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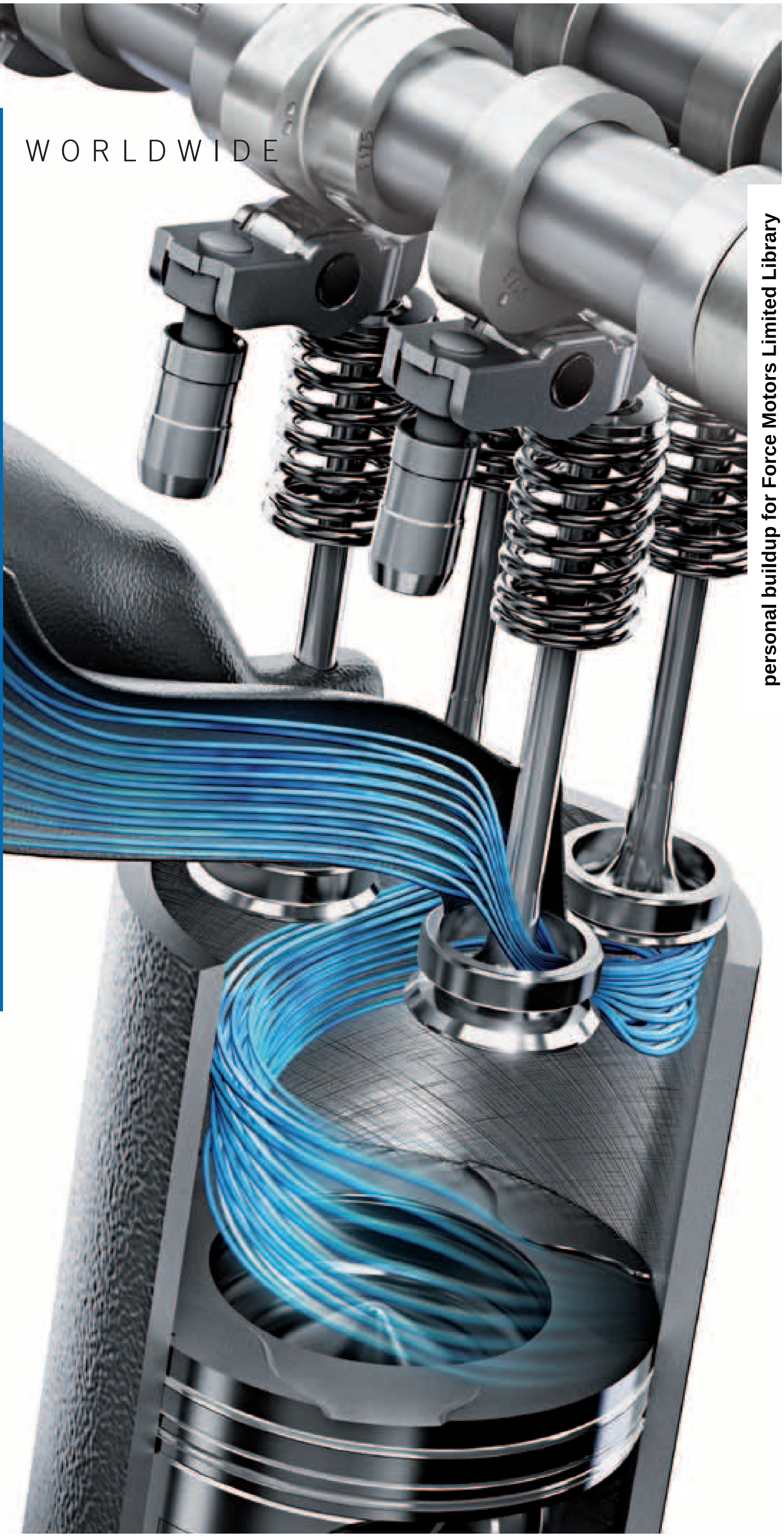
**SERVO-DRIVEN** Piezo Common Rail Diesel Injection System

**OPTIMIZED** Strategic Auxiliary Operation in Commercial Vehicles

**X-IN-THE-LOOP** Engine Testbed for Hybrid Powertrains



# VARIABLE VALVE TRAIN

CONTENTS

COVER STORY

# VARIABLE VALVE TRAIN

**4, 10 I** The mixture formation process still holds many possibilities for significantly optimising emissions and fuel consumption in internal combustion engines. Higher injection pressures, adapted supercharging technologies or controlling the exhaust mass flow still provide plenty of room for further development. Particular potential for improving air/fuel mixture preparation is offered by variable valve timing.





**COVER FIGURE** © Mahle
**FIGURE ABOVE** © [M] mehmetcan / shutterstock

# FAREWELLS

*Dear Reader,*

On my office door at home is a quotation from the memoirs of Benjamin Franklin, who was not only one of the fathers of the US constitution but also a publisher and businessman: "Those who give up liberty to gain security will deserve neither and lose both."

It is, of course, by no means the case that the editor-in-chief of MTZ has no freedom in the sense of freedom of speech. But the duties involved in bearing economic responsibility for a large media family at an international publisher of specialist magazines always take priority. Or to put it another way: the journalistic work – the very reason I once chose this career – has to take a back seat.

Therefore, after just over five years, I am stepping down as target group manager all of the automotive media published by Springer. As this position also includes being editor-in-chief of MTZ, it is with deep regret that I also have to leave this exciting job behind. In future, I will accompany the rapid technical changes in engine and powertrain technology as a freelance journalist.

You are already well familiar with my successor: Wolfgang Siebenpfeiffer, the active publisher of our magazines, will take over as editor-in-chief on an interim basis until a permanent replacement can be found.

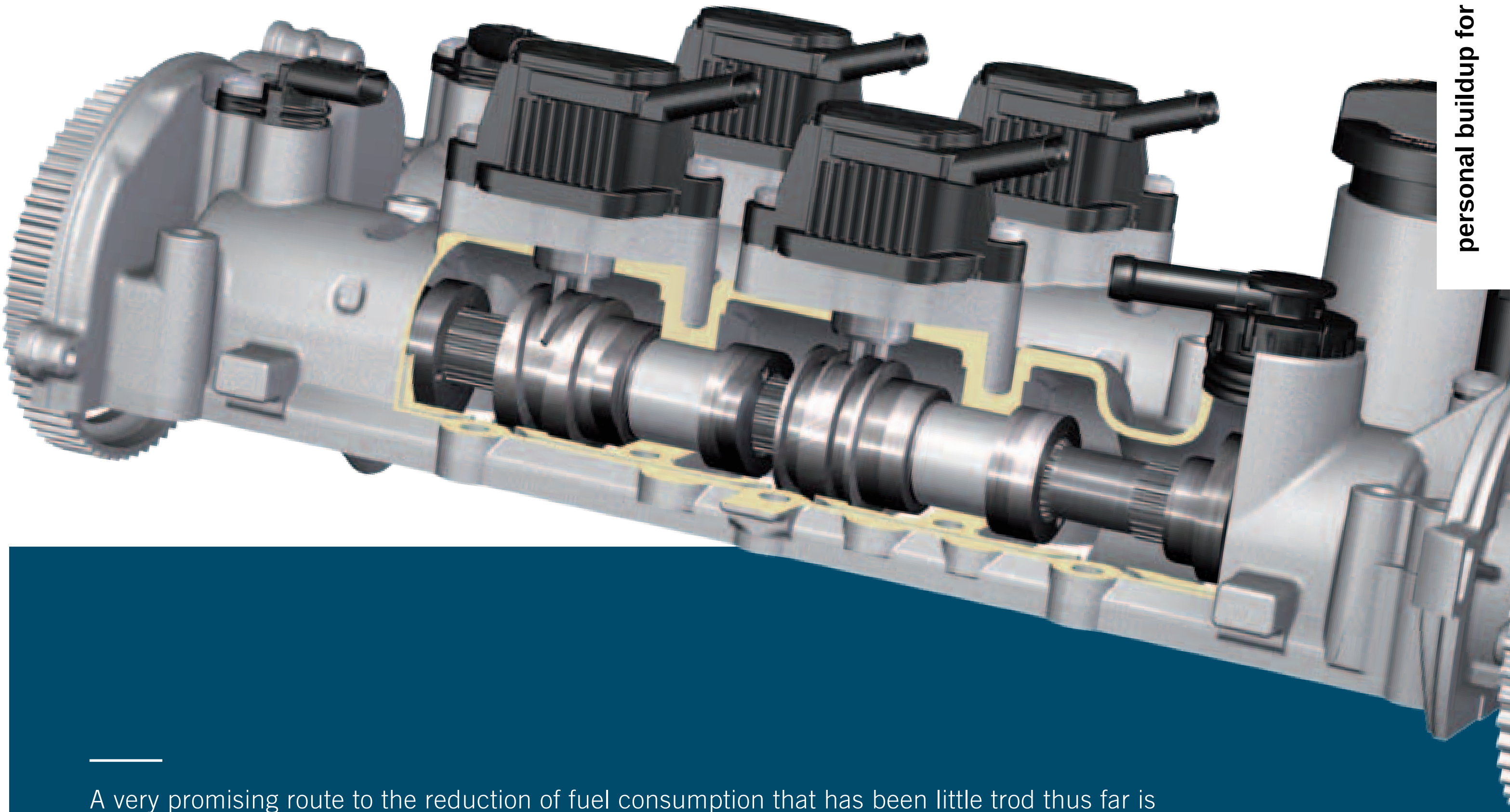
At the same time, but for very different personal reasons, my deputy Ruben Danisch is also leaving the company. He will in future take on new responsibilities as a technical writer in the communication department of a major German supplier. I would like to thank him in particular for his committed work over the past years.

And I would finally like to thank you most sincerely for the confidence and loyalty you have shown towards me. I hope that you will continue to enjoy MTZ as the leading magazine for engines and powertrains.

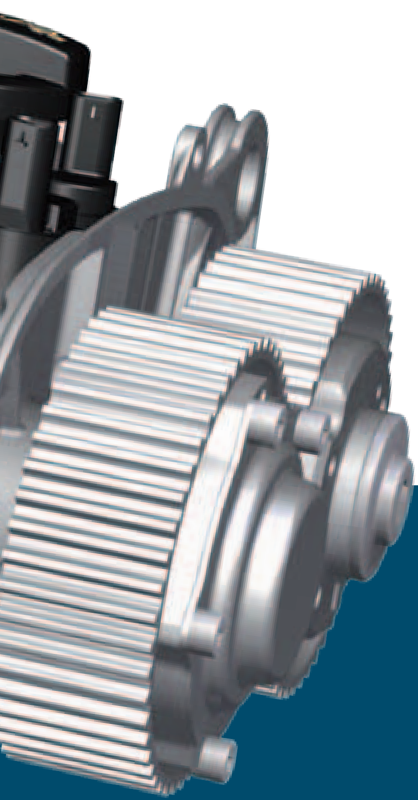
**JOHANNES WINTERHAGEN,** Editor-in-Chief
Frankfurt/Main, January 2012

# THE 1.4-L TSI GASOLINE ENGINE WITH CYLINDER DEACTIVATION





A very promising route to the reduction of fuel consumption that has been little trod thus far is that of cylinder deactivation under partial load. The new 1.4 TSI with petrol direct injection and turbocharging was selected for the first application of this technology in a Volkswagen four-cylinder in-line engine. Within the appropriate map area, the actuation of the inlet and exhaust valves on cylinders 2 and 3 is deactivated and fuel injection shut off.



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## STRATEGY AND ENGINE SELECTION

The ongoing intensification of social, economic and regulatory frameworks is having a substantial influence on driveline development at Volkswagen. In the face of these market conditions, the reduction of fuel consumption and thus $CO_2$ emissions has become a key factor for success. Volkswagen prepared itself at an early stage to face these challenges head on with a wide-ranging BlueMotion Technology strategy that encompasses all the technology modules crucial to the sustainable mobility of the future. The systematic expansion of the technology portfolio for Volkswagen's successful range of TSI petrol engines is therefore also one of the most important tasks for the near future.

One key reason for the selection of the new 1.4 TSI for cylinder deactivation, ❶, is that direct injection is actually helpful in this form of cylinder deactivation because, in contrast to inlet manifold injection, it enables a clear functional split between charge cycle and mixture formation, thus avoiding complications during the switching process. A further factor is the base design of the new engine, which offers excellent prerequisites for the application of this new technology due to its stiff aluminium crankcase and lightweight moving parts (pistons, con rods and crankshaft).

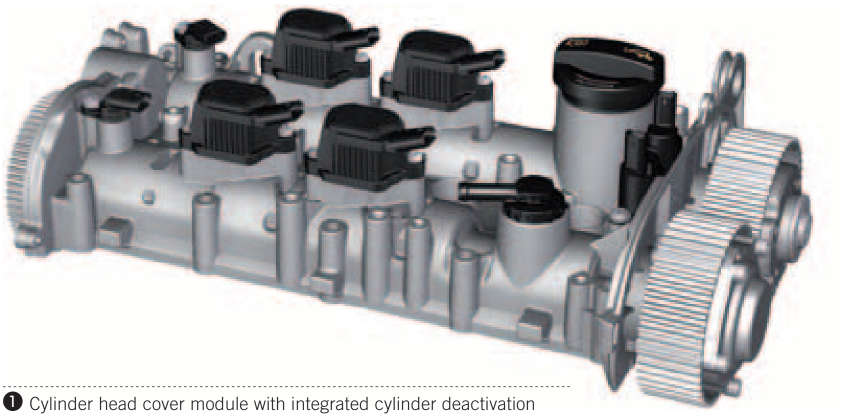
A third reason is the widespread application of this engine throughout the group, meaning the technology can be made available to other users within the group and that substantial synergies can also be achieved in the manufacturing process. The first application of the 1.4 TSI with cylinder deactivation system will be in sporty versions of the Polo and Audi A1.

## FUEL ECONOMY POTENTIAL AND CHALLENGES

The demanding base requirements of the new concept can be formulated as follows:

: fuel consumption of a two-cylinder, but with the smooth running characteristics and performance of a four-cylinder
: reduction in fuel consumption of 0.4 l/100 km in the NEDC
: reduction in fuel consumption in city traffic of up to 1 l/100 km, equating to around 20 %
: implementation of the technology at economically justifiable costs.

Volkswagen is entering new territory with the application of cylinder deactivation on the 1.4 TSI. Due to the substantial challenges associated with addressing vibration excitation, the technology has never before been used in Europe on four-cylinder engine in high-volume production. The fundamental difficulty in cylinder deactivation is not the balancing of masses, which of course remain at a typical four-cylinder level despite cylinder deactivation, but the doubling of the ignition period from 180 to 360° CA. Some very sensitive adjustment was called for in order to achieve the development team objective that the driver should notice the deactivation only at the fuel pump.



❶ Cylinder head cover module with integrated cylinder deactivation

## TECHNICAL APPLICATION AND FUNCTIONAL OPERATION

A switching technology for valve operation that has been in use for several years in various four and six-cylinder engines is the AVS (Audi valvelift system). Thus far, the technology has been used to vary the lift of the inlet and exhaust valves in two stages in accordance with performance demands.

The principle of AVS technology is applied to cylinder deactivation on the 1.4 TSI, although a further development with a double actuator was necessary in order to integrate the switching function into the package space of a small 1.4 l engine. The components were devised and engineered for production under a joint initiative between Wolfsburg Development and Component Development at the Salzgitter manufacturing plant. Alongside cost-effective production, the focus was on a high degree of mechanical durability and reliability, as well as low weight.

With a systematic approach to weight reduction and through the application of state-of-the-art simulation techniques, the additional weight was limited to around 2.0 kg.

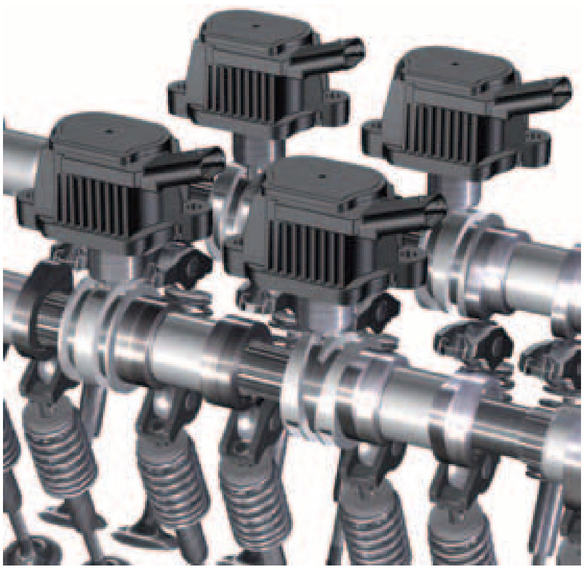
The so-called cam sections for cylinders 2 and 3, which are the ones to be deactivated, are designed to be movable, ❷. They take the form of barrels 68.65 mm in length and toothed on the inside. They are mounted on the externally toothed base shaft, which is made from class C35R hardened steel, and can be moved axially by 6.25 mm. The involute gearing with 24 teeth is designed to bear the load along its flanks. The gear teeth on the cam sections are made using a material removal process, while a forming process is used on the toothed shaft. Once they have been slid onto the toothed shafts during module assembly, the fixed cams are secured axially with cylindrical pins.

Each cam section carries four cams for the two valves that it actuates, arranged in adjacent pairs. One has a conventional full profile that follows the same valve lift curve as on the standard engine. The other is a zero-lift cam with a 360° base circle. The cams are made using category 100Cr6 roller-bearing steel. At the end of the cam sections are the shift gates made from 42CRMo4 steel alloy.

Machined into the outer sides of the shift gates are Y-shaped spiral grooves. Both pins from the two-pin actuators integrated into the cylinder head cover slot into these grooves from above. This layout represents a significant step forward from the Audi Valvelift System, which features separate single-pin actuators on S-shaped groove geometries located at the front and rear ends of the cam sections. This layout enabled the reduction in the installed length of the cam sections required for application in the 1.4-l engine.

The spatial restrictions led to a compact execution of the Y spiral groove and to a narrow separation between the two shift pins, representing another new design feature in the Volkswagen cylinder deactivation system. A further demand on the actuator was the modular construction of the coil assemblies.

Each of the actuators' cylindrical pins has a diameter of 4.0 mm and is also made from roller-bearing steel. The axial route travelled by the pins measures 4.2 mm.



❷ Deactivation units on cylinders 2 and 3

The contour of the shift gates guarantees a unidirectional arrangement of the pins that precludes overshoot. The actuators have been laid out as a bi-stable system, positioned safely and securely in both the retracted and engaged end positions. This is facilitated by magnetic clamping of the armature assembly in both end positions.

The mechanical switching process is completed within half a revolution of the camshaft. For the deactivation of cylinders 2 and 3, the pins for the deactivation system are actuated and fired into the grooves with the slide ramps to the zero-lift cams. The pins are deployed via inertia switching of the coils. Extremely short actuation times are dependent on engine speed, ranging from 72 ms at 1400 rpm to 28 ms at 4000 rpm, and are sufficient to release the armatures from the end position. The switched pins are now in the front end position.

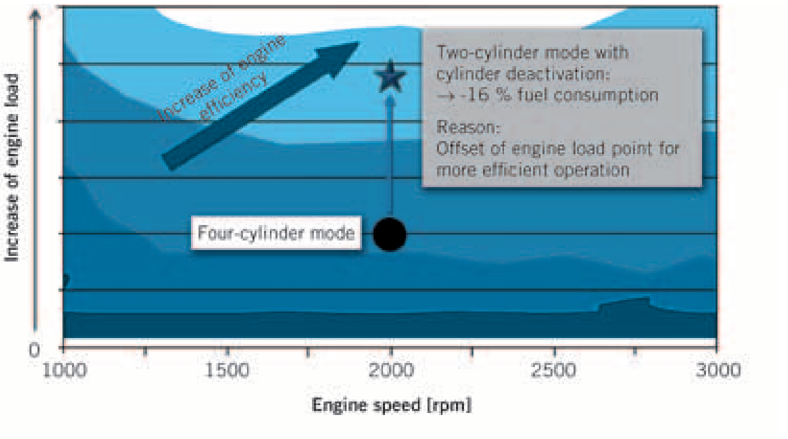
When the axial offset of the cam section is completed, the pins are pressed back into the retracted end position. This occurs through reset ramps at the ends of the Y grooves. A reset voltage is generated within the actuators and measured by the engine control unit for evaluation and subsequent diagnosis of the cylinder deactivation system. This solution dispenses with the need for an additional sensor for confirming successful completion of the switching process.

As soon as the cam sections for cylinders 2 and 3 have reached their end positions for the deactivation system they are locked in place by spring-loaded balls. In this system condition, the zero-lift profiles rotate against the rocker arms. They do not actuate them and the valve springs hold the inlet and exhaust valves in the closed position. The driving torque for the valve train is reduced by around one half.

In order to end cylinder deactivation mode, the pins for full engine mode are actuated and deployed. They push the cam sections back to their original position. As soon as this axial shift is completed, the pins are pushed back into their normal position in the actuators by the reset ramps at the ends of the grooves. The full cam profiles now take over actuation of the rocker arms.

The rocker arms were also redesigned for use with cylinder deactivation. Their cam rollers have a diameter of 21.0 mm, a width of just 5.1 mm and run on a

❸ Effect of cylinder deactivation on engine mapping




set of 14 needle rollers. The fully hardened stud has a diameter of 6.39 mm. Engineers were able to reduce friction by using two roller bearings on the front camshaft mounts, as the front mount in particular is subject to heavy loads from the timing assembly.

One notable challenge for the engineers arose in realizing technology that could be applied in a modular fashion. In general, the cylinder head cover for the cylinder deactivation engine is designed to be fully interchangeable with the cover on the standard engine. The major difference was limited to the fixing points for the actuators and the inner bearing mounts for the camshafts. The mounts are, like the cylinder head covers, made from AlSi9Cu3 pressure cast alloy. Despite the presence of the actuators, the distance between the engine and bonnet required for pedestrian protection remained unaffected.

## INTELLIGENT LOAD CONTROL

A decisive factor in developing the cylinder deactivation system was a new kind of intelligent load control. All switching (from four-cylinder mode into deactivated mode and back again) is completed without any fluctuation in torque. To this end, the inlet manifold pressure is adjusted to the level required for deactivation mode. During the charge cycle, the ignition timing is retarded in line with the charge volume in order to remain torque-neutral.

When the desired charge is achieved, first the exhaust valves and then the inlet valves on the second and third cylinders are deactivated. No further injections occur after the final charge cycle, sealing fresh air inside the combustion chamber. During the next compression phase, this charge of fresh air results in minimal compression pressure inside the combustion chamber, making the switching process smoother.

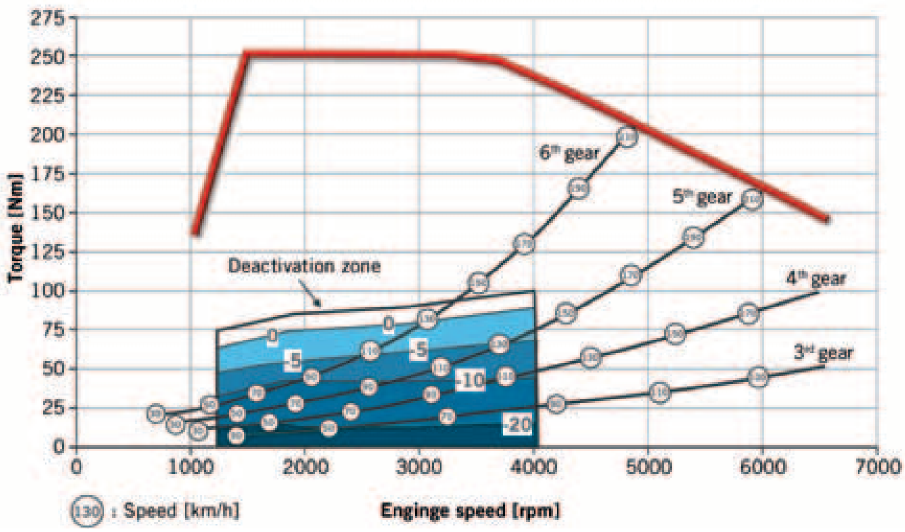
The efficiency of the two active cylinders 1 and 4 is increased because the operating points move to higher loads, ❸. Engine friction in relation to engine speed remains largely constant, while effective power output increases. This heavily de-throttled operating mode results in lower charge cycle losses, improved combustion and lower cylinder-wall heat losses.

The activation of cylinders 2 and 3 occurs in the same order as their deactivation. First the exhaust valves and then the inlet valves are reactivated, sending the trapped air into the exhaust line. The resulting dilution of the exhaust gas is balanced by fuel injection into cylinders 1 and 4. This enables the sensor-based Lambda control to continue operating as normal.

## MATCHING ENGINE CONTROL WITH DRIVING STYLE PATTERNS

Cylinder deactivation occurs in a map area that is used frequently within average customer driving patterns. The lower rev limit was set at 1250 rpm as, beneath this mark, deactivation mode would result in too much cyclic irregularity. The upper limit was determined at 4000 rpm in order to maintain moderate actuator shifting forces. In third gear, the cylinder deactivation zone starts at around 30 km/h and, in fifth and sixth gears, ends at around 130 km/h.

The possible torque in deactivation mode was set at an upper limit of between 75 and 100 Nm depending on engine speed. The knock limit and ignition retardation in deactivation mode mean that optimum




❹ Operating area of cylinder deactivation within engine map

fuel consumption can no longer be achieved at higher torque levels, leading consequently to all four cylinders being activated. At a standstill, the engine is switched off altogether via the automatic start/stop function.

In order to attain greatest fuel-efficiency benefit, cylinder deactivation is applied not just under partial load, but also during trailing throttle conditions. The reduction of braking moment leads to a considerably longer trailing throttle phase, during which fuel injection is deactivated. As soon as the driver activates the brake pedal, the cylinder deactivation mode is cancelled in order for all four cylinders to support the braking effect under trailing throttle. Cylinder deactivation is also suppressed during downhill coasting, as the full engine braking effect is generally desired under these conditions.

The driver is shown the two-cylinder operating mode in the on-board multifunction display, if he/she calls up the current fuel consumption. Without this information, the deactivation mode would barely be detectable as the 1.4 TSI maintains very good acoustic characteristics throughout.

A crucial factor in the engine's excellent vibration characteristics is its base design with a stiff drivetrain construction and lightweight moving parts, as well as its transverse mounting position. When it comes to engine mounts, the same units can be used as those featured in the TDI engines. The dual-mass flywheel was adapted specially to take into account the torsion spring characteristics. It aids the character of the cylinder deactivation engine with a very soft first stage for the deactivation mode and a stiff second stage for high-load operation in four-cylinder mode.

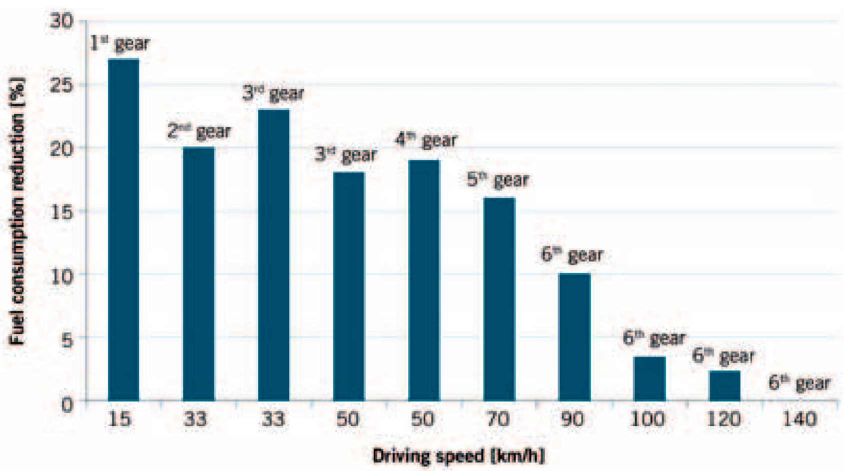
In order to minimize the widely differing exhaust gas pulses between full and deactivation mode, the front and rear silencers in the exhaust system have differently sized resonators and volumes. The length of the pipework was also adapted to suit.

## CONCLUSION

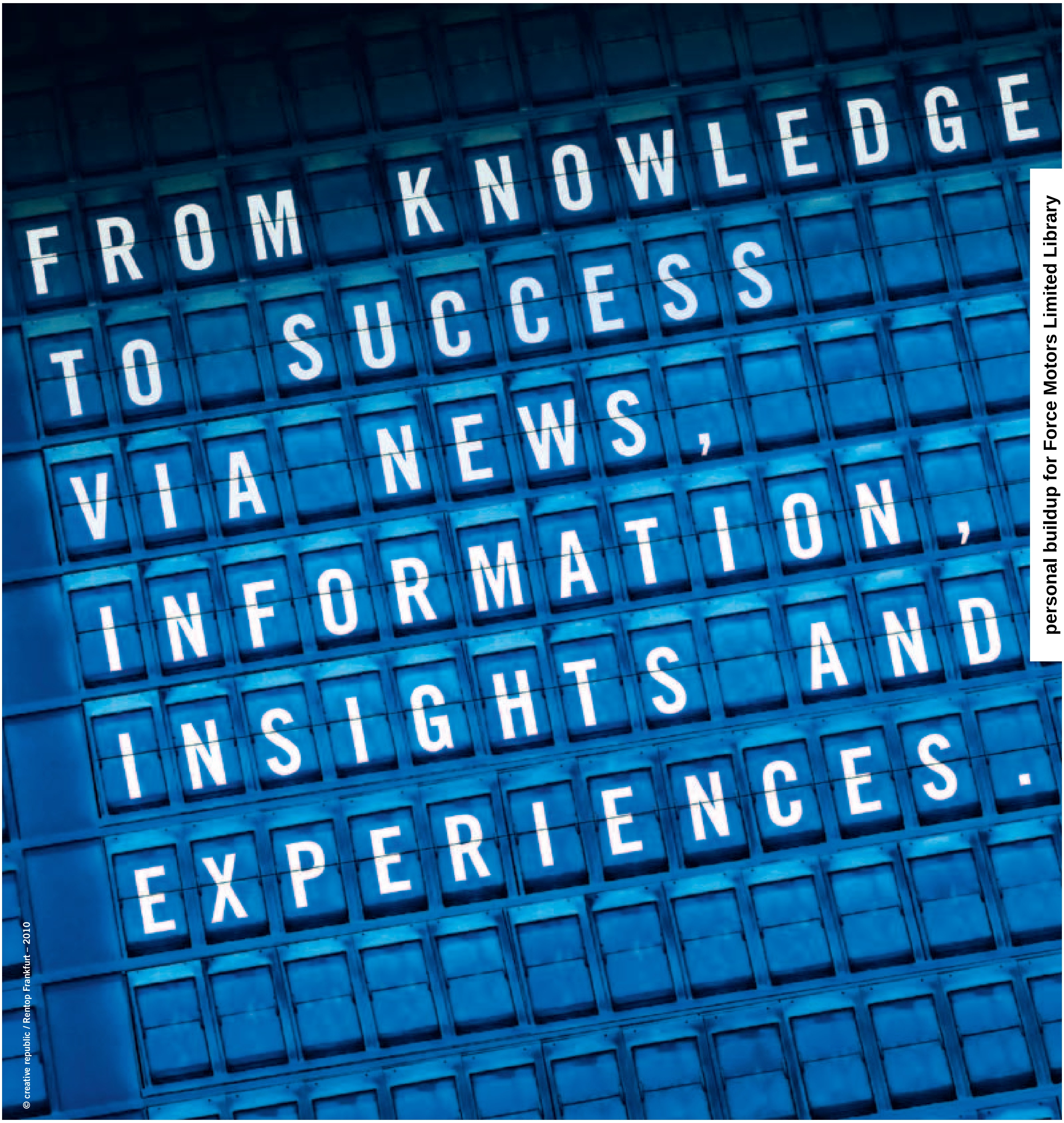
The data produced by the 1.4 TSI with cylinder deactivation proves that it is possible to combine ambitious fuel consumption targets with high power and torque output within the TSI strategy, ❹. The cylinder deactivation technology is an important factor in achieving Volkswagen's $CO_2$ fleet targets. The engine fulfils all the requirements set out in the specification document. Its fuel consumption in the NEDC is lowered substantially by 0.4 l/100 km, which equates to a reduction in $CO_2$ emissions of 8 g/km. If you were to include the start/stop function, which switches off the motor during idling, the savings increase to around 0.6 l/100 km.

With the appropriate driving profile, cylinder deactivation achieves a considerably higher fuel-saving advantage than in the standard consumption test. At moderate speeds in city traffic in particular, as well as cross-country, savings of between 10 and 20 % are possible, ❺. It is not until higher speeds of more than 120 km/h that the load level in the two active cylinders reaches the point where four-cylinder operation once more represents the most fuel-efficient mode.

The first application of cylinder deactivation will occur in 2012 in new sporty versions of the Polo and Audi A1. In these vehicles, the 1.4 TSI has a power output of 103 kW, with a maximum torque in four-cylinder mode measuring a constant 250 Nm between 1500 and 3500 rpm. Further developments were made in parallel to the base engine. It will be possible to apply cylinder deactivation variably within the new engine family bearing the acronym EA 211.




❺ Reduction in fuel consumption due to cylinder deactivation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# POTENTIAL OF VALVE TRAIN VARIABILITY IN PASSENGER CAR DIESEL ENGINES



Variable valve control systems are state-of-the-art technology in today's gasoline engines, although their application in diesel engines has also become the focus of recent studies. These systems offer one possibility of resolving the conflict of objectives between a further reduction in engine-out emissions and an improvement in fuel efficiency. Mahle has examined their potentials on the basis of the Cam-in-Cam variable camshaft for the intake valves.

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## MOTIVATION AND BACKGROUND

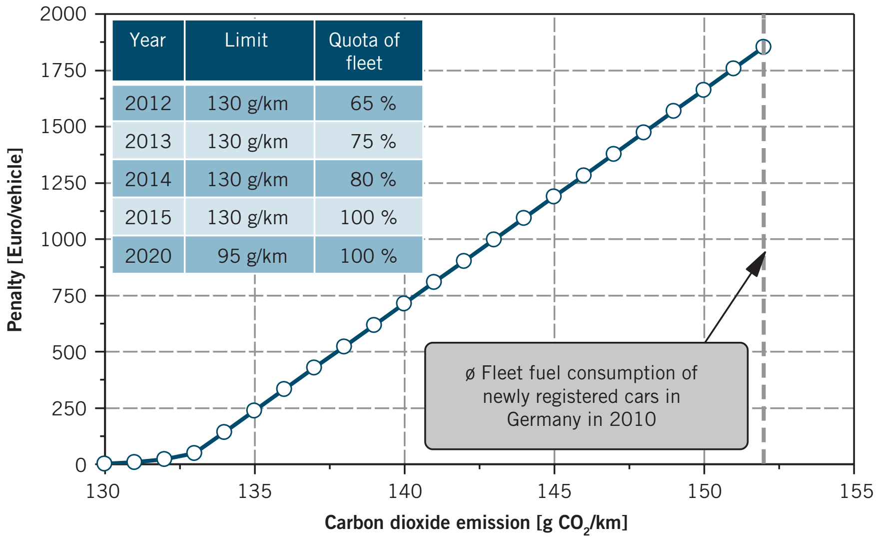
In addition to the introduction of the Euro 6 emissions standards for newly registered vehicles starting in 2014, there is now also a legal regulation by the European Parliament (EC directive no. 433/2009) regarding the reduction of $CO_2$ emissions. It stipulates a reduction of carbon dioxide emissions for the entire vehicle fleet of any respective manufacturer to 130 g/km in the NEDC starting in 2012, and 95 g/km starting in 2020. Noncompliance with these objectives results in penalties. ❶ shows the amount of these so-called incentive taxes [1] when exceeding the stipulated $CO_2$ limit for vehicle fleet fuel consumption, which must be indicated by the manufacturer for each vehicle sold. Until 2015, an initial staggered approach for compliance with the limit comes into effect. While 65 % of the vehicle fleet must comply with the limit in 2012, this percentage rises to 100 % by 2015.

Besides the further reduction of raw emissions, the goal of future technology development is therefore the reduction of fuel consumption. Valve train variability could offer one option to achieve these competing goals. These systems are considered state of the art for gasoline engines, but they are also focused on in current research for diesel engines [2, 3]. One conventional method for the reduction of particulate matter (PM) is the creation of a swirl movement of the flow inside the cylinder by means of a swirl flap, which is located in the flow port. The flap closes the flow port in order to lift the air mass flow over the intake valve of the swirl port. A charge motion can also be created by means of intake variability in diesel engines, which eliminates the use of a swirl flap and the associated throttle losses. In principle, such intake variability is conceivable through cam profile switching or the delay of the individual intake valve lifts.

Alignment of the intake valve control times by means of early or late intake valve closing (EIVC/LIVC) lowers the $NO_x$ emissions and enables a potential reduction of $CO_2$ emissions. By decreasing the effective compression ratio, an additional part of the compression can be transferred to the compressor. This results in the realization of lower compression end temperatures and thus lower nitrogen oxide emissions.

## DESIGN AND FUNCTION

Mahle is able to utilize the proven Cam-in-Cam (CIC) series technology in order to create such intake variability as well as the described effects regarding closing points and charge motion in a cost-effective manner and with optimal installation space requirements. With CIC, two camshafts are assembled together within the installation space of one common camshaft. The fixed cams and the adjustable cams are on the outer camshaft tube, the latter are connected to the inner shaft. The inner shaft can be adjusted relative to the outer shaft by means of a conventional




❶ Penalty per newly registered vehicle depending on vehicle fleet fuel consumption [1] cam phaser, i.e., two adjacent cams are turned toward each other.

For DOHC engines, this enables the adjustment of the two intake valve control times relative to each other. In order to examine the cycle potential and full-load suitability of various variable camshaft configurations on the intake side, several variants were tested in extensive 1D cycle simulation with a predictable PM and $NO_x$ model [5]. Four different variants, as shown in ❷, stood out as candidates for engine testing. The most interesting results were achieved for CIC V1 and CIC V2, which will be described in more detail below. Based on these examinations, the following two variants emerge as the leading options:

: CIC V1: Cam-in-Cam on the intake side, camshaft with baseline cam profile and delayed action of the intake valve on the flow port side
: CIC V2: same configuration as CIC V1 with reduced opening duration at unaltered maximum lift.

The valve lifts of the variants are shown in ❸ as a schematic diagram.

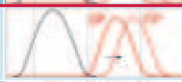
❹ shows the simulation results for the swirl coefficient, the effective compression ratio $\varepsilon_{eff}$ and the volumetric efficiency $\lambda_l$ as a function of the adjustment of the intake valve of swirl and flow ports in a representative partial-load operating point (1500 rpm, 3.2 bar $p_{me}$). The phasing of the intake valves of the flow port has the greatest effect on the swirl number parameter. The cylinder flow swirl number rises with the increased shift of the flow port intake event toward late.

This increases the potential for the reduction of PM emissions. Phase adjustment can be used to steplessly control the desired charge motion. Nevertheless, the swirl values of the conventional operation with swirl flap are achieved. An adjustment of the intake valve event of the swirl port to the flow port does not lead to an improvement of the charge motion. Neither does the valve lift duration show any significant effect on the achievable swirl level.

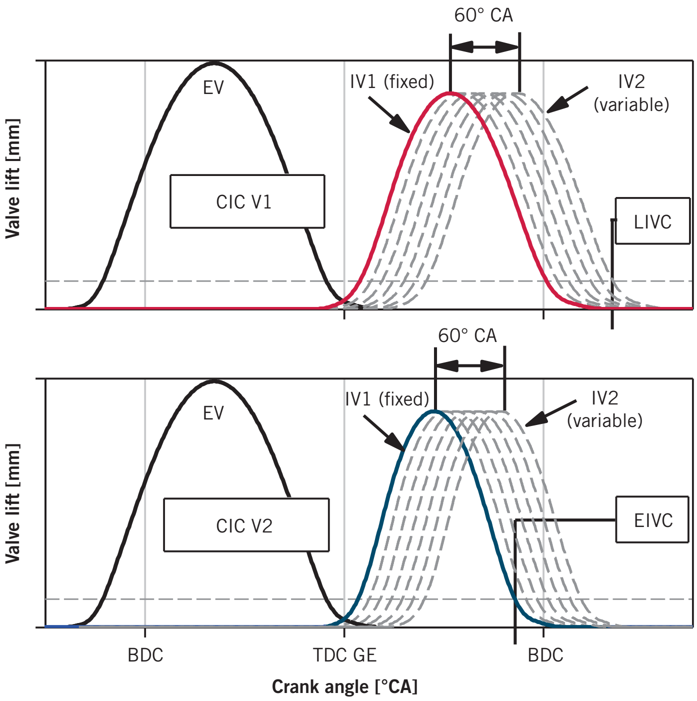
Furthermore, in addition to the swirl effect, the effects of an early or late intake valve closing (EIVC/LIVC) are evident. In the case of CIC V1, the $\varepsilon_{eff}$ weakens along with increasing phasing due to the late intake valve closing. As for CIC V2, there are less significant effects on the effective compression ratio. EIVC is realized at 0° phase setting and the compression is therefore below the base value of 16.5. If the intake valve lift of the flow port is shifted toward late, the $\varepsilon_{eff}$ initially increases (area of baseline intake valve closing) and then decreases toward high phasing (moderate LIVC). Analogously to $\varepsilon_{eff}$, the ratio of the volumetric efficiency $\lambda_l$ is as follows. The result of this investigation is that the combination of Atkinson effect with simultaneous change of the swirl number, in particular, unlocks further potential. A simple adjustment of the closing point at constant swirl, e.g., as realized with CIC V4, is less effective concerning emissions.

## TEST BENCH DESIGN AND TEST RESULTS

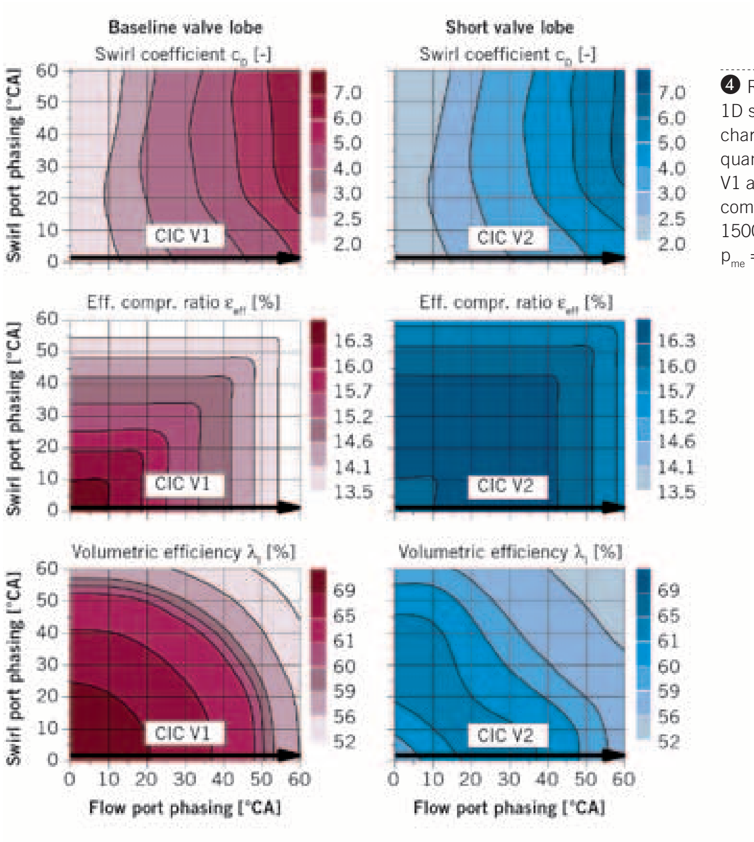
A Euro 5 diesel engine was used for the stationary tests. In series application, this engine features a common rail fuel injection system, exhaust gas turbocharging

| | Configuration | Intake valve opening (all full lift) | Intake valve flow port (FP) | Intake valve swirl port (SP) | Property, opening/ closing timing |
|---|---|---|---|---|---|
| Cam phaser |  | As baseline | Variable | Same position as IV flow port | LIVO + LIVC (both valves) + swirl increase |
| CIC V1 |  | As baseline | Variable ("flow port phasing") | Fixed, same IVO as baseline | LIVC + swirl increase |
| CIC V2 |  | Duration shortened by 35° CA | Variable ("flow port phasing") | Fixed, same IVO as baseline | EIVC, LIVC + swirl increase |
| CIC V3 |  | Duration shortened by 35° CA | Variable ("flow port phasing") | Fixed, IVO 30° CA late | LIVO + LIVC + swirl increase |
| CIC V4 |  | Duration shortened by 35° CA | Fixed, same IVO as baseline | Variable ("swirl port phasing") | LIVC constant swirl |

❷ Overview of the engine-tested variable valve train configurations on the intake side (additional variants were excluded in previous 1D simulations)




❸ Intake variability of the variable camshaft from Mahle (schematic diagram)



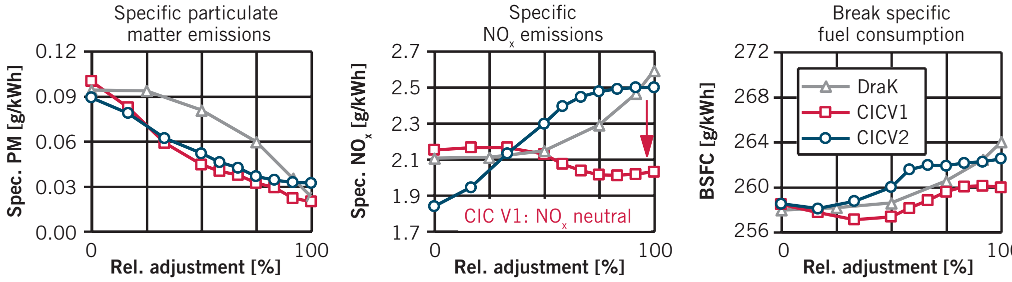

❹ Results of the 1D simulation: characteristic quantities of CIC V1 and V2 in comparison at 1500 rpm and $p_{me}$ = 3.2 bar

with VGT, and cooled high-pressure EGR. The conventional baseline swirl flap module (one swirl flap per cylinder) served as reference system for the increase in charge motion. For investigation purposes, the engine is equipped with full indexing on a standard test bench [6]. The engine control is handled by the prototype control unit Mahle Flexible ECU. This provides the possibility to freely adjust a multitude of engine parameters as well as apply additional technologies and software functions. Custom implemented software functions are, for example, controlling the timing of 50 % mean burnt fraction or adjusting the indicated brake mean effective pressure of each individual cylinder [7].

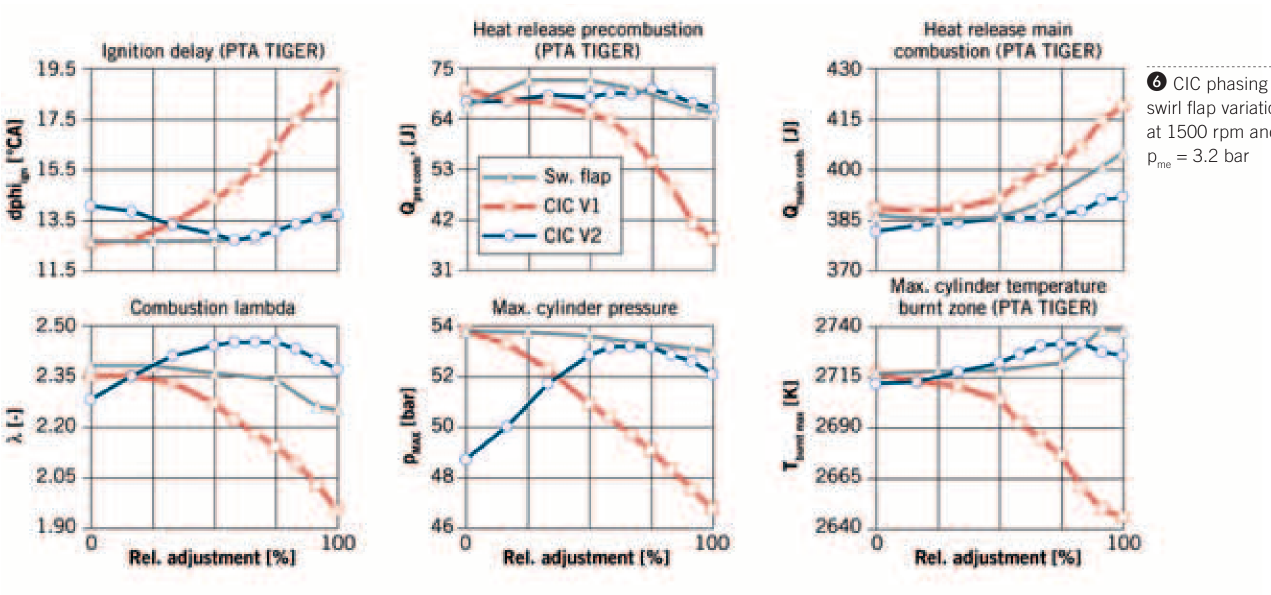
The potential of the variable camshaft versus the conventional swirl flap operation is demonstrated by the results of an exemplary partial-load operating point at n = 1500 rpm, $p_{me}$ = 3.2 bar. For comparison purposes, the adjustment range is shown in a standardized fashion. In both cases, 0 % relative adjustment corresponds to a fully opened swirl flap or respectively an intake valve phasing of the flow port equaling zero. In this case, the swirl number is approximately 2.2 in both cases. A relative adjustment of 100 % corresponds to a maximum swirl generation by means of a 100 % closed swirl flap or a maximum valve phasing of 60° relative CA at a comparable swirl number of approximately 4.6.

❺ shows a quantitatively comparable reduction of PM emissions for all three variants at maximum adjustment. The adjustment only results in a minimal change in fuel consumption of maximal 2 % for CIC V1. The swirl flap and CIC V2 behave similarly concerning $NO_X$ emissions. The conflict of objectives between PM and $NO_X$ emissions, however, can be circumvented by CIC V1. CIC V1 makes it possible to reduce the PM emissions by means of swirl increase in the combustion chamber without increasing the $NO_X$ emissions. This is due to two different mechanisms: On the one hand, valve phasing increases the swirl in the combustion chamber, on the other hand, late closing of the intake valve (LIVC) leads to a reduction of air mass, effective compression ratio, and maximum cylinder pressure. ❻ shows that the maximum cylinder pressure decreases along with the increasing phasing of CIC V1, and that the temperature $T_{burnt\ max}$ is reduced in the burnt zone. Along with the reduction of the effective compression ratio, the ignition conditions in the cylinder change as well. For CIC V1, the ignition delay is prolonged along with increasing phasing. The effect of the swirl flap on the ignition delay, however, is marginal. Along with the increasing ignition delay, the transformed fuel amount (heat transformation) decreases in the precombustion and is shifted to the main combustion, ⑥. The peak pressure increased by the heightened premix percentage is overcompensated by the reduction of the effective compression ratio.

❼ shows the combustion processes of the three variants at 100 % relative adjustment and illustrates the increased ignition delay as well as the heat transformation shift from the precombustion to the main combustion. The control of the timing of 50 % mean burnt fraction was realized via




❺ Swirl flap variation and CIC phasing at 1500 rpm and $p_{me}$ = 3.2 bar




❻ CIC phasing and swirl flap variation at 1500 rpm and $p_{me}$ = 3.2 bar

the main injection timing. Both pre-injections are identical for all three variants.

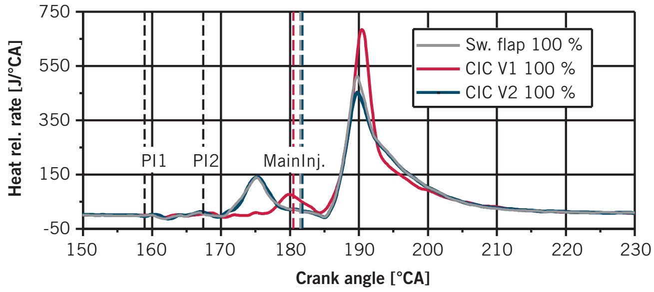
In order to examine the effect of the various precombustions on the main combustion, an additional test was carried out to align the combustion process of the precombustions of CIC V1 and swirl flap by varying the injection start and the amount of fuel. While conditions were the same concerning injection timings and precombustion heat release, the differences in the main combustion partially persisted. The extended ignition delay and the higher premix percentage of CIC V1 can only be partially achieved by application measures of conventional technologies, such as the swirl flap. Only the combination of decreased compression ratio and extended ignition delay enables an $NO_x$-neutral decrease of soot emissions. The resulting increase of the cylinder pressure gradient as a limiting element has to be mentioned at this point, as it contributes to an increased combustion noise. A pressure gradient of 4 bar/° CA is reached at n = 1500 rpm and $p_{me}$ = 3.2 bar at maximum valve phasing with an EGR rate of 22 %.

In order to evaluate the effect of valve phasing in the EGR application range, an EGR variation at n = 1500 rpm and $p_{me}$ = 3.2 bar with constant rail pressure and timing of 50 % mean burnt fraction is exhibited for all CIC variants, ❽. For this purpose, the EGR rate is raised from 0 %, using the closed EGR valve as a starting point (point A in ⑧), until the valve is completely opened (point B in ⑧). The cross section area of the turbocharger's VGT is constantly held in the baseline position. After opening the EGR valve to 100 %, the VGT cross section is gradually decreased (point C in ⑧). The resulting rise in the scavenging pressure gradient ($p_{30}$-$p_{21}$) thus leads to an increase of the EGR rate. ⑧ shows the EGR variations at swirl flap position 100 %, CIC position 100 % (V1 on the left, V2 on the right), as well as swirl flap and CIC 0 % (without swirl measures). The distinctive minimum of the respective BSFC $NO_x$ trade-off lies in the range of the fully opened EGR valve. The decrease of the VGT cross section in the range of the highest EGR rates leads to a rise in BSFC.

The previously discussed positive effects of CIC V1 on emissions and fuel consumption can also be observed during operation with high EGR rates. CIC V1 as well as V2 show a fuel consumption advantage compared to the closed swirl flap across the entire variation range. The conflict of objectives between PM and $NO_x$ emissions can also be observed for the EGR variation with swirl flap and CIC V2. The trade-off also behaves similarly due to the nearly identical combustion process for swirl flap and CIC V2. The CIC V1 effects influencing the combustion process, particularly the higher percentage of premixed combustion, leads to an enhanced EGR compatibility of CIC V1. In spite of an increase of the EGR rate to more than 40 %, the PM emissions stay on an equally low level across the entire range of variation. For swirl flap and CIC V2, however, the same EGR variation leads to an increase of PM emissions.

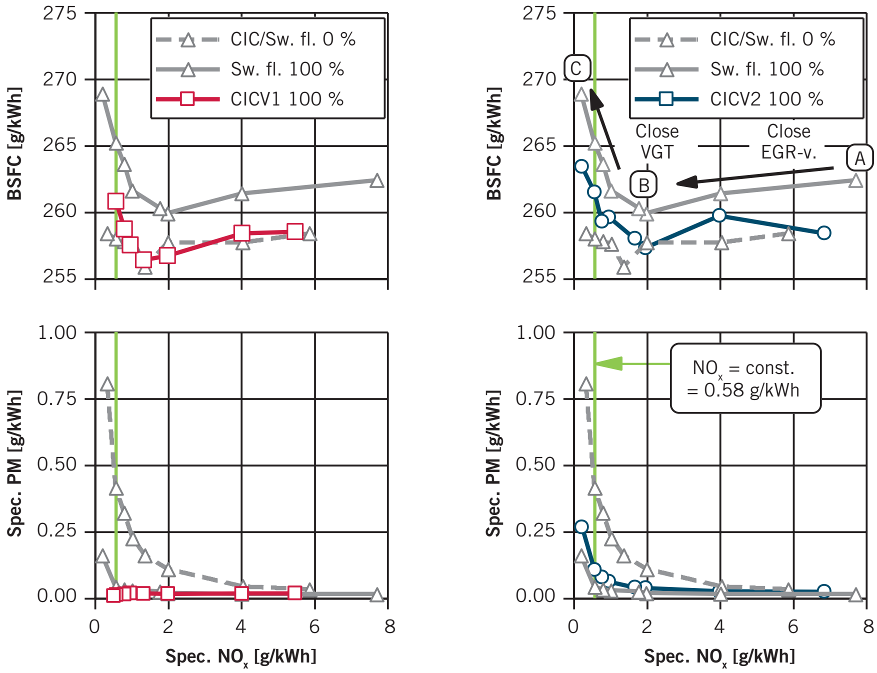
## CYCLE POTENTIAL

Based upon the results from the EGR variations with variable camshaft technology, an operational strategy with a course of action according to the procedure outlined in ⑧ for further NEDC relevant operating points in order to illustrate the consumption potential of variable camshaft under




❼ Comparison of combustion processes at swirl flap 100 % and CIC 100 % (1500 rmp, $p_{me}$ = 3.2 bar) consideration of the Euro 5 emissions requirements is used. For this purpose, a driving cycle simulation for the test engine in its typical vehicle application in GT drive is used. For the respectively derived operating points, which together represent approximately 80 % of the cycle-relevant fuel consumption (operating points in ❾), first the optimal setting parameters of the Cam-in-Cam in a 1D cycle simulation by means of GT power is predetermined. The reference values of technologies in accordance with the Euro 5 emissions level, which are subsequently determined on the test bench in a steady-state engine at operating temperature, serve as a data basis for the cycle simulation. The cycle simulation incorporates a heating curve for the frictional loss of the engine. The deviations in raw emissions due to warmup are not shown. The EGR rate is set to achieve Euro 5 $NO_x$ values for the variable camshaft as well as for the swirl flap variant. Through additional adjustments of the injection start, the injection pressure, and the VGT position, almost identical $NO_x$ and PM emissions for both configurations are achieved for all operating points.

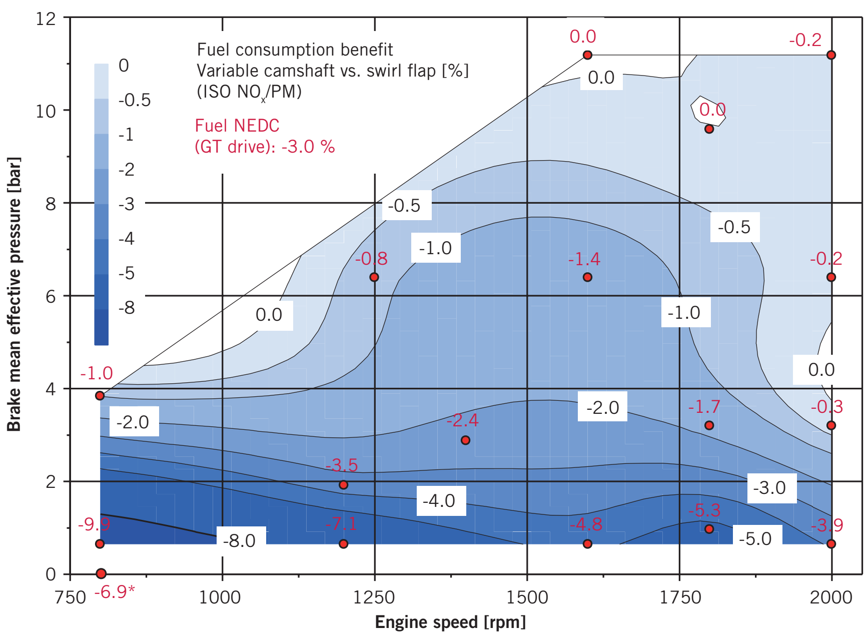
The consumption results in the base application cycle with swirl flap in the described model are comparable to the values published by the manufacturer. Using this model with the steady-state test data for the variable camshaft, a consumption advantage of 3 % can be realized under the basic condition of raw $NO_x$ emissions in the range of Euro 5. ⑨ shows the determined consumption advantages in the NEDC relevant range of the operating map. The most significant advantages occurred with low loads and when idling. The frictional loss of the phaser is approximately 10 W (hydraulic performance due to engine oil requirement) and is included in the values presented.




❽ EGR variation at 1500 rpm, 3.2 bar $p_{me}$

## SUMMARY AND OUTLOOK

❿ shows a summary of the examined variants, which were evaluated by means of the effects swirl, effective epsilon, and gas exchange work. Besides the CIC V1 and




| Speed | Engine load | Diff. BSFC (ISO $NO_x$/PM) |
|---|---|---|
| rpm | bar | % |
| 800 | 0.0 | -6.9* |
| 800 | 0.6 | -9.9 |
| 800 | 3.8 | -1.0 |
| 1200 | 0.6 | -7.1 |
| 1200 | 1.9 | -3.5 |
| 1250 | 6.4 | -0.8 |
| 1400 | 2.9 | -2.4 |
| 1600 | 0.6 | -4.8 |
| 1600 | 6.4 | -1.4 |
| 1600 | 11.2 | 0.0 |
| 1800 | 1.0 | -5.3 |
| 1800 | 3.2 | -1.7 |
| 1800 | 9.6 | 0.0 |
| 2000 | 0.6 | -3.9 |
| 2000 | 3.2 | -0.3 |
| 2000 | 6.4 | -0.2 |
| 2000 | 11.2 | -0.2 |

*Idling point: $\Delta\dot{m}_{fuel}$ [l/h]

❾ Potential of variable valve train in the NEDC

| | Configuration | Swirl | Effective compression ratio | Gas exchange work | Effects |
|---|---|---|---|---|---|
| Cam-phaser |   | + | + | – | **: Considerable BSFC disadvant.** : PM reduction and LIVC (no $NO_x$ reduction), as LIVO, improved engine warmup |
| CIC V1 |  | + | + | + | : PM reduction : $NO_x$ reduction (LIVC) : BSFC reduction |
| CIC V2 |   | + | o | o | : PM reduction or $NO_x$ reduction with EIVC : Lower BSFC with EIVC, higher BSFC at high air mass flows |
| CIC V3 |   | + | o | – | : PM reduction : LIVO, $T_{Exhaust}$ rises : Improved engine warmup : BSFC disadvantage |
| CIC V4 |   | – | o | o | : No PM reduction : $NO_x$ reduction with EIVC : Lower BSFC with EIVC, higher BSFC at high air mass flows |

⑩ Summary of the examined variabilities on the intake side

V2 introduced in this article, additional variants were examined in order to study an expansion of the potential when using of a double-acting phaser, which introduces a second variance and offers adjustment options according to the operating maps as outlined in ④.

The variant CIC V4, ②, offers an Atkinson effect (LIVC) without changing the swirl. CIC V4 behaves similarly to CIC V2 concerning additional engine effects. The emissions behavior regarding PM, however, is less advantageous. In addition to the swirl, variant CIC V3 realizes a late opening of the intake (LIVO), which has a positive effect on the warmup behavior of the engine. Exhaust gas temperatures could therefore be raised by up to 50 K. This, however, goes along with significantly increased fuel consumption. Swirl can also be created by means of a conventional phaser. LIVO of 60° CA results in an even higher swirl level, compared to what is achievable with a swirl flap or CIC. The combination of LIVO and LIVC results in a filling reduction and significantly increased gas exchange losses. An advantageous reduction of $NO_x$ cannot be obtained, since the compression end temperature as well as the maximal combustion temperature rise. The warmup behavior improves with significantly increased fuel consumption.

Based on the studies conducted by Mahle, the application of a double-acting phaser on the intake side offers only limited additional potential. It is easier to realize a variability to raise the exhaust gas temperature by means of a conventional diesel control flap (lossy).

The variable camshaft Cam-inCam in the shows configuration is able to demonstrate additional potentials regarding the reduction of emissions and consumption for modern passenger car diesel engines and thus contribute to the compliance with future emissions requirements. The following potentials were determined on the engine test bench during operation with CIC in comparison to conventional swirl flap operation:

: reduction of PM emissions analogous to swirl flap
: reduction of $NO_x$ formation by late closing of the intake (decreased effective compression ratio) and therefore an improved PM/$NO_x$ trade-off
: reduction of fuel consumption due to advantages in gas exchange and decrease of peak pressures (LIVC).

The trend toward lowest compression ratios and enabling of lowest possible raw emissions makes it difficult to ensure acceptable engine operation during the cold start phase. Even for this challenge, CIC represents an effective technology, since it has a stabilising effect on combustion, especially in the minimal load range.

CIC offers additional areas of application on the exhaust side, with the aim to contribute toward exhaust gas management that is independent of fresh air and to accelerate or improve the catalytic converter light-off, or the light-off and the regeneration of downstream components for exhaust gas aftertreatment.

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# SERVO-DRIVEN PIEZO COMMON RAIL DIESEL INJECTION SYSTEM

The requirements to be met by future diesel engines represent major challenges for fuel injection technology: Fuel consumption, emissions and noise development are to be further reduced without impairing driving enjoyment. To address these challenges, Continental has developed a new fuel injection system that features a high level of precision and accuracy. The key component is a servo-driven injector that is operated in a closed control circuit.

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## MOTIVATION

Since its introduction to the market, piezo-electric actuation technology has been instrumental in improving the efficiency of direct injection (DI) diesel engines. The level of switching accuracy and speed facilitated by piezo technology has helped to pave the way for complex multiple injection patterns, thereby improving engine-out emissions levels and reducing combustion noise.

However, despite the success of this technology, modern diesel engines require further progress on all levels as they increasingly have to compete against turbocharged DI gasoline engines. Therefore the good low-end torque must be maintained while engine noise needs to be further refined. Since piezo injectors have already reached a high level of mechanical precision, Continental aims to achieve further progress by using closed-loop control. At the same time the hydraulic efficiency is dramatically increased in comparison to previous piezo injectors. In the following an overview of the system, its key components, and the related control functions is given.

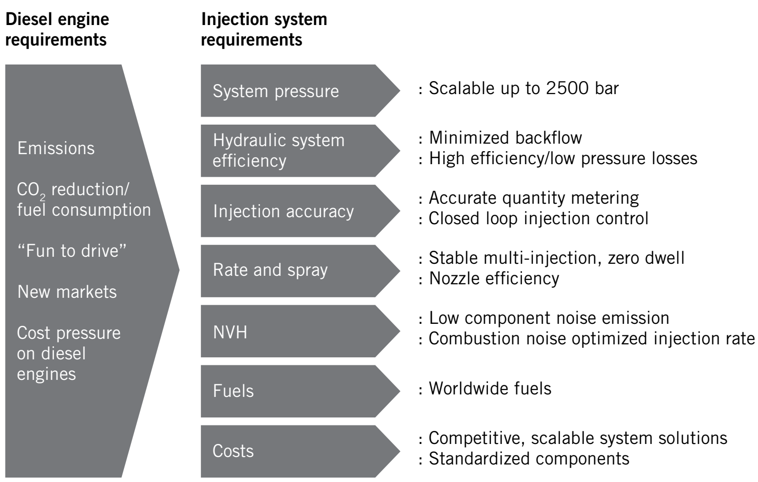
## REQUIREMENTS FOR DIESEL INJECTION

❶ shows the correlation between the above-mentioned engine requirements and the injection system requirements. Continuing a long-standing trend, the new system was developed for higher fuel pressure levels of up to 2500 bar. To increase efficiency, the fuel return flow from the injector and pump were minimized. Increased requirements in terms of injection accuracy demanded improved injector stability and closed-loop control via the engine control unit to correct the injection duration. Minimized hydraulic dwell between two injections is another prerequisite for flexible shaping of multiple injection patterns. Optimization of the nozzle resulted in improved spray efficiency and mixture distribution which helps to improve fuel consumption and emission levels.

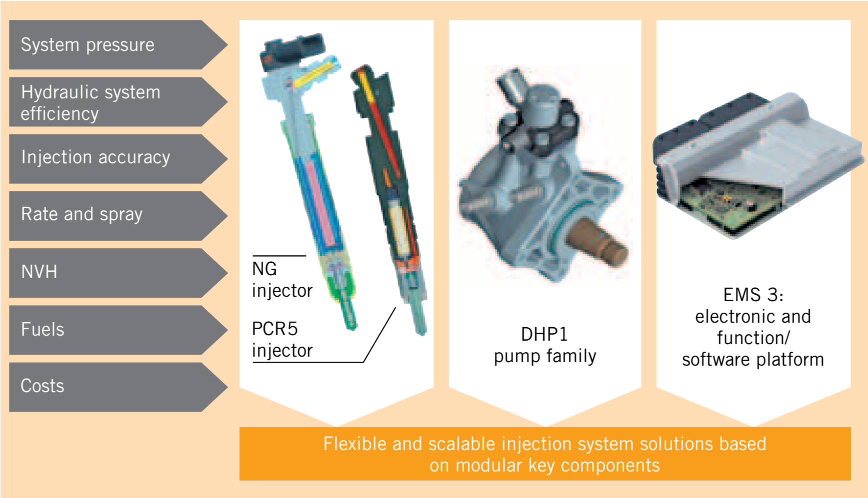
The reduction in noise emissions of injector and pump plus the ability to reduce the combustion noise result in reduced engine noise. The system's robustness towards various fuel qualities is a welcome property within worldwide diesel applications. Last but not least, cost pressures and quality awareness had to be addressed. The new injection system solution is therefore based on standardized and scalable components. Between them they cover a wide range, spanning engine applications from strongly cost-driven to high performance.

The key elements of the Continental diesel portfolio are shown in ❷:

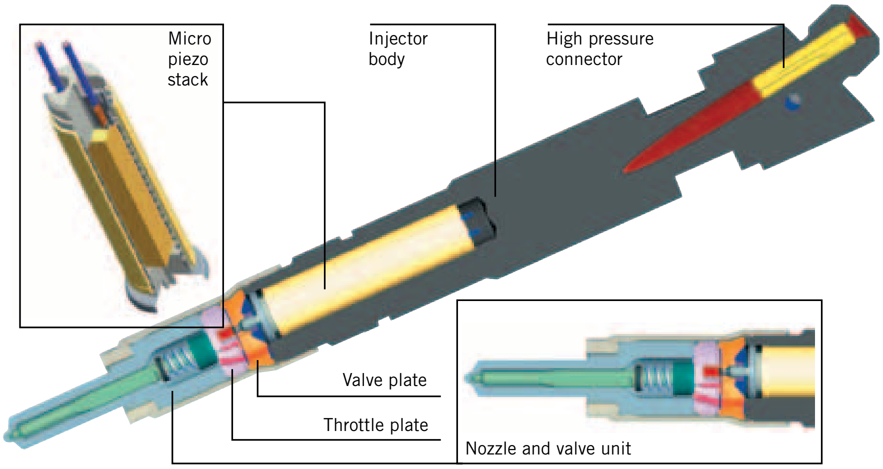
: The new inline servo-valve PCR5 injector has a minimized backflow and a closed-loop control.
: The direct drive NG injector with mechanical coupling between actuator and nozzle needle, which also facilitates real-time closed-loop control and advanced capabilities for injection rate shaping. As this injector has already been presented [1] it is not covered here.




❶ Breakdown of diesel engine requirements to injection system requirements




❷ Key components which are assembled to build scalable system solutions



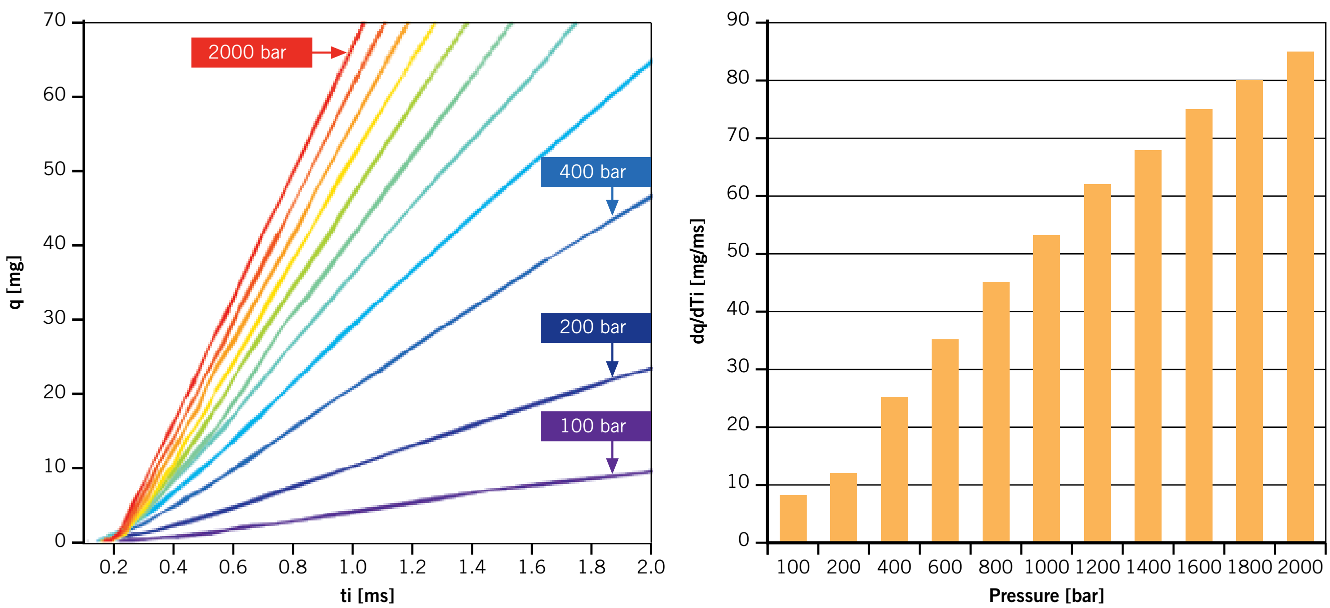

❸ Main design features of the PCR5 injector

- The DHP1 pump, featuring a roller shoe driven single-plunger pump with digital inlet valve, is the basis for a whole pump family.
- The open and scalable electronics and software platform EMS3 (Engine Management System 3) contains functions for the fuel injection system and for the overall engine management like, for example, air path and exhaust gas after treatment control. EMS3 comprises standardized solutions for both gasoline and diesel engines [2].

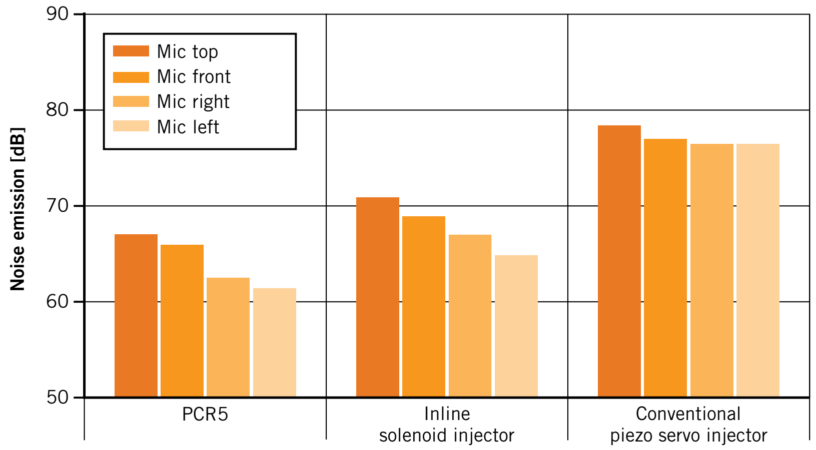
## NEW INLINE INJECTOR

The new servo-driven inline injector is actuated by a very compact and powerful piezo drive. Compared to a conventional piezo actuator, the new micro stack requires only one fourth of the volume. This compact packaging within the injector was used to maximize the internal hydraulic volume. As the micro stack is so powerful, the rail pressure can also be significantly increased. In a first development step the injector will be capable of handling 2000 bar. However, its general layout offers the potential for up to 2500 bar of injection pressure. An overview of the injector design is depicted in ❸.

Optimizing the hydraulic configuration of the injector led to the best possible trade-off between injector return flow and injection characteristics. The PCR5 injec-




❹ Injection characteristics of the PCR5 injector; ti map (left), ti map gradient (right)



5 Measured injector noise in a semi-anechoic chamber

tor is free of permanent leakage, thus allowing a rail pressure increase without negative impact on system efficiency. In addition the injector has the pressure hold capability that is required by start/stop systems. Compared to Continental's previous servo injector, the total fuel return flow was reduced by approximately 80 %. This progress can be exploited by using a smaller high-pressure pump. 4 shows that very linear, fully ballistic gain curves were achieved at the same time. The gain curve gradients were reduced to < 100 mm³/ms, ensuring a very stable and reproducible injection quantity behavior. The actuator and valve design were dimensioned to ensure a stable opening and closing behavior under all operation conditions. Furthermore the design is robust against low quality fuels and deposits.

Using the new piezo micro stack paved the way for an overall more compact product. If required, the PCR5 injector can have as little as 17 mm shaft diameter along its whole length. It therefore contributes to downsizing concepts. Nevertheless, the injector's high pressure volume was maximized, which lowers quantity deviations across multiple injections. To achieve future emission targets, the nozzle tip geometry was also optimized. Tapered spray holes with a cone factor of up to 5 are one measure to improve emissions. Reduced sac volume and improved seat geometry for optimized de-throttling are additional parameters which are currently being reviewed for their potential benefits.

On top of improving the injector's core functions, its NVH performance has also been improved significantly. This is due to the reduced energy demand and an optimized current profile as well as the inline configuration. As a result the injector noise has been lowered to a level below that typically found in solenoid inline injectors. This sets a new standard irrespective of the actuation principle. The corresponding noise measurements in an acoustic chamber can be seen in 5.

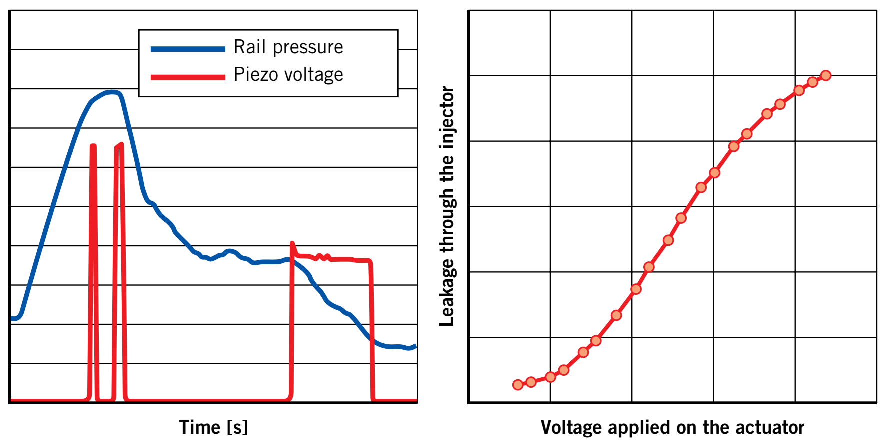
## INJECTION CONTROL FUNCTIONS

In order to fully exploit the injection pressure potential of up to 2500 bar, and to support complex multiple injection patterns, the fuel quantities need to be more accurately metered. For this purpose, the piezo actuator has much more potential to offer than a solenoid actuator. Two control functions have been implemented:

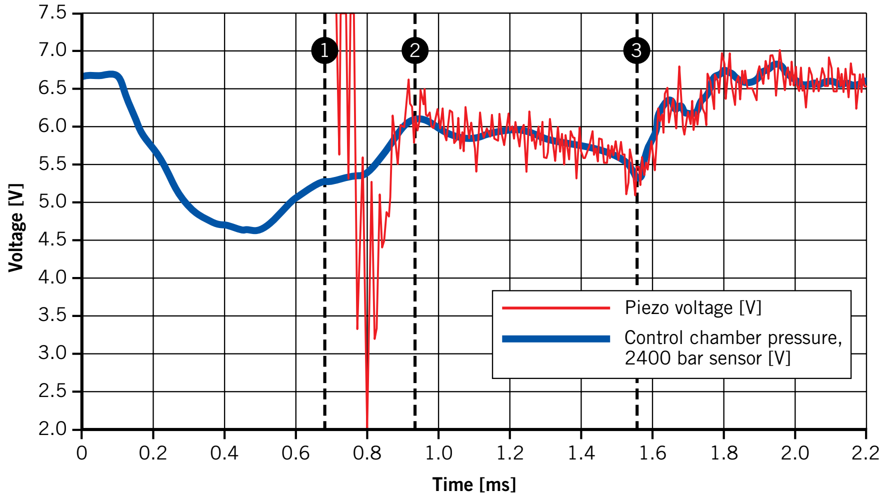
- Injector Control Valve Adaptation (ICVA)
- Needle Motion Control Function (NCTL).

The ICVA function is used to monitor the valve characteristics. By applying different voltage levels to the piezo stack, it is possible to control a variable valve lift. When the valve opens, the resulting leakage reduces the pressure in the system. This pressure-drop can be measured by a conventional rail pressure sensor, 6 (schematic diagram). For that purpose a test pulse is applied in a time frame where there is no pump activity and no injection, 6 (left). Therefore, the pressure behavior which is measured at this moment is only a result of the flow drained via the valve. Monitoring the rail pressure gives information on the valve characteristic, 6 (right) and is used to adapt the command of the piezo in order to precisely compensate manufacturing tolerances and lifetime effects (drift) of the valve unit.

The ICVA function is fully focused on the valve unit. In addition, the PCR5 system has an integrated NCTL, based on a feedback signal delivered by the piezo actuator. In a servo-driven injector, the needle motion generates a change of the control chamber volume. Due to this effect, the pressure in the control chamber is closely linked to needle motion. When the valve is closed, there is almost no flow through the outlet throttle, and the pressure on the valve is almost equal to the pressure in the control room. As the piezo stack is in direct contact with the valve, it is possible to use the piezo as a pressure sensor, which detects the pressure inside the control chamber. 7 focuses on the closing of valve and needle. At position 1 the servo valve has been



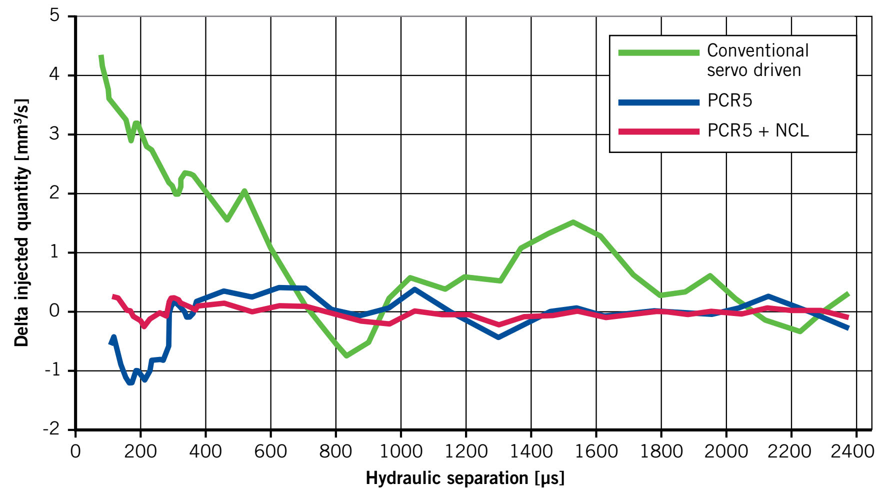
6 Schematic diagram of the valve characteristics




❼ Measured piezo voltage compared to control chamber pressure

closed, the inlet is filling the control chamber. At position 2 the pressure in the control chamber is sufficiently high to initiate the closing of the needle. This causes an increase of the control chamber volume, and the pressure decreases slightly. At event 3 the needle is closed, which stops the change of volume, and the pressure increases due to the refilling through the inlet throttle. For this measurement the injector was instrumented with a control chamber pressure sensor. As shown in ⑦, the same signature can be acquired from the piezo actuator voltage. Thus, detecting this signature provides information to the engine control unit (ECU) on the needle closing time for all injections and all injectors.

As the piezo stack is a superposition of many layers, the output signal is amplified by the stack design and does not need any specific amplification as would be required with standard pressure sensors. Therefore, the piezo signal accuracy has the quality to build a real-time control function which monitors injection duration and significantly improves the fuel metering, for example, during multiple injection as shown in ❽. Adding NCTL to the PCR5 injector, which itself is already robust against pressure waves, shows stable multiple injection patterns with up to +/-0.2 mg/stroke with a hydraulic separation as low as 60 µs. The combination of injector and needle control function makes it possible to attain minimized hydraulic separations, thereby achieving a marked improvement in the performance of the injection system [3].



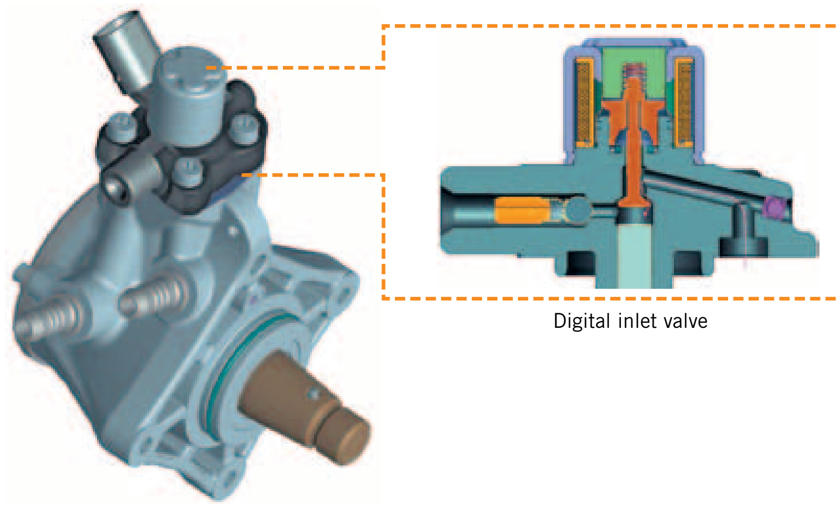

❽ Injection quantity difference of 2nd injection dependent on hydraulic separation time with NCTL function (800 bar/pilot-main pattern)

## HIGH-PRESSURE PUMPS

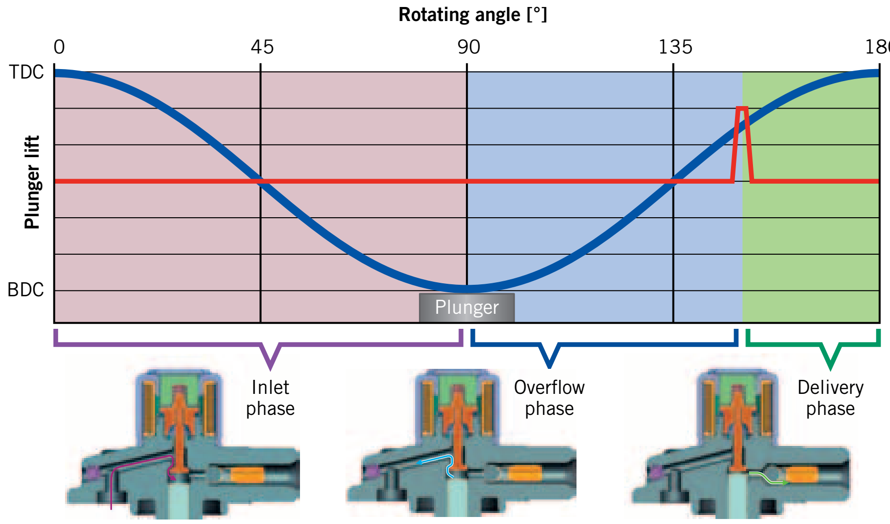
To cope with a wide range of engine applications Continental has developed a new high pressure pump family. The basis for this family is the single plunger pump (DHP1) with a roller shoe drive and a digital inlet valve (DIV) as shown in ❾. Compared to conventional pumps, the new inlet valve replaces two valves at the same time, the electromechanical analog volume control valve (VCV) and the mechanical inlet valve. The digital inlet valve technology reduces the number of moving parts and lowers the susceptibility to low fuel quality. The valve is currentless open, hence leading to a safe engine shut-down in case of failure.

The DIV is mounted directly on the top of the pump cylinder to achieve a compact pump design with minimized cylinder dead volume. The working principle is shown in ❿. During the inlet phase the plunger chamber is always filled completely without any de-throttling. After the plunger has reached bottom-dead-center the overflow phase begins. Fuel which is not needed is pushed back into the supply. The rail pressure controller triggers the actuation of the DIV depending on the consumption at the actual operating point. Now the delivery phase starts and lasts until top-dead-center of the pump is reached. The main advantages of the DHP1 are:

: The design supports pressure capability up to 2500 bar.
: The concept allows for compact packaging; the pump weight is on benchmark level.
: The DIV technology ensures a better plunger filling and thus an improved hydraulic efficiency. This means that the required presupply pressure to the pump can be lowered. The power consumption can therefore be reduced, which in turn lowers the $CO_2$ emissions.
: The DIV facilitates a very immediate control of the required fuel quantity and thus strongly reduces control deviations, especially in transient driving conditions.




9 Diesel high pressure pump DHP1




10 Digital Inlet Valve (DIV) working principle

## CONCLUSIONS

The new servo-driven piezo common rail injection system offers the efficiency and accuracy to fulfill upcoming engine requirements, to optimize engine-out emissions and thus to minimize the need for exhaust gas after treatment. The PCR5 injector and DHP1 offer a pressure potential of up to 2500 bar. The absence of permanent injector leakage and its pressure-hold boost the overall system efficiency and fully support the widespread use of the start/stop function. Continental has leveraged its long experience with piezo technology to improve the injector in two important aspects:

- Firstly the miniaturization of the actuator and of the servo hydraulics allows significantly reduced fuel backflow and a reduced high pressure pump sizing.
- Secondly the tolerances of the injection system have been improved by using the piezo actuator as a sensor to build closed-loop control of the injected fuel quantities.

The improved accuracy permits the use of extended injection patterns without additional sensors. With dwell times below 100 µs the system flexibly supports complex multiple injection control. In addition, the minimized actuator noise and the system's potential contribution to reduced combustion noise address a worldwide challenge to the diesel engine's success. In combination with the DHP1 pump family and the open and scalable EMS3 electronics and software platform, this makes for an efficient diesel injection system which can be applied across a wide range of passenger car models developed for Euro 7 emission levels and beyond.

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# LOWER FUEL CONSUMPTION WITH NEW OXYGEN SENSORS

Owing to ever more stringent emission limits, exhaust sensors are gaining an increasingly important role. Driven by new legislation, the requirements by car makers regarding size, durability and accuracy of the sensors are continually on the rise. Bosch continues to optimize oxygen sensors to meet these requirements. In 2011 the new generation LSF Xfour oxygen sensor entered mass production.

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## LEGISLATION

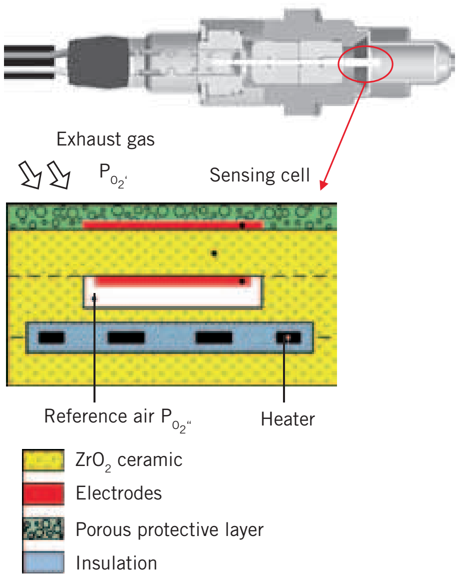
Adoption of the Euro 5 and Euro 6 standards in Europe and Brazil, for example, require lower $NO_x$ emissions. The new Bin2 and LEVIII limits at USA set particularly stringent requirements. $NO_x$ emissions for Low Emission Vehicles (LEV) may not exceed 44 mg/km. For Super Low Emission Vehicles (SULEV) the limit is only 12 mg/km.

## HISTORICAL PERSPECTIVE

Bosch has been manufacturing oxygen sensors since 1976. A oxygen sensor measures the oxygen concentration in the exhaust pipe. It allows precise control of the operating conditions of combustion engines and catalytic converters. In April 2011 Bosch launched the new generation LSF Xfour oxygen sensor. The design is smaller, more durable and more accurate than the predecessor generation. Additionally the sensor makes its signal available faster than the predecessor generation.

## MOTIVATION

More than 90 % of the harmful hydrocarbon emissions (HC) are generated before the engine and the catalytic converter have reached operating temperature. After reaching operating temperature, the emissions are very low. Therefore the start-up phase of sensor and converter must be shortened, by cutting the light-off time of both devices. This allows early optimization of the air-fuel mix to ensure operation at $\lambda = 1$. $\lambda = 1$ denotes the optimal chemical, stochiometric ratio between combustibles and oxidizables to ensure the cleanest possible exhaust gas. Therefore a high accuracy of the sensor signal and a fast readiness after engine start is necessary.




❶ Layout of a planar switching-type sensor

## FUNCTION

The heart of a oxygen sensor is the ceramic sensor element made of zirconium dioxide in which two electrodes and the heater are embedded. Exhaust gas is in contact with the outer electrode of the sensor through a porous ceramic protection layer. A second, inner-electrode is exposed to the ambient air through a reference channel. When the zirconium oxide is heated to high temperature, the sensor element generates a Nernst voltage. The voltage value depends on the partial pressure difference of oxygen between the two electrodes, ❶.

The difference of the oxygen partial pressures at the two electrodes, the temperature, the molar constant, and the Faraday constant enter the Nernst equation, Eq. 1.

**EQ. 1**

$$U_N = \frac{RT}{4F} \ln \frac{P''_{O2}}{P'_{O2}}$$

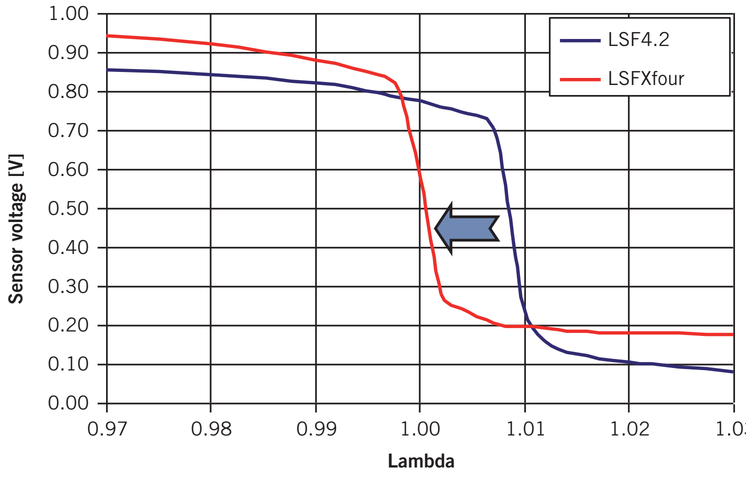
The dependence of voltage on air-fuel ratio (lambda) is called the characteristic curve of the oxygen sensor. This characteristic depends on the sensor temperature.

## TEMPERATURE CONTROL OF THE NERNST CELL

By precisely controlling the Nernst cell temperature, the accuracy of the oxygen sensor can be greatly enhanced. The internal resistance of the Nernst cell is a precise gauge of the sensor element temperature. Hence, the inner resistance of the Nernst cell is used as the control parameter for the heater power control loop.

## CATALYTIC REACTION IN THE PROTECTION LAYER

The electrodes are not only sensitive to oxygen but also to hydrogen concentra-



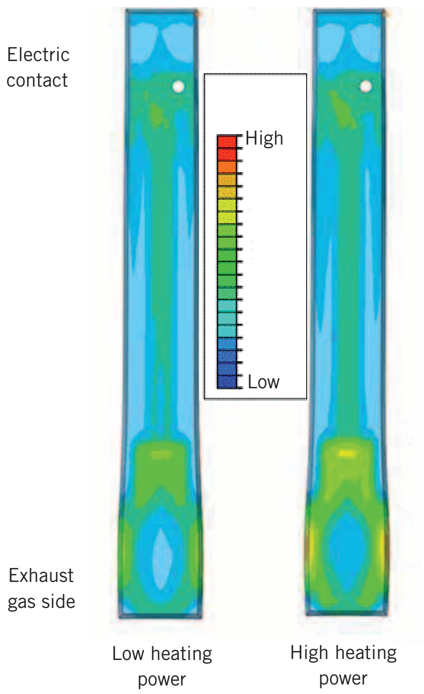

❷ Optimized lambda-characteristic curve of LSF Xfour

Sample variants were manufactured based on the simulation results, ❸. A sensor signal availability of less than 5 s after key-on was achieved. Compared to LSF4.2 the time until the sensor is ready could be reduced by half. To make use of the fast light-off both the engine specific application and the robustness of the sensor against water is relevant.

tions in the exhaust gas. Theory suggests that the inflection point of the characteristic sensor curve occurs exactly at $\lambda = 1$. In practice this inflection point can be shifted to the lean side due to the mix of various exhaust gas components. Small hydrogen molecules diffuse faster through the protection layer of the outer electrode than larger oxygen molecules. Therefore the oxygen concentration near the electrode differs slightly from that in the exhaust gas stream. To counteract this effect, chemical additives are added to induce a catalytic reaction in the protection layer. Hence the new generation LSF Xfour has a near perfectly accurate characteristic curve, ❷.

## EARLY SENSOR READINESS DUE TO STRONG HEATER POWER

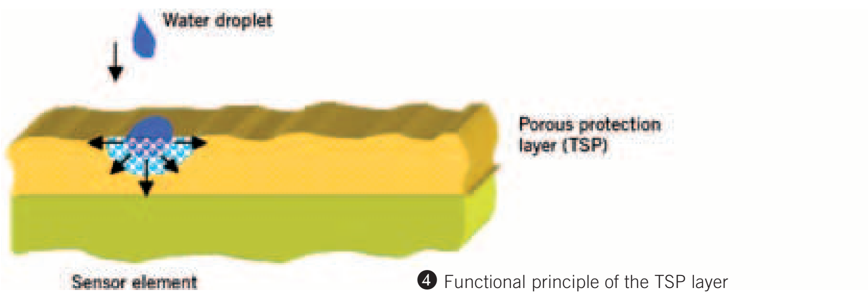
To achieve the earliest possible signal availability after key-on, the oxygen sensor needs a high-power heater to facilitate a very fast heat-up. The LSF Xfour heater is designed to heat up the measurement cell quickly and homogeneously. Heat transfer within the ceramic and heat exchange through radiative exchange with the environment was considered in the sensor design. The ceramic is designed to minimize mechanical and thermal stress during the heat-up process. The temperature distribution and the heat balance of the ceramic sensor element were simulated with real gas-flow conditions including the influence of the protection tube. Numerical simulation was implemented to find the best parameters for heater layout. Statistical methods were employed to calculate failure rates for stress loads.



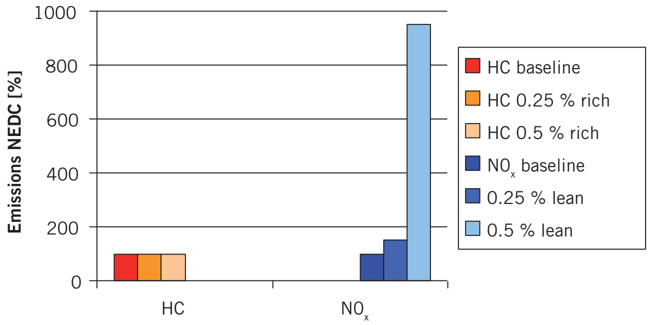

❸ Mechanical stress of the sensor depending on the heating power

## SENSOR RESILIENCE AGAINST WATER DROPLET IMPACT

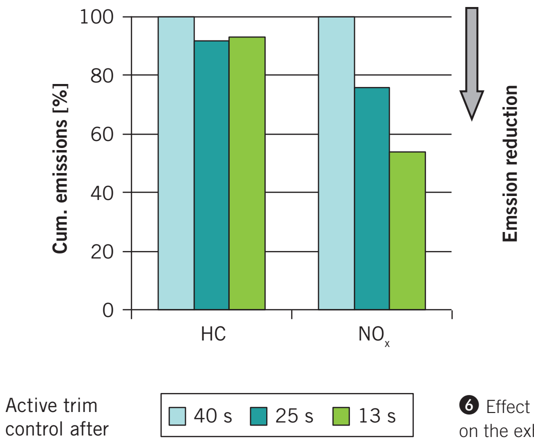
Fuel combustion generates water which can condense in the cold exhaust pipe during the start-up phase of the engine. When one of these water droplets hits the ceramic sensor element, it leads to local cooling of the surface at impact by 300 °C within just a few milliseconds. The resulting stresses can lead to mechanical failure of the ceramic sensor element. The car makers react by introducing a delay time to warm up the exhaust pipe. Bosch has optimized the metal protection tube of the oxygen sensor and has introduced a porous ceramic coating on the sensor element, which minimizes the consequences of water droplet impact. This coating is referred to a thermo shock protection or TSP. The impact of a water droplet depends on its size and the element surface temperature. To optimize the application of the sensor, Bosch has developed an application tool to measure the water droplet number and size in the exhaust gas. With this tool the water droplet distribution in the exhaust pipe during start-up can be measured over time. Thus, the optimal point-in-time to heat up the oxygen sensor can be determined. The combination of these technologies has let to a significant reduction in the delay time before the oxygen sensor can be turned on, ❹.




❹ Functional principle of the TSP layer




❺ Result of a shift in the characteristic curve on the exhaust gas




❻ Effect of delayed activation of trim control on the exhaust gas

## FLEXIBILITY CONCERNING THE MOUNTING POSITION

Another improvement is the reduction of length. Given this fact and higher temperature robustness more flexibility of the mounting position is the result.

## SUCCESSFUL RUNS TO PROVE REDUCED EMISSIONS

Improved accuracy and faster signal availability of the new generation LSF Xfour gives the potential to significant reduction of HC and $NO_x$ emissions. This has been proven in various test runs. Initially, a test was conducted with an application consisting of a wideband sensor (oxygen sensor with a linear characteristic over a wide range of oxygen concentration) upstream and a LSF Xfour sensor downstream of the catalytic converter. In this configuration, the $\lambda = 1$-point of the wideband sensor is corrected by the signal of the LSF Xfour (trim control). Hence, the LSF Xfour sensor determines the system's accuracy. A lambda shift of only 0.5 % towards “rich” raises HC emissions only slightly. The same shift towards “lean”, however, will increase the $NO_x$ emission load dramatically. Fluctuations of the exhaust gas mix into the lean or rich domains can be greatly reduced by using the new generation LSF Xfour oxygen sensor, ❺.

Early signal availability of LSF Xfour is also advantageous. The $NO_x$ emissions were reduced by half in this test when lambda trim control was running 13 s after key-turn-on (in comparison with a reference test with a delay time of 40 s), ❻.

The new generation LSF Xfour was also extensively tested mounted upstream of the catalytic converter. The greatly improved accuracy at $\lambda = 1$ is an advantage, because the composition of the exhaust gas mix can be controlled within a very narrow range between rich and lean conditions. Due to smaller amplitudes of lean-rich excursions, which need to be compensated by the reactions within the catalytic converter, the closed loop cycles are tighter and faster.

## OXYGEN SENSOR FOR THE FUTURE

In summary, by launching the new generation LSF Xfour, Bosch manufactures oxygen sensors with a number of significant improvements: The characteristic function is more accurate, the lambda signal is available faster after key-on, the closed loop cycles are accelerated and the excursions from $\lambda = 1$ are minimized. The robustness is enhanced by a new protection tube and a porous ceramic TSP coating. The small sensor size allows the car makers a large degree of flexibility when integrating the LSF Xfour sensor into their systems.

## OUTLOOK

Development of new oxygen sensors is continuing at Bosch to meet all future emission legislations and customer requirements. Bosch has equipped Diesel engines with oxygen sensors since 2002. The oxygen sensor LS-Diesel, scheduled for launch in 2013, is tailored specifically towards meeting the requirements of Diesel engines. These requirements are a very high lean range accuracy, good soot particle resilience, continuous signal availability at low gas temperatures and high flow rates and resilience against accumulation of engine oil and Biodiesel. Continuous signal availability at low exhaust gas temperatures and resilience against soot accumulation require a specific and sophisticated design trade-off. This has been implemented in a novel way in the new oxygen sensor LS-Diesel.

The LS-Diesel will feature the same compact design and a TSP coating similar to the design chosen for the LSF Xfour. The LS-Diesel will enable the customer to choose a preferred connector. Hence, the LS Diesel combines Diesel-specific innovations with various established design elements. This accords the greatest possible flexibility and ease of tests and assembly during manufacturing to both supplier and OEM.

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## INITIAL SITUATION

The reduction of fuel consumption, $CO_2$ emissions and low operating costs are burning issues for manufacturers and customers of commercial vehicles. Continuously enforced $CO_2$ regulations for heavy duty trucks will further increase the necessity to optimize fuel consumption of the entire power train of commercial vehicles worldwide. Since the long distance truck has the biggest impact on $CO_2$ savings in the field of highway transportation, it is obvious to concentrate on this segment for fuel reducing measures.

Optimization of the fuel consumption of commercial vehicles has always been a high priority for truck manufacturers as well as end users. Innovative technologies like waste heat recovery, hybridization, electrification, and similar concepts will be only available in a medium term or are currently not very attractive to customers due to high additional product costs. This begs the question which technologies can be used in the short term at reasonable costs to further increase the efficiency of the power train for long distance trucks?

## SHORT TERM SAVING POTENTIAL

Due to the long standing optimization of the internal combustion engines of commercial vehicles in terms of friction reduction and thermo-dynamic efficiency, the remaining improvement potential is quite small. But in the entire field of the auxiliary devices the situation looks quite different. An optimum layout, integration and control strategy of the entire auxiliary system, a measurable fuel saving can be quickly achieved relatively easily and cost effectively.

In this context, the expression “auxiliaries on-demand” gains a special importance; it expresses the strategy that auxiliary devices should deliver the required performance only when it is really needed. The primary focus is on the following components:

- multi-stage switchable or fully variable controllable fan
- multi-stage switchable or fully variable controllable coolant pump
- variable flow rate of the oil pump
- switchable oil jets for piston cooling
- detachable air compressor
- on-demand activated hydraulic power steering pump.

# OPTIMIZED STRATEGIC AUXILIARY OPERATION IN COMMERCIAL VEHICLES

Ancillary units are a considerable factor in terms of vehicle efficiency. AVL has analyzed the influence of optimized strategic auxiliary operation on the fuel economy of heavy duty trucks. As a result, fuel consumption and $CO_2$ emissions of long haul trucks could be reduced by approximately 3 to 4 % compared to current technological standards.



The interaction between the different engine subsystems and their increasing complexity demands for a consideration on a system or complete vehicle level. AVL List has developed a new methodology [1] that enables the evaluation of the global optimum for the estimated behavior of complete vehicles in a relatively short development time and with reasonable effort. Central building blocks of this development methodology are:

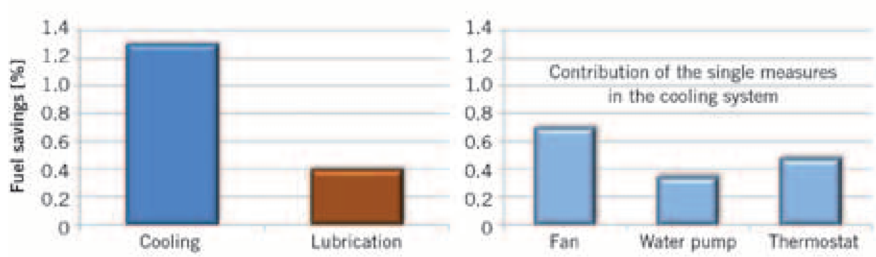
- modeling of the entire system on a vehicle level
- modeling of the IC engine and its central subsystems: combustion, cooling and lubrication system
- model adjustment on the base of vehicle and/or test bed measurements
- development of an optimum operating strategy to identify the maximum fuel saving potential
- writing of a software code for the engine control management on the base of a developed operating strategy
- validation of the evaluated saving potential on the test bed trials and in the vehicle.

Central part of this methodology is a simulation environment that features a modular structure and thus allows an adjustment to different requirements and development stages – from physical models for the concept evaluation and detailed system layout up to real time models (HiL, MiL) [3]. The corresponding model parameterizations can be transferred automatically ensuring consistency of the development process, an essential prerequisite for quality and efficiency [2].

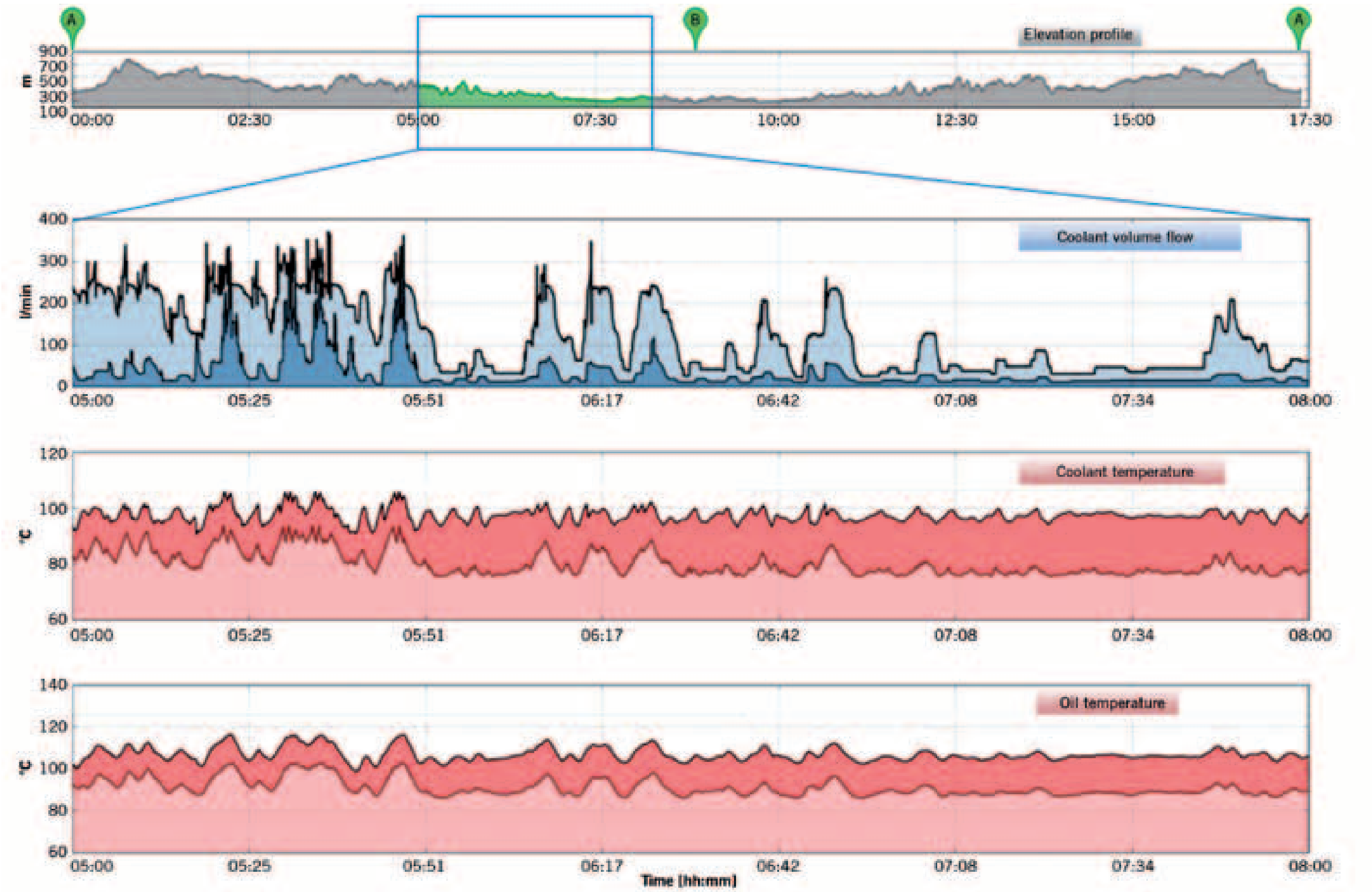
## OPTIMIZATION OF FUEL CONSUMPTION VIA THERMAL MANAGEMENT

The optimization of the cooling and lubrication circuit of the engine is focused on two main aspects: pump drive performance according to actual demand means – only as much coolant and oil should be pumped as is actually needed to stay within the individual component limits – and optimized thermal management for the entire engine. All over all fuel savings up to 2 % at average environmental conditions (e.g. 25 °C) can be achieved by the consequent optimization of the engine and vehicle thermal management, ❶.

In typical European long distance truck operation the engine is operated under part-load conditions most of the time. However, coolant flow and the drive for the coolant pump are designed for a sufficient cooling under “worst case” full load conditions. A significant fuel saving can be achieved for part-load operation by an on-demand controlled coolant pump allow-




❶ Fuel savings by thermal management measures [%]




❷ Influence of the thermal management on media temperatures during driving cycle

ing consequently the maximum increase of the coolant temperature. The same also applies to the lubrication system where oil jets should ensure sufficient cooling of the pistons under full-load conditions. Under part-load conditions these oil flows can be switched off and in parallel the performance of the pump drive can be significantly reduced.

The fan in a commercial vehicle is of special importance although it is activated for a relatively short time (10 to 20 %, depending on route) in a typical long distance cycle only. Due to the high fan power installed (about 30 kW) the fan should be activated only when all other measures cannot ensure sufficient cooling.

Fuel savings up to 0.8 % are possible if the fan drive is decoupled from the engine by using multi stage or actively controllable fully variable clutches. Thus small and medium cooling requirements can be accomplished without engaging the fan. As parasitic losses of a disengaged fan can be up to 0.2 % of the entire fuel consumption this status should be optimized or ideally set to zero.

Despite all optimization measures that will generally raise the level of the coolant and lubrication temperatures, a reserve is needed for the buffering of short term cooling requirements, ❷. This reserve may be necessary when, for example, the retarder brake is activated while driving downhill. A model based control algorithm must account for the relatively high system inertia due to the large coolant volume of a heavy duty truck.

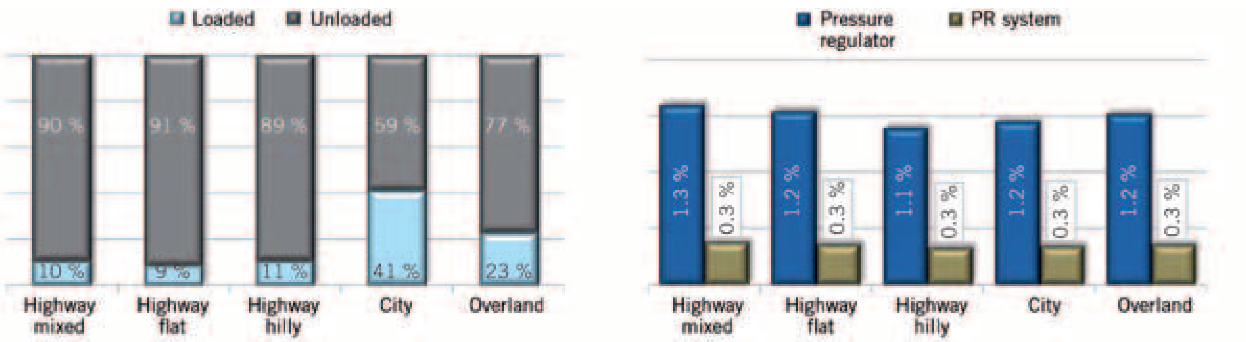
## AIR COMPRESSOR AND POWER STEERING PUMP OFFER A SIGNIFICANT POTENTIAL

At cruising conditions on highways both the power steering pump and air compressor are seldom activated or need to deliver their full power, ❸. Frequently the auxiliaries operate at a reduced level or at idling causing unnecessary but significant power losses.

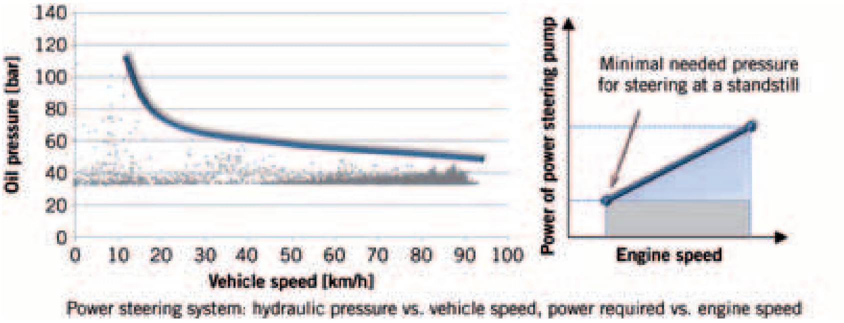
Conventional air compressors without power limitation during idling produce a savable power loss of up to 1.3 % of the entire fuel consumption during the entire drive cycle. Air compressors with power reduction functionality during idling, as used in modern long haul trucks, could still save up to 0.3 % of fuel by a complete decoupling of their drives.

In typical long haul cycles the full power of hydraulic power steering pump is only seldom required, ❹, since the need for steering support and the corresponding drive power significantly decreases for increasing vehicle speeds.

Under real driving conditions the direct coupling of the power steering pump to the engine speed causes significant losses, ④ (right), since the maximum steering power has to be available already at idle speed. For this reason it makes sense to design controllable steering aid systems




❸ Load spectrum and saving potential for the air compressor



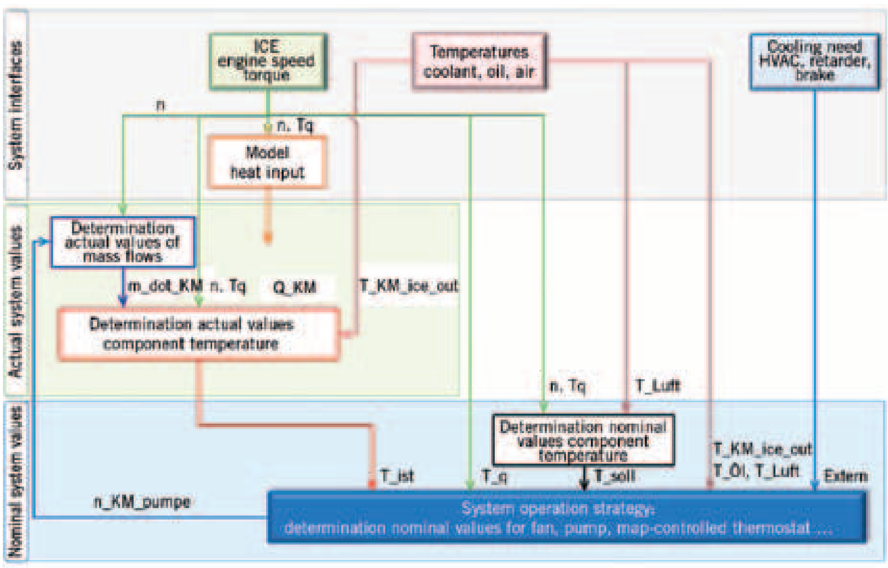

❹ Load spectrum/oil pressure of a power steering pump during a typical driving cycle

and to reduce the provided hydraulic pressures during higher engine and vehicle speeds. Implementation of such strategies can save up to 1 % of fuel over the entire drive cycle.

## CONTROL IS EVERYTHING

For a full exploitation of the entire fuel saving potential, it is necessary to approach system and component limits without exceeding them, to maintain reliability and durability. A model based approach mapping the entire system must include sufficiently accurate control of various actuators and must account for considerable system inertia, ❺.

In view of a global system simulation, AVL has developed functions for a model based system control. These functions allow the prediction of the system behavior based on actual load conditions with sufficient accuracy. It will reduce the needed safety margins of the individual components without influencing the reliability and durability under real driving conditions. The coupling of a model based control and a global system simulation allows the calibration of the previously developed control software with




❺ Control scheme for the thermal management operation strategy high degree of accuracy So that on the engine test bed or in the vehicle only minor adaptations or optimizations remain.

## SUMMARY

Optimization of auxiliary devices in combination with an optimized control strategy that accurately meets the demands has the potential to reduce the fuel consumption and $CO_2$ emissions of long haul trucks by approximately 3 to 4 % compared to current technological standards. For other commercial vehicles (delivery trucks, city busses, off-road applications etc.) [4] savings up to 5 % should be possible, depending on the share of part load operation.

How much of this potential can be realized in practical operation depends on how accurate the entire system is mapped in the engine control software and how close the component limits can be approached.

Saving potentials of this approach, the relative ease coupled with short term feasibility, and the low additional system costs are attractive for both manufactures as well as customers. When analyzing the total cost of ownership a return of the investment should be possible in a period of less than half a year.

## FUTURE PROSPECTS

In the future the control of auxiliary devices could be coupled with GPS systems or toll collection systems. Dependent on the approaching section of the route and additional parameters, like traffic, weather etc., auxiliaries could be pro-actively controlled. By such measures the coolant temperature could be temporarily reduce before the next hill is reached and an overheating of the system could be avoided even without starting the fan thus saving fuel. Similarly, just before the next downhill section is reached the system could lower the coolant temperature to have a sufficient buffer for the coming heat input of the retarder.

AVL List has demonstrated that fully electrified auxiliaries will offer further potential for optimization since parasitic loads could be finally eliminated. However, the cost development of the electrification of commercial vehicles has to be monitored. Fully electrified auxiliaries will reach a status in terms of reliability, costs and power density where the integration into long haul trucks will make sense. However, the major proportion of the identified potential of "auxiliaries on-demand" can be exploited even with mechanical or partly electrified solutions that are already available – if they are combined with an intelligent control strategy.

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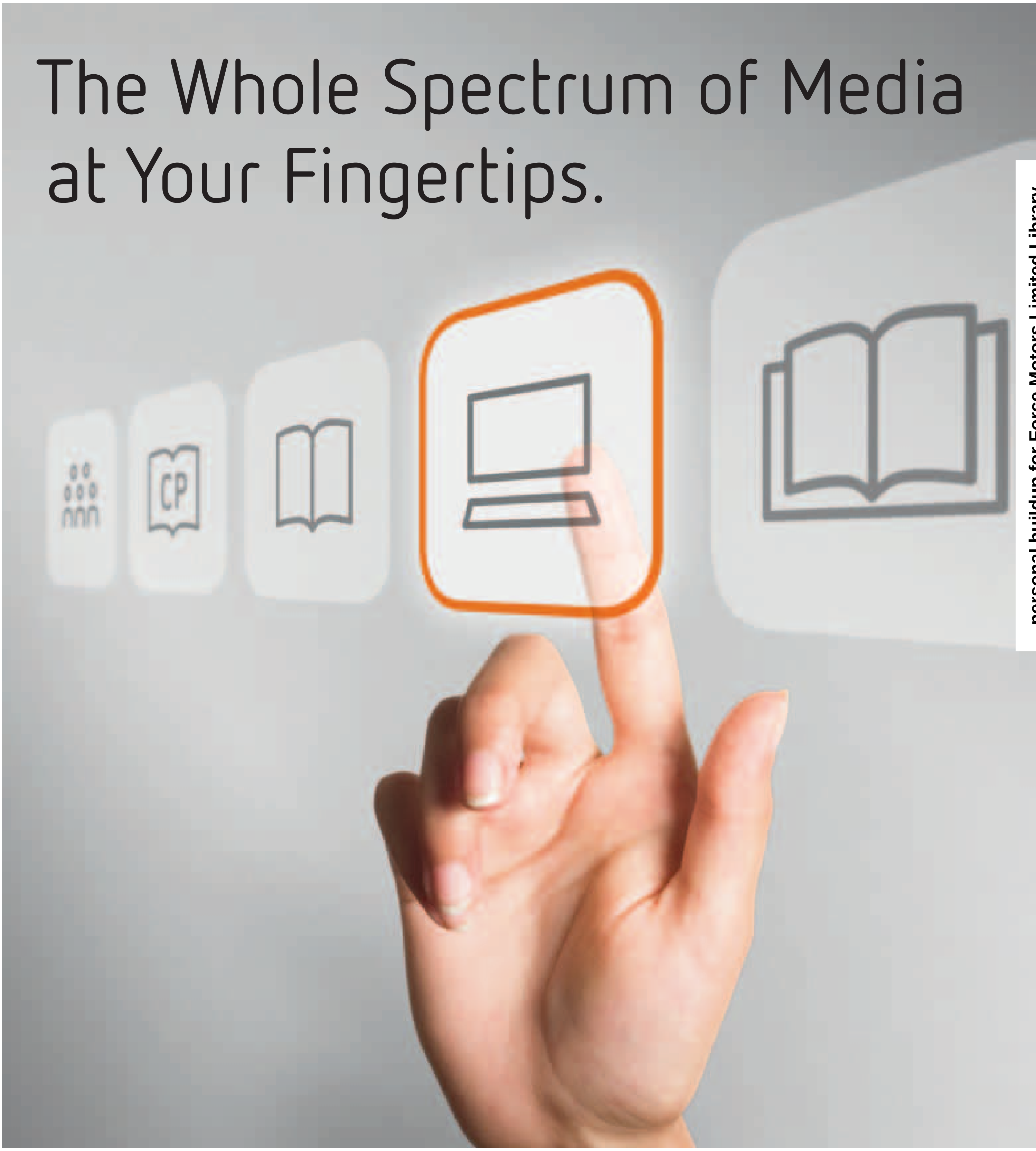
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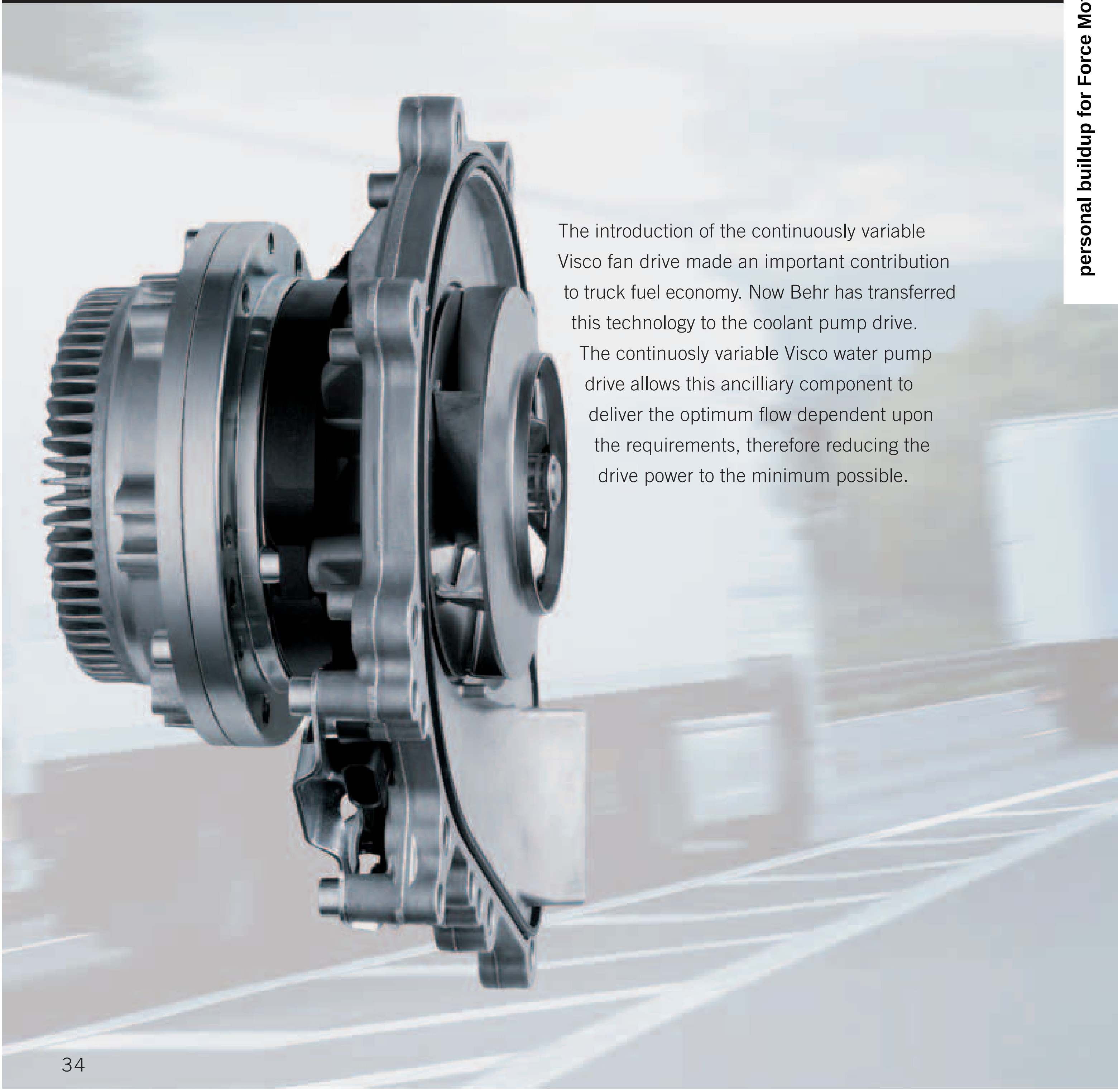
# VISCO COOLANT PUMP
## DEMAND-BASED FLOW RATE CONTROL





The introduction of the continuously variable Visco fan drive made an important contribution to truck fuel economy. Now Behr has transferred this technology to the coolant pump drive.

The continuosly variable Visco water pump drive allows this ancilliary component to deliver the optimum flow dependent upon the requirements, therefore reducing the drive power to the minimum possible.

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## REDUCING THE DRIVE POWER REQUIREMENT

Up to now water pumps world wide have almost always been powered by direct, uncontrolled drives. Consequently, the delivery and the power consumption have been directly related to the engine speed, which, in many conditions, results in an unecessarily high flow rate. The potential to reduce the flow rate thus the drive power is inherent within the Visco water pump.

With the introduction of each new engine emissions legislation level it has become more and more difficult to reduce the emissions and, at the same time, reduce or even maintain the engine fuel consumption. However, reducing the fuel consumption and, therefore, the $CO_2$ emissions is of primary importance: it could become in Europe the focus for legislation following the Euro VI emissions legislation. Additionally, the truck manufacturers represented by the ACEA have committed to a reduction of the fuel consumption of future trucks: 20 % per tonne.kilometre by 2020. These trends are motivating the investigation of all potential areas for improving.

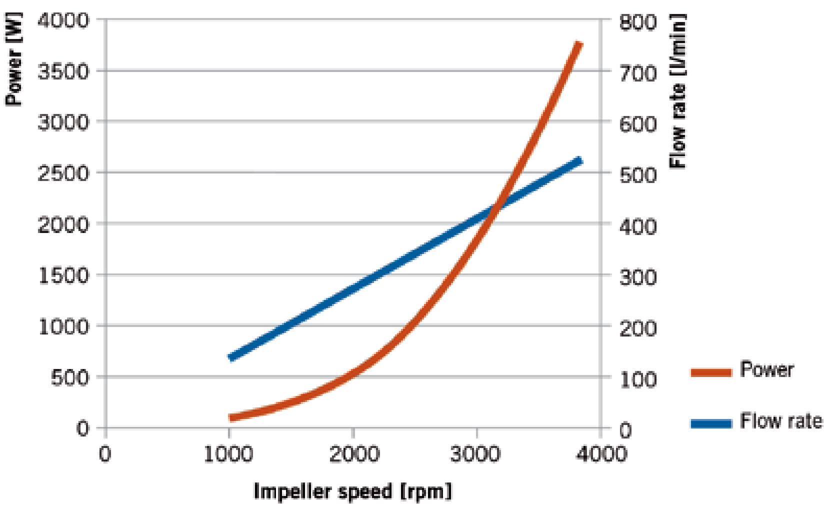
An important area for such measures to reduce consumption is that of the power required to drive the engine ancillaries. In the field of engine cooling, it has been possible to realize a significant benefit during the last decade by switching from bimetal to electronically controlled Visco fan drives. Since, owing to the ram air flow through the radiator, the fan engagement is required only for a relatively small proportion of the time, the biggest savings were realized by reducing the internal losses of the disengaged fan and by a precise and quick engagement of the fan at high coolant temperature levels. Here, only small additional potentials for optimization still exist in relation to the transition from Euro V to Euro VI.

The objective first and foremost is to optimize the efficiency of the individual components: notable advances in this respect have been made recently in the area of the coolant pump, through altering the shape of the impeller, and in particular by using covered impellers. As a result, hydraulic efficiencies of over 60 % are now perfectly feasible; an increase over and above this, however, will require an ever greater outlay.

A much greater potential for savings, however, has been identified in the intelligent demand-based regulation of ancillary components. Prior to the introduction of Euro IV, only rigid drive coolant pump concepts had been explored and these were belt-driven directly from the crankshaft. Since the nominal flow rate for the coolant pump is typically designed for the most critical operation that can be assumed, all operting points that differ greatly from the most crtical permit a reduction in flow rate. With regard to the engine block and cylinder head, the determining parameter for the coolant flow is primarily the prevention of local boiling, particularly when there is a low flow rate, this is most critical under heavy engine load, i.e. at or close to maximum torque. The situation is similar in systems with coolant cooled exhaust gas recirculation, where the considerable differences in temperature between the coolant and the exhaust mean that attention must be paid to the risk of boiling. Heat transfer in the radiator is, however, less strongly dependent on the coolant flow rate. Modern radiators are saturated on the coolant side even with moderate flow rate values, thus a reduction of the flow rate would be possible at many operating points.

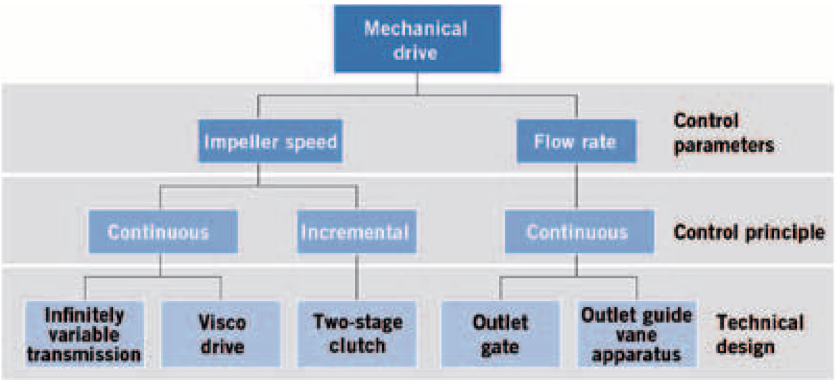
The coolant pump, which since it is a radial flow device, is a hydraulic machine just like the fan. When components such as these are operated, the flow rate increases more or less linearly with increasing engine speed; power consumption, however, increases to the third power, ❶,

❶ Flow rate and power consumption of truck water pump

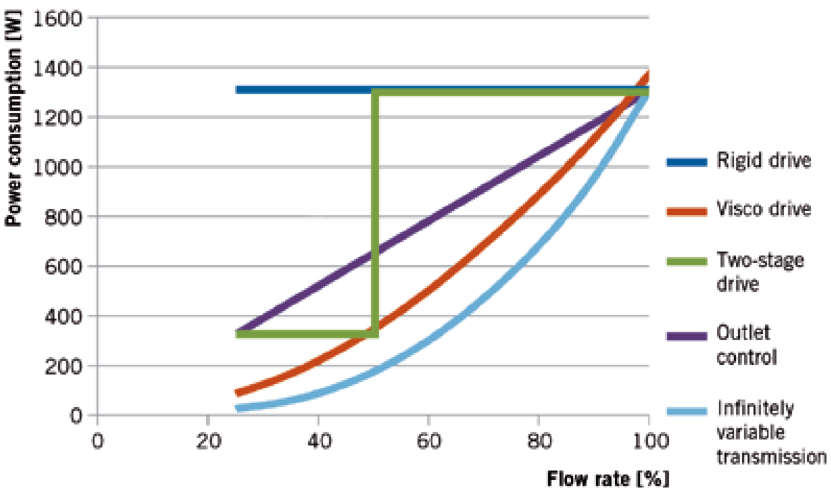


i.e. if the current operating point requires only half of the flow rate of the peak rating, the speed can be reduced to half of the maximum and the shaft output at the impeller is reduced to an eighth. The impeller speed is, therefore, a major consideration in reducing the drive power required.

When a pump design with variable flow rate is used, the maximum flow rate can be configured, more so than in a rigid design, to cover load peaks in individual components. As a consequence of reducing the flow rate in normal operation. Even if the variable system is designed to cover higher load peaks, the mean power consumption of the system may decrease significantly when compared with a rigid design that has no reserve for the higher load peaks.




❷ Mechanically driven concepts for truck water pumps




❸ Power balance comparison of the considered concepts at 1200 rpm engine speed

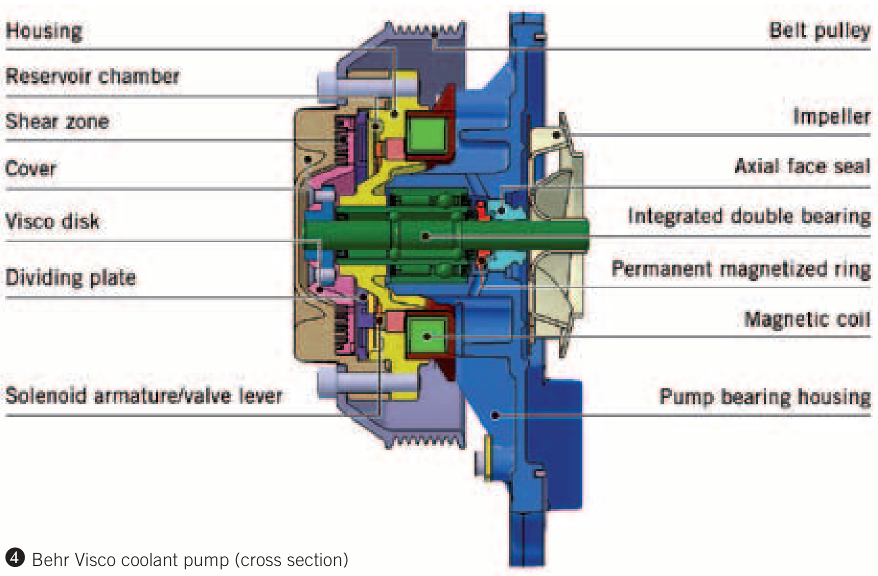
## SELECTION OF THE TECHNICAL CONCEPT

Since, in trucks, continuous pump drive inputs of up to and over 4 kW may be required during engine operation, purely electrical drive systems are out of the question for the introduction of Euro VI. The current electrical architecture of vehicles with 24 V on-board power supply does not permit the generation and transfer of power at this level. Accordingly, only mechanically driven concepts are considered here, ❷, but this includes concepts that directly control the flow rate by restricting or diverting the flow. In a second group of concepts the flow rate is regulated indirectly by way of the impeller speed; this group includes principles with continuous and incremental control: alongside the Visco principle, continuous approaches include an infinitely variable transmission unit. The established two-stage principles use an electromagnetic switchable clutch to activate the direct fixed drive; with the clutch open, the impeller is driven via an eddy current, which reduces the impeller speed by a preconfigured ratio via slip.

A comparison of the theoretical potential savings of the various concepts, ❸, based on a frequently used engine speed in heavy trucks (1200 rpm) identifies, on the one hand, the rigid drive system as the basis and, on the other, the infinitely variable transmission as the theoretical ideal, when 100 % efficiency and an unlimited setting range is assumed. The Visco drive comes closest to this theoretical ideal, the differences to the infinitely variable transmission unit result from the slippage. Indeed, there are no further efficiency related losses in the Visco drive system, since there are no non-moving parts or further drag losses in the torque path. In concepts with control via the outlet geometry, the drive torque at best reduces linearly with the volume flow rate, such that if the volume flow rate is reduced to half the drive torque is also reduced by half. With the two-stage concept, downshifting to the reduced flow rate is possible only if the required flow rate corresponds to less than the maximum flow rate with the lower stage. If the flow rate requirement of the engine increases again, the maximum engagement becomes necessary once more. Rapid shifting between part and maximum flow rate is not desirable, since this would lead to a sharp increase in the alternating pressure load on the components of the cooling circuit. A slow shifting speed should likewise be avoided, since, owing to low thermal inertias, localized boiling could occur in seconds at specific locations in the coolant circuit. The two-stage concept is therefore only directly comparable with the Visco concept as regards the design points of maximum and reduced flow rate.

## TECHNICAL IMPLEMENTATION OF THE CONCEPT

The Visco coolant pump from Behr, ❹, can, in principle, be housed in the packaging space of a rigid pump. No modifications are necessary on the coolant side and the control unit can be spatially integrated in the belt pulley. Since the impeller and belt pulley speeds differ, an advanced bearing system is required. The Behr design uses an integrated double bearing, with the two bearing surfaces arranged

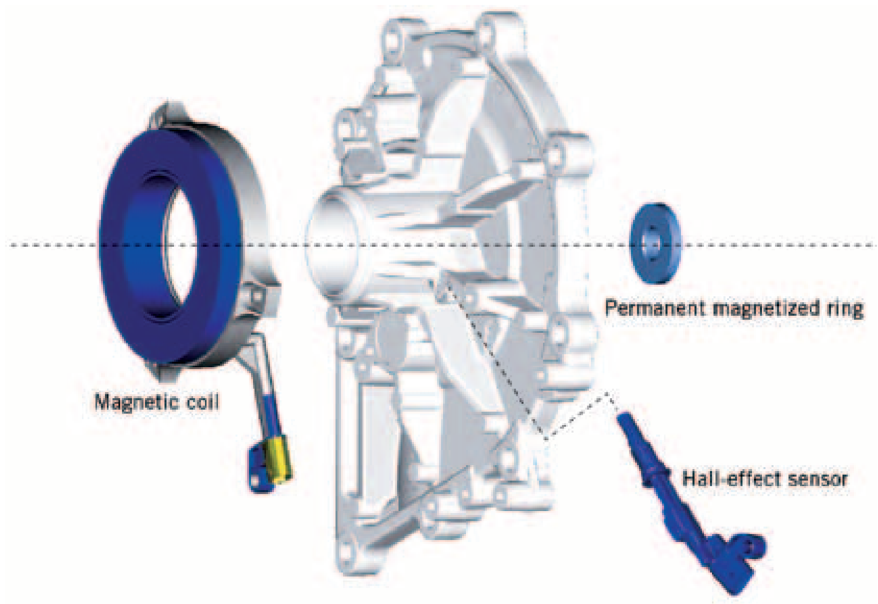



❹ Behr Visco coolant pump (cross section)

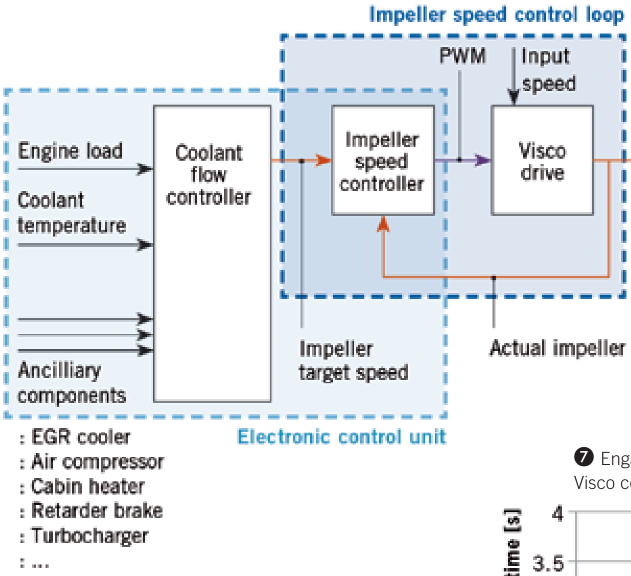
coaxially. The central shaft supports the impeller at one end and the Visco disk, which is driven by shear forces, at the other. The middle ring of the bearing provides the outer ring for the inner bearing surface and the inner ring for the outer bearing surface. This middle ring supports the housing of the Visco drive unit, via which the forces from the belt pulley are channelled. The outer ring of the bearing is pressed into the bearing housing of the pump, and, as such, only the outer bearing surface is responsible for transferring forces from the belt drive. This integrated bearing design facilitates the optimal utilization of the installation space.

The torque for the impeller drive is transferred via the shear zone of the drive. A two-chamber principle can be followed to facilitate the control of the level of silicone oil in the shear zone, thus the degree of engagement. An oil reservoir provides a second chamber alongside the shear zone and the exchange of oil between the two chambers can be regulated using a dual-acting solenoid valve. The solenoid valve is actuated by a magnetic coil attached permanently to the bearing housing of the pump, ❺. This transfers the magnetic field across a radial air gap to the control unit housing, on which the terminal profiles for the ring-shaped solenoid armature on the side facing the solenoid valve are positioned; the zone between the poles of the base structure is magnetically insulated. When current is applied to the coil, the solenoid armature is drawn against its self-canceling spring load and the silicone oil is delivered into the oil reservoir; the impeller rotates with a minimal degree of engagement, approximately 25 %. When the current is removed, the solenoid valve assumes the opposing position because of the return spring; the silicone oil is delivered into the shear chamber and the maximum degree of engagement of the impeller of at least 95 % is achieved. This also ensures fail-safe behaviour in the event of an electrical failure: the impeller rotates with the maximum possible degree of engagement. The degree of engagement can be continuously varied between the minimum and maximum values through cyclical switching of the magnetic coil and pulse-width modulation (PWM), typically with a frequency of 2 to 4 Hz.

The impeller speed is regulated in a closed control circuit, ❻, and a Hall-effect sensor for speed monitoring is installed in the pump bearing housing for this purpose. This sensor records the impulses of a continuously magnetized ring attached behind a floating ring shaft seal to the

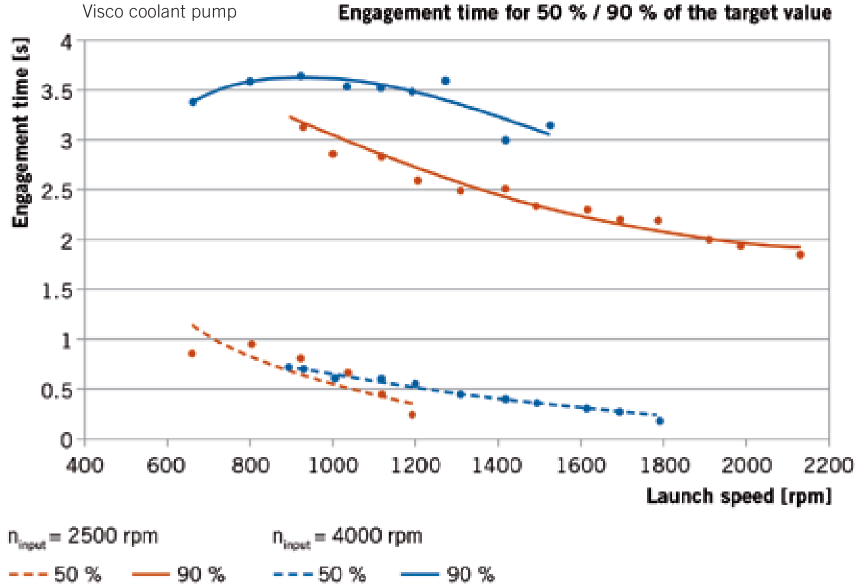



❺ Electrical components of the Visco coolant pump




❻ Control configuration of an engine cooling concept with Visco coolant pump

❼ Engagement times of the Visco coolant pump




impeller shaft. The actual recorded impeller speed is compared to the target speed in the control unit's speed regulating module; the PWM set variable for the control unit is modified according to the direction and amount of deviation in order to minimize these parameters.

The recording of the actual impeller speed is also an important parameter for maintaining the operational reliability of the engine; if the pump speed is too low or the pump is stationary, the engine's output must be restricted and corresponding warnings must be communicated to the driver. The impeller target speed is established by a higher-level control unit and is subject to various engine and vehicle specific parameters. The chief parameter in this respect is the engine load, to which the heat release is related. Other such parameters include the coolant temperature and the differences in coolant temperature between specific locations. A decreasing coolant volume flow rate leads to an increasing temperature difference across the heat transfer path. Furthermore, various secondary components may require a specific coolant flow rate, which is determined in a higher level controller; these include the exhaust gas recirculation cooler (EGR cooler), air compressor for the brake system and the retarder brake system.

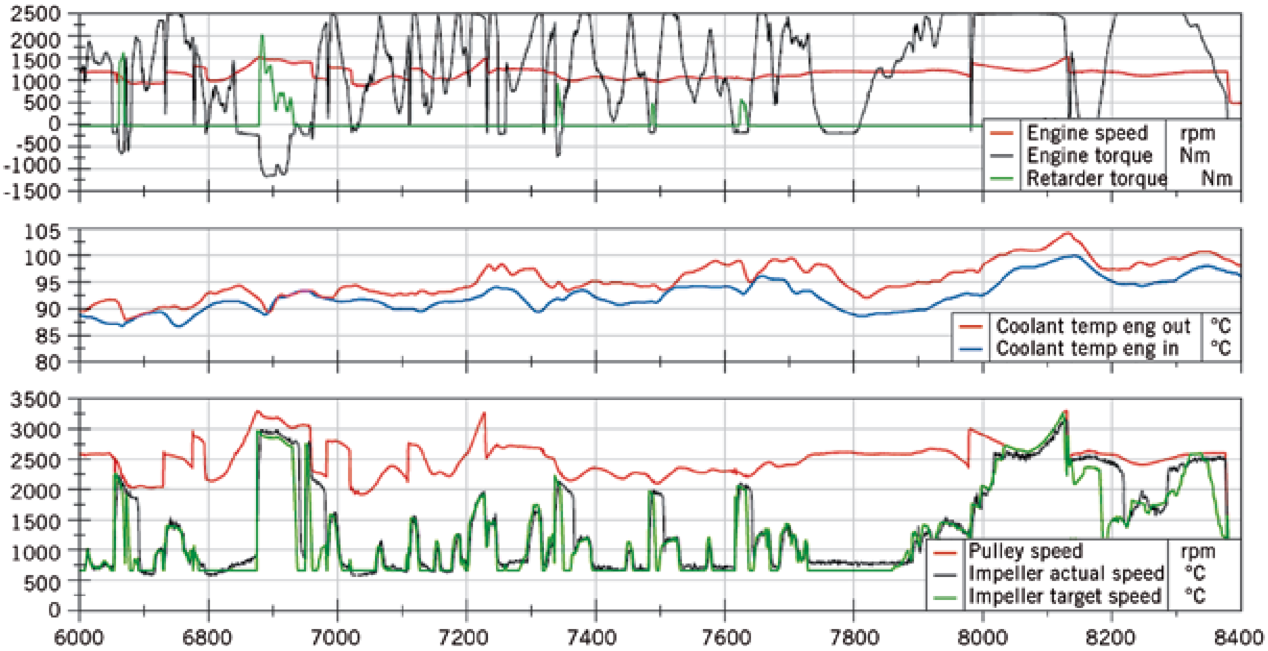
## CONTROL DYNAMICS

The engine and vehicle specific requirements for coolant volume flow rate have different dynamic properties. The release of heat in the engine or from the exhaust gas recirculation cooler at maximum load results in the maximum rate of coolant temperature increase in specific areas. To prevent localized boiling, the coolant flow rate must be quickly increased in such situations. Studies have demonstrated the engagement properties of the Visco coolant pump to be sufficient in all cases in such circumstances. The engagement times, ❼, are dependent on the engine speed and the drive output speed at the beginning of the engine acceleration, with 50 % of the end value reached after no longer than 1 s, and 90 % after 3.5 s.

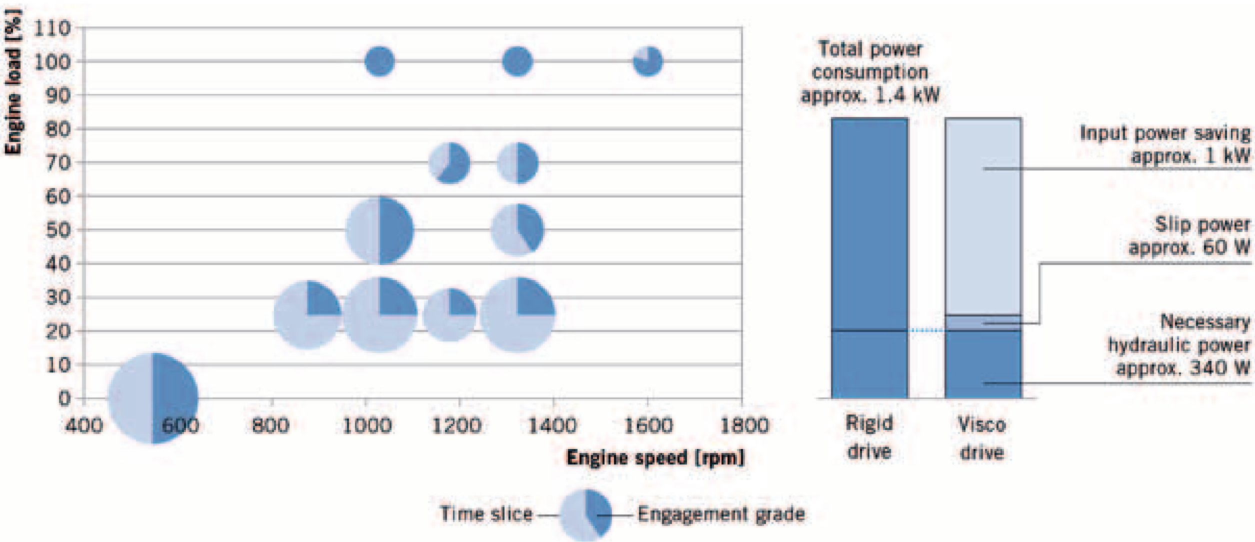
Measurements in vehicle, ❽, demonstrate the good tracking of the actual impeller speed to the target speed. It can be seen that there is only a slight deviation between the actual and target impeller speed for various degrees of engagement. Notable deviations only arise following a full engagement, owing to the hysteresis in the Visco drive unit. The minimum engagement of the pump is often also sufficient during typical engine transients around the mean loading, this results in the high potential fuel saving.

## FUEL SAVINGS

Assuming an engine speed spectrum normal in long-distance heavy-duty trucks and a conservative distribution of the load dependent engagement of the coolant pump, a reduction of mean drive power from 1.4 to 0.4 kW can be calculated, ❾. Trucks with a gross vehicle weight of 40 t require an engine output of 100 kW to overcome driving resist-




❽ Measurements in vehicle with the Visco coolant pump




❾ Calculation of the fuel saving potential: load profile according WHSC; time slice of each duty point according circle area; engagement grade of the Visco drive at each duty point according circle graduation

ance, primarily air drag and rolling resistance, at a speed of 80 km/h on the flat. Accordingly, reducing the pump drive power by 1 kW at 100 kW engine output results in a corresponding fuel saving of 1 %. This value is highly dependent on the usage profile of the vehicle, the long haul application represents the best opportunity to reduce the speed of the water pump. In more challenging conditions, for example driving in hilly terrain with long stretches under full engine load and with braking, pump must be operated more often at full load. However, the potential saving when compared to a traditional directly driven pump may still be around 0.5 %. These values have been substantiated through vehicle field tests.

## SUMMARY

The potential fuel saving of the Visco coolant pump in heavy-duty trucks is impressive. The extremely robust Visco transmissions concept functions wear-free: the torque is transferred via hydraulics. Regulation in accordance with the fail-safe principle and the use of a speed sensor ensure high operational reliability. Integration in the existing installation space is problem free. All validation and field trials have been completed successfully, and the first series production application began in the second half of 2011.




# FUEL PUMP WITH BRUSHLESS MOTOR FOR HIGH VIBRATION APPLICATIONS

Electrifying components in commercial vehicles can be a challenge. High vibration levels, for instance, amplify carbon brush and commutator erosion in a conventional electric motor. To offer a solution Federal-Mogul has developed the brushless fuel pump. This BLDC pump has successfully passed 10,000 h of operation in B100 fuel.

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## BOUNDARY CONDITIONS IN COMMERCIAL VEHICLES

Commercial vehicles are known to put component technologies to the ultimate test. Rigorous requirements such as one million miles and ten years lifetime will reveal each and every downside of a new technology. Electrification, though a successful trend in the automotive industry in general, provides an example. Originally the fuel pumps in on-highway or off-highway commercial vehicles were driven off the engine itself. As electrically operated pumps are more efficient, brushed direct current (DC) fuel pumps have replaced many of the engine-driven fuel pumps in order to reduce the direct load on the engine and the overall energy consumption.

However, medium and heavy duty diesel engines for on-road or off-road vehicles tend to produce a high level of vibration. Together with differing qualities of diesel fuel (sulfur, dirt, ester origins, lubricity) this can reduce the lifetime of the fuel pump. The combination of aggressive fuel and vibration is particularly likely to cause brush/commutator erosion. This can reduce the pump lifetime from a traditional minimum requirement of 6000 h to as little as several hundred hours in certain applications.

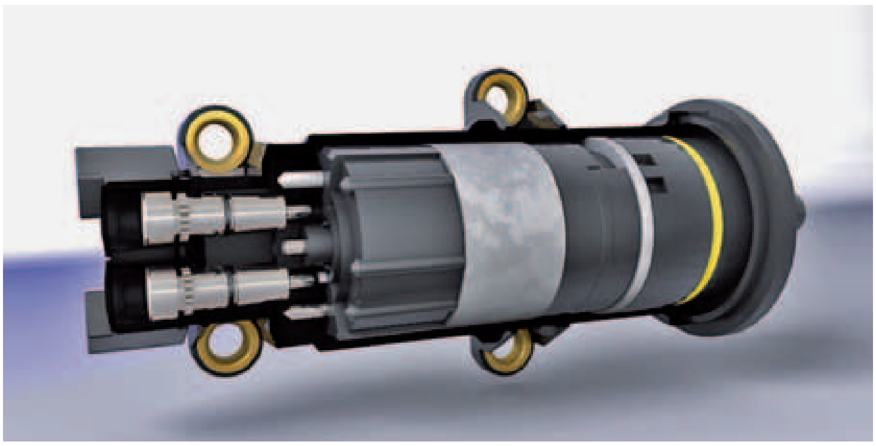
In other words brushed DC fuel pumps are not automatically lifetime components on heavy duty diesel engines. Federal-Mogul therefore developed the roller vane style fuel pump which is driven by a brushless DC motor (BLDC), ❶. The new pump has well over 10,000 h of lifetime in 100 % U.S. biodiesel and a much higher efficiency than brushed versions. Currently the new pumps are delivered to a leading international engine manufacturer and to Tier 2 suppliers where they go into fuel delivery modules. This paper explains the approach to pump design and the principle of electrical operation.

## DESIGN CHALLENGE

Brushes in a DC motor are always subject to wear. The durability of sliding contacts depends on their design, the material, their size, and the application's boundary conditions. Some factors have been identified by Federal-Mogul during intense testing to accelerate contact erosion:

High vibration levels can cause the carbon brush to lose contact with the copper or carbon commutator surface. This momentary separation can cause two arcs: The first occurs when the contact is lost, the second when it the contact is re-established. If the energy content of the arc is sufficient, it will result in local surface damage.

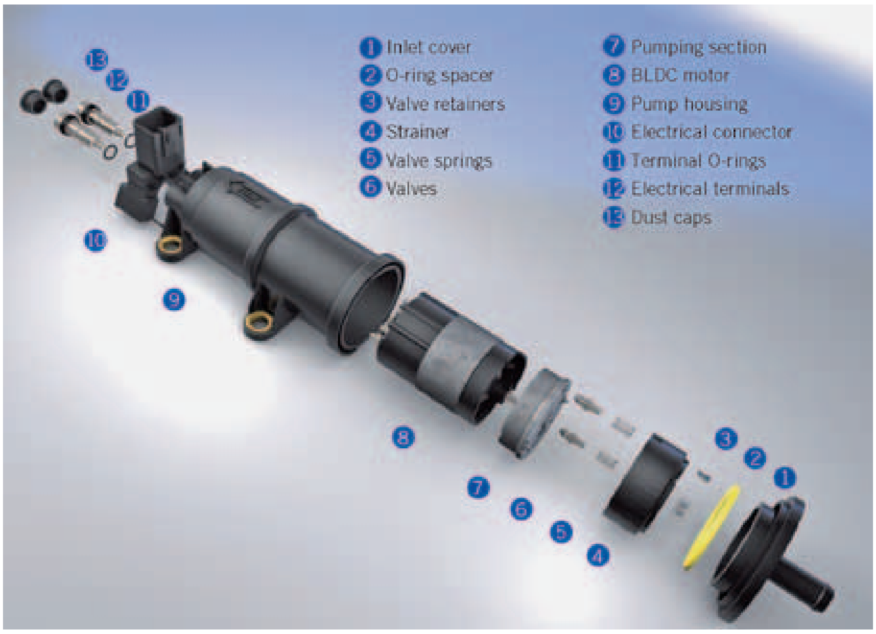
Aggressive fuel types can actually create two forms of erosion on the motor brushes and commutator: One type of erosion is mechanical in nature due to dirt, contamination or aggressive chemicals such as sulfur that reduces the lubricity found naturally in the fuel. The contaminated fuel or lower lubricity fuel can result in accelerated wear of the sliding partners. US/NATO JP military fuels are one example of a lower lubricity fuel. Another form of erosion is electrical in nature. Aggressive fuel types with high conductivity can further accelerate electrical erosion by amplifying the intensity of the arc between the brushes and the commutator.



❶ Cross section of the new brushless pump; contact-free commutation prolongs the component's service life, improved magnetic materials increase the efficiency

Although a 12 V automotive system that is not exposed to high vibration levels will not normally result in a lot of arching damage, similar 24 V produce arching with higher energy intensity which can result in greater damage due to the increased voltage being applied.

In many commercial vehicle applications some or all of the above conditions apply. As the actual “load collective” is hard to predict, it is just as difficult to predict the level of erosion which is amplified by vibration. Even the traditional requirement of 6000 h minimum engine runtime is difficult to fulfill with a brushed DC motor. In the meantime the requirement has become tougher and is often 10,000 h of pump life.



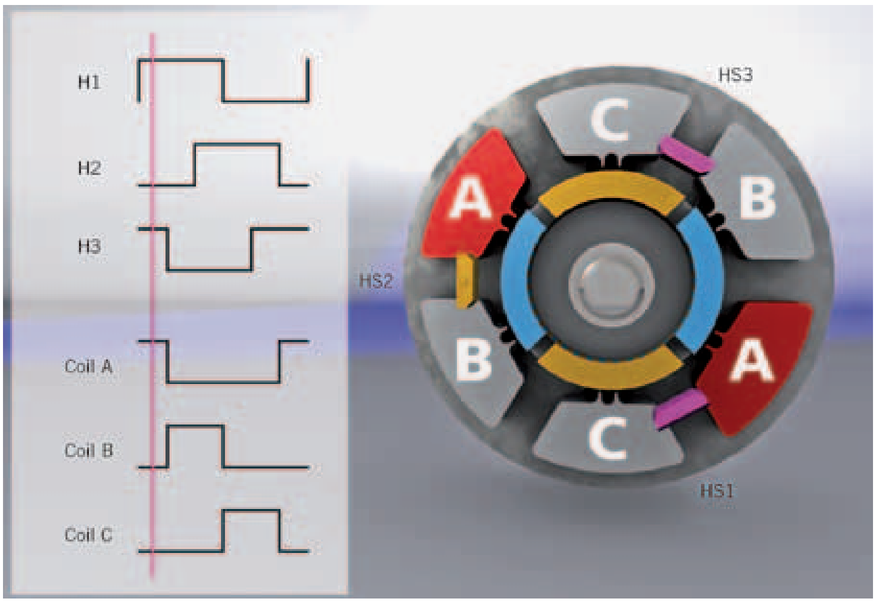

❷ Principle design of the pump

### PUMP DESIGN AND CONTROL ELECTRONICS

The pump dimensions were defined to cover an application range of diesel engines with 4.4 up to 15.2 l of displacement. The engines within this span typically deliver an output ranging from 56 to 470 kW of power. The applicable emissions regulations are either Tier IV Interim (Stage IIIB) or Tier IV Final (Stage IV).

For practical reasons one of the development targets concerned packaging: The new pump technology had to fit in the installation space of existing pumps. This was seen as a prerequisite to facilitate a drop-in replacement of brushed pumps wherever pump failure requires this. As BLDC motors need electronic commutation, the control circuit board and electronics also had to fit in roughly the same package size. In order to keep the added cost low, it was an early decision to focus on a simplistic electronic design that also has the required reliability even if the pump is installed directly on the engine. As a consequence to these requirements the new brushless fuel pump functions without micro controller. Three BLDC technology patents were filed during development. They cover key aspects of the motor itself, the manufacturing method, and the assembly process.

❷ shows the internal design of the in-line pump version. The control circuit board is self-contained within the BLDC motor. Three hall sensors on the bottom of the PCB (facing the motor) detect the poles of the four permanent magnets of the rotor. If a hall sensor detects the presence of a pole underneath, it will activate one of the four field effect transistors (FETs) on the top of the control circuit. A voltage is applied to the FET gate which permits the flow of a high current from the FET source to the FET drain and on to the stator coil opposite the rotor magnet. This current generates an electric field in the stator coil which is repelled by the rotor’s magnetic field. This effect causes the rotor to spin. As the moving pole travels forward and comes underneath the next hall sensor in the direction of spin, the pole proximity will be detected and the process repeated. ❸ shows the hall sensor signal pattern and the resulting current flow to the coils. By using the high-grade material neodymium for the rotor’s permanent magnets, the pump’s pressure and flow capabilities are 30 % higher when compared to conventional pump designs with brushed motor.




❸ Hall sensor signal pattern and resulting current flow to the stator coils

The new pump has fuel-lift capability (priming) and can be packaged into many orientations, ranging from in-tank, to in-line, and to filter assembly integration. However, the BLDC fuel pump can also be placed on the engine itself where it is exposed to the highest level of vibration. The rotor shaft is supported by plain bearings made with a phenolic resin which was specifically developed for this type of fuel-contact application.

## OPTIONS FOR ENGINE APPLICATION

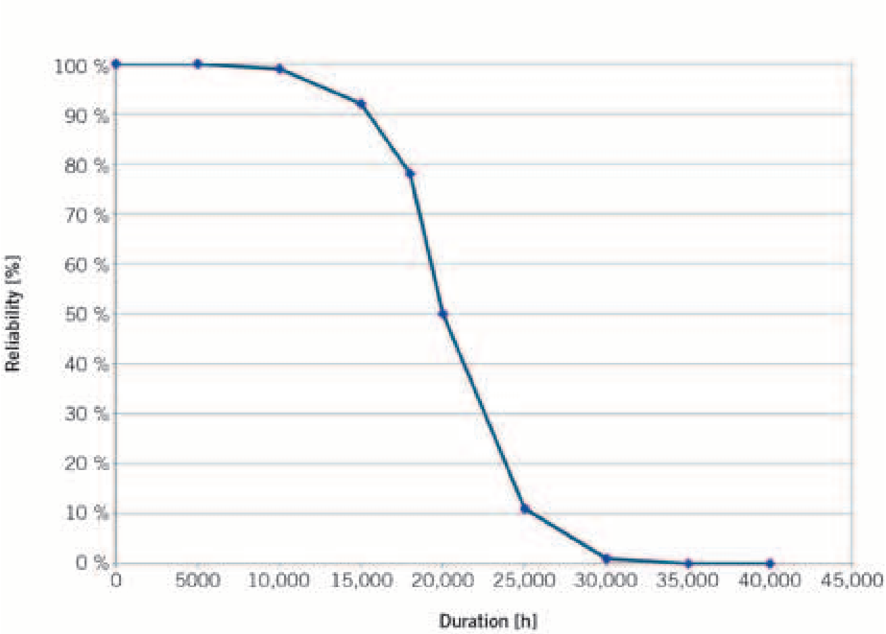
During operation the pump spins at a constant speed and delivers a constant flow of diesel fuel to the high-pressure pump that feeds the injectors. A return line downstream of the high-pressure pump feeds the excess fuel back to the tank. The amount of fuel flow is adjusted during pump application depending on the maximum engine requirement. The parameters that are adjusted during the application process include the motor, the valve assembly, the pump section and the level of friction.

The winding inside the BLDC motor itself can be changed to adjust the speed at which the motor runs. By assuring that the maximum power is available at the most efficient speed range, the motor draws only a low current. Also there are two alternative pump sections with different volumetric capacity which differ in overall height by 3 mm. By adjusting the spring loads within the valve assembly, the pressure can be defined at which the valve cracks open to let fuel flow back: Depending on the spring load this will either happen at low pressure or at high pressure. This regulates flow and pressure.

Federal-Mogul has developed a software tool which is used to specify the above parameters and many more in order to define the pump performance to meet customer requirements far ahead of manufacturing the first pump sample.

## DURABILITY TEST RESULTS IN DIFFERENT DIESEL FUELS

The new pumps have been validated on many levels. The development program included thermal environment testing, electrical component validation, and fatigue durability testing. In addition to over 10,000 h of lab testing, BLDC fuel pumps have proven their durability in diesel engine applications since 2008. A fleet of test vehicles has completed over 1 million on-highway miles, including fleet operation in the USA and Canada. However the most important single result is the pump life in aggressive fuel grades. Therefore unit operation durability was tested with diverse fuel qualities:




❹ BLDC pump wear-out reliability tested within a 12 V application

In ultra low sulfur diesel (ULSD; ≤15 ppm of sulfur) the pump units worked smoothly for over 15,000 h. In pure European biodiesel (EN 14214) the pumps continued to function after 10,000 h of operation. Over 10,000 h of operation were also achieved for 100 % North American biodiesel (B 100 fuel, Soy Methyl Ester – SME). Engine manufacturers usually specify no more than 20 % of biodiesel content in diesel fuel for their durability requirements.With JP fuels (JP 8) the pumps surpassed over 2500 h of operating time, which exactly meets the requirements for this extremely tough type of military fuel. Western Canada diesel fuel is known as a potential reason for fuel pump failure on commercial vehicle diesel engines because the aggressive fuel can attack the brushes. The new BLDC pumps operated for over 2000 h with this fuel grade. With Japanese Toyu fuel (a kind of kerosene fuel) the pumps also achieved over 2000 h of operation.

The test hours listed above represent the duration of the fuel durability test based on customer requirements and when parts were removed for teardown evaluation. The hours do not indicate product failure.

❹ gives an example of the wear-out failure reliability for a 12 V application with 100 % European biodiesel (EN 14214). As no contact separation and arcing occur with the brushless design, no significant difference could be seen between the test results for 12 V and for 24 V.

## CONCLUSION

The trend to electrify ancillary components is very likely to continue in the commercial vehicle field as well. However, as vibration dictates the life of a brushed DC motor, electrifying fuel pumps, for instance, require a particularly rugged design. The new brushless pump provides a durable and reliable option to electrify the fuel pump. The principle of electronic commutation that avoids wear in general also eliminates amplified wear caused by contact separation and arcing. Therefore the brushless pumps have a much greater durability on medium to heavy duty diesel engines with high levels of vibration. Owed to the electronic commutation the pump's durability is the same for 12 V and 24 V applications. To make the new pump available to existing diesel engine applications, where problems originate from the high levels of vibration, a simplistic, compact, and very robust design was developed.

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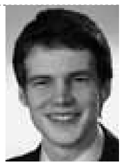

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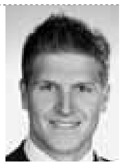

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# DEVELOPMENT AND OPTIMIZATION OF HYBRID POWERTRAINS AT THE X-IN-THE-LOOP ENGINE TESTBED

The development of hybrid powertrains is a challenging task for vehicle manufacturers. On the one hand there are requirements imposed by legislation and customers leading to different and in many cases concurrent development targets. On the other hand development engineers face a wide extent of degrees of freedom, which determine the vehicle's behavior and characteristics. An important tool to master the complexity of the development task consists in the integration of real drive units in the simulation. The TU Darmstadt has worked out a concept for this integration and is investigating the possibilities it presents for system tuning and the use of optimization methods.

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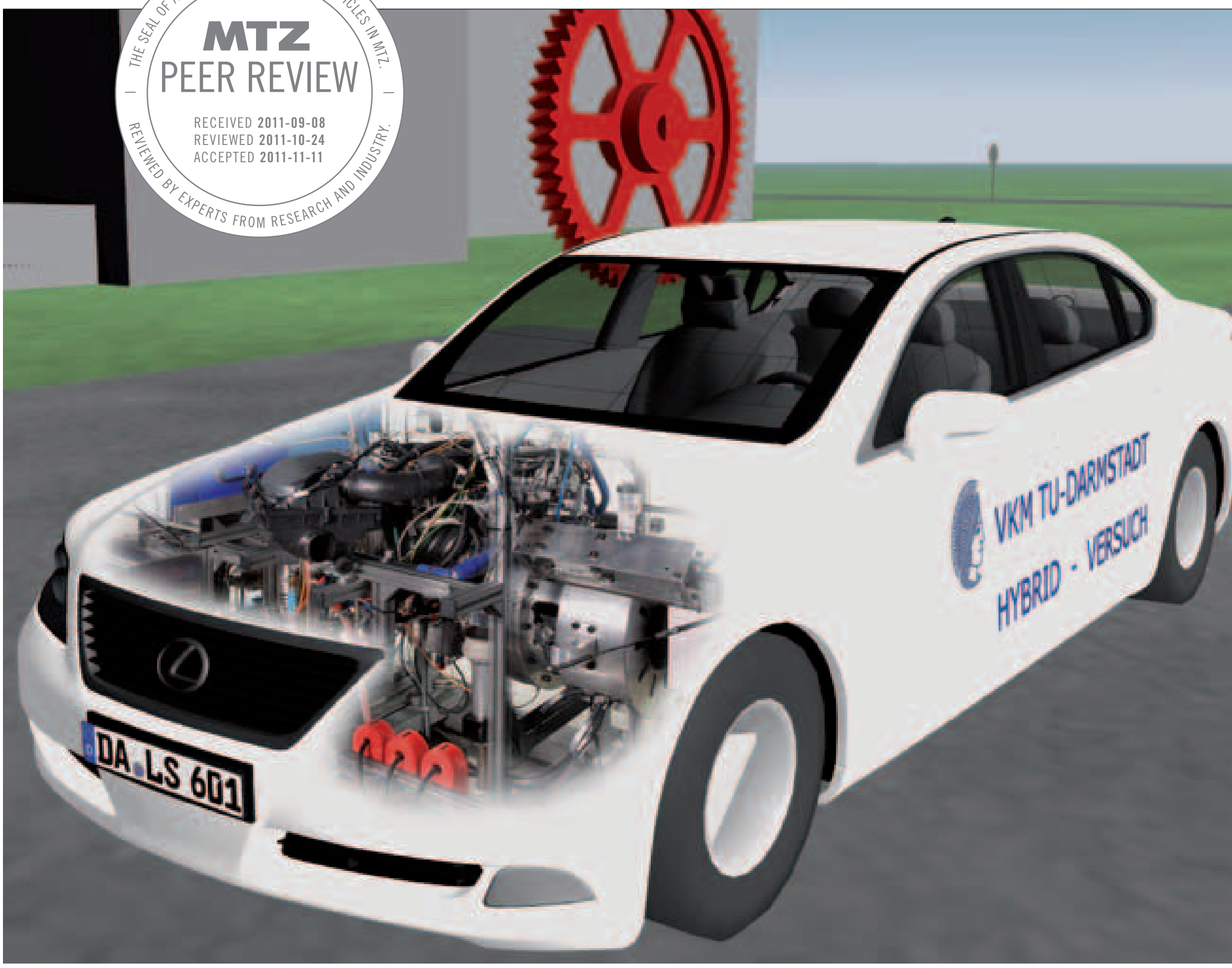


## 1 MOTIVATION

The targets for hybrid vehicle development are derived from the demand for high vehicle efficiency, durability and drivability as well as low pollutant emissions and vehicle costs. The scope of possibilities for system tuning ranges from the components' dimensions and adjustments to their mechanics and application to functions and parameters of the hybrid operating strategy.

The partially adverse effect of these measures on the different development targets can be illustrated by an example: In a vehicle with a power split or a parallel powertrain configuration low driving power demand can be supplied permanently by the internal combustion engine (ICE) or alternately and in different combinations by the ICE and the electric motor (EM). Using the ICE exclusively it is set to run mainly in low load points and consequently generally with low efficiency. Sharp increases in the power demand can only be met with limited dynamics and may lead to increased emissions, depending on the type of ICE. However, installing sufficiently dimensioned electric components und using them with a corresponding strategy the powertrain's response to load steps can be improved and emissions due to transient ICE operation can be reduced. These improvements in drivability and emissions are accompanied by increased costs or strain of the electric components, which may result in reduced durability. Furthermore, the high usage of electric components can result in reduced efficiency. If the necessary electric energy has to be provided by charging the battery with the ICE, additional effort will be generated due to losses in the EM and the battery. To which extend this additional effort can be compensated by a higher ICE efficiency (load point shifting) depends strongly on the driving profile, which is mainly influenced by the driver input.

The interaction between the different possibilities of intervention and the development targets and the thereby ensuing complexity of the system tuning is aggravated in the example given before when pure electric driving is permitted. More electric power is needed and the problems related with turning off the ICE – the cooling down of the exhaust gas system and elevated input of oxygen to a three-way catalytic converter – require adequate measures [1]. These concerns and additionally drivability requirements have to be met in the design and application of all powertrain systems.

Due to the complexity of the development task the possibilities of purely analytical approaches are limited. Driving simulations present the flexibility needed to test numerous variants. However, in the advanced phases of the development process one cannot rely on pure simulation, as current simulation model quality does not allow an accurate evaluation of the quality of a powertrain configuration, especially when considering the development targets emissions and drivability. It is recommended that, for the advanced development of drive units as well as the function development and application of the hybrid operating strategy, the drive units are set up on an engine testbed and integrated into an in-the-loop vehicle simulation (XiL), which operates in realtime. This integration of the real components into the vehicle simulation should be applicable as easily as possible and usable in current testbed systems.

## 2 INTEGRATION OF REAL DRIVE UNITS INTO THE VEHICLE SIMULATION

Generally, there are two ways to combine simulation and test on an engine testbed. On the one hand, pure simulation can be used to create load profiles and provide these profiles as input for the testbed controllers, on the other hand simulations can be set up to run in realtime at a testbed. The first method implies that the simulated drive unit's response to the input variable and its dynamics correspond to the real system with its current application, for example a ICE. The testbed dynamometer can set the speed according to a calculated speed profile and the load can be applied in an open loop control, for example setting the load input for an ICE. If no precautions like a precontrol logic are taken, a closed loop torque control to set the engine load leads to deviations between the actual and the set load points, as the two controllers for speed and torque are applied to control systems with different response times [2].

As the model accuracy necessary for the open loop control of a ICE cannot be obtained with acceptable effort, and possible changes to the unit under test should be investigated as well, the Institute for Internal Combustion Engines and Powertrain Systems, together with AVL, worked out a concept to upgrade existing engine testbeds and to equip them with a realtime vehicle simulation [3, 4]. The idea to use vehicle simulation at a testbed had been realized before, as testbed automation systems facilitate running certification cycles with a vehicle simulation. Yet the need to test hybrid powertrains brings about new requirements, especially due to the start-stop operation and the demand for high flexibility of the simulation models and flexibly customizable test scenarios with a simulated environment. Furthermore, the realtime simulation should be able to interact with enhanced units under test and their control units [5].

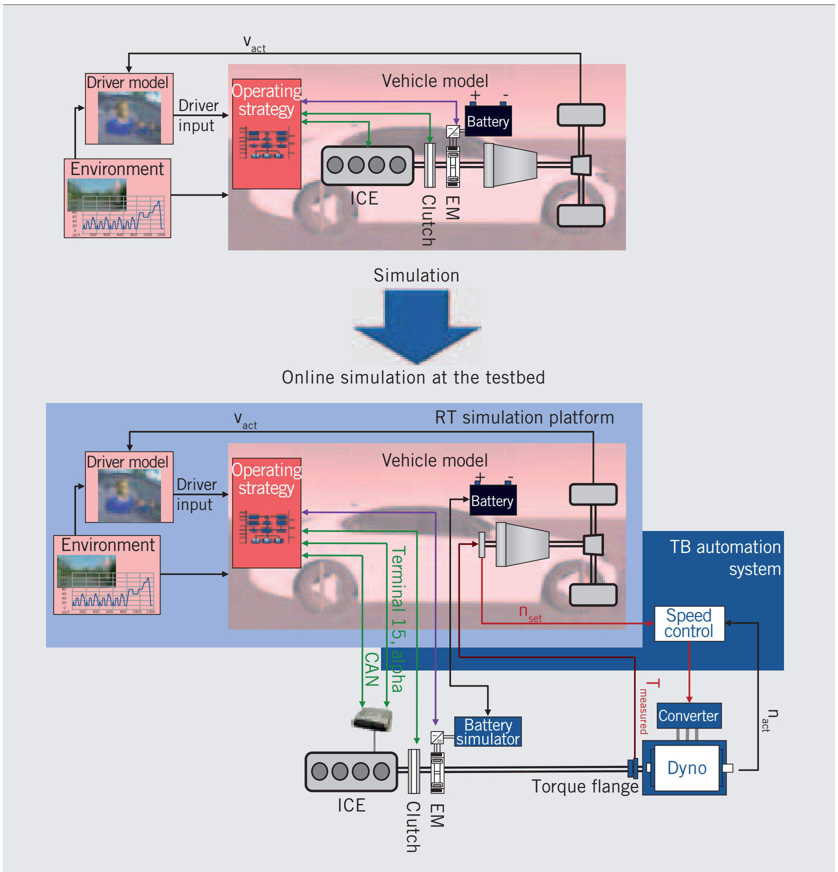
❶ presents the integration of a vehicle simulation at the XiL engine testbed at the Institute for Internal Combustion Engines and Powertrain Systems. The unit under test is a ICE to which an electric motor can be mounted, connected to the ICE by a clutch with low stiffness. The EM is powered by a battery simulator, a DC voltage source with parametrizable internal resistance, and a controller, developed at the institute, which allows varying the torque and power regime. Hence the testbed set-up enables virtual vehicle tests with a real ICE (engine in-the-loop – EiL) or a real EM as well as the integration of a P2-hybrid powerpack into a vehicle simulation.

The simulation runs on a realtime system, which is connected to the testbed automation system via CAN. The operating strategy (located on a Hybrid Control Unit – HCU), which is implemented as a part of the corresponding vehicle model, provides the input for the drive unit installed on the testbed. Partly these signals are passed on via additional signal lines, partly they are led through the automation system, thereby facilitating the implementation of security functions.

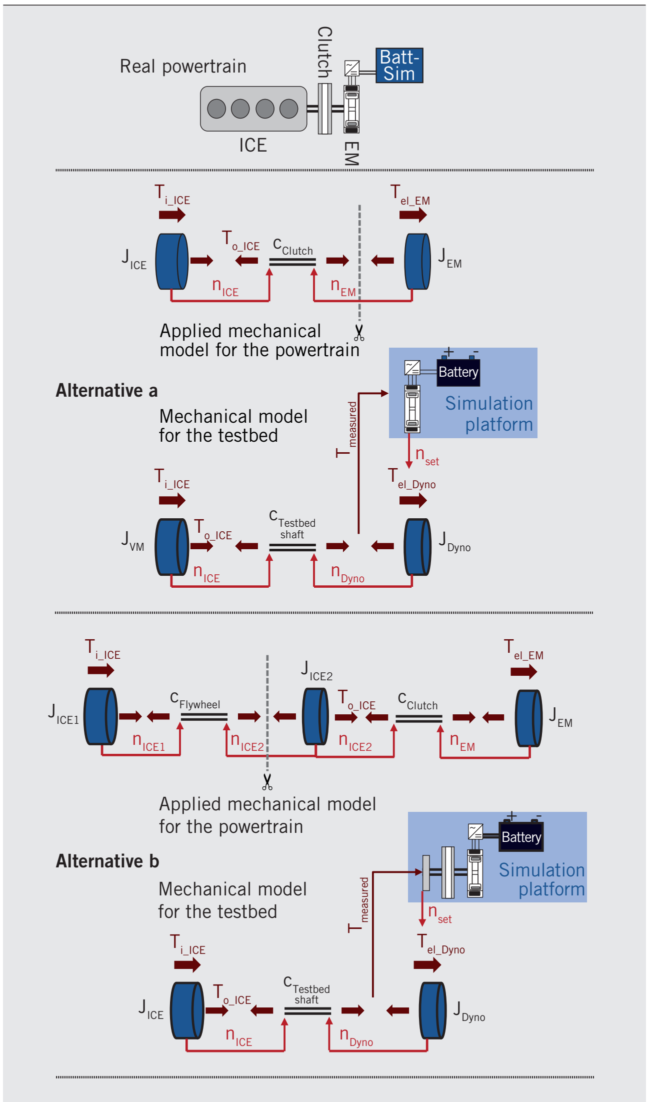
The installed powertrain components are mechanically coupled to the virtual part of the powertrain via the testbed shaft and the dynamometer (Dyno). The mechanical output of the unit under test has to be submitted at any time to the torque and speed that are present in a real vehicle with the current output torque. Due to technical principles deviations occur at an engine testbed: From the measurement of one quantity (torque or speed) the other quantity has to be calculated in the vehicle model and has to be regulated at the testbed with the dyno control. Setting this second quantity is delayed due to dead times in signal transmission and limitations in the dynamics of the testbed control. Ways to reduce these delays interfere with the attempt to keep the testbed set-up flexible and simple and facilitate the integration of the XiL method to existing testbeds. At the XiL testbed of the Institute for Internal Combustion Engines and Powertrain Systems the set values for the testbed are transmitted via the CAN connection, whereas the measurements for speed and torque can be sent directly to the realtime system.

The limitations of the dynamics of the testbed control are results of the mechanical configuration of the testbed. The unit under test and the dyno form a system susceptible to oscillations, which are mainly excited by the pulsating torque output of an ICE. To avoid constant operation in resonance the eigenfrequency is placed below the engine's idle speed, for a four-cylinder engine in a range from 10 to 15 Hz. Hence, the input to the dyno control is delayed when showing up on the unit under test, and torque or speed signals of the unit under test cannot directly be used as input signals to the dyno control. Compensatory approaches like a dyno control on an estimated unit under test torque require low signal dead times and are therefore difficult to reconcile with the target of a simple and flexible coupling of the realtime system to the testbed. For these reasons the concept presented here opts for a speed controlled dyno and for measuring the torque with a torque flange at the testbed shaft and providing it to the vehicle simulation. Accordingly, the interface to the testbed in the simulated powertrain has to be an inertia at the output side of the corresponding drive unit. The speed of this inertia has to be transmitted to the testbed. The drive unit installed on the testbed and the testbed shaft replace the corresponding drive unit and the nearby shaft element in the simulation model. The deviations to a real powertrain and which quantities can be investigated accurately in XiL tests will be evaluated in the following.

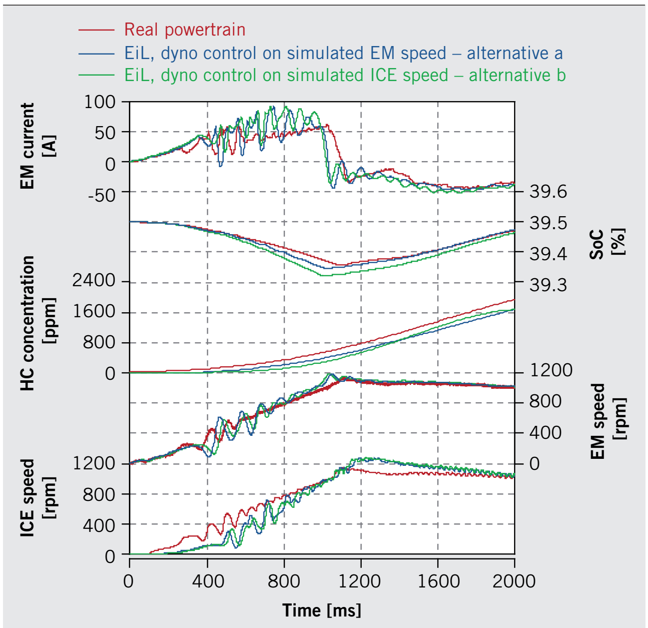
The reference is a testbed set-up with a real ICE and a real EM, only coupled with each other by a clutch. With this set-up one can represent the charging of the battery in serial hybrids. The battery is represented by the battery simulator and the corresponding battery model. In the following this testbed set-up is the reference for the EiL testbed. ❷ presents the mechanical model of the reference – the real powertrain – and how it is implemented in the EiL concept. In the alternative a the input side of the electric motor is the interface between the simulated and the real components. Consequently, the testbed shaft replaces the real clutch and its




❶ Implementation of the vehicle simulation at the XiL engine testbed at the Institute for Internal Combustion Engines and Powertrain Systems




❷ Possible interfaces between the simulation and the testbed for the implementation of EiL in the contemplated example




❸ Comparison of the engine start for real powertrain and EiL simulation



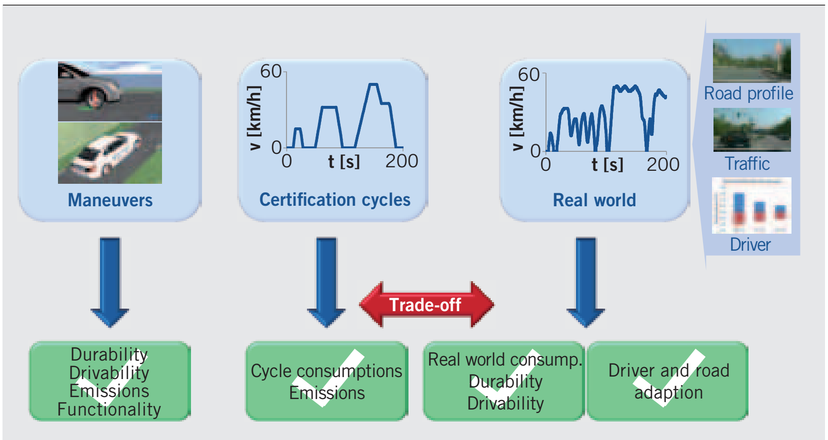
mechanical behavior resembling the dynamics with different damping. If, however, the inertia of the ICE is used as the interface between simulation and real component (alternative b), an additional inertia for the speed calculation has to be added to the system, which changes the vibration characteristics of the mechanical system.

For the comparison of these two alternatives the EM drags the ICE to 1000 rpm and is afterwards used as a generator to load the battery. Representing this kind of engine start with an EiL set-up is challenging because of the unsteady ICE torque exciting resonance vibrations in the critical speed range. The specific application of the ICE for this testrun enables the injection for an engine speed higher than 900 rpm. The torque demand for the ICE is 50 Nm. Prior to this, it was stopped in a defined manner, so that there is residual gas in its exhaust gas system.

❸ shows deviations in the speed and current curves, which can be related to the differences in the mechanical set-ups and to the different damping values and to a higher total inertia respectively. Besides, the deviations can be explained by the delayed setting of the speed values on the dyno. Above all the speed overshoot is evident. Consequently, for a short time the real ICE converts more power than calculated in the corresponding real time simulation, as this calculation is based on the set speed rather than the actual speed. However, in relation to the energy conversion in an average engine runtime these deviations as well as the differences in the power demand for the engine start are small (< 1 kJ) and negligible. In the emission measurements one cannot identify an effect provoked by the different engine start-ups. So one can conclude that, if the development targets energy conversion and emissions are considered, an analysis on an XiL testbed with both presented interfaces to the simulation is feasible. Nevertheless, the differences in the speed curves between the completely real components and the EiL set-up postulate that for a drivability examination or detailed power contemplation it is necessary to install the adjacent components on the testbed or to use a highly dynamic testbed concept, as described in [2, 6].

## 3 SIMULATION ENVIRONMENT AND TEST SCENARIOS

It is necessary for the simulation environment for XiL tests to run in realtime and be configurable flexibly with powertrain models and hardware I/Os. Further requirements result from the test scenarios. Typically, powertrain systems are tested with speed profiles for longitudinal dynamics, like certification cycles. Furthermore, the functionality of the complex interaction between the operating strategy and the real drive units should be tested and applications for the corresponding control units and the strategy should be found. Therefore it is necessary to be able to flexibly set realistic as well as extreme input values to the vehicle. Moreover, the operating strategy should be fed with the same input signals as in a real vehicle. Hence the simulation environment must be able to calculate in three dimensions so that maneuvers as electric break-

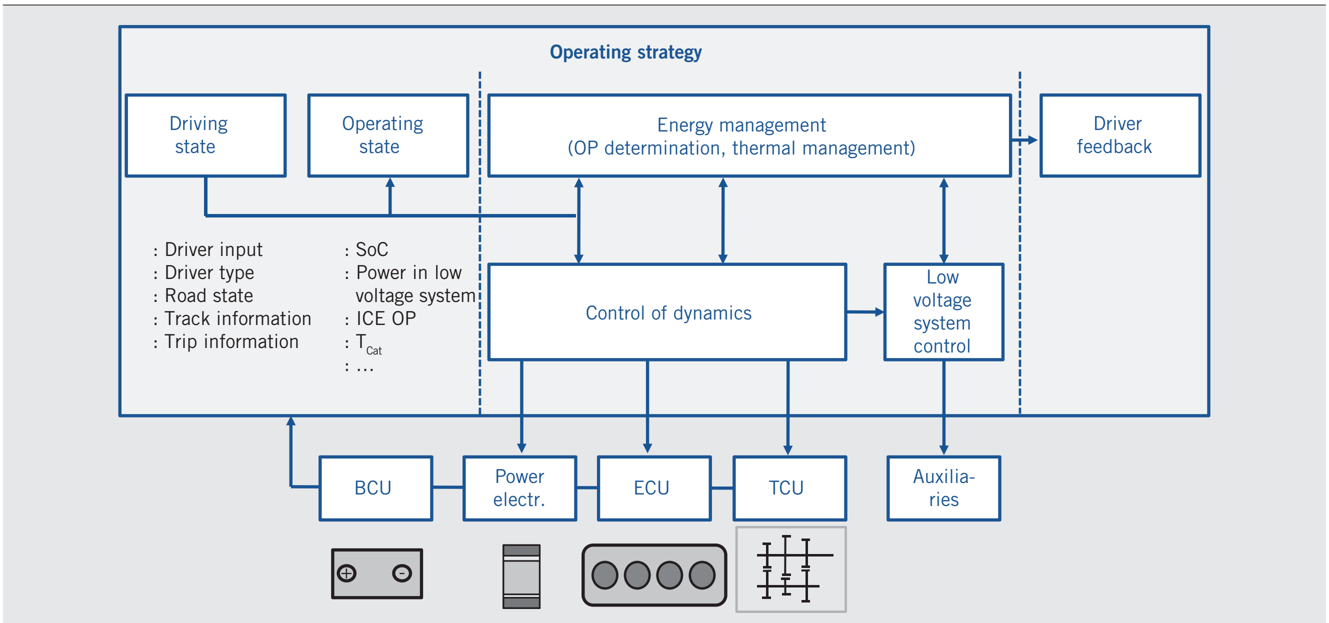



❹ Overview over tests scenarios and development targets

ing during cornering are represented correctly. For the testbed presented here the test environment AVL InMotion powered by IPG CarMaker was chosen. It meets the requirements mentioned before and can be used consistently for pure simulation as well as for simulation at powertrain or chassis dynamometer testbeds. With this environment a set of testruns, which represent real vehicle tests on reference tracks in and around Darmstadt, were created, taking into account the traffic obstacles and giving the opportunity to choose between different types of drivers [2].

With a simulation environment of that kind it is possible to address not only cycle results, but also realistic testrun results as development targets. One can also test operating strategy functions like a driver or road adaption and quantify their effect with measurements at the drive unit, ❹. Cycle tests can be used primarily for the first functionality tests and the determination of global strategy parameters, which refer to the runtime of the drive units and the power demand for them. For these tests it is sensible to use an EiL testbed set-up, mainly in order to record the emissions correctly. The application of the dynamic interaction between the different drive units can be realized most efficiently with specific maneuvers which represent a desired driving situation. A testbed set-up with several real drive units increases the quality of the test results. In order to be able to facilitate parallel development for the different development tasks it is proposed to structure the operating strategy modularly and divide it mainly into a global (Energy management) and a dynamic part (Control of dynamics) [7], ❺.

❻ presents a successful functionality test for the P2 powertrain with the powerpack testbed set-up on the TU Darmstadt reference city cycle. The ICE is started by closing the clutch when the strategy identifies a high power demand. In the next steps of the development the interaction of the different drive units can be improved using specific maneuvers in order to reduce the drop of the powerpack output torque at the engine start.




❺ Structure for an hybrid operating strategy

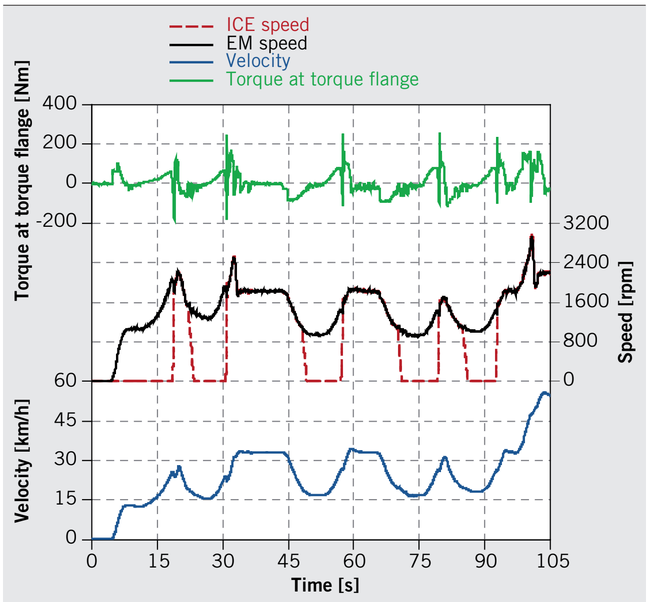
## 4 APPLICATION SCENARIOS

Problems like the not optimized start of the ICE in hybrid vehicle operation reveal the potential of the XiL concept, as adaptions to the drive unit can be tested together with the operating strategy in an early phase of the development process. In addition to the optimization of the operation in quasi-static load points like the application of an Atkinson cycle [8], one adjusts the start-stop application of the engine, as shown for a power split powertrain in the following. In ❼ the two different start applications are presented, one focusing on high comfort (hybrid start initiated by the vehicle system), the other one focusing on fast torque build-up (driver-initiated hybrid start) [9]. In the upper part of ⑦ the operating strategy initiates a comfort orientated engine start because of a low battery SoC. By evacuating the intake manifold, enabling the injection at high engine speed and retarding the ignition angle, the torque build-up is smoothed and therefore has little influence on the vehicle speed. In the lower part of ⑦ the operating strategy starts up the engine as fast as possible because of a driver input demanding high power. Injection is enabled at 600 rpm, the throttle is opened the ignition angle is advanced. Consequently, the first combustion cycles present high peak pressures and the engine can contribute immediately to the vehicle acceleration. However, these settings provoke higher vibrations in the engine mounts ($a_z$). Such tasks directed at the tuning between operating strategy and engine application can be realized at the XiL testbed even before the engine control unit (ECU) is equipped with an adapted interface, due to the use of the standardized CAN calibration protocol (CCP) for the communication between the realtime system and the control unit [10]. Alternatively, a rapid control prototyping to bypass the ECU can be used or the communication can be realized with the protocol ilinkRT in combination with an ETK control unit [11].

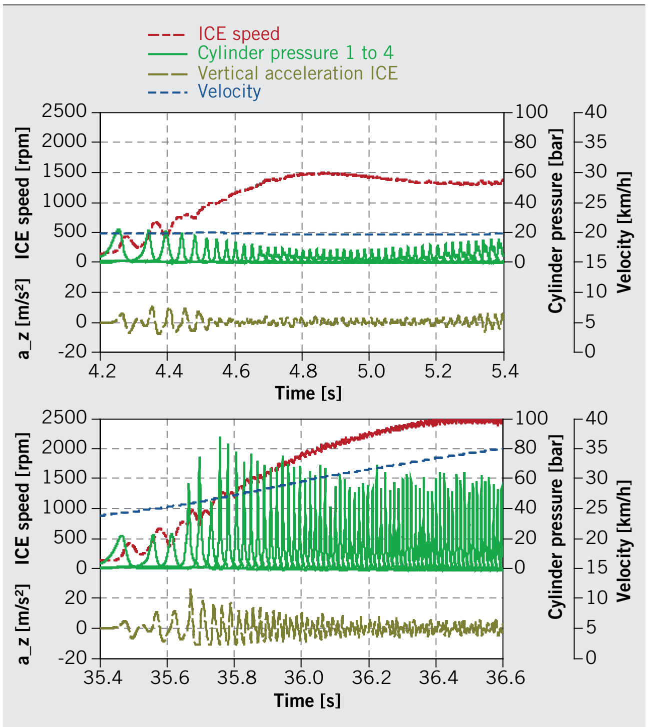
As mentioned at the beginning, an analytical approach towards parametrizing the global energy management of an operating strategy is difficult because of the complex correlations. The algorithms applied on control units apt for vehicle use are heuristic, which implies that different thresholds have to be parametrized in an application process [12]. As EiL or XiL testbeds are tools which present the influence of parameters on the development targets (consumption, emissions and durability) in a reproducible way, their availability facilitates the use of optimization methods for parameter determination. Direct optimization, i.e. the control of the testbed by an optimization algorithm, is very costly and time-consuming due to the simulation in realtime. Therefore it is proposed to apply a model based optimization with previous design of experiment (DoE), ❽, [13]. [7] presents a basic example, in which strategy parameters are optimized. The target is the weighted sum of measured fuel consumption, measured emissions and the battery stress based on a damage model. To avoid information losses due to the combination of different targets, a multi-criteria optimization algorithm can be used. The result of such an optimization is the pareto set, a set of optimal configurations [14]. ❾ shows such a pareto set, which is obtained from parameter variations for a driving simulation. Knowing this pareto set makes it possible to implement an adaptation into the operating strategy that activates optimal parameter configurations depending on the driving situation.

## 5 SUMMARY

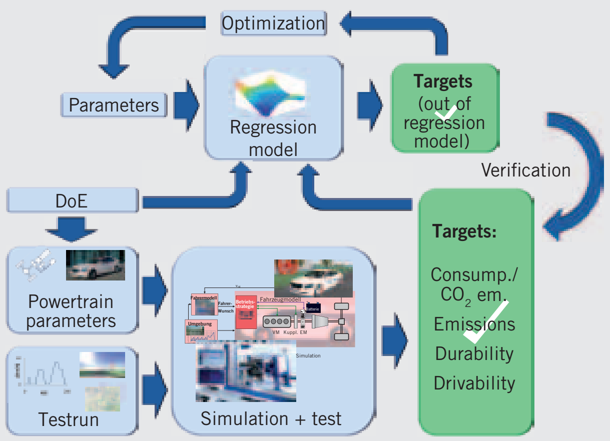
The concepts presented in this paper are aimed at the holistic optimization of the powertrain in a wide range of the development process. The flexible EiL and XiL methods with scalable hardware integration support the development in the early concept-orien-



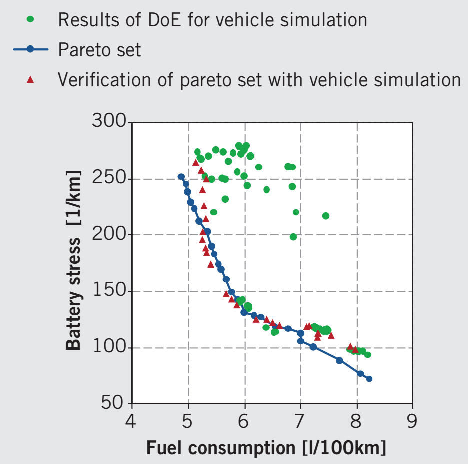

❻ XiL simulation with real combustion engine (ICE) and electric motor (EM) on the first part of the TU Darmstadt city cycle




❼ Application of different engine start types in a power split powertrain; hybrid start initiated by the vehicle system (top), driver-initiated hybrid start (bottom)




❽ Scheme of the process of a model-based simulation at the XiL engine testbed




❾ Pareto set as result of a multi-criteria optimization of parameters in a simulation model of a power-split hybrid



tated phases with reliable consumption and emission measurements using a real ICE and a real exhaust gas system, in later phases with tests of prototypes of drive units and in the integration phases with testruns for the control unit application and durability tests of powerpacks with realistic load profiles. The fact that the testing environment at the testbed is conditionable offers advantages, especially regarding the development and application of functions: reproducibility and the possibility of automation. The automation reduces development time and costs and, in combination with the presented approach of a systematic optimization of control unit parameters, secures high powertrain quality while mastering the high number of variants. The high comparability of the in-the-loop method facilitates the validation of the optimization results in various real-world scenarios with alternating constraints, given the complex interaction of operating strategy, combustion engine, electric motor and battery.

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