for fuel cells, the seal's persistent resilience after temperature exposure should certainly be viewed as an important factor.

Also advantageous is the ratio between the required tightening torque and the sealing action. Here, LEM seals achieve best values compared to other sealing technologies, since the sealing action is achieved at lower bolt forces and thus potentially produces less warping and lower flange material stress, ❹.



❸ Creep relaxation (ASTM F38): the silicone sealing elements of LEM seals demonstrate a very good recovery after compression

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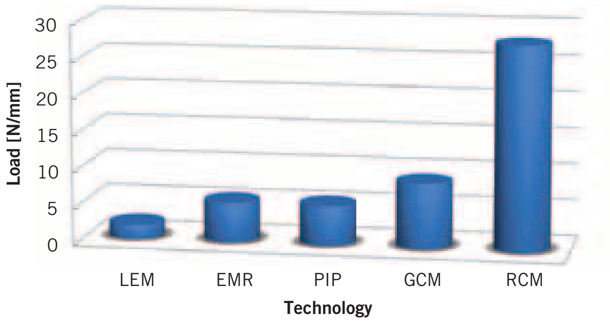

❹ Load-to-seal testing: the ratio between the required linear loads and the effected seal (EMR = Edge Molded Rubber, PIP = pressed-in-place, GCM = graphite-coated metal, RCM = rubber-coated metal)

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SUITABILITY OF LEM TECHNOLOGY FOR FUEL CELLS

Due to its comparably high power density, the low temperature fuel cells with proton exchange membrane (NT-PEM) (also referred to as a polymer-electrolyte membrane-fuel cells, Polymer Electrolyte Fuel Cell, PEFC) has special potential for development of automotive solutions. The individual cells, which are only around 2 mm thick, have a very complex symmetrical structure consisting of two bipolar plates as electrodes, two gas diffusor systems as well as a polymer membrane in the middle as a solid electrolyte for ion exchange.

The cell's entire system is sensitive to catalytic exhaust substances such as CO and sulfur compounds. At the same time, water is produced in the oxidation of the fuel. Along with the supply of clean fuel and clean air, a secure seal in each individual cell, protecting it from the environment and neighboring cells, is therefore decisive for the cell's lifespan. Since a single cell only provide up to around 1 V of current, up to 200 cells are connected in a stack in a row for instance. The high number of the seal clearance makes the importance of suitable sealing technology clear.

HT-PEM fuel cells (High Temperature-Polyelectrolyte Membrane fuel cells) with an operating temperature of around 160 °C, which distinguish themselves in contrast to NT-PEM through using around 30 % less of necessary system components (for example, lower coolant use and no gas-humidification), have a somewhat lower power density. Technically they are however a possible area of application for LEM sealing technology, just as NT-PEM.

What speaks for the LEM technology is that it makes low seal clearance with a secure seal possible even at low tightening torque. The possible topography of the sealing elements is ideally suited for components with long paths between the tightening bolts that hold the stack together. With respect to the structure of the cells and the materials used in them, low tightening torques can possibly contribute to lower warping over all, or to reduced mechanical stress on the individual cells. On top of all this, the stack can be disassembled without being destroyed, if needed (serviceability).

The height of the resulting seal clearance in practice limits the selection of the type of seal that can be used for fuel cells, since the numerous seal clearances in a stack would add considerably to the over all building height. Depending on the technology with which it is being compared, a LEM seal can reduce the individual seal clearances by 50 % and more. In fuel cell stacks, that has an especially noticeable effect: If, for example, a stack of this kind consists of a number of cells ranging into the hundreds of cells connected in a row, the space savings through low-stacking seals can be significant.

SUMMARY AND OUTLOOK

For fuel cell technology, LEM technology opens up new possibilities, combining minimal seal clearance heights, a secure seal at minimal tightening torque and easy access to the stack, ❺.

At the start of 2010, there were already more than 20 million LEM seals in automotive use, so that one can truly call this a comprehensively proven technology. Typically it solves sealing problems where other technologies fail.

PROPERTY	FIPG	LEM
SEAL CLEARANCE HEIGHT	+++	+++
RIGIDITY OF SEALING BOND	++	++
MEDIA COMPATABILITY	+++	+++
WEIGHT	+	+
TOPOGRAPHY	---	+++
PROCESS RELIABILITY	+/-	+++
CYCLE TIME	+	++
SUITABILITY FOR ASSEMBLY	+/-	+++
ACCESS/MAINTENANCE	---	+++

❺ Comparison of the technical suitability of FIPG and LEM for sealing fuel cells

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COMPARISON BETWEEN INTERNAL COMBUSTION ENGINES AND SIMULATED ELECTRICAL PROPULSION OF TAXIS

The term “triumphal procession of electromobility” produces an enormous expectation at the end users side, but intensive development work has still to be done. The transition from today´s vehicles towards electric propulsion is not to be expected to take place overnight but will start with slot applications. Energietechnik Wallner has analysed the potential of taxis.

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START INTO ELECTROMOBILITY

With more than 40 million vehicles on the roads in Germany, even the goal for 2020, that one million thereof will be electrical, will contribute only little to the reduction of CO_2 emissions. For the beginning, the concentration on certain areas is necessary [1]. But where shall we start? The power supply companies are waiting with their investments in the infrastructure for the many electric vehicles to be sold, the vehicle manufacturers are waiting for a widespread refilling infrastructure, all are waiting for standards [2], a true chicken and egg problem.

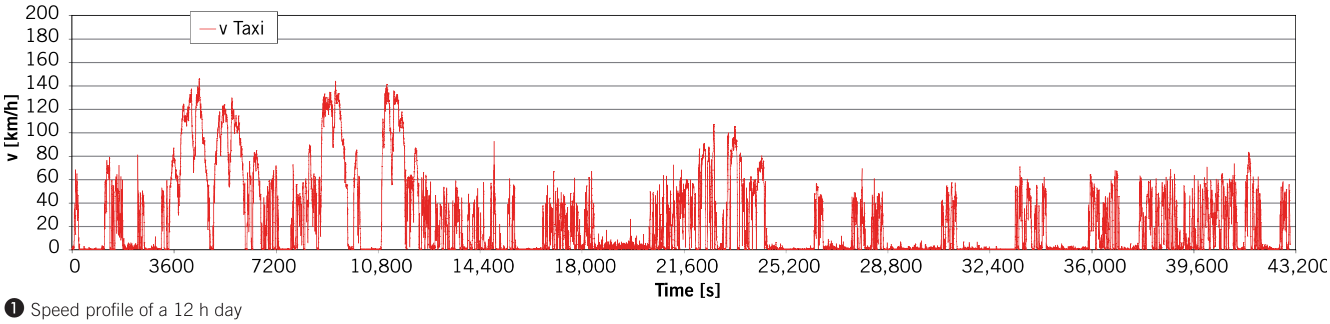
A good starting point is the commercial transport sector. There, the electric propulsion is in good use for years. The British milk floats operate by night – and quiet because they are electric. Fork lifts are driven by the millions in buildings without any exhaust – because they are electric. Many more innovative solutions are under way.

Of the about 750,000 vehicles licensed in Munich, only about 3400 are taxis (i.e. less than 0,5 %), but they are responsible for over 3 % of the CO_2 emissions, due to their high mileage and therefore are especially interesting for the electric propulsion.

PREREQUISITES FOR THE PROJECT

To create real numbers for the evaluation, the company Energietechnik Wallner GmbH has made the investment, to log to the split second the distribution of the energy requirements at real taxis driven by diesel engines and derived thereof the dimensioning of electric driven taxis.

Taxis are a means of transport for passengers and their requirements have to be fulfilled of course (e.g. for climate control), but the decision for purchasing and operating the vehicles are made by the taxi owners according to economic facts. A taxi is therefore especially in the area of conflict between attractiveness of



❶ Speed profile of a 12 h day

the vehicle and its cost effective operation. Taxis are operated by two types of drivers:

- self-employed taxi owners, who typically have a single vehicle that is used privately from time to time
- pure taxi drivers, who get a car provided, that possibly is operated in multiple shifts.

This market is driven by a strong competition. In the foreground is the optimization of the turnover, the requirements for the availability of the vehicles are therefore very high. The driven distances are varying, on extreme days (fairs, the Oktoberfest etc.) less waiting at the taxi stands and empty trips occur. Therefore, these have to be accounted for separately.

The vehicle is means for work and place of work at the same time. Therefore both groups of data, the drive data automatically gathered via GPS, and the personal views and consumption relevant usage at the waiting stands, noted by the drivers on questionnaires, were stored. The driven distances and speeds were logged and the respective power requirements and the potential for recuperation during breaking were calculated. The results were drive characteristics, where times for waiting at stands were included. An example is shown in ❶. An enlarged section with larger distances and higher speed (highway drive) is shown in ❷. The logging of the vehicle speed via GPS enables also the assignment of the drive routes to their topology, ❸, so that the influence of hills could be taken into account.

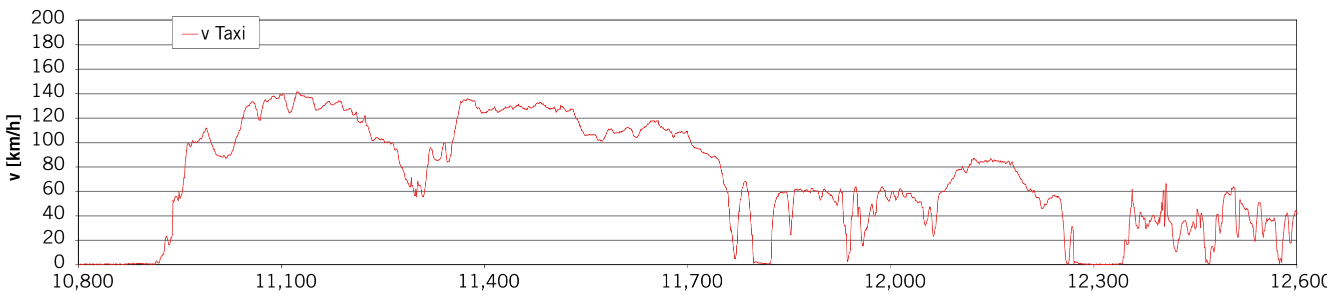
The connection to the vehicle CAN bus would allow to log additional interesting data from the vehicle. As series-production vehicles were used, no modification should be made for the measurements.

MEASUREMENT PROCEDURE

These first measurements are showcases – a comprehensive analysis is still to be done. The measurements were taken at various working days. To ensure a broad coverage (inclusion of extreme days), a respective measurement plan should be setup for a further execution. The speed profiles were stored during typical operations. The waiting times at taxi stands were marked and commented by the drivers. In the following a comparison with extreme days and a rating of the rides were done.

ANALYSIS

First of all a plausibility check of the logged data was performed, e.g. leaps in the data (passages through tunnels) were interpolated. Derived from the logged speed profiles the ratings for the power of acceleration and braking were calculated and illustrated in a chart power over time, ❹. The result was data important for the evaluation and dimensioning of the simulated vehicle. The consumption during the measurements served as a rough comparison and a verification of the calculated data. With this data an exact analysis of the operating conditions is possible. Also the anticipated energy consumption of an electric vehicle may be described. The dimensioning of the optimal electric drivetrain including recuperation is possible. Various layouts of the energy storage system (super caps, battery etc.) may be calculated and compared during the simulation process. The exact logging of the times spent at the taxi stands allowed to estimate the influence of recharging on the dimensioning of the energy storage. The energy storage may be minimized for the typical requirements, extreme days may be handled with range extenders (80/20 rule). The dimensioning may be done according to cost aspects, an optimal mixture of the storage technologies is possible. The mixture of the various energy storage systems allows the reduction of the battery capacity in favor of super caps. The purchasing cost of the vehicle is optimised.



❷ Highway ride – city rides

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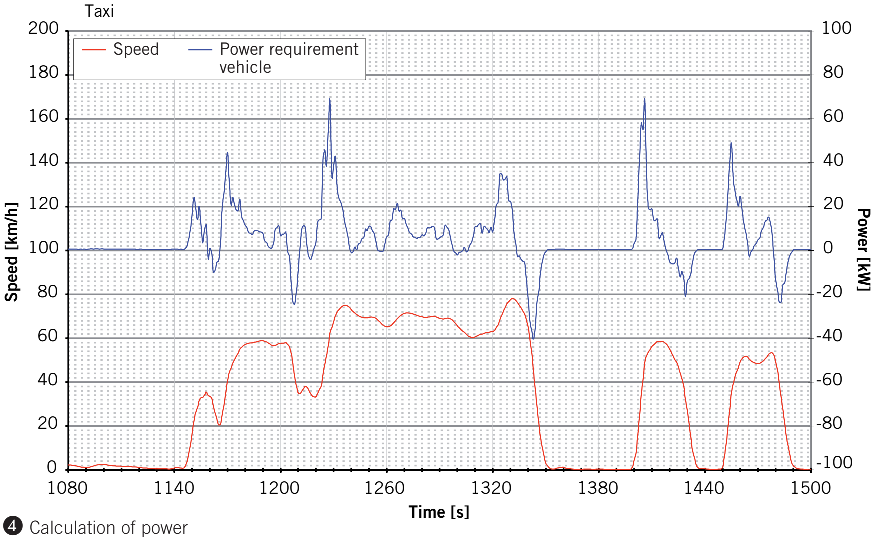
❸ GPS measurement of the rides



❹ Calculation of power

RESULTS

From the vehicles requirements (acceleration power, speeds, braking power, times and distances of rides etc.) the requirements for the propulsion and storage systems are derived, ④. These are important findings for vehicle manufacturers, city planners, taxi owners and electricity suppliers, to select sensible and market conform options of their offerings. The various influencing factors are shown in their mutual dependence. A profile showing the strain for the battery is possible. The comparison with other vehicle concepts (e.g. vehicles powered by LPG, hybrids) is possible.

Along with the pure propulsion oriented data other data may be logged if there is a connection to the vehicle's CAN-bus. Energy intensive applications like climate control may be analysed in their distance reducing influence. Additional usage during the waiting times at the taxi stands (like climate control, video applications) is of no harm if there is a recharging concept chosen, as they occur while the vehicle is on the mains. The recharging during the unproductive waiting times is therefore welcomed by the drivers. Valuable hints regarding the possible usage of electric taxis are shown from the driver interviews, e.g. the inductive charging is considered interesting but not an absolute must. There has to be made a tradeoff between the organizational requirements for conductive charging and the higher cost for inductive charging.

OPEN QUESTIONS

The present results have shown in principle that drive cycle profiles may be derived from measuring the real usage of vehicles. To show the relevance for specific questions, further, more detailed, investigations are necessary. Such further investigations might show:

: potentials for electric driven taxis for OEMs
: profiles of loads on specific vehicle components
: ROI analysis of infrastructure investments for city planners and power supply companies
: assembly of benefit arguments (e.g. for the purchasing or using of an electric taxi, for the taxi guest).

More investigations are therefore interesting for OEMs, communities and energy provider [3].

THE FUTURE OF ELECTROMOBILITY

Electric vehicles and their operation may not be seen independent from the electric infrastructure. To successfully stabilize the power grids, more and more important due to the volatile renewable energy sources, electric vehicles and their energy storage is particularly important [4]. Segmented markets of the commercial transport sector and their requirements, like e.g. taxis, are a good chance to guide this change actively.

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CHAIN TENSIONERS AS AN EXAMPLE OF AUTOMOTIVE DESIGN-TO-COST

Hydraulic chain tensioners have a remarkable influence on the dynamics of timing chains in automotive application. A failure of the tensioner may lead to a serious damage of the engine. The demands to the hydraulic chain tensioner are especially high at efficient low-friction engines. At present, several types of hydraulic chain tensioners, ranging from simple ones to complex ones, are available in the market. However, the dynamics of simple tensioners can be quite similar to the dynamical behavior of complex variants in a typical operation range, by choosing appropriate parameters. In this work done at the TU München, a design-to-cost strategy is proposed to design a hydraulic tensioner, which fits the dynamical demands at optimal costs.

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1 INTRODUCTION

Under the influence of increasing cost pressures, it is essential to design a hydraulic tensioner which fits the dynamical demands at optimal costs. For this purpose, a design-to-cost strategy is proposed. After the definition of the technical specifications (2), which determines the demands to the hydraulic tensioner, the technical solutions (3) are discussed. Then the topics Design-for-Manufacturing-and-Assembly (4) and target costing (5) are addressed. The analysis of the dynamics of hydraulic chain tensioners is based on the comprehensive experience of the Institute of Applied Mechanics, Technische Universität München, in experiment, simulation and theory of chain tensioners in timing assemblies [1, 2, 4].

2 TECHNICAL SPECIFICATIONS

During engine operation, hydraulic chain tensioners need to perform two fundamental tasks. The pre-stressing of the chain has to be ensured during the entire life cycle of the engine in order to adjust the elongation of the chain and worn sprockets. Moreover the vibrations throughout the resonance run of the timing mechanism must be damped. The tensioner is mostly acting on the chain using a tensioning blade.

Thus, the chain tensioner has to assure a certain force despite of the operation state and the load and it has to introduce damping in the range of the resonance frequencies. The detailed demands to the tensioning element depend on the characteristics of the chain drive. For the characterization of the tensioner dynamics the frequency-dependent plunger force and the area of the hysteresis loop, which is the damping work, is used in the following.

It should be denoted, that the tensioner has also a remarkable influence on the level of efficiency. A high tension in the chain drive causes high friction forces and may affect the acoustics negatively.

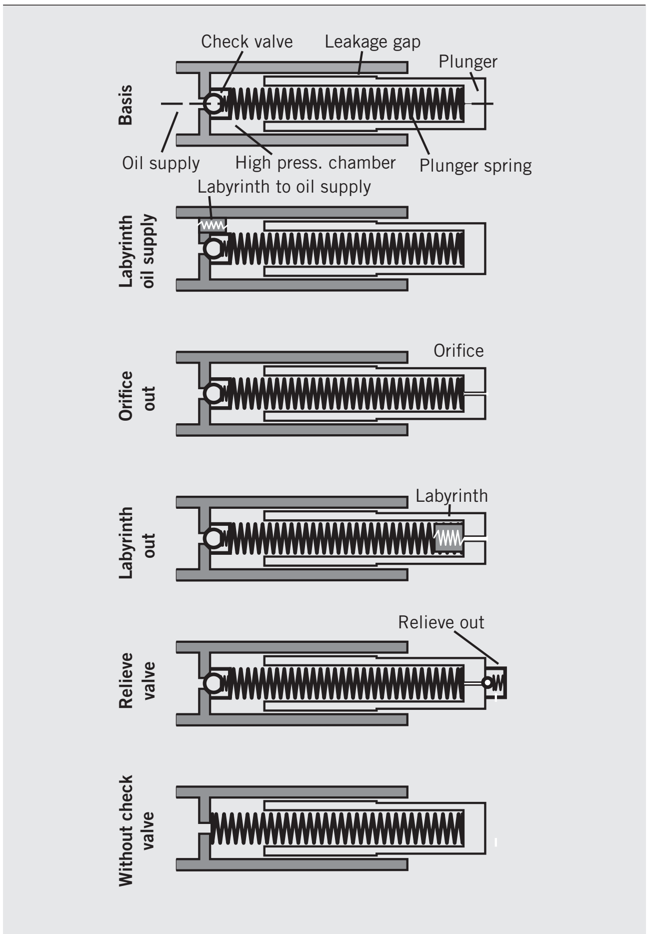
3 TECHNICAL SOLUTIONS

The functional principle is a combination of a spring-loaded plunger and a hydraulic tensioning and damping system, which increases the tension force. The damping effect is caused by leakage flows. Six typical variants of hydraulic chain tensioners, which are available in the market are picked out and analyzed in the following, ❶. The description of every type is following the variant “basis”.

The type “basis” comprises a high pressure chamber bounded by housing and plunger, a check-valve mounted to the oil supply and a leakage gap out of the high pressure chamber. A plunger spring assures a static pre-tensioning, which is necessary at the start of the engine, when there is no oil pressure. During the operation, two phases can be distinguished: the compression and the expansion. In the compression (or damping-) phase, the plunger retracts due to the contact forces of the tensioning plate and counteracts the plate’s motion. The check valve impedes the backflow of the oil in the supply line at high chamber pressures so that the pressure level in the high pressure chamber increases. The enclosed oil is pressed out through the leakage and the desired damping effect is generated. In the expansion phase, the plunger extends as a result of the spring force and the oil pressure. The pressure in the high pressure chamber decreases, the check valve opens and oil flows back from the supply line into the chamber.

Tensioners of the type “labyrinth oil supply” have an additional tube connecting the high pressure chamber and the oil supply. The flow is retarded by a labyrinth, which is a tiny spiral-shaped tube.

The variants of the type “orifice out” and “labyrinth out” have a connection between the high pressure chamber and the environment. In the first case, the flow is retarded by an orifice, in the second case by a labyrinth. There is a significant difference in the pressure loss, which influences the dynamics. For a detailed discussion of the dynamics of labyrinths in chain tensioners, see [1]. The connection to the environment also assures ventilation. This prevents air bubbles in the high pressure chamber in case of a dif-



❶ Common types of hydraulic chain tensioners

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ficult mounting orientation, which would cause a significant decrease of the force level.

In some cases tensioners of the type "relieve valve" are used, which have a pressure relieve valve in the high pressure chamber. This valve opens at a certain pressure and thus limits the maximum plunger force independent of temperature and load.

Finally the rarely used type "without check valve" should also be listed for the sake of completeness. This hydraulic tensioner has no check valve. This causes very low plunger forces especially at low load frequencies.

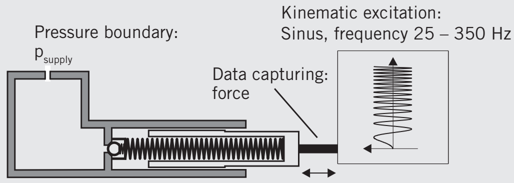
For the analysis of the dynamics of the discussed variants, the simulation tools developed at the Institute of Applied Mechanics are used. Emphasis has been put on the modeling of the hydraulic medium including nonlinearities, the pressure resistances and impacts in the check valve. The simulation models have been aligned and verified in the investigated operation range using a test rig performing various experiments, [1]. The boundaries of the simulation are shown in ❷. The chain tensioner is supplied with oil by a supply chamber with a size typical for the supply tubes in an engine. The supply chamber is connected to oil with constant pressure separated by an orifice. The plunger is connected to a kinematic excitation, which is a sinus with linear increasing frequency (25 to 350 Hz). Different amplitudes have been used in the simulations (0.1e-3 m, 0.25e-3 m, 0.5e-3 m). The oil temperature is 70 deg Celsius, the oil supply pressure is 4 bar and the air content is 1.3 per cent. For a detailed discussion of the effects of the temperature, oil type and the supply pressure see [3].

3.1 COMPARISON OF THE DYNAMICS

An overview of the dynamics of the discussed types is given in ❸. Plots of the maximum force of each working cycle over the excitation frequency are shown on the left column, the value of the area inside the hysteresis loop over the excitation frequency on the right column. The area inside the hysteresis loop corresponds to the damping work, [3]. An excitation amplitude of 0.25e-3 m has been used for this comparative overview. The variants differ in added or removed components with respect to the type "basis". The parameters of the main components are unchanged. The characteristic trend of the plunger force amplitudes and the damping work comes from the coactions of frequency-dependent inflow and outflow.

The force amplitudes of the type "basis" are rising until a frequency of 60 Hz is reached. Then the value remains a constant high level. The damping working is decreasing by the frequency, which indicates lower damping a high frequency.

The additional connection to the oil supply causes a significant decrease in the force level of the type "labyrinth oil supply". The force amplitudes at the lower and higher end of the investigated frequency range are falling characteristically. The decrease at low frequencies is interpreted as the flow out of the high pressure chamber into the supply chamber. The resistance of the labyrinth is increasing overproportional by the frequency, see [1]. At high frequencies the amount of oil, which can flow into the high pressure chamber during the short time period of an open check valve, is decreasing. This results in falling amplitudes. It should be denoted, that the labyrinth has a very high resistance at low temperatures, which results in a force level similar to the type "basis". The damping work over the excitation frequency shows that the mid-frequencies are very well damped.



❷ Simulation set-up

The dynamics of variant "orifice out" is similar to the dynamics of the type "labyrinth out". In the latter case, the resistance is higher, which results in a higher force level.

The hydraulic chain tensioner of the type "relieve valve" has a significant constant force level if the pressure in the high pressure chamber is periodically hitting the cracking pressure. A disadvantage of this variant is, that the relieve valve is introducing additional oscillations and impacts into the system, which may have a negative influence on the overall dynamics of the chain drive.

The type "without check valve" has particularly low forces at low frequencies.

3.2 POSSIBILITIES FOR DYNAMIC ADAPTION

An empirical table, ❹, summarizing the main influencing parameters has been derived from many experiments and simulations, which have been performed at the Institute of Applied Mechanics, Technische Universität München. A subjective assessment of selected parameters regarding the influence on the force amplitudes is given, which should help to adjust the desired dynamics. Additionally, an assessment of the sensitivity is added.

In general, the force is increasing by an increasing excitation amplitudes at all cases except type "relieve valve". In most cases there is a proportional correlation. Tensioners of type "relieve valve" have no increase of the force amplitudes if the cracking pressure is reached. It should be denoted, that it is hardly possible to configure the type "without check valve" to reach forces of more than 1000 N at low frequencies.

The dynamical behavior of the discussed variants and the possibility for dynamic adaption by the influencing factors opens up various possibilities for the design of hydraulic chain tensioners. A skilled selection of the type and adaption of the parameters helps to keep complexity low at a defined dynamical behavior. For example, there is the possibility to increase the force amplitudes by inserting a filling element in the high pressure chamber. This reduces the volume and results in a higher stiffness. Sophisticated chain drive simulations, which comprise the detailed modeling of the hydraulic tensioner [2], can support the design process and help to identify problems like force amplitude drops or too high oil consumption.

4 DESIGN-FOR-MANUFACTURING-AND-ASSEMBLY (DFMA)

The assessment of influencing parameters given in ④, offers the possibility to design a hydraulic chain tensioner dedicated to cost optimal production and assembly.

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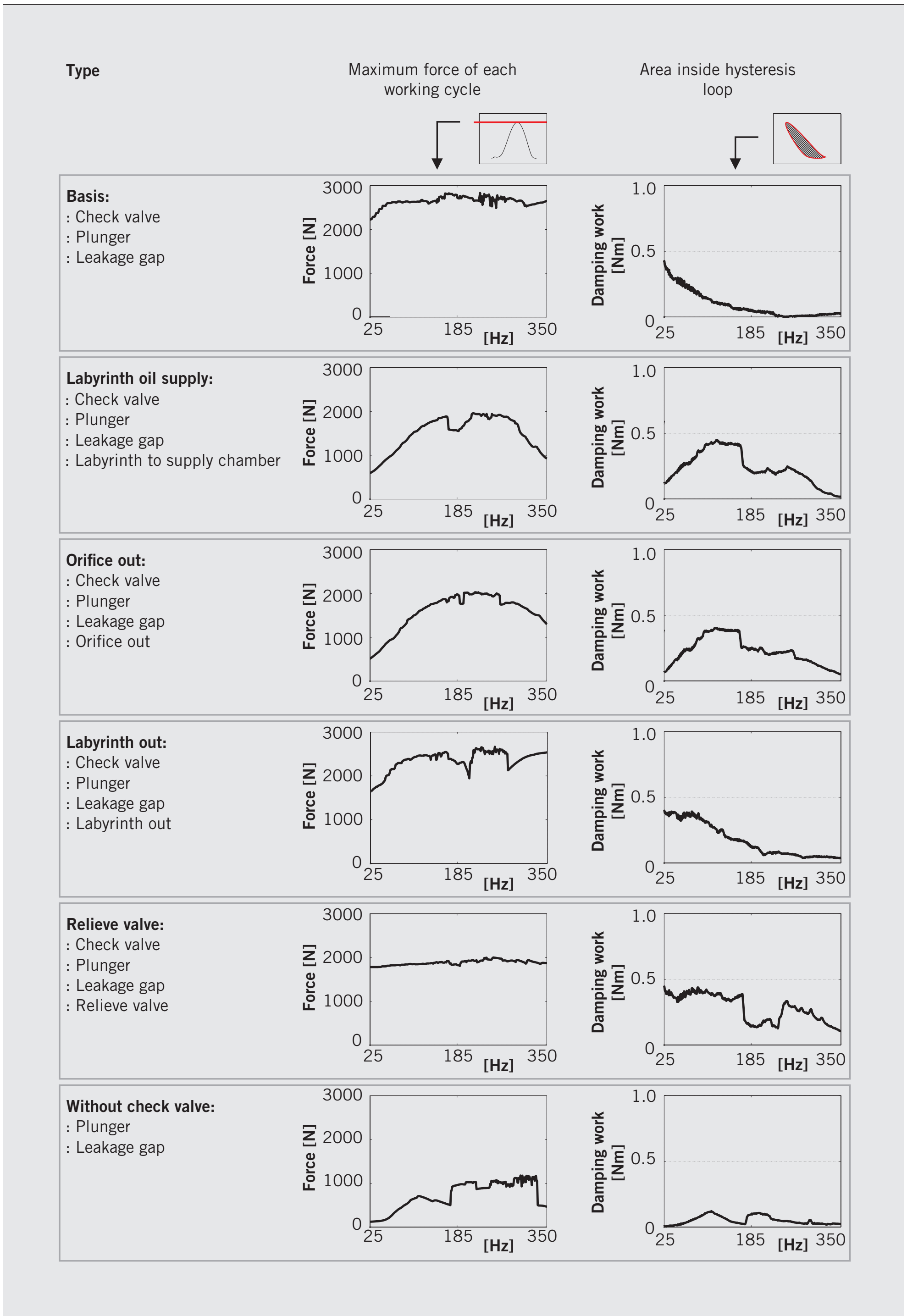
Elements, which have a remarkable influence on the dynamics and a high sensitivity and therefore must meet tight tolerances, are: check valve, relieve valve, labyrinth, orifice and leakage gap. On the other hand, parameters with an inferior sensitivity are: volume of the high pressure chamber, length of the leakage gap, surface quality of the supply tube, diameter of the supply tube, surface quality of the high pressure chamber and stiffness and prestressing of the plunger spring. The tolerances of these elements can be optimized regarding the costs.

5 TARGET COSTING

For the further evaluation of possible technical solutions, it is crucial to derive a detailed fact base on the manufacturing cost of each variant with a clean sheet calculation. This allows for an identification of the most cost optimal solution and is a basis for discussion with potential suppliers.

The following main steps have to been taken, partially in an iterative way:



❸ Evaluation of the simulation-based analysis

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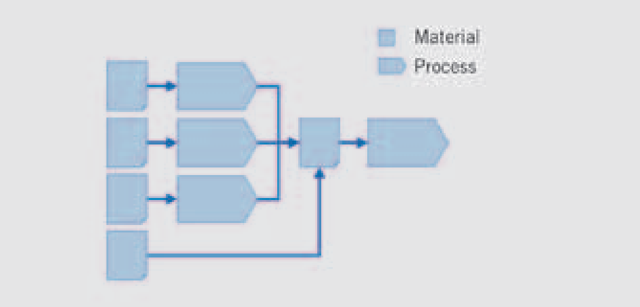
PARAMETER	INFLUENCE ON FORCE	SENSITIVITY	POSSIBLE SIDE EFFECTS
ELEMENTS OF CHECK VALVE	High	High	
RELIEVE VALVE PRESTRESSING	High	High	Additional vibrations and impacts
RESISTANCE OF LABYRINTH OIL SUPPLY	Medium	High	Low resistance –> increased backlash into supply chamber
RESISTANCE OF ORIFICE/LABYRINTH OUT	Medium	High	Low resistance –> increased oil consumption
PLUNGER DIAMETER	Medium	Medium	Large diameter –> possible filling problems
LEAKAGE GAP WIDTH	Medium	Medium	Large gap –> increased oil consumption
HIGH PRESSURE CHAMBER VOLUME	High	Low	Large volume –> possible filling problems
LEAKAGE GAP LENGTH	Low	Low	
SURFACE QUALITY OF SUPPLY TUBES	Low	Low	Possible problems if diameter is too small
SUPPLY TUBES DIAMETER	Medium	Low	Possible problems if diameter is too small
SURFACE QUALITY OF HIGH PRESSURE CHAMBER	Low	Low	
PRESTRESSING/STIFFNESS OF PLUNGER SPRING	Low	Low	

❹ Influencing parameters

: setup of a structured bill of material
: definition of the value stream map
: definition of material and process cost
: definition of overhead charges
: compilation of a consistent overall calculation and discussion with potential suppliers.

The bill of material distinguishes purchased parts from parts manufactured at the supplier. This differentiation is relevant for the application of different overhead expense rates for material and process. For the setup of the bill of material, clear system boundaries have to be defined in order to insure comparability of different solution architectures. Based on this, the value stream map, ❺, is defined. It contains all necessary process steps at the supplier, including required resources, e.g. machine types and/or labor. It is crucial to assume an optimal-realistic process in order not to factor in inefficiencies. The value stream map additionally flags all steps with value-added of the supplier. Commercially available calculation tools structure this process, include common material cost and machine rates, as well as labor cost for various production locations. The use of such software tools accelerates the calculation and offers a good foundation for supplier discussions. Alternatively exist different databases, partially free of charge on the web. In any case, the expertise of an experienced cost calculator is required. After the material and process cost are defined, overhead charges for SG&A and potentially development at the supplier (if not invoiced separately), as well as profit of the supplier need to be defined. These charges can be either based on total cost or on the value-added. Both ways have their justification: the first one is simple to go, usual values can partially be derived from the supplier's P&L. Charges on value-added are more useful when the vertical integration of the supplier is relatively low. After the calculation, a discussion with individual suppliers is reasonable. This allows for the reduction of calculation uncertainties and assures feasibility due to early involvement of suppliers.



❺ Value stream map, illustrative

6 CONCLUSION

A design-to-cost strategy to develop a cost optimal hydraulic chain tensioner, which meets the dynamical requirements, is presented. The dynamical analysis of various technical solutions is based on the comprehensive experience in the experiment, theory and simulation of hydraulic chain tensioners at the Institute of Applied Mechanics. The dynamical behaviors of the discussed variants as well as the influencing factors are treated, which are used for the design and cost optimization. Finally the commercial supplier issue is treated, which provides the information necessary for the selection of the cost optimal chain tensioner type.

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NOISE-CONTROLLED DIESEL ENGINE

Cylinder pressure development is considered to represent a key variable for describing processes inside the engine. Engine management based on cylinder pressure can provide the means to optimize, control and monitor the release of heat in any individual cylinder. However, cylinder pressure sensors are limited in their suitability for use in mass production. This is why the aim of the “Noise-Controlled Diesel Engine” project (No. 1003, AIF No. 323 ZBG) funded by the Forschungsvereinigung Verbrennungskraftmaschinen e. V. (Research Association for Combustion Engines – FVV) is to examine how consumption, noise and limiting exhaust emissions can be improved by integrating appropriate acoustic sensor signals into the diesel engine management system.

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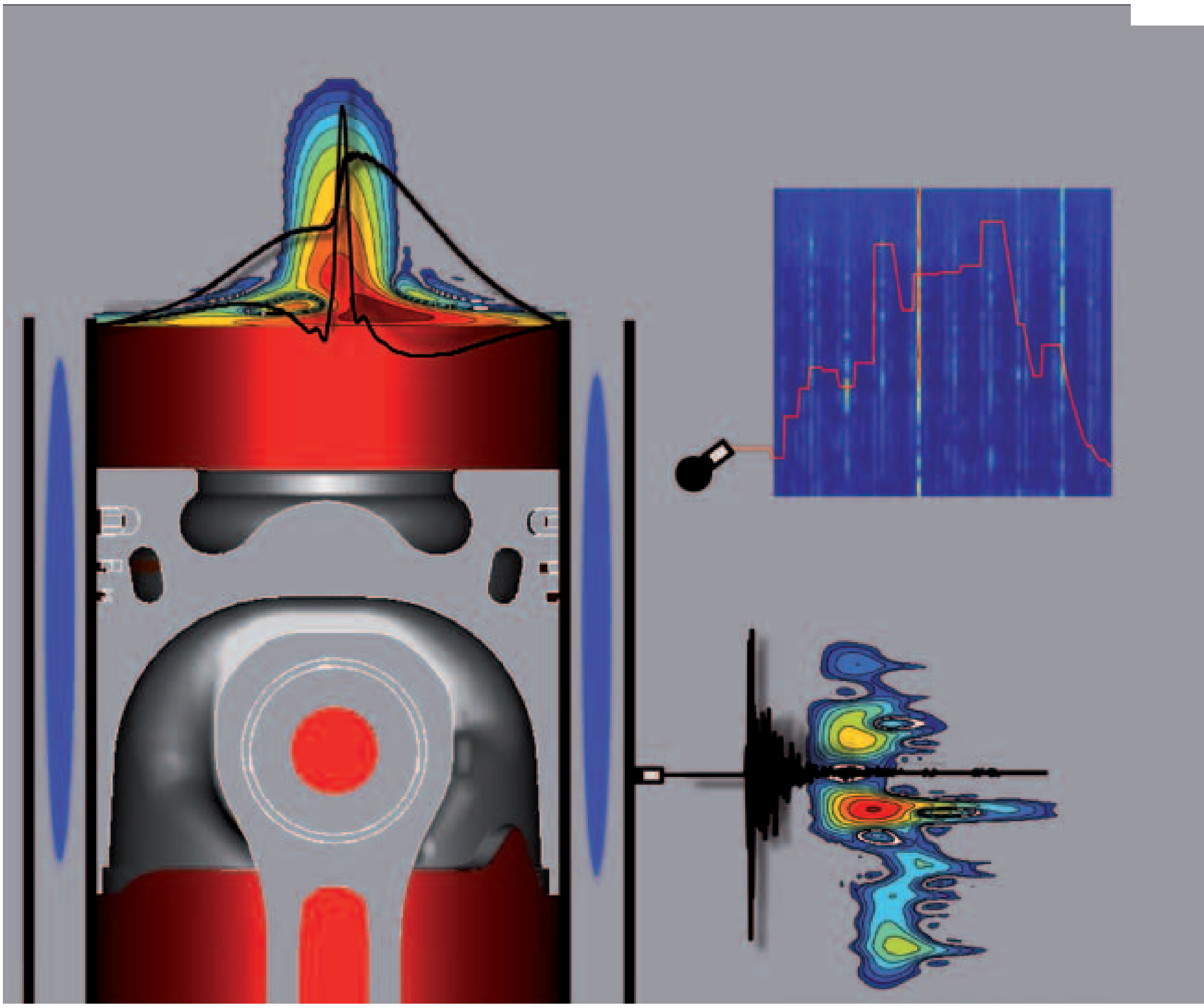


1 INTRODUCTION AND PROJECT OBJECTIVE

Today's diesel engine passenger cars are defined by low emission (fuel consumption) and dynamic driving behavior. During the starting and warm-up phase as well as in the lower load and engine speed range, the combustion noise from a diesel engine dominates over other noise sources as a result of a long ignition delay. Employed almost exclusively today, common rail injection technology provides the capability of extensively influencing this noise by means of one or two pre-injection events. At the same time, however, even greater priority must be given to meeting the exhaust-emissions limits in the load and engine speed range relevant to exhaust emission and consumption testing while keeping consumption as low as possible.

The statutory regulations on exhaust emissions and, in future, on consumption must be satisfied. The increased use of exhaust gas after treatment systems, however, produces greater degrees of freedom inside the engine. These can be used for reducing combustion noise. This leads to the idea of a noise-controlled diesel engine while taking other boundary conditions into account.

The research objective at the center of the "Noise-Controlled Diesel Engine" project is to find out which improvements can be made in relation to consumption, noise and limiting exhaust emissions by integrating suitable acoustic sensor signals into the diesel engine management system, ❶. Achieving these goals demands detailed signal analysis. This reveals which combustion, cylinder pressure, injection and engine noise information is contained in a structure-borne sound signal and provides answers on how to extract it.

If signal processing can offer the means to determine significant features of combustion and/or cylinder pressure as well as engine noise emissions using structure-borne sound, a noise-based engine management system can be produced that delivers benefits similar to those of cylinder-pressure-based management without the need to employ cylinder pressure sensors with their critical aspects of price, accuracy, reliability and lifecycle.

Combining different features from the structure-borne sound signal then provides the control unit with information on air-borne sound (indirectly through parameters), the injection system and combustion, making it possible to investigate new strategies for controlling a diesel engine. However, as the target variables of consumption, exhaust emissions and engine noise diverge, optimization must be weighted.

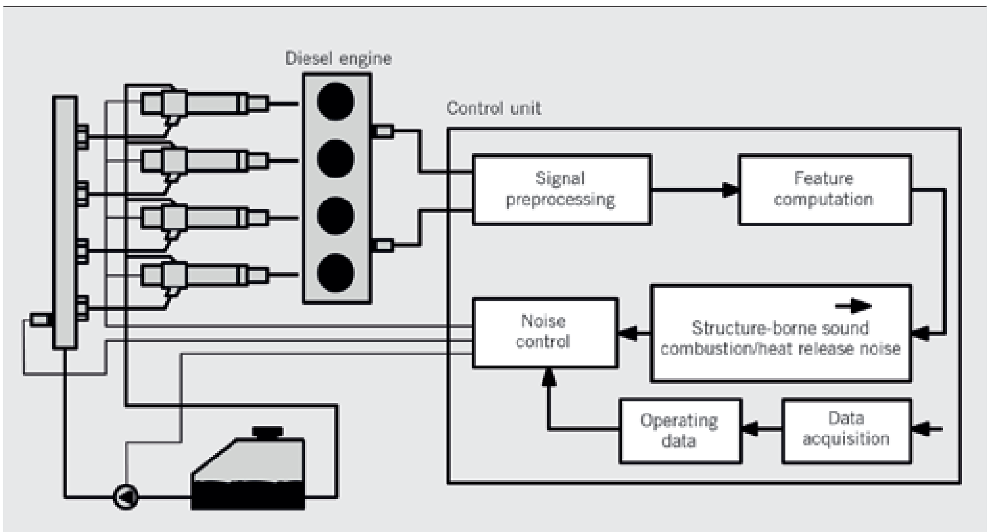
The control strategy necessary for doing this is being examined within the project, with its suitability verified on the basis of an example application.

After describing the test set-up, Section 3 illustrates how to compute significant features from structure-borne sound as the basis for determining combustion parameters. In this context, it also describes the preliminary studies relating to time and frequency range as well as subsequent extraction of combustion timing. This is followed by an explanation of the method used to compute the noise features for evaluating the inconvenience caused by diesel knock. The results are then summarized whereupon attention turns to considering further activities.

2 TEST SET-UP

For the purpose of testing, a modern four cylinder production diesel engine with DI common rail system was set up on the acoustics test bench of the Institute for Mobile Systems at the University of Magdeburg and fitted with measuring equipment.

Three microphones were positioned in the far field on the intake, exhaust and upper side. Proceeding from preliminary studies, 18 sites were furthermore determined for positioning structure-borne sensors at which good cylinder pressure transmission behavior is given. These positions on the intake and exhaust side of the



❶ Diagram showing the principle behind the noise-controlled diesel engine with noise parameters integrated into a diesel engine's electronic control system

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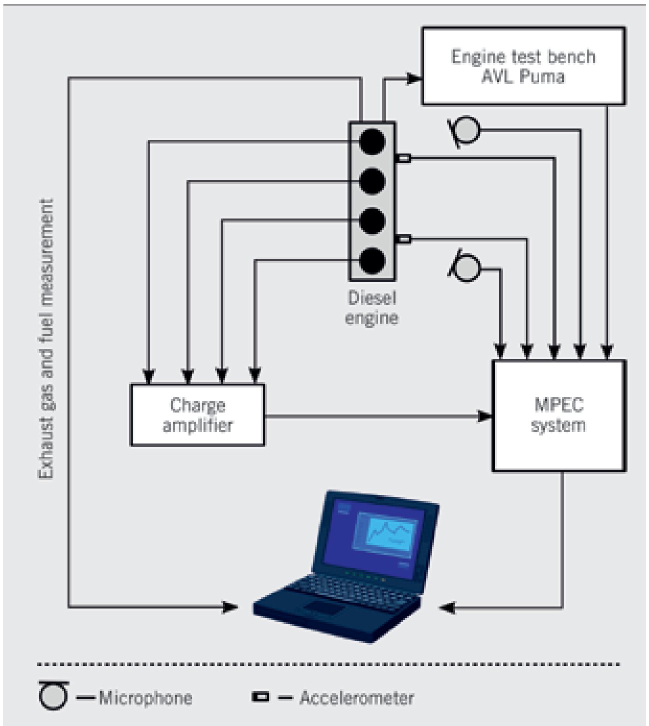
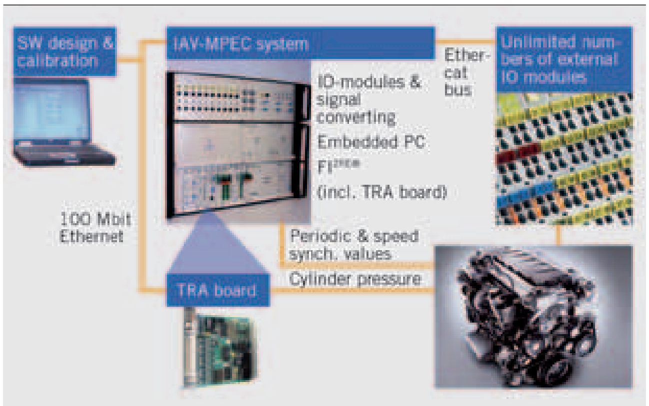


❷ Test-bench set-up with IAV-MPEC for integrating algorithms from Matlab/Simulink



❸ Hardware components of IAV-MPEC

engine block as well at the front end of the engine bearing and in the vicinity of the injection valves above the cylinder head were used for further investigations.

The engine was fitted with four cylinder pressure sensors and an ammeter clamp at the injector of cylinder 1 for recording cylinder pressures and for generating a reference signal for injection. For the initial test series, the engine was controlled using a calibration control unit providing the capability of accessing all internal control units. To obtain maximum flexibility in terms of controlling the diesel engine and to permit fast integration of new algorithms, IAV GmbH's MPEC (Modular Prototyping Engine Controller) Rapid Prototyping System [11] programmed with Matlab/Simulink was then integrated in the test bench set-up for controlling the test engine, ❷.

The MPEC system, ❸, is based on embedded PC technology and has an operating system offering real-time capability. To evaluate the engine sensors and control the actuators, the prototype control unit is equipped with user-configurable I/O modules connected via Ethercat field bus. IAV's FI2RE (Flexible Injection and Ignition for Rapid Engineering) injection control unit is a further key component of the system, its primary function being to actuate the injection system in synchrony with crank angle.

The TRA (Thermodynamic Realtime Analysis) card integrated in FI2RE is the equivalent of an 8-channel indication system and permits process-synchronous recording of cylinder pressure, microphone and structure-borne sound signals. Programmable with Matlab/Simulink, a DSP initially evaluates the thermodynamics of the cylinder pressure signals and computes the features of the acoustic signals in real time. The computed parameters are then transferred from the TRA card via CAN bus to the IAV-MPEC software running on the embedded PC. This is where the cylinder-pressure-based combustion features or acoustic features are fed into the control loops of the diesel engine control system on a cycle-resolved basis. Managing the engine on the basis of cylinder pressure provides the capability of optimizing, controlling and monitoring heat release development for each individual cylinder. This has the purpose of further reducing the resultant emissions.

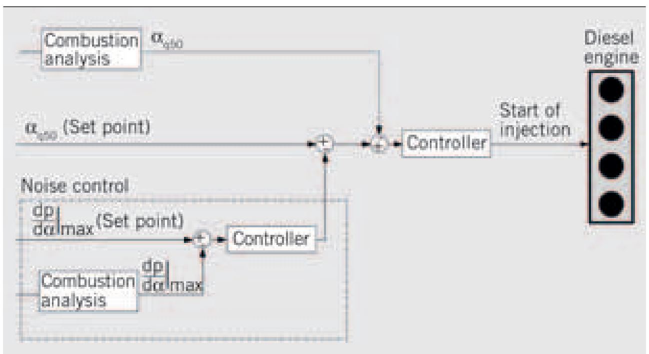
❹ shows a schematic diagram of the structure underlying the cylinder-pressure-based engine management system as implemented by IAV GmbH in the prototype control unit (IAV-AC3 control system).

Controlling the main center of heat release is the best method for stabilizing heat release development. The start of injection valve actuation is timed for each individual cylinder on the basis of the main center computed and a given setpoint value.

The principal function of the noise controller is to limit or reduce the maximum energy conversion rate, particularly benefiting noise emission. The noise controller acts in response to a correction of the setpoint value for controlling the main center of heat release.

3 COMPUTING FEATURES OF COMBUSTION PARAMETERS

In cylinder-pressure-based engine management, combustion is controlled on the basis of combustion timing. For noise-based engine management, therefore, this variable must be computed from structure-borne sound signals. This can be done by modeling



❹ Schematic diagram of a cylinder-pressure-based engine management system of the type implemented by IAV GmbH in the IAV-MPEC

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cylinder pressure from structure-borne sound signals. The results of initial work on obtaining information on the low-frequency component of cylinder pressure by evaluating a structure-borne sound signal were published by Azzoni back in 1997 [1]. For spark ignition engines, Wagner [12] has presented a physically motivated approach to reconstructing the pressure signal independently of engine speed. Other approaches employ cepstral techniques [4] or neuronal networks [10] for reconstructing the cylinder pressure signal. The idea pursued in the project is not to model the entire cylinder pressure curve but to compute the necessary features directly.

3.1 PRELIMINARY INVESTIGATIONS RELATING TO TIME AND FREQUENCY RANGE

The options for analyzing structure-borne sound in the engine management system essentially depend on the quality of the sensor signals. In an initial step, the measurement positions (sensor sites) were determined by way of experiments. For this purpose, cylinder pressure, structure-borne sound and air-borne sound were measured side by side on the engine test bench. In a second step, it was possible to ascertain the time and frequency ranges likely to provide useful information. Wigner-Ville distribution was used for selecting the frequency ranges. The advantage over other time frequency methods is high time frequency resolution. This makes it possible to match up the points and frequency ranges of noises present in the signal with a high degree of accuracy. Wigner-Ville distribution is computed by Fourier transformation of the temporary auto-correlation function [8]:

EQ. 1

$$W_{xx}(t, f) = \int_{-\infty}^{\infty} x(t+\tau)x^*(t-\tau)e^{-j2\pi f\tau}d\tau$$

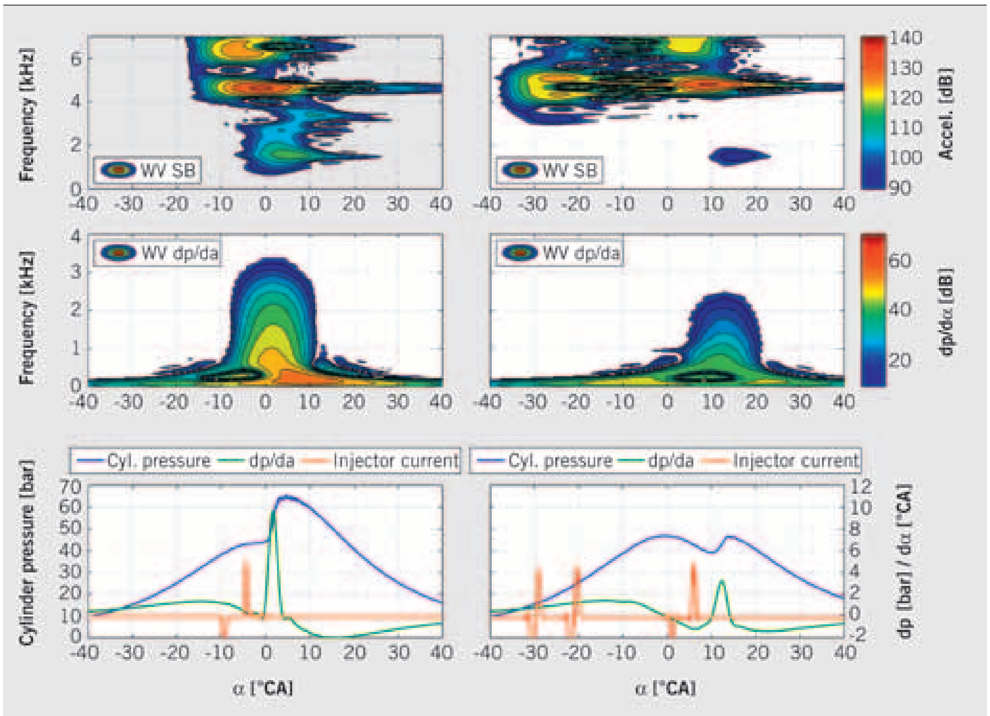
The Smoothed Pseudo Wigner-Ville Distribution (SPWVD):

EQ. 2

$$SPW_{xx}(t, f) = \int_{-\infty}^{\infty} g(t-t') \int_{-\infty}^{\infty} h(\tau)z(t'+\tau)z^*(t'-\tau)e^{-j2\pi f\tau}d\tau dt'$$

was used successfully for investigating interference noises and their influence on structure-borne sound since this produces an even better time frequency resolution and is better at suppressing interference terms [6]. Structure-borne noise is excited by direct and indirect combustion noise as well as mechanical noises. The cylinder pressure gradient has a significant influence on the structure-borne sound signal. Initial preliminary investigations that varied injection and engine speed were able to reveal significant features in the frequency range from f = 0.5–3 kHz. The following explanations relate to the acceleration pickup on top of the cylinder head. ❺ shows the Smoothed Pseudo Wigner-Ville analysis of the structure-borne sound signal (top), cylinder pressure gradients (center) as well as the time signals for cylinder pressure, cylinder pressure gradient and injector current (bottom). The left-hand part of ⑤ shows the analyses for operating point n = 1500 rpm, M = 40 Nm without pre-injection event, the right-hand part operating point n = 1500 rpm, M = 50 Nm with main injection and two pre-injection events.

The SPWV analyses of the structure-borne sound signal clearly show the signal energy between f = 0.5–2 kHz and its direct correlation with the cylinder pressure gradient. The correlation is plain to see both for individual injection events as well as for multiple injection events. Noise also increases as cylinder pressure gradients rise. In the upper frequency range, injection noise is between f = 3.5–10 kHz. The SPWV analyses of cylinder pressure gradients also show that signal energy rises as the gradient increases. The



❺ Time-frequency analyses using Smoothed Pseudo Wigner-Ville distribution, top: SPWV for the acceleration signal, center: SPWV for the cylinder-pressure gradient, bottom: time signals for cylinder pressure, cylinder-pressure gradient and injector current; left: Operating point n = 1500 rpm, M = 40 Nm, main injection event only, right: operating point n = 1500 rpm, M = 40 Nm, main injection and two pre-injection events

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sensor position for a combustion position feature was selected by determining the maximums for each frequency data point from the SPWV and, in the next step, correlated with the maximum pressure gradient. The acceleration pickups on top of the cylinder head exhibit high correlation with the pressure gradient which is why they were used for extracting the combustion timing feature.

3.2 EXTRACTING THE COMBUSTION TIMING

After determining the frequency range and sensor position that reveal significant features of combustion noise, it is now necessary to implement a real-time algorithm for determining these features. The frequency range relevant to combustion was extracted by means of a low-pass filter. The position is then ascertained for the structure-borne sound signal's maximum amplitude (SBS_{max}). The search is restricted to the range after main injection so as to mask out the pressure gradients and interference caused by the pre-injection event. As combustion noise correlates with the energy conversion rate [9], the main center of heat release is computed at constant load using the following model:

EQ. 3

$$\widehat{\alpha_{q50}} = a \cdot SBS_{max} + b \cdot n + c$$

where: a, b, c = constants determined by experiments, n = engine speed.

To define the model, injection was varied at different engine speeds and load points. The model was generated at constant load while varying engine speed and injection. The left-hand part of ❻ shows the situation at operating point $M = 50$ Nm without pre-injection event while varying engine speed in the range from $n = 1000 \ldots 2000$ rpm and the start of main injection in the respective engine speed ranges (top). The main centers of heat release as well as the maximum structure-borne sound position were determined online. The next step involved determining the model between main center of heat release and structure-borne sound offline (⑥, center). The root-mean-square error (RMS) is: $e_{rms} = 0.36$ °CA. The model was validated with a speed ramp in the range $n = 1000 \ldots 2000$ rpm (bottom). Here too, good correlation is shown to exist between measured and modeled main center of heat release.

The right-hand part of ⑥ shows the situation at operating point $M = 160$ Nm with pre-injection event while varying engine speed in the range from $n = 1000 \ldots 2000$ rpm and the time of main injection in the respective engine speed ranges (top). It was also possible to generate a model at this load point for estimating the main center of heat release (center). The RMS error is: $e_{rms} = 0.41$ °CA. Once again, the model was validated with a speed ramp with good correlation between measured and modeled main center of heat release (bottom).

It was possible to show that at selected operating points, structure-borne sound signals provide a good basis for estimating the main center of heat release.

4 COMPUTING NOISE FEATURES

During the starting and warm-up phase as well as in the lower load and engine speed range, the combustion noise from a diesel engine dominates over other noise sources. The structure-borne sound signal is to provide the basis for gaining an objective assess-

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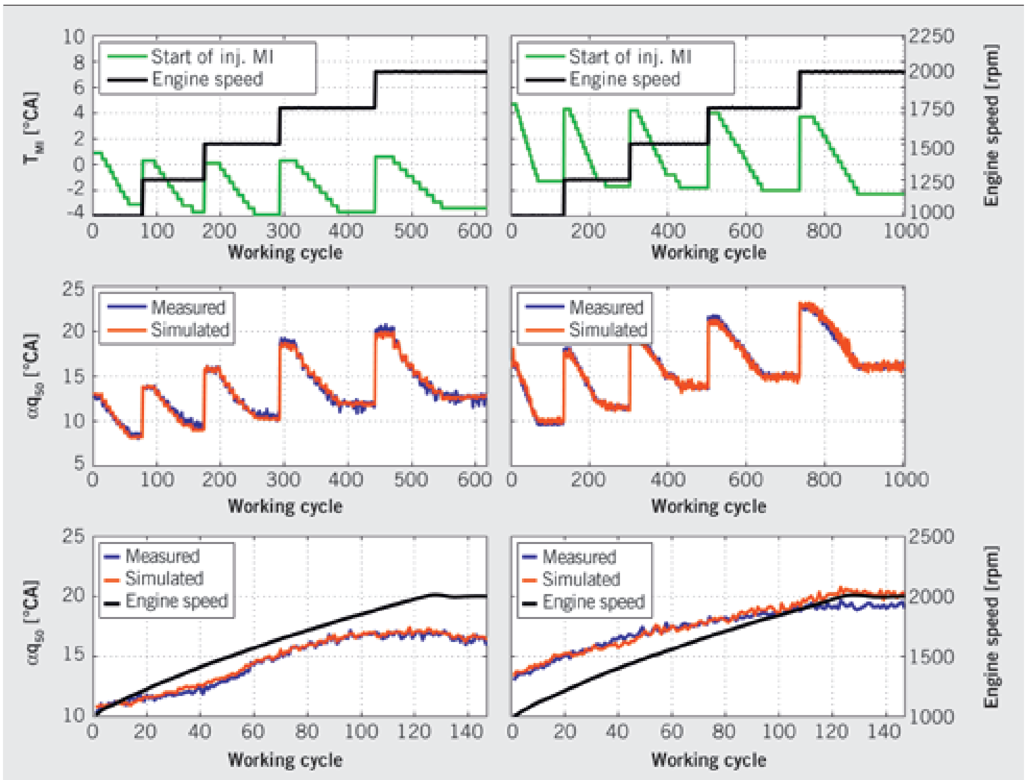
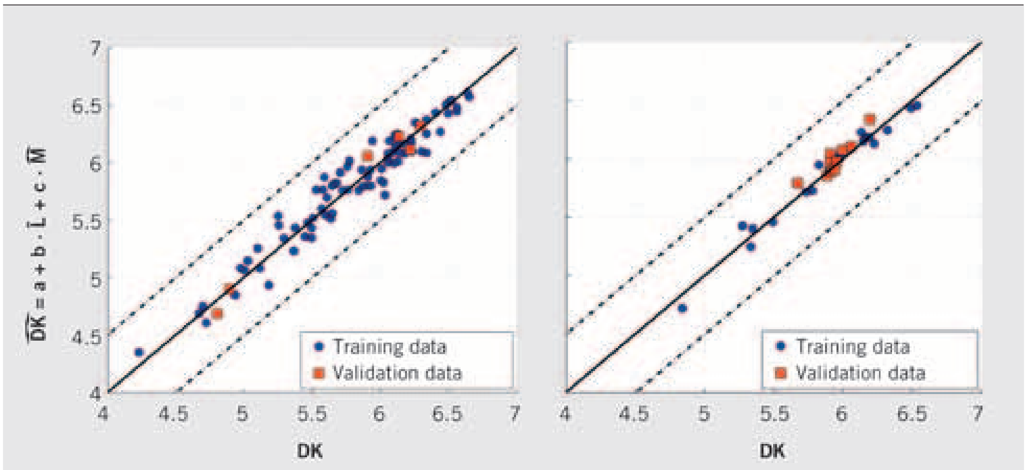


❻ Left: varying engine speed and injection at constant load ($M = 50$ Nm) without pre-injection event (top), comparison of main centers between measurement from cylinder pressure signals and structure-borne sound model (center), verification with speed ramp (bottom); right: varying engine speed and injection at constant load ($M = 160$ Nm) with pre-injection event (top), comparison of main centers between measurement from cylinder pressure signals and structure-borne sound model (center), verification with speed ramp (bottom)

❼ Plot showing correlation between the diesel knock rating from the microphone (DK) and the rating modeled from the structure-borne sound sensor ($\widehat{DK}$) on the exhaust side: on the left for the first dataset ($n = 1500$ rpm, $M = 50$ Nm), on the right for the second dataset (n, M varied)



ment of the inconvenience caused by diesel noise and integrating it into engine management. The impulsive noise contained in "diesel knock" is perceived as being particularly irritating in this context. As part of FVV's "Objectifying Subjective Assessments" project, subjective assessments were taken as the basis for developing objective parameters to evaluating irritation from noises [7]. The evaluation criteria were drawn up here for air-borne sound signals. Loudness (L) and modulation (M) must be computed for determining irritation. The psychoacoustic parameter of loudness describes the ratio of the volume of two sounds, with a doubling of the loudness level equating to a doubling of the loudness perceived [5]. The modulation parameter (M) expresses the impulsive noise content of rattle [2]. The rating (DK) of diesel knock in the part-load range is computed from both quantities as follows:

EQ. 4
$$DK = a + bL + cM$$

where: a, b, c = constants determined by experiment

and is classified on a scale of 1 (not acceptable) to 10 (not identifiable). The new idea behind this project [3] uses a regression model to determine the rating indirectly:

EQ. 5
$$\widehat{DK} = a + b\hat{L} + c\widehat{M}$$

The loudness rating ($\hat{L}$) is calculated by computing the loudness of structure-borne sound signals (L_{SBS}) and engine speed (n):

EQ. 6
$$\hat{L} = a_0 + a_1 L_{SBS} + a_2 L_{SBS}^2 + a_3 n$$

The regression model can now be taken as the basis for determining the loudness rating for air-borne sound signals from structure-borne sound signals and engine speed. The modulation rating ($\widehat{M}$) is furthermore computed by evaluating the modulation spectrum of structure-borne sound signals (M_{SBS}) and engine speed (n):

EQ. 7
$$\widehat{M} = a_0 + a_1 M_{SBS} + a_2 M_{SBS}^2 + a_3 n$$

This means that the regression model can be used for determining the modulation rating for air-borne sound signals from structure-borne sound signals and engine speed.

4.1 RESULTS OF COMPUTING NOISE FEATURES

Design of experiments (DoE) with subsequent evaluation provided the basis for computing irritation from structure-borne sound signals for operating point $n = 1500$ rpm, $M = 50$ Nm. For the 86 measurements conducted, those starts of injection and fuel quantities injected in the main and pre-injection events were varied that were identified in advance as being relevant influencing variables. A second test series was also conducted, with engine speed being varied in the range of $n = 1000 \ldots 2000$ rpm and load being varied in the range of M $M = 50 \ldots 120$ Nm. The timing of main and pre-injects events was also varied.

The results of determining irritation from structure-borne sound signal are presented for the first dataset ($n = 1500$ rpm, $M = 50$ Nm) in the correlation plot on the left of ❼. The results obtained for the second set, for which engine speed and load were varied, are shown in the correlation plot on the right of ⑦. The correlation plots relate to the rating computed from the microphone (DK) and modeled from the structure-borne sound sensor ($\widehat{DK}$) on the exhaust side. The maximum error between modeled and computed diesel knock was below five percent for both datasets.

Evaluation using the DoE method provides a good basis for showing the divergent tendencies of the acoustic and engine-related parameters determined. By way of example, this is shown for the measurement points at $n = 1500$ rpm, $M = 50$ Nm in ❽. The curves for the exhaust gas parameters, the rating for diesel knock as well as specific fuel consumption exhibit the expected profiles. However, this diagram also reveals the necessity to weight optimization of the target variables since a good rating for diesel knock comes at the expense of high specific fuel consumption and very high concentrations of hydrocarbons in exhaust gas. Although the converse case shows that these characteristic values fall as the rating worsens, the concentration of nitrogen oxide rises sharply.

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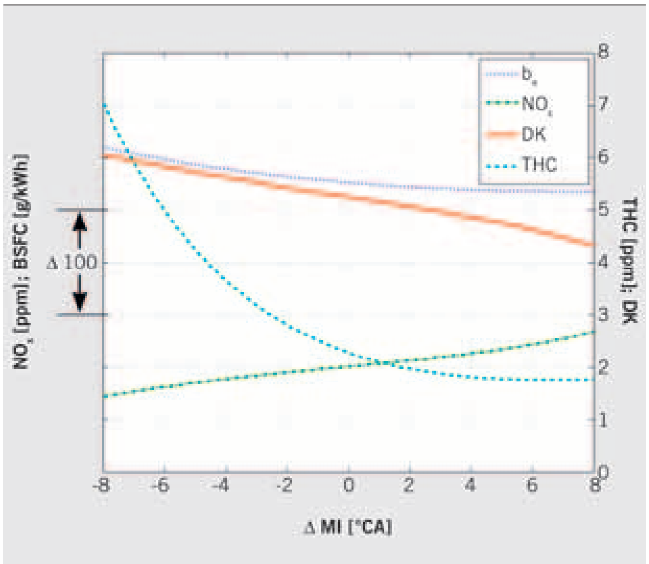
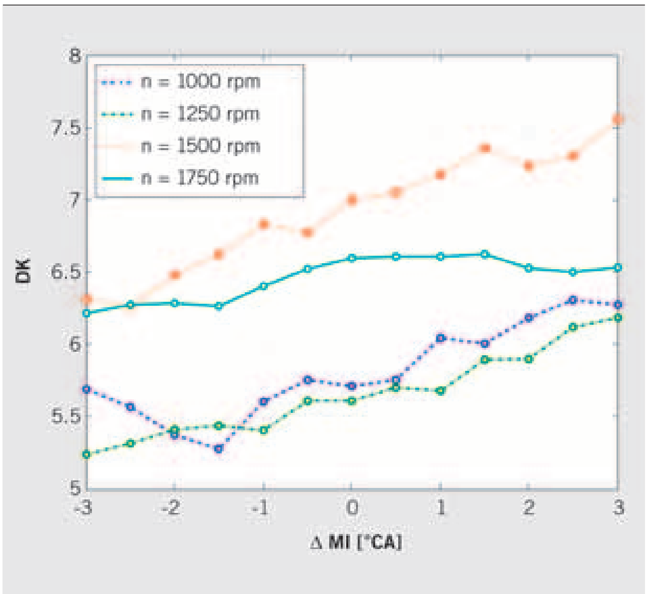


8 Influence of main injection timing on acoustic and engine-related parameters at $n = 1500$ rpm, $M = 50$ Nm constant injected fuel quantities and pre-injection timing



9 Effect of changing main injection timing on diesel knock rating at different engine speeds

In terms of noise control, noise is currently assessed on the basis of evaluating the cylinder pressure gradient $dp/d\alpha|_{max}$. Noise is corrected by altering main injection timing. However, this assessment does not always reflect the subjective perception of noise. This is why assessment in future is to be based on the objectified subjective assessments presented here. With this aim in mind, measurements were conducted at various operating points. Proceeding from the actual state, this was done by varying the point of main injection at different engine speeds. 9 shows the curves for the diesel knock rating from the microphone on the exhaust side as a function of changing the start of injection for the main injection event. At low engine speeds, the diesel knock rating is worse than it is at high speeds because the impulsive noise content is more pronounced there. The results show a potential of up to an entire diesel knock rating. This also demonstrated that noise control can be based on objectified parameters.

5 SUMMARY AND OUTLOOK

The central aim of research is to realize a cylinder-pressure-based engine management system with structure-borne sound sensors and find out what improvements this makes in relation to consumption, noise and limiting exhaust emissions. The main center of heat release is required for the engine management system. For selected operating points it was possible to reveal that the main center of heat release can be emulated using a model comprising structure-borne sound features and control unit variables.

The maximum cylinder pressure signal gradient is used as the manipulated variable for controlling noise in the engine management system. This is to be replaced with objectified subjective assessments. To achieve this objective, the irritation from air-borne sound signals was modeled from structure-borne sound signals. Modeling took place at operating point $n = 1500$ rpm, $M = 50$ Nm

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and also by varying engine speed and load. The error was below five percent in both cases.

In modeling the main center of heat release, it is necessary to examine whether the position of the main center of heat release can also be modeled for the other three cylinders. Further operating points must be studied to assess the universal validity of this approach.

Modeling irritation from structure-borne sound signals also delivered good results both at operating point at $n = 1500$ rpm, $M = 50$ Nm as well as by varying engine speed and load. Here, it is necessary to determine whether irritation can be controlled by using the start of main ignition as the manipulated variable.

To attain these goals, the irritation-related models and algorithms must be produced with real-time capability in Matlab/Simulink and implemented in IAV's MPEC. Finally, the entire engine management system must be evaluated in relation to consumption, noise and limiting exhaust emissions.

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