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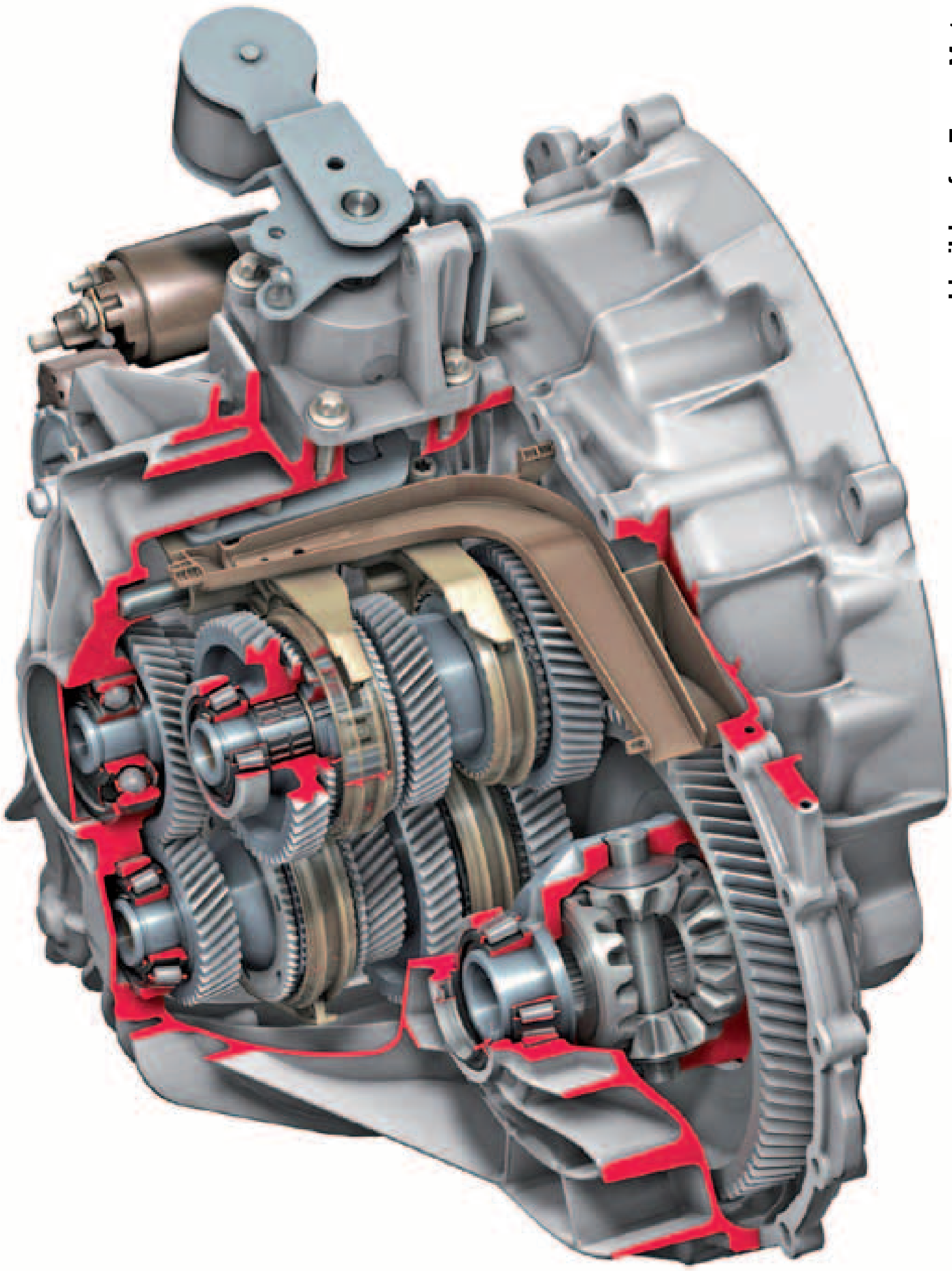
USE OF SIMULATION for the NVH Development of Brake Pads

DRIVER MODEL for Virtual Control System Development

CONTROLLED Pendulum-slider Oil Pump

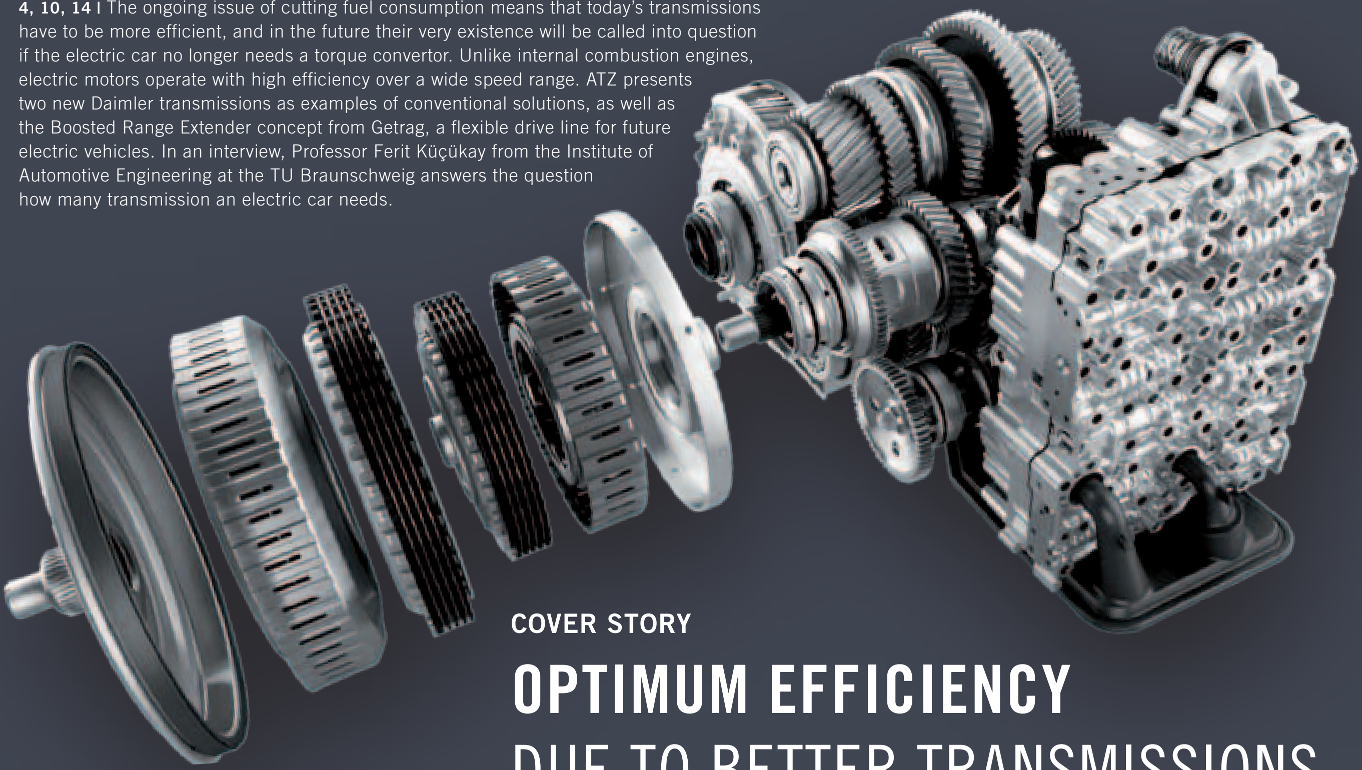
/// INTERVIEW
Ferit Küçükay
TU Braunschweig



OPTIMUM EFFICIENCY
DUE TO BETTER TRANSMISSIONS

CONTENTS

4, 10, 14 I The ongoing issue of cutting fuel consumption means that today's transmissions have to be more efficient, and in the future their very existence will be called into question if the electric car no longer needs a torque convertor. Unlike internal combustion engines, electric motors operate with high efficiency over a wide speed range. ATZ presents two new Daimler transmissions as examples of conventional solutions, as well as the Boosted Range Extender concept from Getrag, a flexible drive line for future electric vehicles. In an interview, Professor Ferit Küçükay from the Institute of Automotive Engineering at the TU Braunschweig answers the question how many transmission an electric car needs.



COVER STORY

OPTIMUM EFFICIENCY
DUE TO BETTER TRANSMISSIONS

A FIRST STEP

Dear Reader,

Yesterday, a first step was taken in eliminating an important argument used by critics of electromobility. The opening of the hybrid power plant run by Enertrag, Total and Co. in Dauerthal, Brandenburg (Germany), means that electricity can now be buffered. Wind power is stored in the form of hydrogen, thus preventing distribution networks from being overloaded at peak production times and still providing enough power during windless periods and at night.

As the first of its kind in the world, the hybrid power plant combines the energy resources of wind, hydrogen and biogas. The electricity generated in three wind power plants with a nominal output of 6 MW is used to produce "CO_2-free" hydrogen in a 500 kW pressure electrolyser. This "green" H_2 is stored and can be used as required in a 700 kW combined heat and power generation unit. Powered by this regenerative energy, electric cars would be genuinely climate-neutral.

Until now, electric cars have been accused of basically not saving any CO_2 at all because they rely on coal-, oil- and gas-fired power stations for their electricity. Even an electric car like the Mitsubishi i-MiEV emits an astonishing 93 g/km of CO_2 (see page 48) with the current German fuel mix – whereas it really should be zero. Petrol-driven small cars like the VW Polo Blue Motion and the Kia Rio, at 87 and 85 g/km respectively, emit about the same amount of CO_2. The Dauerthal project means that electric cars can now operate entirely, and not only locally, with zero emissions.

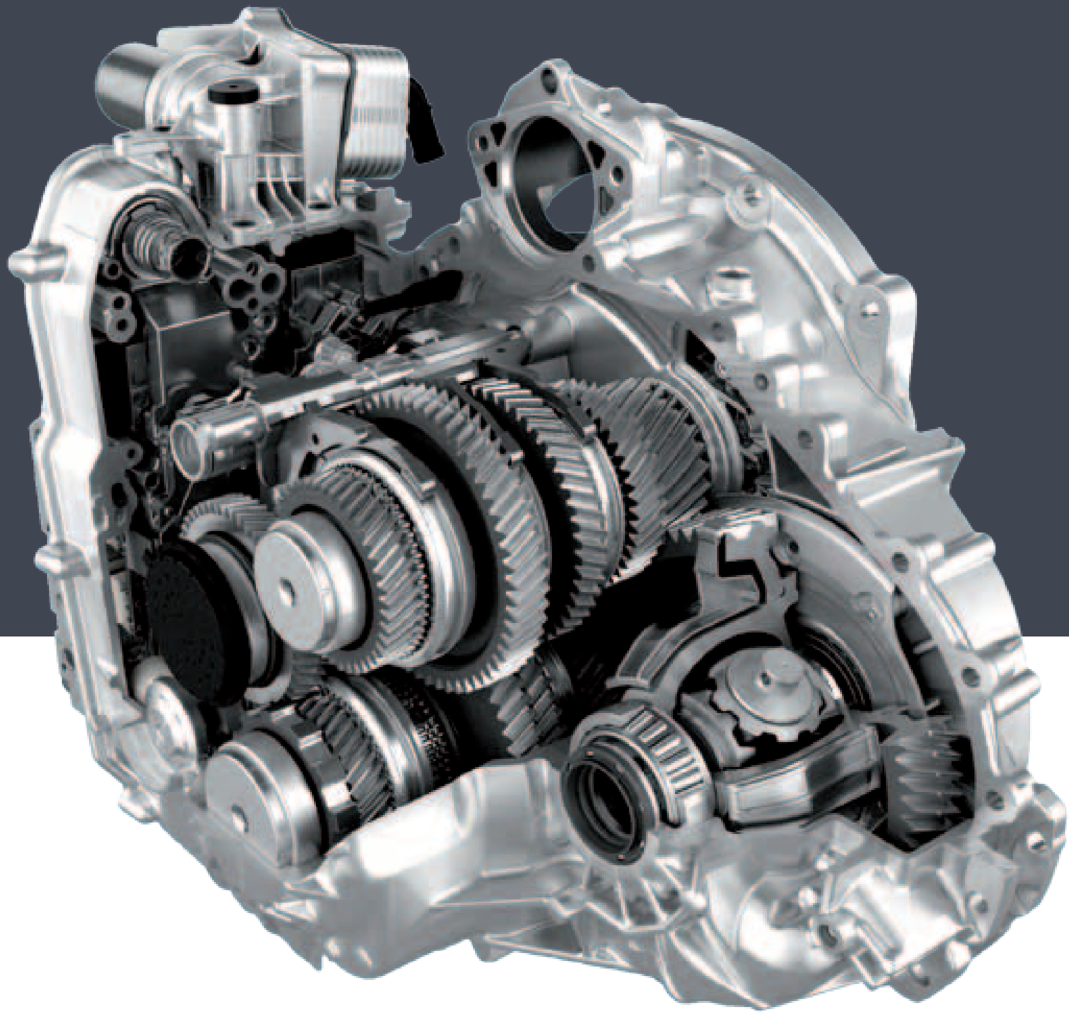
Although Audi's similar e-gas project has not progressed quite as far – the power plant is not scheduled to go into operation until 2013 – its technology goes one step further. The hydrogen produced is not only stored and used to generate electricity in a combined heat and power plant during windless periods. In addition, Audi combines it with CO_2 to produce methane, the basis for natural gas. This CH_4 can then be used to power conventional natural gas cars. That would be the ideal CO_2 cycle.

So there is no lack of concepts and ideas. It's just a question of actually taking the right steps.

I wish you a Merry Christmas and an enjoyable and relaxing festive season. May you start the New Year with renewed energy and inspiration.

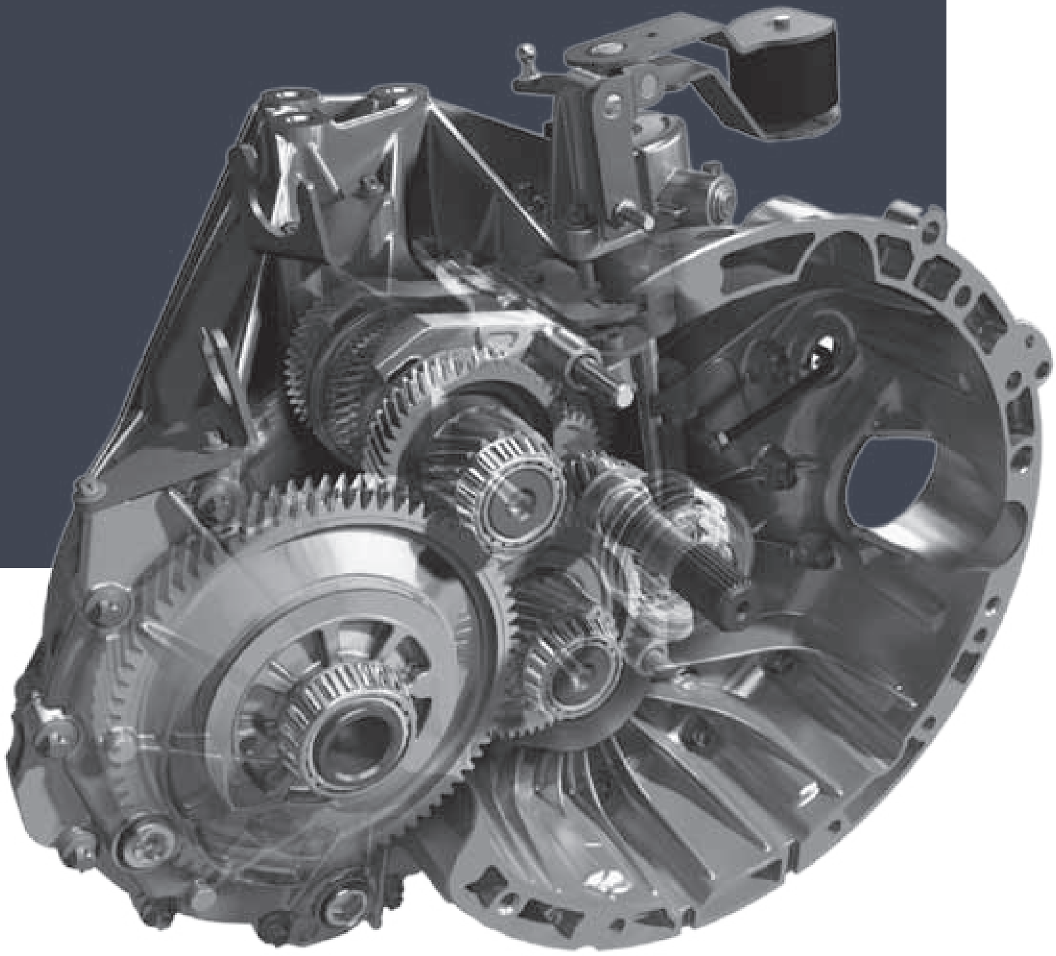
Michael Reichenbach

DIPL.-ING. MICHAEL REICHENBACH,
Vice Editor-in-Chief
Wiesbaden, 26 October 2011



EFFICIENT FRONT-TRANSVERSE TRANSMISSIONS FROM MERCEDES-BENZ

While introducing the new Mercedes-Benz B-Class, also new transmissions are developed. With the 7G-DCT seven-speed dual clutch transmission and SG6-310 six-speed manual transmission, the key objectives low CO_2 emissions, maximum variability and short design were fulfilled. In comparison to other transmissions on the market, the 7G-DCT uses an oil-cooled multi-plate clutch instead of the dry dual clutch in order to deal with the required torque capacity of 350 Nm. The 7G-DCT replaces the CVT Autotronic.



AUTHORS

DR.-ING. RALF WÖRNER
is Entire Project Leader Transmissions Development 7G-DCT and SG6310 at Daimler AG in Stuttgart (Germany).

DR.-ING. ANSGAR DAMM
is Project Leader Development 7G-DCT at Daimler AG in Stuttgart (Germany).

DR.-ING. RALPH EBERSPÄCHER
is Leader Manual Transmissions Driveline at Daimler AG in Stuttgart (Germany).

DR.-ING. CARSTEN GITT
is Leader Development Hybrid Transmissions and was until September 2011 Leader Development Light Transmissions at Daimler AG in Stuttgart (Germany).

FROM THE GROUND UP NEW

As part of the IAA Motor Sow 2011, Daimler presented the successor model series to the Mercedes-Benz B-Class as the first model to feature the new vehicle architecture "New Generation Compact Cars" (NGCC). Tailored to the overall vehicle concept and layout, the 7G-DCT seven-speed dual clutch transmission and SG6-310 six-speed manual transmission were developed from the ground up for front-wheel drives. The process incorporated the following key objectives:

- creating an efficient overall concept incorporating a high degree of efficiency to fulfil current and future requirements in terms of CO_2 emissions
- ensuring maximum variability through the use of a modular gear set and common parts – as far as technically practical
- compact design and low gross weight, tailored to the general vehicle packaging requirements
- a very smooth gear change and precise shift response
- integration of additional functionalities such as the intelligent stop-start function, the electrically operated parking lock and reverse gear lock along with provision for additional drive system technology.

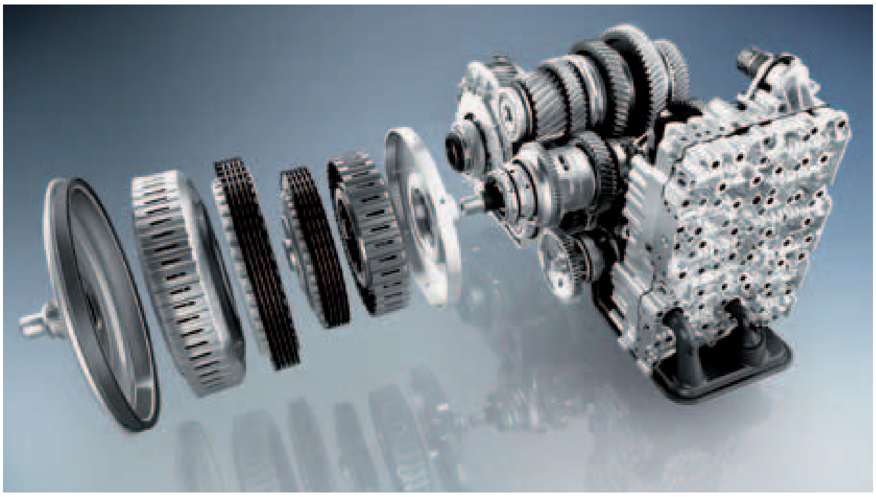
Both transmissions will be offered in combination with the OM651 and M270 four-cylinder diesel and gasoline engines at various levels of development. In the following the 7G-DCT dual clutch transmission and the SG6-310 manual transmission are presented more in detail.

INITIAL SITUATION

For the first time, the OM651 diesel engine as well as the completely new M270 gasoline engine used from the C-Class to the S-Class will be available in the compact vehicle segment of Mercedes-Benz Cars, the NGCC. A prerequisite for the installation of these systems was the further development of the sandwich construction method used to date, which features a hollow floor assembly and a forward-sloping engine. The evolved concept known as the “Energy Space” enabled the installation of the powertrain in a conventional position in the NGCC architecture without sacrificing the advantages of the sandwich design regarding spatial ergonomics, safety and provision for alternative drive technologies.

In view of the implications of these general technical and design conditions, it was decided not to use the transmission from the predecessor model. Taking into account the front-transverse configuration and the overall vehicle concept, a spur gear layout incorporating an adapted differential configuration in Q 1.2 was identified as the ideal solution for the new model series. Integral in this context and appropriate to the competitive environment, a dual clutch transmission was identified as a customer-optimized solution in addition to the manual transmission. Analysis showed that there was no product available on the market that could comprehensively provide the functions relating to CO_2 targets and packaging requirements as defined in the requirement specifications.

Based on these insights and in view of the expertise accumulated by Mercedes-Benz in the area of transmissions over decades, it was decided to develop and produce the 7G-DCT in-house. Similarly, a comprehensive competitive analysis was also carried out for the SG6-310. Looking at the requirement specifications, it is clear that an internal implementation of the manual transmission is also the logical solution – one that also enables synergies to be realised from the combined production of key assemblies within Daimler AG. These include components such as output shafts, idler gears, shift sleeves, differentials as well as the production of the casing sections for both transmissions.



❶ Built-up of the 7G-DCT dual clutch transmission with seven speeds and wet clutch

7G-DCT SEVEN-SPEED DUAL CLUTCH TRANSMISSION

The new 7G-DCT replaces the Autotronic [1] continuously variable automatic transmission (CVT) previously used in the B-Class. ❶ shows a built-up of the transmission. Its dry weight is around 81,2 kg. Based on the experience gained in the area of automatic [2] and manual transmissions, it was possible to implement the new product concept within a very short space of time. A unit whose innovative gear set concept, ❷, facilitated an extremely compact design was presented within three years.

❷ Gearset of the 7G-DCT

❸ Oil-cooled multi-disc clutch as double clutch of the 7G-DCT

Similar to transmissions already available on the market, the outermost of the two concentric clutches is connected to the un-even gears (1, 3, 5 and 7). The inner clutch operates gears reverse (R), 2, 4 and 6. The decision to employ an oil-cooled multi-plate clutch, ❸, instead of the dry dual clutch now so prevalent in the market was influenced by the following criteria:

- Low moment of inertia: The required torque capacity of 350 Nm (with the option to be increased) leads to significantly higher weights and moments of inertia for dry clutches than in the multi-plate clutch. This results in limited driving dynamics. Furthermore, the multi-plate clutch offers a simple option to increase torque through modified assembly using friction plates.
- Thermal robustness: With a permissible combined vehicle/trailer weight of over 3000 kg, the NGCC vehicles are designed to offer the driver a high level of reliability even in critical situations, when manoeuvring, stopping and moving off on inclines. Moreover, the SUV-like vehicles planned as part of the NGCC series place exceptionally high demands on the thermal robustness of the starting device.

A key differentiating characteristic of the new gear set is the use of a double gear chain as well as the highly integrated R-gear using the existing gear ratios. This permits the extremely short design of the 7G-DCT compared to competitor transmissions with an equivalent torque capacity. It was possible to combine gears 5 and 7 as well as gears 4 and 6 using a single pinion gear drive each. Despite the resulting compulsory combination in these gears, it was possible to achieve a smooth, progressive gear-ratio step in conjunction with the design of the final drive.

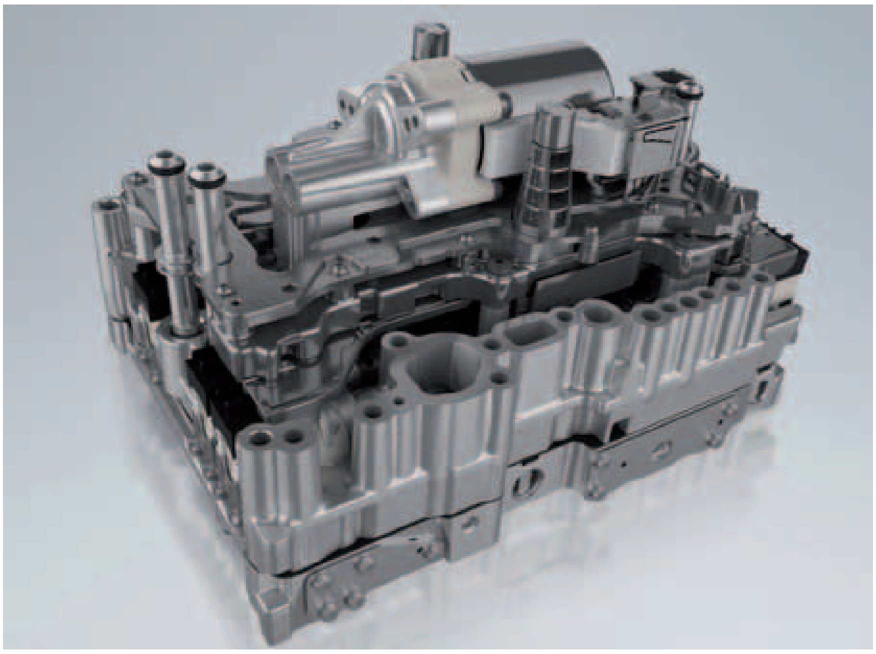
A completely new design was chosen for the reverse gear. Many systems employ a separate reverse gear shaft. Although this provides freedom in terms of the gear ratio used, it leads to additional weight, the need for extra installation space, and higher costs. Other systems use the gear ratio of second gear to transfer torque from there to the drive shaft not directly connected to second gear, thereby reversing the direction of rotation. The disadvantage of this design is that the gear ratio of the reverse gear may be relatively high, depending on the design of the gear set. In the 7G-DCT, torque was also transferred from second gear to the other drive shaft in order to change the direction of rotation. However, the output is not transferred directly from there to the output shaft via the final drive, but first using the ratio of third gear back to drive shaft 1. From here, torque is transferred onwards via first gear to the output shaft. Despite the constraints resulting from the gear chains of gears 4/6 and 5/7, this approach achieves an acceptable gear ratio for reverse gear.

The combination of two gear set configurations with two different final drive gear ratios enables the transmission to be adapted to meet all requirements arising within the NGCC series. Consequently, small gasoline engines are equipped with a gear ratio spread of 7.142, while the larger gasoline and diesel engines achieve optimum power delivery with a gear ratio spread of 7.990. ❹ shows the table of the gears and gear ratio spreads.

The shift sleeves for shifting gears are actuated by innovative shift actuators which are fitted as modules and integrate the hydraulic function. The enclosure fitted to the differential transmission reduces churning losses in the transmission and improves efficiency. Reduced oil foaming also results in improved acoustics and excellent shift quality under even the highest load scenarios.

FOR GASOLINE ENGINES		FOR DIESEL ENGINES	
1st Gear	15.943	1st Gear	15.943
2nd Gear	10.038	2nd Gear	10.038
3rd Gear	6.927	3rd Gear	6.359
4th Gear	4.915	4th Gear	4.335
5th Gear	3.606	5th Gear	3.205
6th Gear	2.771	6th Gear	2.501
7th Gear	2.232	7th Gear	1.995
Reverse gear	12.807	Reverse gear	13.950
GEAR RATIO SPREAD	**7.142**	**GEAR RATIO SPREAD**	**7.990**

❹ Table of the gear ratios and gear ratio spreads for the 7G-DCT



❺ Fully integrated hydraulics control unit of the 7G-DCT

❻ Vane oil pump of the 7G-DCT

MECHATRONICS WITH TWELVE SENSORS IN THE CONTROL UNIT

The fully integrated hydraulics control unit, ❺, for the 7G-DCT is based on a thick-film substrate acting as a circuit board on which the unpackaged components are fixed using a conductive adhesive. The central element is a 32-bit microcontroller with integrated Flash memory and RAM. In addition to the evaluation electronics for the sensors and output stages for the solenoid control system, it was also possible to place the noise suppression and actuation for the electrical oil pump directly on the circuit board. This assembly technology can cope with a very broad temperature range.

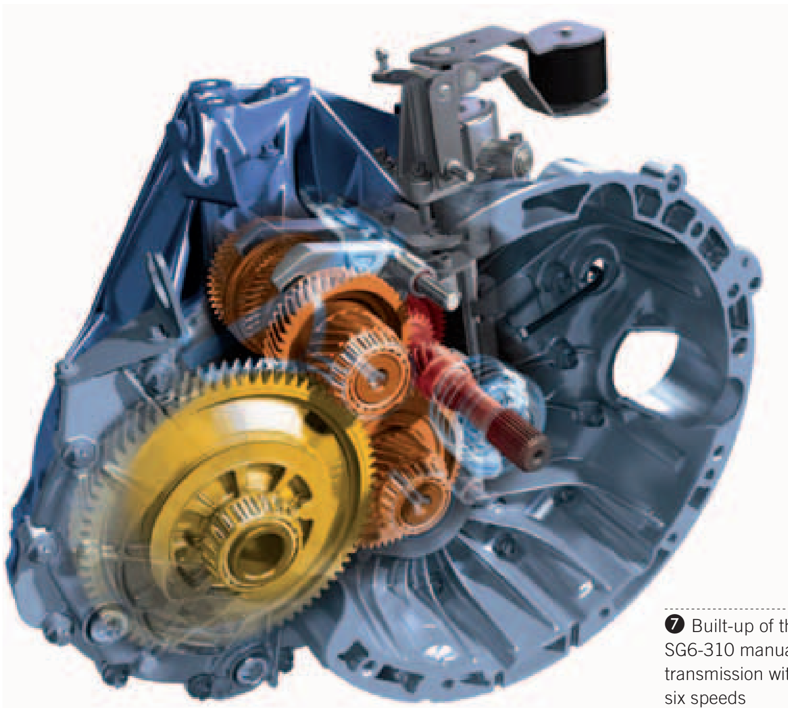
A particular focus during development was on minimizing the electrical interfaces. As a result, five position sensors, three speed sensors as well as two pressure and temperature sensors are fully integrated into the control unit. The connection to the circuit board is established using flex foil and aluminium wire bonds. The connections for the electric oil pump, the nine solenoid valves and the solenoid actuator of the 7G-DCT are also integrated in the control unit in the same way. As a result, there is no need for a complex wiring harness. The on-board connection for the 7G-DCT is reduced to a five-pin plug.

Similar to the measures already implemented in Mercedes-Benz planetary gear sets, an electrically driven auxiliary oil pump, ❻, in vane type was also integrated in the dual clutch transmission. Whereas this pump served as an add-on solution in previous applications, it has been highly integrated into the 7G-DCT, both structurally and functionally.

Consequently, this auxiliary oil pump is available not only for the intelligent stop-start function to keep the transmission permanently pressurized independently of whether the engine is running. It can also be used to boost the mechanical pump during normal driving. This provides scope to further reduce the size of the vane cell pump, which already exhibits excellent efficiency for design reasons, thereby significantly improving the overall efficiency of the transmission.

In addition to the functions outlined above – stop/start support and boosting – the auxiliary pump has two further functions. If the cooling oil requirement increases significantly, the cooling flow requirement of the clutch is regulated using the auxiliary pump. Since sufficient quantities of oil are already available in these areas due to the engine speed, the boost function does not need to be used for this purpose. The auxiliary oil pump therefore fulfils the function otherwise performed by an additional control valve.

The fourth function is closely linked to another feature of the 7G-DCT, the park-by-wire system. So that the parking lock can be engaged and disengaged even when the engine is not running, every park-by-wire function needs an additional source of power that not dependent on engine speed. By integrating the auxiliary oil pump, this function could be implemented without using an additional actuator. To sum up, despite the wide range of functions offered by the 7G-DCT (seven gears, wet dual clutch, park-by-wire), the



❼ Built-up of the SG6-310 manual transmission with six speeds integration of the auxiliary oil pump means that only eight control solenoid valves and one shift valve need to be installed. A further developed version of the 7G-DCT will also be offered with all-wheel drive along with higher power and torque outputs.

Taking everything into account, the overall concept of the 7G-DCT achieves fuel savings of 9 % in the NEDC compared with its predecessor transmission, the Autotronic. As a result, the transmission performs on a par with the manual transmission for all applications within the NGCC.

SG6-310 SIX-SPEED MANUAL TRANSMISSION

The new SG6-310 six-speed manual transmission for front-wheel drives was developed jointly with the 7G-DCT. ❼ shows the design of the transmission. Designed as a three-shaft transmission with an overall length of 357 mm, the SG6-310 is extremely compact. The unit's dry weight is around 46 kg. The drive shaft features a fixed-floating bearing, whereas the two output shafts and the differential gear system use tapered roller bearings. All tapered roller bearings are mounted according to the so-called Set-Right principle and consequently do not require individual adjustment. Gears 1, 2, 5 and 6 are shifted on the lower output shaft, while gears 3, 4 and R are shifted on the upper one. Multiple use of gear pairs contributes to a compact and weight-optimised design. Gears 4 and 5 feature a shared fixed gear, while the idler gears of gears 1 and R mesh directly with one another, doing away with the need for a reverse gear shaft or reverse idler gear.

To meet the exacting requirements regarding ease of shifting, triple cone synchromesh systems are used in gears 1 and 2. All other forward gears feature a dual cone synchromesh system, with reverse gear using a single cone system. The four weight-optimised cast aluminium shift forks are guided by means of slide bearings on just two shift rods, which also offer benefits in overall weight terms.

OVER-TORQUE FUNCTION AND PLASTIC GEARSHIFT BRACKET

The necessary process of detecting when the transmission is in neutral for the stop-start function is implemented by sensing a magnet attached to the shift shaft. Activation of the reversing lamp is also triggered in the same way. The SG6-310 currently has a torque capacity of 310 Nm, which can be increased to 330 Nm for short periods in the over-torque function. As a result, outputs of up to 100 kW can be handled in conjunction with diesel engines and up to 115 kW with gasoline engines.

At market launch, the transmission SG6-310 will be offered in two series of ratios which were intensively optimized during development with regard to fuel consumption and ease of shifting as well as start-up and acceleration response. ❽ shows the table of the gear ratios and gear ratio spreads in the diesel and gasoline versions for the gears 1 to 6 and the reverse gear.

The gear lever is mounted in a plastic gearshift bracket with a mechanical lift lock for the reverse gear and connected to the shift module at the transmission end by two Bowden cables for the shift and selection movement. Carefully coordinated decoupling elements on the Bowden cables combine with a specifically designed locking mechanism for the internal gearshift to enable a shift sequence that is perceived by the driver as extremely fluid and precise. The specially designed hydraulic clutch system supports the excellent ease of shifting through minor adjustments to pedal force, optimum sensitivity and exceptional NVH characteristics.

FOR GASOLINE ENGINES		FOR DIESEL ENGINES	
1st Gear	14.438	1st Gear	13.182
2nd Gear	8.169	2nd Gear	7.339
3rd Gear	5.203	3rd Gear	4.695
4th Gear	3.614	4th Gear	3.322
5th Gear	2.733	5th Gear	2.429
6th Gear	2.337	6th Gear	1.976
Reverse gear	12.994	Reverse gear	12.031
GEAR RATIO SPREAD	**6.171**	**GEAR RATIO SPREAD**	**6.656**

❽ Table of the gear ratios and gear ratio spreads for the SG6-310

SUMMARY

The newly developed transmission portfolio represents an important part of the technological leap achieved by the new B-Class from Mercedes-Benz as the first model to feature the new vehicle architecture "New Generation Compact Cars" (NGCC). In the case of the 7G-DCT dual clutch transmission, it was possible to integrate additional technology modules. The company's first dual clutch transmission combines the use of a wet starting device and the resultant very high level of robustness and comfort when pulling away with a highly efficient electro-hydraulic control unit and, for the first time, added integration of the park-by-wire and intelligent stop-start functions. The combination of an electric and mechanically powered oil pump also helps the transmission set new benchmarks in terms of efficiency.

Following on from the high standard of precision and smoothness achieved by the predecessor of the SG6-310 six-speed manual transmission, development work on the new transmission focused on further optimization. This led to yet another improvement in shift quality coinciding with even lower shift force as well as further optimization of friction loss and weight.

In total, both transmissions contributed to the achievement of current and future requirements as regards CO_2 emissions. Moreover, the variable gear set modules also guarantee the future viability of the concept in other drive variants within the NGCC.

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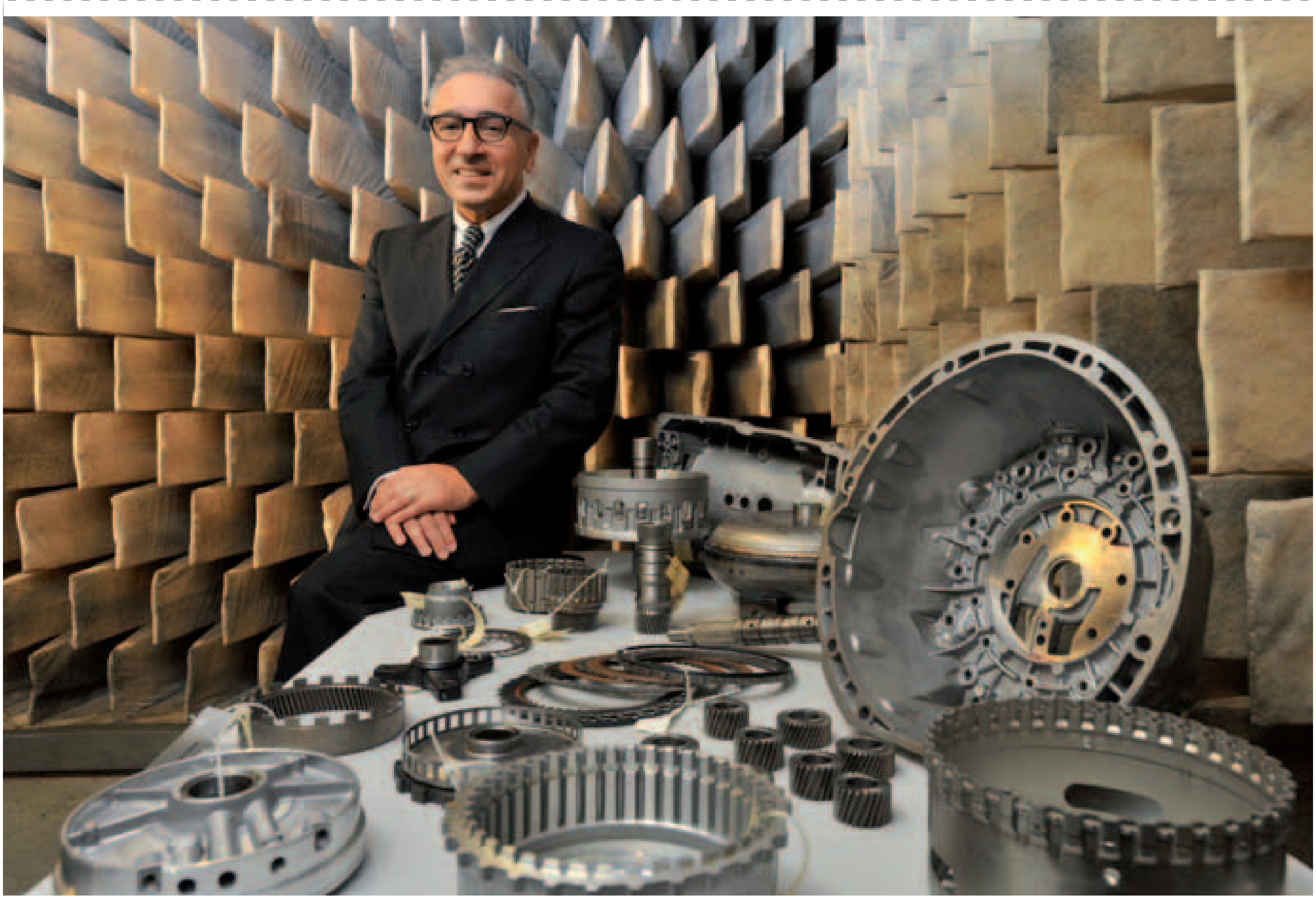


"I HAVE NO WORRIES THAT THE TRANSMISSION WILL DISAPPEAR WITH THE ELECTRIC CARS"

When the first electric cars were designed, there was a fear that, with the introduction of the ideal powertrain in the form of an electric motor that develops high torque from a standstill, there would no longer be any need for a transmission. But now it seems that even three gears are possible. ATZ discussed these and other questions in an interview with the Director of the Institute of Automotive Engineering, Professor Ferit Küçükay, from the TU Braunschweig.

Prof. Dr.-Ing. Ferit Küçükay (born in Istanbul in 1953) initiated the International Transmission Symposium together with the organisers CTI ten years ago. From 5 to 8 December 2011, around 1000 participants and 90 exhibitors are expected in Berlin. Küçükay studied at the TU Munich until 1977. After completing his doctorate and habilitation on the subject of powertrain and transmission dynamics at the same university, he joined BMW in 1985. There, he held various development and management positions, including head of automatic transmission development. In 1997, he became Director of the Institute of Automotive Engineering at the TU Braunschweig, where, in addition to the subjects of requirement management, chassis and driver assistance systems, he focused his research work in particular on hybrid and electric drive systems. He is also a consultant, co-publisher and member of the advisory board for various conferences and magazines, and is a member of the board of directors of the Automotive Research Centre Lower Saxony (NFF).

ATZ _ The transmission industry is facing major changes due to the electric car. How many gears can I still offer as a transmission supplier?

KÜÇÜKAY _ Even if the electric car really becomes predominant one day, I am still not worried that the transmission will disappear. That is due to the following fact: if you need to drive up a steep incline in a fully loaded small car, you need approximately 1500 Nm of torque at the driving axle, and for the top speed on the motorway you need about 1500 rpm at the driving wheels. Such an electric motor would probably have the same mass as the vehicle itself. A transmission with a gear ratio of 10 combined with an electric motor with 150 Nm and a maximum speed of 15,000 rpm would be the solution here.

"For electric vehicles, a significant improvement in efficiency is possible with a two-speed transmission."

Could you do that with one gear or are two gears necessary?

I can certainly see possibilities to offer two or even three gears in electric cars. Multi-speed transmissions in combination with an electric drive system certainly make sense for both extra-urban and urban driving. According to our estimates, an improvement in efficiency of ten percent can be achieved in the NEDC by using a two-speed transmission – that is a huge amount! After all, even the electric motor does not offer the best efficiencies at all load and speed points.

The supplier ZF presented its Electric Twist Beam system (a transmission/electric motor unit per wheel) as a new solution at the IAA 2011. What are its advantages?

I see that as a good solution because a integration of functions has taken place. That offers advantages, for example in reducing unsprung masses by integrating the electric driveline into a twist beam rear axle instead of in the wheel. Furthermore, it also creates more space for batteries. Therefore, I can certainly see opportunities for this drive system in the urban mini and micro-car segment.

What are the potentials for improving efficiency in automatic transmissions, for example by using spur gears?

You mustn't only think of the intermeshing gear wheels. After all, the transmission contains more loss-causing components which, when taken altogether, account for an efficiency of 80 to 95 percent related to the NEDC. So there is a huge improvement potential here. The aim is to further improve the efficiency of the transmission by reducing tooth surface roughness and by using dog clutches instead of multi-disk clutches, an oil pump which is used on-demand or fixed/floating bearing arrangement. The transmission also contributes more to improving the efficiency of the entire powertrain by ensuring that the drive machine can always work in an operating range with optimum efficiency. The transmission must therefore have a high gear ratio spread, which means that it also requires several gears to ensure that drivability is not impaired. According to our calculations, gear ratio spreads of 9 to 10 and the same number of gears certainly make sense.

Are nine or ten gears still comfortable for the driver?

Yes, if the gear shifts are carried out without the driver noticing them. And that works better if there are smaller steps between gears.

What about the efficiency of such a transmission?

It doesn't have to be lower if one considers the development of stepped automatic transmissions. The skill lies in keeping the internal efficiency of the transmission within a good range by using as few shift elements as possible and using friction-optimised components. Today's eight- and nine-speed transmissions have only five or six shift elements, and therefore the same number as some five-speed automatic transmissions. With more gears that are shifted automatically, it is easier to operate the drive machine efficiently in all driving situations. For that reason, the driving cycle fuel consumption of an automatic vehicle is often better than that of a vehicle with a manual transmission. The afore-mentioned advantages concerning comfort and fuel consumption have meant that the proportion of vehicles with automatic transmissions has risen to 50 percent of the entire production volume – in spite of additional costs.

Daimler has just presented its first dual-clutch transmission, which will replace the Autotronic CVT in the B-Class. Does that mean that the CVT is out of the running?

Dual-clutch transmissions are definitely an interesting alternative, not only to the CVT but also to an automatic transmission. But the CVT is by no means out of the run-

Ferit Küçükay in conversation with Dipl.-Ing. Michael Reichenbach, ATZ Vice Editor-in-Chief

The share of automatic transmissions has increased to more than 50 percent today, Küçükay confirms

ning. Perhaps in Germany, but not in Japan. One in every seven automatic vehicles produced is fitted with a CVT, and the trend is upwards. The reason is that the CVT has great and growing popularity in the Asian-Pacific region, but not in Europe. Jatco is investing in CVT technology – and has been doing so for years. Considerable improvements can be achieved with regard to package space, costs and efficiency, for example by using a smaller variator combined with a two-speed transmission.

Is there a need for the new torque converter automatic transmission from ZF and Mazda at all in the smaller front/transverse sector, where dual-clutch transmissions are dominant?

On a worldwide scale, stepped automatic transmissions are dominant in the front/transverse sector. But that needn't remain the case, of course. Dual-clutch transmissions are competitive in all respects, as Volkswagen's DSG system has proven. Competition is good for business. Torque converter automatic transmissions would certainly have not achieved their good state of development today if they hadn't been under massive pressure from dual-clutch transmissions. At the same time, dual-clutch transmissions are making progress. The efficiency of dry dual-clutch transmissions currently sets the benchmark in this class.

"Gear ratio spreads of 9 to 10 and the same number of gears certainly make sense."

What is your favourite transmission and your favourite car?

As far as the transmission is concerned, I can't give a general answer. For long-distance driving, I enjoy the advantages of a torque converter automatic transmission, whereas in a sporty roadster I want to enjoy the driving experience by shifting gears manually. My favourite car is a VW Beetle from 1953, the year I was born, and I often drive it to work. I am fascinated by the simplicity of the car, which still gets you from A to B even today.

Professor Küçükay, thank you very much for this interesting interview.

INTERVIEW: Michael Reichenbach
PHOTOS: Frank Bierstedt

FLEXIBLE POWERTRAIN FOR ELECTRIC CARS

The previous trend of always fitting more expensive technologies in the internal combustion engines and in the transmission is no longer sustainable due to the customer's demand for economical vehicles. Getrag is breaking away from this previous trend with their Boosted Range Extender concept for hybrid and electric cars. Economical solutions, which have been combined in this concept in both transmission as well as the internal combustion engine, will satisfy the customer requirements regarding economical consumption, adequate range and excellent driving characteristics.

AUTHORS

DIPL.-ING. STEFAN HÜPKES
is Head of the Next-generation Technology Department at Getrag Ford Transmissions in Cologne (Germany).

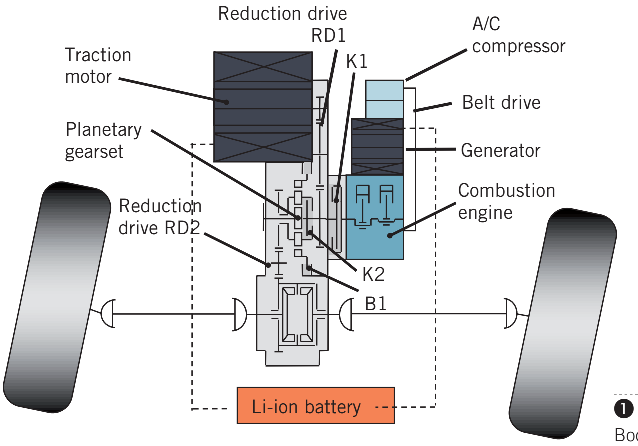
DIPL.-ING. GEORG HILLEBRAND
is Head of the Software Development Group at Getrag Ford Transmissions in Cologne (Germany).

DIPL.-ING. WOLFGANG BRINGS
is Member of the Boosted Range Extender Project Team at Getrag Ford Transmissions in Cologne (Germany).

DEVELOPMENT OF NEW POWERTRAINS

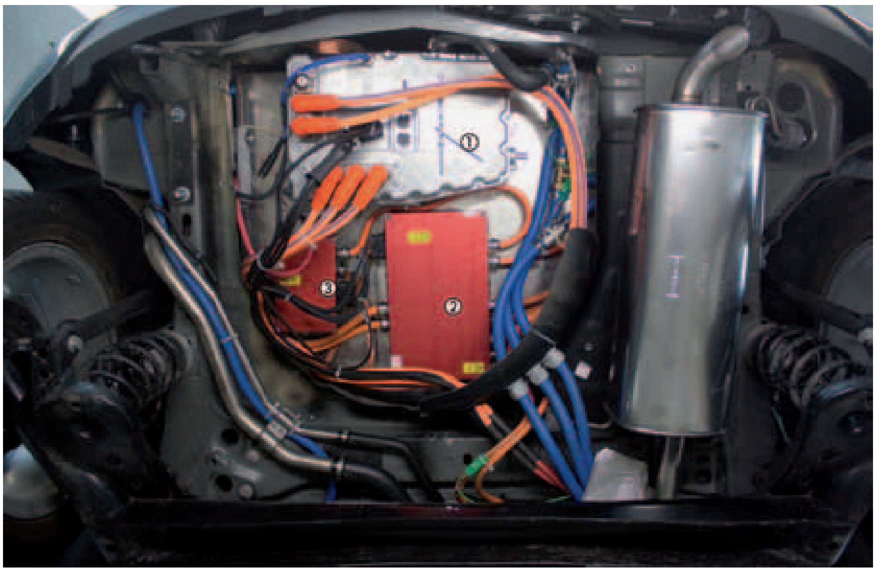
The German Federal Cabinet adopted an "Electro-mobility" governmental programme in May 2011. The government emphasises the national and international interests in development and market launch of electro-mobility and this support started with the provisions in the Economic Stimulus Package II totalling 500 million Euros for 2009 to 2011. The main focus is on the development of new powertrains, which must fit in the available space and meet the customer requirements regarding performance, safety, reliability and costs [1].

Getrag has already started to develop a new powertrain concept called Boosted Range Extender in order to fulfil these customer requirements. The name consists of both a stretching of the travelled



❶ Principle of the Boosted Range Extender

❷ Engine space with combustion engine (1), power electronics (2) and Boosted Range Extender transmission (beneath the power electronics)



❸ Conversions made underneath the trunk – power electronics with DC/DC converter (1), high voltage supply distribution box (2) and isolation monitor (3)

distance and an operation of the combustion engine parallel to the electric motor. The target of the concept is to present the known driving functions together with an economical powertrain. In particular, the achievable normal driving range should be around 600 km without having to use a very large and expensive battery. Also the fuel consumption for long journeys should be less than the consumption of a conventional powertrain.

The development and construction of a new hybrid transmission has started after the initial concept evaluation and was already displayed as a cut-away model at IAA 2009. This powertrain concept has now been fitted in a testing car as a working prototype and as the pilot operation.

CONCEPT DESCRIPTION

The Boosted Range Extender concept consists of a two-gear planetary transmission linked with an electric motor for short journeys up to 50 km and a combustion engine, which can also be coupled mechanically to the powertrain. This will make the range of the Boosted Range Extender comparable with the range of a conventional powertrain.

Additionally a small 15-kW generator is connected to the combustion engine and this can be switched in via the drive belt and a clutch. The system allows

: pure electric driving
: a serial hybrid mode and
: a highly efficient parallel hybrid mode.

The two ratios of the transmission enable the electric motor and the generator as well as the engine to be downsized. Benefits regarding weight, costs and space will be realised compared to other range extender concepts. An additional benefit of the two ratios is the increased gradeability [2, 3].

❶ shows the principle of the Boosted Range Extender. The electric motor is coupled to the sun gear of the two-gear planetary transmission via the reduction drive RD1. This enables the electric motor's speed level to be defined independently from the combustion engine. This also allows high-speed motors, which have an improved performance/weight and costs ratio, to be used.

The planetary carrier is linked to the differential via a second reduction drive RD2, which defines the operating range of the engine.

Brake B1 and clutch K2 are needed for the two ratios of transmission. In second gear the sun and planetary carrier are coupled thus operating at the best possible efficiency. The engine can also be coupled to the sun gear via the dog clutch K1. Therefore the two ratios are also useable for the engine as well.

The generator and the air-conditioning compressor are driven via a belt drive. The first of two couplings in the belt drive enable the generator to be coupled to the engine and to work as the starter. The second allows the generator to drive the air-conditioning compressor, which enables cooling in electrical mode without having to use an additional device.

POWERTRAIN COMPONENTS

The powertrain described before has been installed in an actual Ford Fiesta as testing car, ❷. A three-cylinder gasoline engine with 1.0 l displacement is used as the combustion engine (1). The traction motor with 82 kW maximum power is fitted underneath the power electronics (2). It was not necessary to make any structural modifications to the car due to the compact design of the Boosted Range Extender unit.

The Lithium-polymer high voltage (HV) battery with 14 kWh was fitted in the car's trunk and mounted in a robust frame, which was bolted to the car. This ensures adequate crash safety inside the prototype vehicle. The combined converter / DC/DC converter (1) used for the generator as well as the HV distributor (2) and the isolation monitor (3) were fitted in the spare wheel well, ❸.

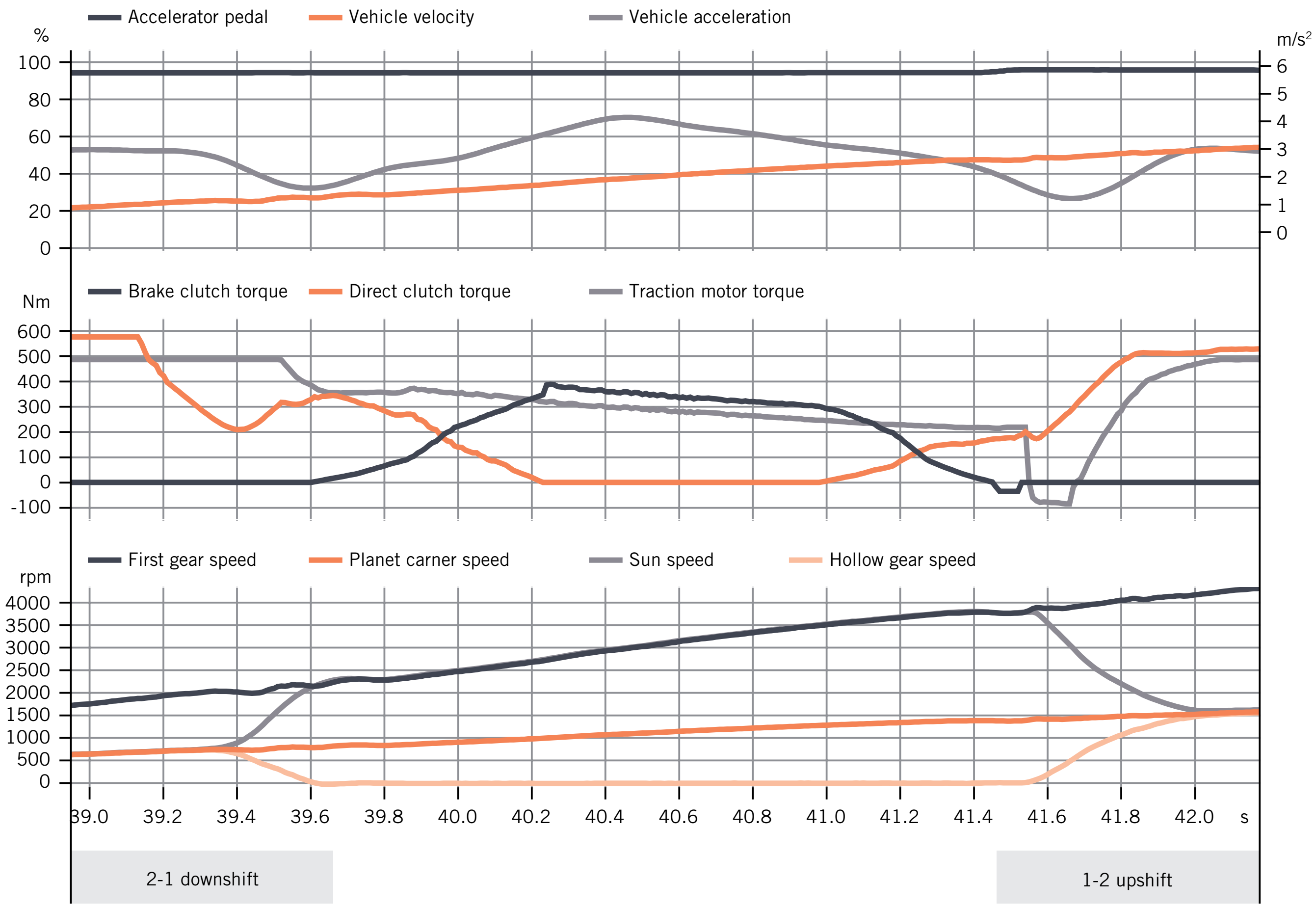
COOLING

Since the temperature level of HV battery and the other HV components strongly differ, two additional cooling circuits had to be fitted in the car. One circuit cools the HV-battery and the other cools both converters as well as both E-machines. The cooler is mounted in front of the engine cooler, so that no additional fans would be needed. An electrical pump was installed to replace the mechanical water pump of the engine, in order to ensure that adequate cooling could be realised.

CONTROLLER AND SOFTWARE

All of the newly integrated HV-components communicate via a separate CAN bus. A Microautobox by dSpace is used as the master controller, which uses the vehicle CAN bus and the range extender CAN bus as well as controlling various other hardware inputs and outputs. Getrag has programmed all of the range extender functions in Matlab/Simulink and they have been autocoded similar to the company's Powershift transmission functions.

After careful analysis of the CAN messages in the original vehicle we were able to set up the system so that no warning or fault messages were displayed on the car's instrument cluster and the driver assistant



❹ Gear shifting process when 2-1 downshifting and 1-2 upshifting in electric mode systems (ABS, ESP and ASR) continue to operate without any special tuning.

A CAN-suitable display with a touch screen and measurement data-logger function was installed in the car as control interface, so that the driver could easily enter driving function details, such as operating mode, engine power or recuperation procedures.

BRAKING AND RECUPERATION SYSTEM

As the range extender will primarily be working in electrical mode and therefore no vacuum will be present, the vacuum brake booster is supplied by an electrical vacuum pump. An additional sensor fitted on the brake pedal enables the free travel to be used for controlling the deceleration torque during the recuperation phase.

ELECTRICAL DRIVING AND GEAR SHIFTING

The car will run in electrical mode as long as the battery charge is sufficient. The Boosted Range Extender uses first gear for severe acceleration or driving up a slope and second gear for normal loads and high speeds. The planetary transmission and the hydraulically operated wet clutch enable gear changing to be made without torque interruption.

A challenge to the shifting quality is present because of the huge ratio step between the two gears. The characteristics of the electric motor to deliver constant power and therefore to supply the same wheel torque before and after the shift, provide good options for producing unnoticeable gearshifts. The excellent controllability of the electric motor is used in both pure electrical driving as well as in the parallel hybrid mode, in order to produce good gear shifting qualities, ❹.

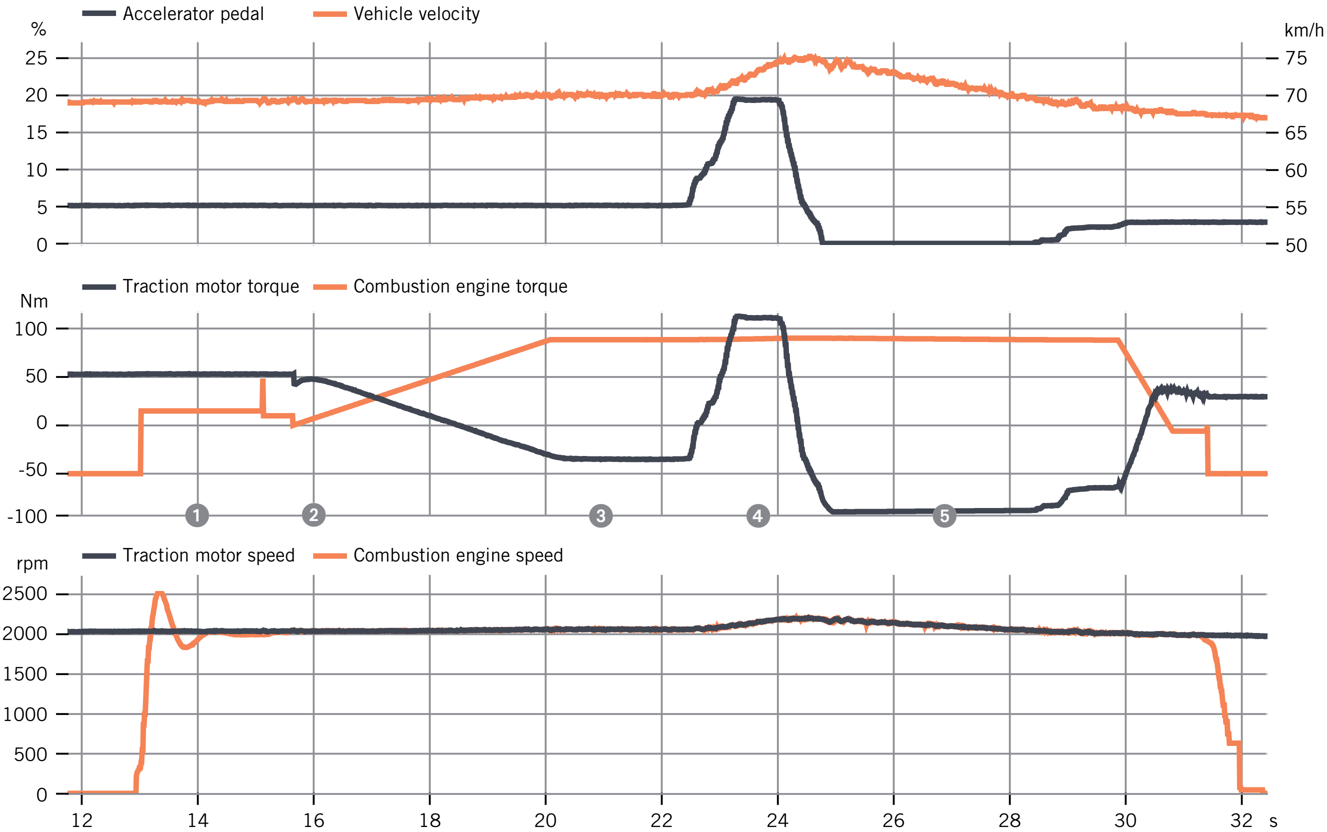
The recuperation option is an important characteristic of the Boosted Range Extender. Up to 30 kW can be recuperated over the entire speed range in principle. Nevertheless the recuperation power is matched to the brake pedal movement and the vehicle speed to achieve an optimal driving feeling.

PARALLEL HYBRID MODE

The Boosted Range Extender runs in parallel mode at higher vehicle speeds and low battery charge. That is possible above 15 km/h in first gear and above 45 km/h in second gear, ❺. First the engine will be started by the generator and brought up to the correct speed (1). It will then be linked to the transmission input shaft via a dog clutch (2). The direct mechanical coupling provides a higher degree of efficiency than in serial mode.

The engine delivers the necessary power to drive the car and to recharge the battery (3). The electric motor provides the necessary dynamics when accelerating or braking (4). The engine also provides power during coasting to recharge the battery (5). This prevents the occurrence of uneconomical operating points.

It is also possible to use all of the power sources simultaneously for driving. This makes clearly higher overall power available, which is sufficient to ensure sporty driving.



❺ Speed/torque developing during a load change in parallel hybrid mode

SERIAL HYBRID MODE

Serial hybrid mode will be used when the battery is discharged (or almost discharged) at low vehicle speeds. This mode provides a poor degree of efficiency as compared to the parallel mode and therefore it is only used sparingly.

The generator starts the engine similar as in parallel mode, but it is not coupled to the transmission by the dog clutch. The output power from the generator will be converted into electrical power and used for recharging the battery and for electrical driving.

For the optimisation of the engine operating points NVH aspects are taken into account. For example at low driving speeds with low loads, high engine speeds or high engine loads are not used for a consistent system behaviour.

SUMMARY AND OUTLOOK

Using the testing car, Getrag could show that the Boosted Range Extender concept fulfils the important attributes regarding driveability, dynamics and gradeability. Excellent gear shifting quality can be exhibited as a result of the good controllability and the dynamics of the electric motor, despite the huge ratio step. The testing car has shown very agile properties in parallel hybrid mode, as the low power engine is supported by the dynamic electric motor.

In a first practical trial held at Großglockner the Boosted Range Extender proved its capabilities also under difficult driving conditions. In a next step the predicted emission value of only 36 g CO_2/km in NEDC shall be verified.

All-in-all the Boosted Range Extender is a fully-fledged powertrain. It fulfils the requirements of a pure electrical car as well as those of a conventional car.

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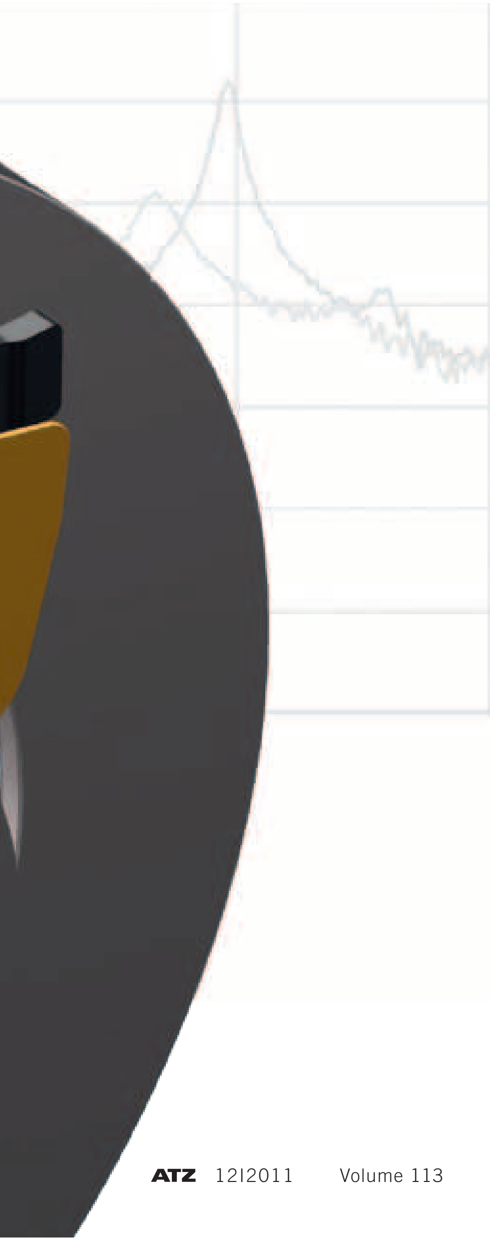
USE OF SIMULATION FOR THE NVH DEVELOPMENT OF BRAKE PADS

The function of a brake as a safety-relevant system is taken for granted. This is why customers are increasingly focusing on aspects such as noise emission and driving comfort as determinants of the perceived driving experience. TMD Friction developed the “Structured Approach to Noise Testing” in order to meet these specific demands on modern brakes.



AUTHORS

DIPL.-ING. WOLFGANG HOFFRICHTER
is Specialist NVH at TMD Friction in Leverkusen (Germany).

DIPL.-ING. ENRIQUE WEGMANN
is Specialist Simulation at TMD Friction in Leverkusen (Germany).

DR.-ING. AXEL STENKAMP
is Manager Physical Research & NVH at TMD Friction in Leverkusen (Germany).

MOTIVATION

As the automotive industry developed, so did the brake systems of passenger cars [1]. A short braking distance alone, however, is only one of various and to some extent contradictory development goals which go far beyond the safety aspect. In particular the ongoing developments in terms of new drive concepts will require additional innovations in the field of braking technology too.

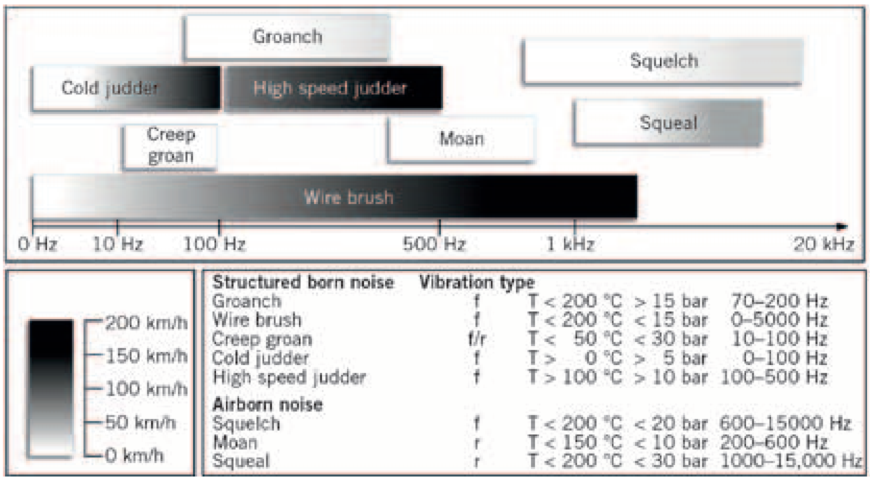
Thus, besides ecological and safety criteria, comfort aspects are increasingly coming to the fore. NVH describes a vehicle's entire comfort behaviour in terms of noise, vibration and harshness during operation. According to the J. D. Power study conducted in 2006, approximately 7.7 % of the customer statements refer to abnormal behaviour of the brake. Approximately 39 % among these customers describe NVH phenomena [2].

CLASSIFICATION OF NVH

A wide range of noise phenomena are to be found in the brake systems established on the market. ❶ classifies the typical NVH effects of a friction-based brake according to the frequency band and the required operating conditions such as temperature, brake pressure and the potential speed in which these phenomena can be observed [3]. The brake NVH phenomena perceivable in vehicles may have manifold root causes, but they all have one thing in common: they are vibrations within the frequency range potentially audible to the human ear that are locally induced during braking.

One major criterion for distinction in the NVH approach is the way in which the vibration is transmitted, hence the names airborne vibration and structure-borne vibration. Structure-borne vibration is introduced into the vehicle body by the brake via the axle. From there it is emitted into the vehicle interior. The vehicle's transfer function is critical to the perception of NVH phenomena and is mainly determined by the damping of the vehicle body and the vehicle body mass: Potentially lighter vehicles such as small cars are more susceptible to structure-born NVH.

In the case of airborne noise, a resonant vibration of large amplitude is pro-



❶ NVH frequency classification

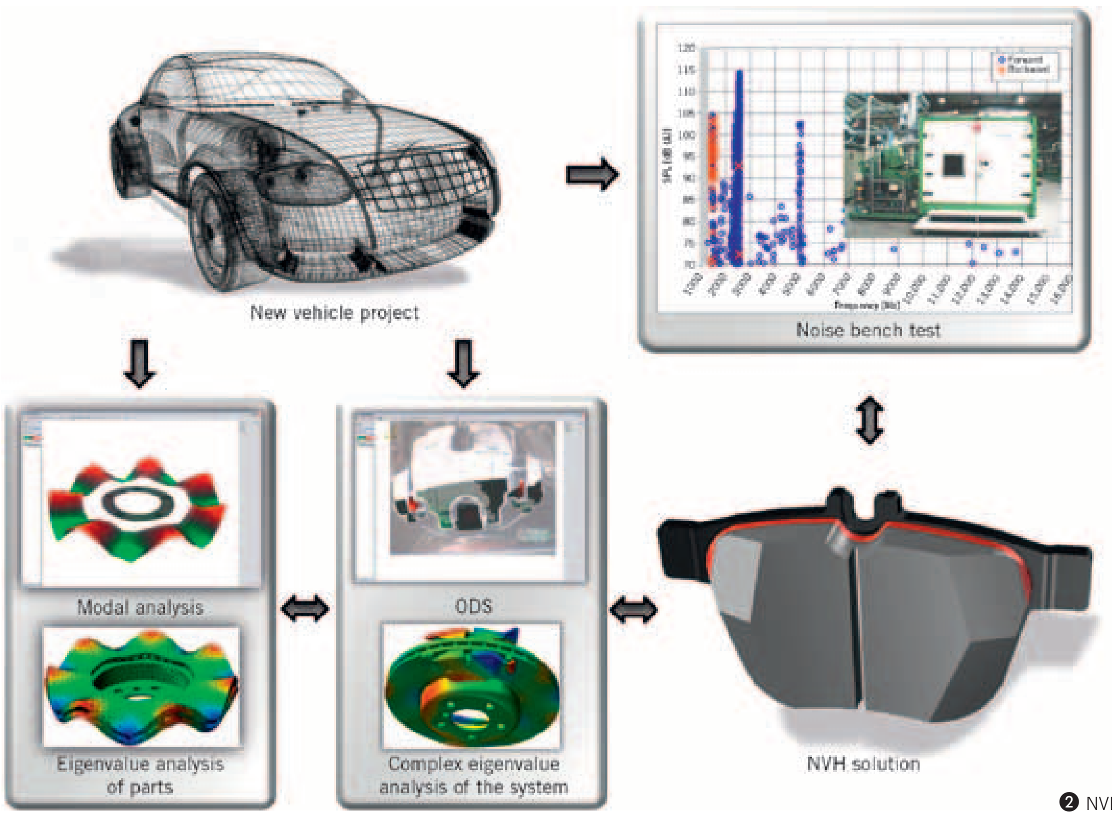
duced in the brake system, directly emitted by the brake and transmitted through the air to the ear. Brake squeal with a frequency of 1000 Hz to approximately 16.000 Hz is a typical airborne noise phenomenon.

Another distinguishing factor is the type of vibration. Here we can distinguish between self-resonant vibrations (①, mode shape r) such as squeal or moan and forced vibrations (①, mode shape f) such as wire-brush or judder.

TOOLS FOR ANALYSIS, TESTING AND DEVELOPMENT

Vibrations during braking are caused by a broadband excitation of the complete system occurring during the friction process. Depending on the specific system characteristic, this natural frictional vibration may increase to the extent that NVH phenomena occur. For this reason, the main aim in the preliminary development phase is to develop friction pairings which cause as little excitation as possible during braking. However, this entails a conflict of goals between friction coefficient and excitation. In most cases, brake pads with a friction coefficient that is high and stable in terms of temperature and velocity are prone to increased excitation. However, pads of this kind are highly popular owing to the pleasant brake pedal feeling during deceleration as well as the more compact size and the lighter weight of some of the brake components (for example the brake booster).

TMD has developed a “Structured Approach to Noise Testing” for the structured execution of any new vehicle project, ❷. Beginning with a free-free measured modal analysis of the individual components, the test benches mainly used in the development of brake pads are noise test benches with complete axle configurations. These noise test benches allow vehicle simulations to be performed in coordination with the customer using artificial matrix



❷ NVH Structured Approach

❸ Configuration of a brake pad

tests or natural endurance test cycles. Different climatic conditions are simulated by means of powerful air-conditioning systems which reconstruct extreme climatic conditions such as high atmospheric humidity (80 to 100 %) or below-freezing temperatures (approximately -10 °C). By default, these test bench simulations include activities such as the recording and analysis of microphone signals in order to describe the specific NVH behaviour.

Operating deflection shape analysis (ODS) using a scanning laser vibrometer is a complementary technique to the modal analysis of the single components which provides a better understanding of the existing mode shapes and deformations in the system during deceleration with an NVH phenomenon.

The conventional metrological analysis methods described in the context of the Structured Approach are uniquely intended to describe the present vehicle system and to develop, in an iterative process, target-oriented noise optimization measures. Moreover, CAE has become an additional important development tool for NVH optimization in the recent past.

In parallel to test bench and vehicle tests, CAE is increasingly being used as an NVH development tool. Finite element (FE) simulation supports the NVH development with respect to the friction lining in order to provide a better understanding of noise phenomena. This simulation allows the three-dimensional deformations and vibrations of the component to be observed in a manner which would require considerable effort using measurement techniques. It is for example used to perform the eigenvalue analysis of the different components and complex eigenvalue analyses of the simplified brake system piston-pad-disc.

The starting point for these simulations is the determination of the material parameters of the different components by adapting the model results of the modal analysis. In addition to the density, five independent elastic constants each for the friction material and the underlayer are adapted for the pad. Furthermore, by measuring the partial compression on the pad in axial direction, the non-linear rigidity of the material is deducted, and this is of relevance for the complex eigenvalue analysis of the brake system piston-pad-disc [4, 5].

The adaptation of the FE-model for example allows a range of geometrical changes made to the pad to be compared. The simulation results thus efficiently provide the NVH expert with a basis for decision-making.

NVH PRODUCT OPTIMIZATION IN THE DEVELOPMENT OF FRICTION MATERIAL

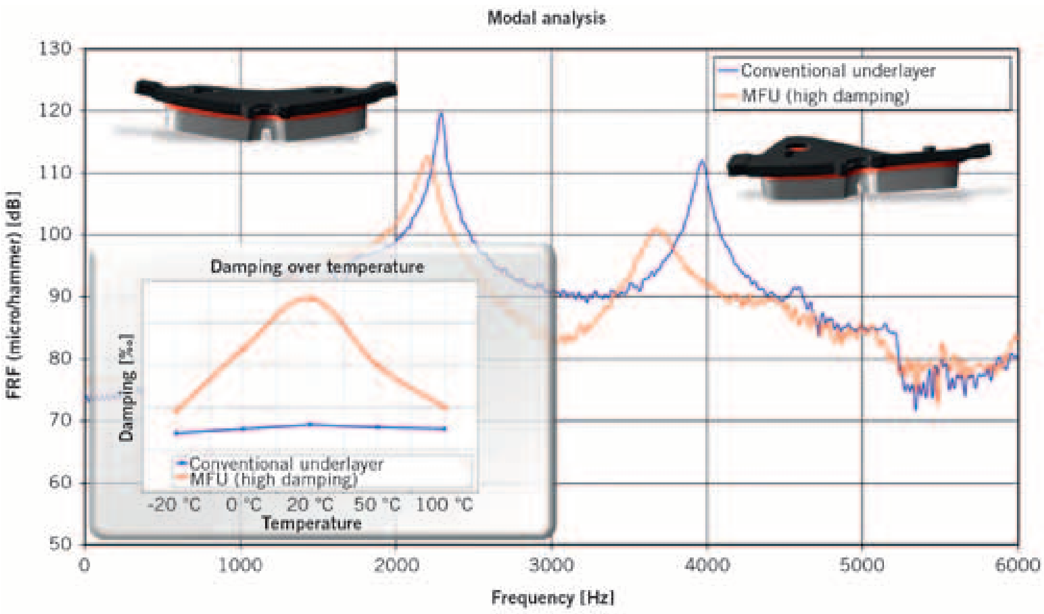
In the course of a vehicle project, TMD as a manufacturer of friction material primarily focuses on selecting friction materials and combining them with a wide range of primary and secondary measures to optimize comfort. Secondary solution measures can contribute to the noise desensitization of brake systems. They for example include damping shims, ❸, on the backplate. The purpose of the damping shim is to withdraw vibration energy from the brake system by additional damping on the one hand and to decouple the component friction lining from vibration on the other.

The term "primary measures" describes all NVH relevant measures that are directly associated with the core components of the product "brake pad", ③. They include the pad geometry, the friction material and the underlayer. Whilst the physical properties of the underlayer remain constant over the wear life of the pad, they change at the friction surface layer over the lifetime. This is due to factors such as wear-induced reduction in thickness and the load-specific conditioning of the friction lining. The underlayer has complex tasks to perform in the pad. It serves not only as a thermal insulator but is also required for the redistribution of stress between friction material and backplate and the comfort behaviour during braking [6].

The multifunctional underlayers (MFU) newly developed by TMD Friction effectively aim at achieving "zero noise emission". The MFU eliminates not only narrow-band resonant squeal but also broadband forced NVH phenomena such as wirebrush without affecting the temperature stability of the brake pad or the pedal feeling. Thanks to the relatively high damping capacity and elasticity in the multifunctional underlayer, the absorptive capacity of the vibration energy produced by friction can be ensured over the entire service life of the brake pad. ❹ illustrates the behaviour of the damping for a complete brake pad with MFU compared to brake pads with a conventional underlayer. Depending on temperature a multifunctional underlayer has a damping coefficient that is up to three times higher than that of conventional underlayers.

The damping values measured can be included in the simulation as additional material properties to support the selection process of the different material combinations for the underlayer and the friction material. ❺ is an example showing the result of a complex eigenvalue analysis from FE simulation for a given friction coefficient. The damping characteristic of the 2 mm thick underlayer was varied in the simulation.

In the present simulation, the positive real values are a measure of the eigen-



❹ Comparison of the underlayers

mode's tendency towards instability and thus a potential NVH phenomenon. The higher the value from the simulation result, the higher the theoretical probability that this eigenmode will produce a noise (squeal). In the case shown here, the high damping of the multifunctional underlayer has a system-stabilizing effect and reduces the instable eigenmodes from six to only one potential phenomenon. This simplified simulation model consisting of piston, pad and disc as well as a model fitting for each component to reflect measured values of the real system serves to support the development process.

CONCLUSION

The "Structured Approach to Noise Testing" developed by TMD Friction is a tool allowing the brake pad to be effectively tailored to the system-specific requirements of any new vehicle. Finite element simulations support brake pad development and offer a wide range of options for three-dimensional system analysis. In pursuit of our goal of achieving zero noise emission, we not only efficiently use a set of geometric and secondary measures; we also use the multifunctional underlayers (MFU) developed by TMD Friction which eliminate various broadband NVH phenomena effectively and permanently.

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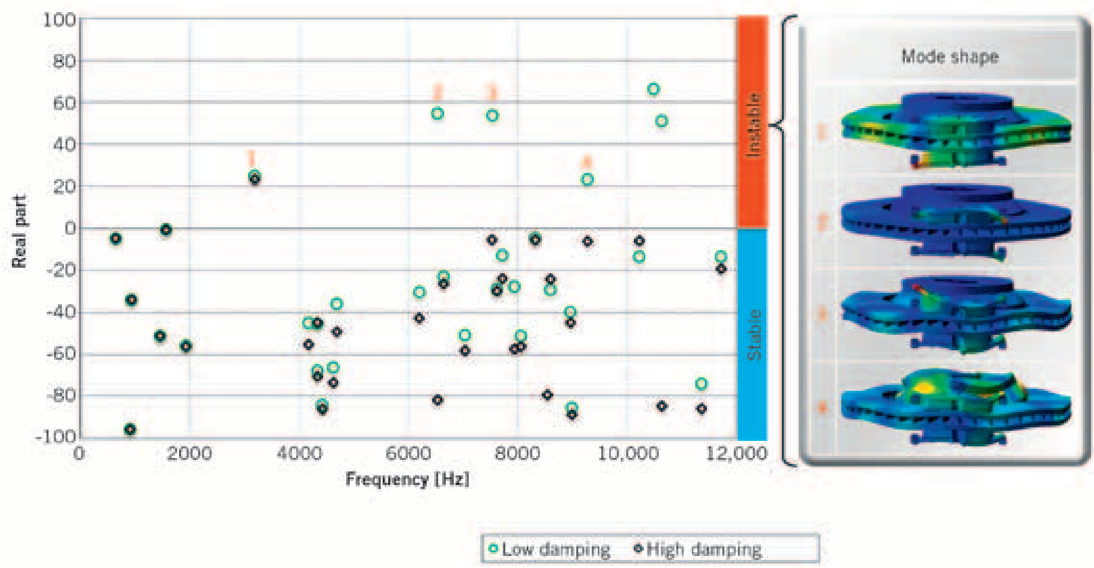
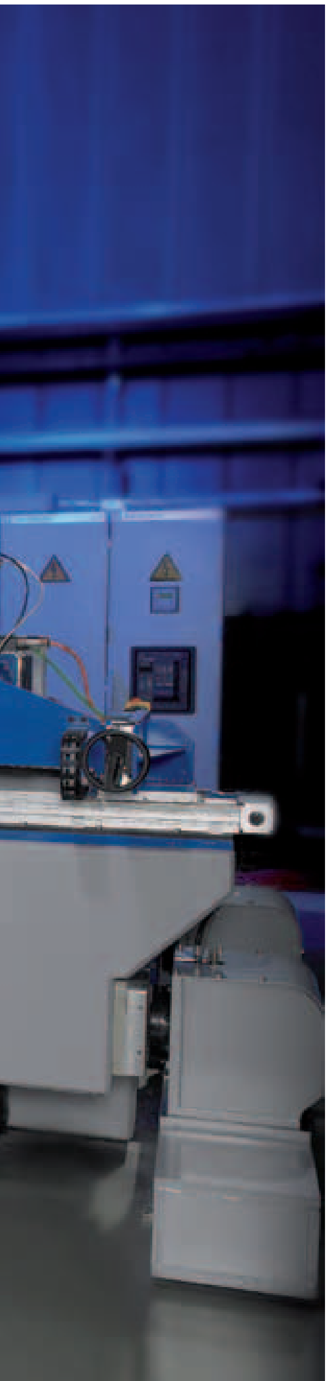


❺ Complex eigenvalue analysis with and without highly damping underlayer

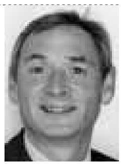
MEASUREMENT OF ROLLING RESISTANCE AND ENERGY EFFICIENCY OF CAR TIRES

The EU regulation 1222/2009, which will come into force in 2012, specifies that car tires must be marked with a label in a user-friendly manner. As the energy efficiency mainly depends on the rolling resistance, standardized measuring methods will be necessary to categorize the tires. The Test Systems business area of ZF Friedrichshafen AG has therefore developed test benches for measuring rolling resistance. These are based, with regards to torque methods, on a precise measurement of the torque with HBM torque flanges.



AUTHORS

DIPL.-ING. (FH) ROLAND BÖSL is Sales and Product Manager for Test Systems of Wheels and Tyres at ZF Friedrichshafen AG in Passau (Germany).



DIPL.-ING. KLAUS MEIER is Account Manager for ZF in Passau and Specialist for Torque Measurement at HBM Hottinger Baldwin Messtechnik GmbH in Darmstadt (Germany).

EU REGULATION 1222/2009

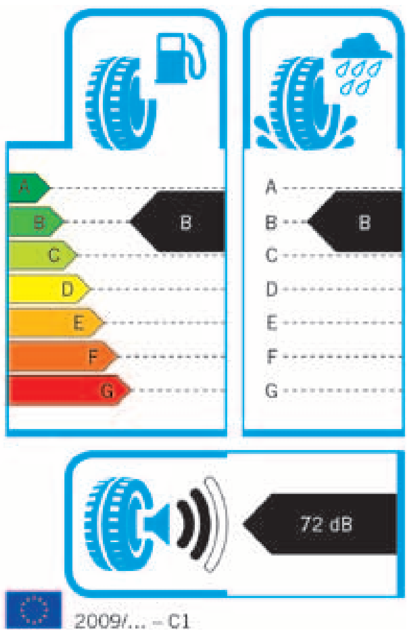
The declared aim of the EU is to reduce CO_2 emissions by at least 20 % by the year 2020. A significant proportion of the CO_2 produced in the EU comes from road traffic. Vehicles with low fuel consumption contribute directly to a reduction of CO_2 emissions. In total, 20 to 30 % of fuel consumption by vehicles is due to the tires – mainly because of the rolling resistance. A reduction of tire rolling resistance can therefore contribute significantly to energy efficiency in road traffic and therefore also to a reduction in pollutant emissions.

The EU regulation 1222/2009 proposes a classification of rolling resistance via label similar to that of household appliances, ❶. A buyer should primarily read, in addition to the rolling noise and the wet adhesion properties, the energy efficiency of the tires. The EU assumes that this will give the buyer the incentive to buy particularly fuel saving tires. This will in turn put pressure on tire manufacturers to develop and offer tires accordingly.

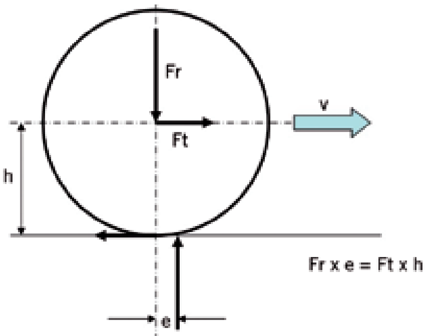
METHOD OF MEASUREMENT FOR ROLLING RESISTANCE

The classification of tires according to rolling resistances presupposes that rolling resistance can be precisely measured. The rolling resistance results from the deformation of the tire as it rolls over a surface, ❷. Test benches that can measure rolling resistance consist of a wheel suspension which presses the tire with a defined force F_r onto a rotating drum. This is driven by an electric motor that is controlled in such a way so as to constantly maintain a specified speed v.

In principle there are three methods of measuring the rolling resistance of a tire on a test bench. The various methods for rolling resistance measurement are described in ISO 28580, to which the EU regulation also refers. In the so-called deceleration measurement, the time taken until the tire coasts down to a standstill from a defined speed is measured. This first method is very sim-



❶ The new EU labeling for car tires is based on the well-known labels for household appliances



❷ The rolling resistance of a car tire is due to the tire deformation and is, amongst other things, dependent on the wheel load F_r

ple from a measurement technology aspect as only time needs to be measured. However, this method is very time-consuming when determining the rolling resistance over a specified speed profile as the tire must coast down each time and then be accelerated again to the new speed value.

The second method measures the forces that occur at the wheel hub. The vertical force component F_t is required to determine the force with which the tire presses on the drum. The tangential force component that is necessary to hold the wheel in position provides a value for rolling friction. The problem with this process is the complete decoupling of both components. The vertical force component is approximately 5000 N while the tangential component is only approximately 50 N. Crosstalk in the magnitude of just 1 % would therefore significantly invalidate the measurement result.

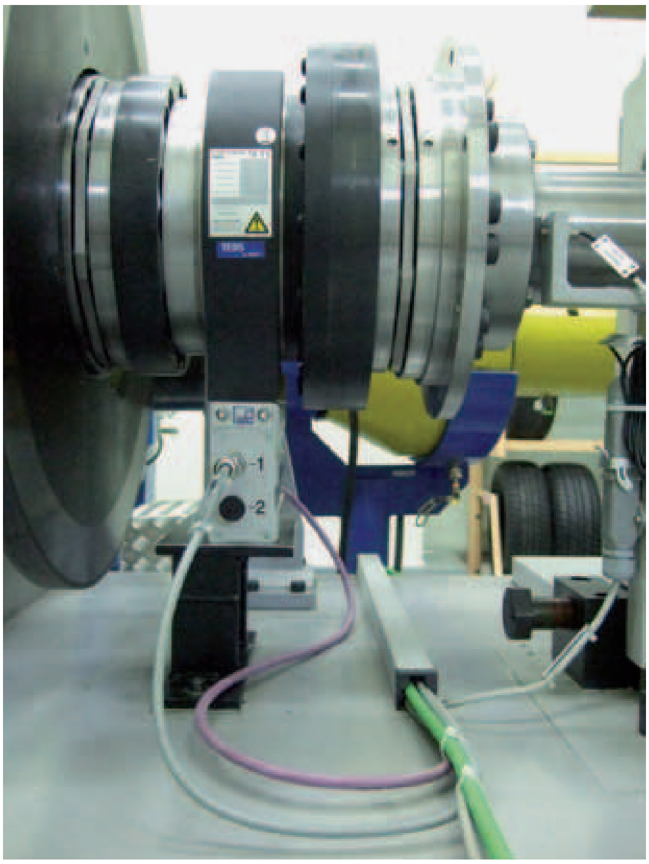
A third method – the so-called torque method – avoids these problems and directly measures the torque that is necessary to keep the speed of the drum, and therefore the tire, constant. A measurement flange mounted between motor and drum determines the torque. The vertical force component is instead measured at the wheel hub.

Another method is in principle possible where the performance of the electric motor is recorded at the inverter of the drive. However, this is actually only an auxiliary construct again aimed at measuring the supplied torque together with the speed. This method has very many dependencies and is therefore extremely imprecise. Calibration is also very difficult.

MODERN TEST BENCH ENGINEERING

With the new EU regulation, there is an increased demand within the industry for tire test benches that are capable of measuring the rolling resistance accurately. Thus, the Test Systems business area in Passau of ZF Friedrichshafen AG has concerned intensively with the measurement of rolling resistance. The measurement of the absolute value for rolling resistance is crucially significant here. Previous test benches were capable of testing various

❸ Tire test benches by ZF measure both the tangential force on the wheel hub and the torques up to 10 kNm transferred from the drive to the drum



❹ The T12 measurement flange from HBM records the current torque and digitalizes the measured value directly in the rotor; the data is then transmitted via a Profibus interface tires in comparison to each other in order to determine which tires had the lowest rolling resistances – but an absolute measurement was only possible to a certain extent.

The tire test benches at ZF measure both the forces at the wheel hub and the torque that must act on the drum to keep the speed constant, ❸. The drum has a diameter of 2 m and is made of aluminum. This has a lower inertia torque compared to drums made of steel. The drive therefore only has to apply a lower torque to accelerate the drum. The electric motor used has a maximum torque of 500 Nm. The torque method has advantages, with reference to measuring the absolute value. A precise alignment between tire and drum is extremely important for the force measurement. Even a small deviation leads to errors in the measurement as the tire is then essentially running "downhill" and so an additional component appears in the form of a downslope force. The torque method based on the machine alignment is therefore more easier to control.

The specialists at ZF are using an in-house developed wheel force dynamometer with hydrostatic bearings. These ensure that crosstalk does not occur between the vertical and tangential components and that a very high measurement accuracy is reached due to friction freedom. A T12 measurement flange from HBM is installed between the electric motor and drum to measure the torque. The measurement flange, which has a nominal (rated) torque of 500 Nm can be operated with speeds of up to 18,000 rpm. The accuracy of the torque measurement is very high. Both the linearity error and the relative standard deviation of the repeatability are just 0.01 %.

MEASUREMENT FLANGE FOR TORQUE DETERMINATION

The T12 torque measurement flange, ❹, consists of a stator and a rotor in which the actual measurement element is integrated. The amplifier is also integrated in the rotor to evaluate the measured signals and transmit them digitally to the stator. The 19 bit resolution provides correspondingly high accuracy. The measured value can then be optionally transmitted onwards via Profibus or CANopen with a rate of up to 4800 measured values per second. Alternatively, a frequency or analog output voltage is available with 0 to 10 V. The energy supply and data transmission between rotor and stator is without contact.

In addition to the high accuracy of the torque measurement, it is this digital data transmission that is a decisive advantage of the T12 measurement flange in comparison to other methods. With this data transmission character, interferences on the signal lead, which can occur in analog data transmission, are completely excluded. The Profibus interface ensures that the measurement flange can be easily integrated in the automation architecture of a test bench. An S7 PLC from Siemens is used for the control system, this controls all processes. A Simadyn module in the PLC is responsible for regulating the drum speed.

The forces in the wheel hub developed by ZF are measured with force measurement cells. MGCplus by HBM measuring amplifiers digitalize the signals of these force transducers and provide them to the system via Profibus. This system also measures the temperature and the air pressure in the tires. Both parameters play an important role in rolling resistance measurement and must therefore be known precisely. The measurement specifications in ISO 28580 set out a precise method for this. The tire must be kept at a temperature of 25 °C for at least three hours before the measurement. The tire must also be filled with precisely specified air pressure for the measurement.

In order to implement a rolling resistance measurement, the tires are pressed with a force of approximately 5000 N – depending on the tire type – onto the drum. This is then accelerated to the required speed. The drum drive is then controlled so that the speed remains constant. The torque necessary for this is measured, resulting in the rolling resistance. In order to eliminate effects caused by friction in the drum bearings and the wheel hub, or effects due to air resistance, the measurement is then repeated with a very low wheel load. This so-called skim load is just high enough to move the tire. Subtracting this value allows the actual rolling resistance of the tire to be determined.

The torque method has significant advantages for the precise measurement of tire rolling resistance. In particular, a slight offset between the drum and tire axes does not lead to a measurement error as is the case in the force method. In addition to the high accuracy of the measurement flange, it also offers digital data transmission where interference during transmission is excluded. The ZF test benches master both methods.

SPECIAL APPLICATION FOR BMW

ZF Passau GmbH has developed and built a universal test bench for tires for the automotive manufacturer BMW. This can be used to measure the rolling resistance of the tire and other important parameters. These include the so-called High Speed Uniformity tests in which the behavior of the tire at high speeds is tested.

However, in contrast to a pure rolling resistance test bench, a drive with a significantly higher torque of up to 10 kNm is necessary. A T12 measurement flange, ④, with a nominal (rated) torque of 10 kNm was used for torque measurement in this test bench. Despite the very high nominal (rated) torque, the measurement flange can reproducibly measure the torque down to 0.1 Nm. This means a precise measurement of the rolling resistance is possible here just like in other tire test benches.

A DRIVER MODEL FOR VIRTUAL CONTROL SYSTEM DEVELOPMENT

Virtual road tests for the functional safeguarding and application of vehicle control systems place high demands on the driver model used. The driver model developed by Tesis Dynaware previously determines realistic targets and switches between open and closed loop control depending on the driving situation. It is a prime example of the development and preliminary adjustment of traction control systems for all-wheel drive vehicles at Magna Powertrain.

AUTHORS

DR. RER. NAT. RAINER FISCHER
is Project Manager for Driver Model Development at Tesis Dynaware GmbH in Munich (Germany).

DR.-ING. TORSTEN BUTZ
until September 2011 was Head of Product Development at Tesis Dynaware GmbH in Munich (Germany).

DIPL.-MATH. MARTIN EHMANN
is Product Manager at Tesis Dynaware GmbH in Munich (Germany).

DIPL.-ING. MARIO VOCKENHUBER
is Development Engineer for Controlled Powertrain Systems at Magna Powertrain AG & Co. KG in Lannach (Austria).

THE PROBLEM

The consistent use of simulation methods at Magna Powertrain allows real road tests to be carried out more efficiently and better results for adjustment variants for traction control systems to be achieved according to customer specifications [1]. For this purpose, the simulation framework Dyna4 from Tesis Dynaware [2] provides models of vehicle, driver and environment which have high mapping quality but low computing time requirements and permit good virtual preliminary application of the all-wheel control units.

Tests based on driving tasks can be put together from any desired sequence of open- and closed-loop manoeuvres, iterative simulations and automated evaluations of the results. This allows variant calculations, controller verifications and virtual adjustments in function development and even control unit testing in Hardware-in-the-Loop (HiL) environments.

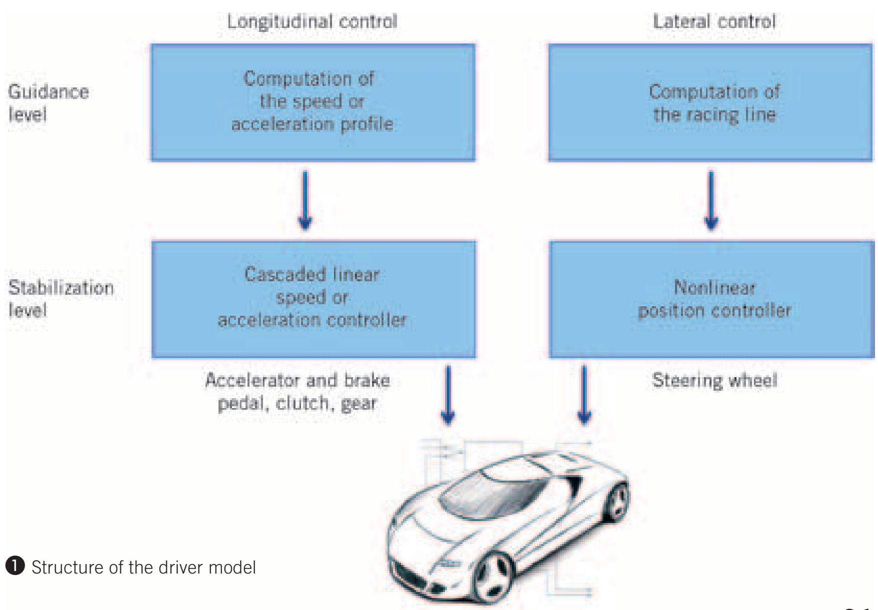
VIRTUAL TEST DRIVER

The implementation of virtual road tests places high demands on the driver model used. This must provide optimum setpoints for the test task in each case and ensure the longitudinal and lateral guidance of the virtual vehicle. The mapping of a wide range of plausible driver behaviours is achieved including realistic action of actuating variables and full use of the traction potential.

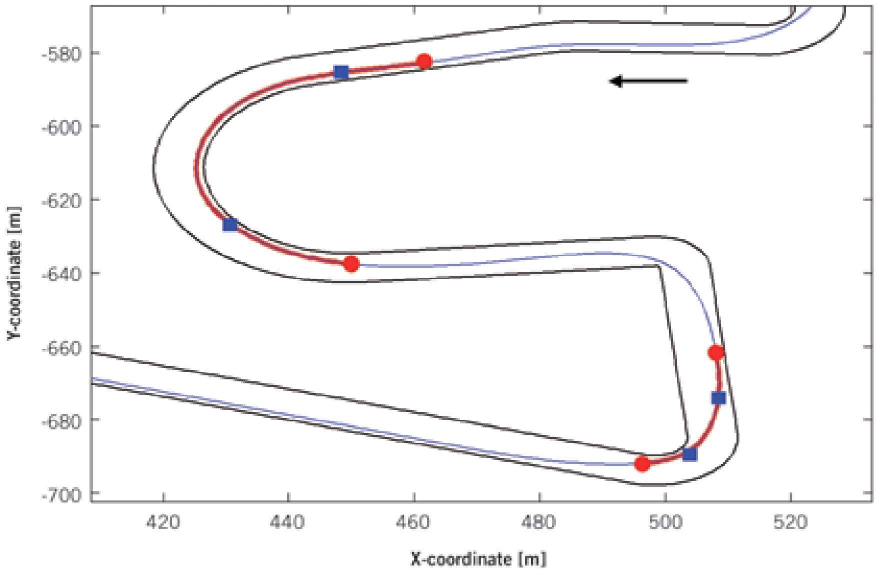
The Dyna4 test driver is a virtual driver model which can be universally used for a wide range of test scenarios ranging from standard and handling tests to extreme driving situations, the simulation of fuel consumption cycles, parking or realistic reactions to surrounding traffic. The modular structure, ❶, is based on the two-level model by Donges, which has been expanded to include a strict separation between longitudinal and lateral guidance and between setpoint calculation and actual control [3].

The specified course for lateral guidance is determined dynamically during the simulation or in a pre-processing step carried out beforehand. This is done via a heuristic method which uses the expert knowledge of test drivers to calculate approximations of time-optimal ideal lines for handling courses and mountain passes [4]. Curves and sequences of curves are identified and optimum trajectories with regard to the acceleration behaviour are computed. At very short computing times, this procedure provides good correspondence with trajectories actually driven and better results than the driver models of the competitors. Extreme driving situations such as those which are for example necessary for control system intervention on the part of an ESP control unit can be simulated in a reproducible way in the HiL test.

The non-linear position controller used for lateral control is based on the theory of non-linear system decoupling and control. Various driver types can be mapped by varying characteristic parameters such



❶ Structure of the driver model



❷ Phase subdivision for cornering on a handling course

as the distance for which the driver can see up ahead.

Depending on the driving situation, the control variables for the longitudinal guidance of the vehicle are determined by means of a cascaded speed or acceleration control which compensates for the non-linearities in engine and powertrain via inverse engine characteristic maps and gain scheduling controllers. The determination of realistic settings for the longitudinal guidance is described in the following section.

MANOEUVRE SUBDIVISION FOR THE LONGITUDINAL GUIDANCE OF THE VEHICLE

The specification of a target speed profile for longitudinal guidance is widespread in practise, but it is not suitable for a large number of applications. In particular when the friction coefficient is low, a pure speed control can lead to unrealistic braking and gear change operations in curves and thus stand in the way of the virtual adjustment of the traction control system to suit the human driver. The approach presented automatically carries out phase subdivision depending on the curve profile of the specified course. As with lateral guidance, curves and characteristic curve points such as the brake point and the end of the curve are identified. For each curve phase, either speed profiles or acceleration profiles for the driving controllers in question or open-loop setpoints are determined depending on the desired driver type.

By way of an example, the manoeuvres “braking” and “operation of the accelerator pedal followed by driving at a constant speed in curves” are outlined for the two curve sections marked, ❷:

: The beginning of the brake phase (red circle) is determined from the specified end of the brake phase (blue square) and the acceleration limits typical for the driver. A deceleration controller is used for slowing down to the desired curve speed. At the end of this phase, the suitable gear for negotiating curves has been engaged.

: Braking and gear change operations are not permitted when driving at a constant speed. The curve speed is determined from the maximum curvature and the lateral acceleration limits typical for the driver.

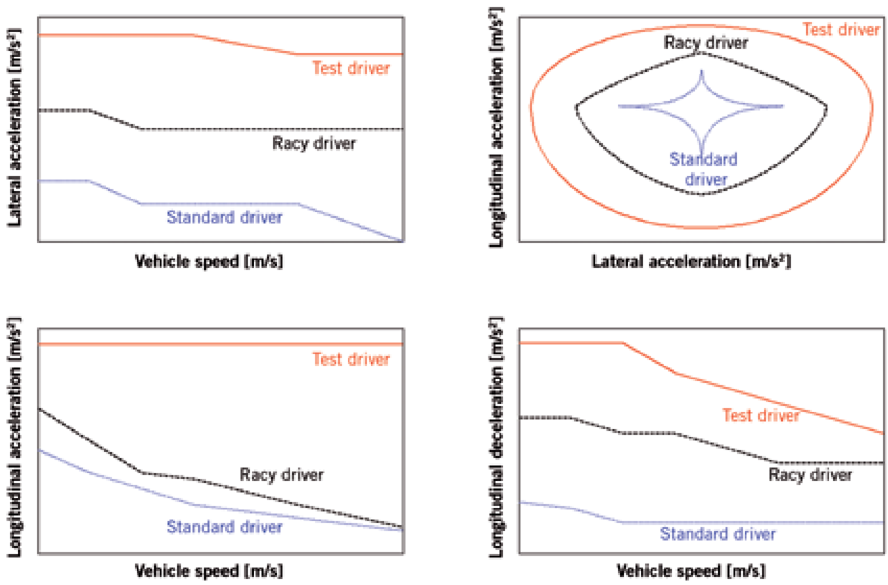
: The beginning of the acceleration phase (blue square) is defined by falling below a lateral acceleration limit when driving at a constant speed. An acceleration controller is used for accelerating until the end of the curve is reached (red circle).

A suitable speed profile is used for the transition between the curves.

MODELLING DRIVER TYPES

Different driving styles are mainly mapped via three-dimensional g-g-v diagrams which reproduce the longitudinal and lateral acceleration limits of different driver types such as standard driver, racy driver or test driver depending on the driving speed, ❸. This also allows degressive acceleration processes and progressive braking processes etc. to be modelled.

The characteristic curves for the acceleration [5] reflect the fact that racy drivers and test drivers achieve much higher lateral and longitudinal accelerations than the average driver does. The longitudinal acceleration and the longitudinal delay tend to decrease as the speed increases,



❸ Speed and acceleration characteristics for average drivers, sporty drivers and test drivers with the precise reduction varying as a function of the driver type.

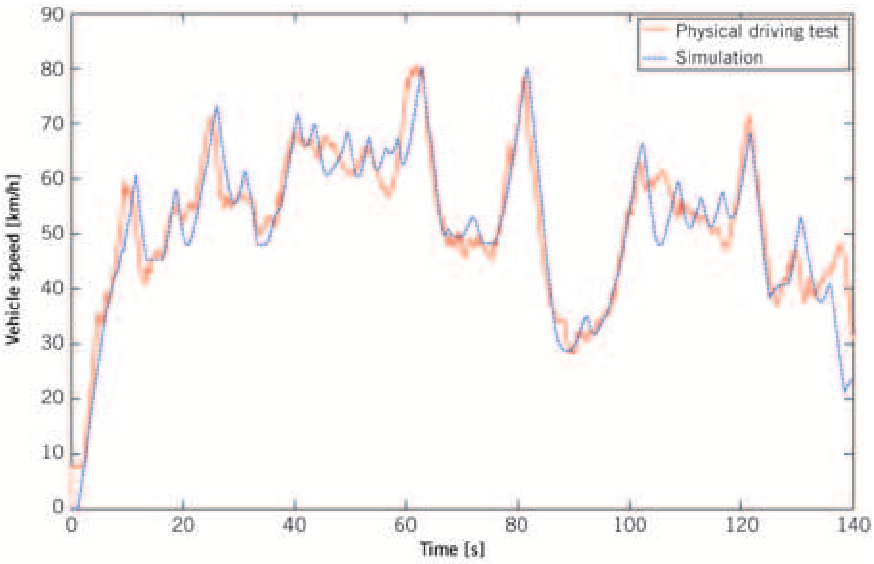
In addition, other characteristic parameters for the subdivision of manoeuvres are used to distinguish between different driving styles. Some examples of this are the end of the brake phase, which is later in the curve when the driving style is racy, the limitations on accelerator and brake pedal gradients as well as dead times at the transitions between phases.

ADJUSTMENT OF TRACTION CONTROL SYSTEMS

In the development of traction control systems at Magna Powertrain, virtual function development is becoming more and more important, and this also includes the reduction of CO_2 emissions [6]. To qualify for series production, the control algorithm has to fulfil a number of criteria from the fields of driving dynamics, traction, fuel consumption and system load. Different adjustment variants can be alternately tested for high and low friction values in the simulation, thus allowing adjustment to be carried out with a high confidence level. As a result of the humanoid model for the longitudinal guidance of the vehicle, even control logics which adapt to the driver behaviour can be tested realistically.

Low friction coefficient conditions are a particular challenge. Here the virtual driver has to achieve a stable driving behaviour even in extreme situations and is only permitted to operate the accelerator pedal and the brake in a moderate, targeted way when the vehicle goes into a skid. The driving speeds for a handling course largely show a correspondence between measurement and simulation, ❹. The phase-by-phase manoeuvre definition depicts the sensitive, dynamic behaviour of an experienced test driver in the operating states braking, load change and operation of the accelerator pedal followed by driving at a constant speed in curves.

Every time the control system is adjusted, the simulation allows the fuel consumption, the driving stability and the traction to be determined in a good correspondence with the test drive. The load on the powertrain results from the effects of the forces and torques on the mechanical components, such as the friction work occurring in the all-wheel clutch and the permanent torque in the axle transmission, which is equivalent to damage. Via suitable weighting, this is included in the total sample as a basis for the evaluation of the operational stability. Tool support from Dyna4 thus allows simulations to be carried out for a wide range of vehicle setups, the results of calculation and the quality of the adjustment variants to be evaluated and documented.



❹ Driving speed during test drive and simulation on a handling course

SUMMARY

The increasing number and complexity of mechatronic control systems in vehicles means that they can no longer be developed and adjusted without the assistance of simulation models and tools. A contribution to this is made by the driver model from Tesis Dynaware, which allows various driver types to be simulated. In the evaluation of traction control systems for all-wheel vehicles from Magna Powertrain, it was possible to successfully meet high demands such as that for control over the vehicle under low friction coefficient conditions. The results for the adjustment of all-wheel systems show good behaviour with regard to subjective evaluation by experienced test drivers and according to objective development criteria.

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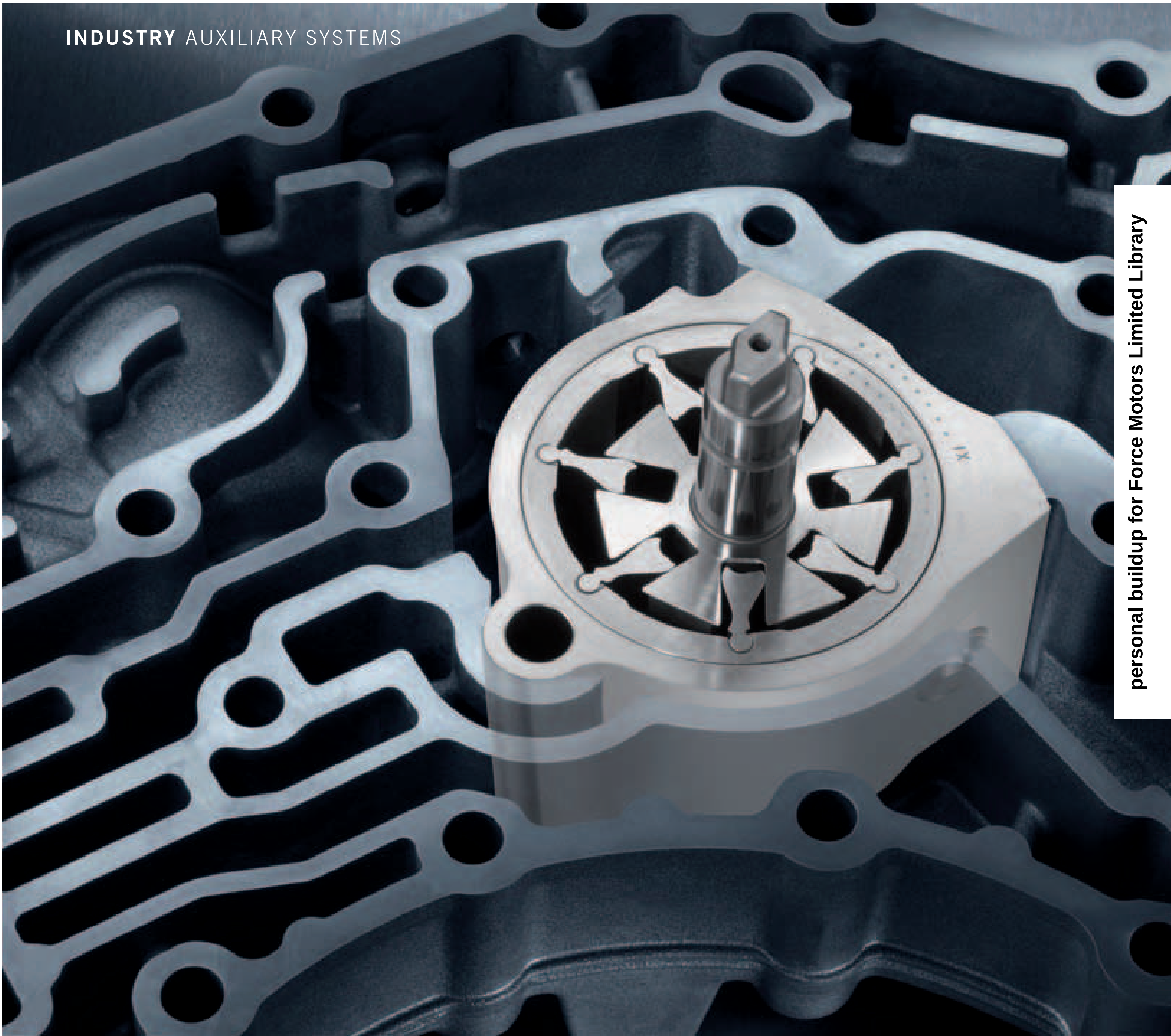


CONTROLLED PENDULUM-SLIDER OIL PUMP FOR THE SUPPLY ON DEMAND OF TRANSMISSIONS

The application of controlled oil pumps to reduce CO_2 emissions in combustion engines is currently the state of the art. In contrast, uncontrolled pump types are still commonly used in vehicle transmissions. Mahle developed a pendulum-slider pump with special control characteristics. Now, it is possible to further reduce fuel consumption in all types of powertrains with automatic transmissions.

AUTHORS

DIPL.-ING. HOLGER CONRAD
is Head of Product Development in the Segment Pumps at Mahle Filtersysteme GmbH in Crock (Germany).

DIPL.-ING. (FH) RÜDIGER KNAUSS
is Head of Pre-development Design Product Development Europe Filtration and Engine Peripherals at Mahle Filtersysteme GmbH in Stuttgart (Germany).

MORE AUTOMATIC TRANSMISSIONS

The number of vehicles fitted with automatic transmissions will increase further over the next few years. One reason for this trend is the required increase in powertrain efficiency by tuning the engine speed to the optimal shifting points and gear ratios. Another optimisation measure is the hybridisation of vehicle powertrains, which requires an automatic transmission in order to cleverly switch between combustion and electric motor operation.

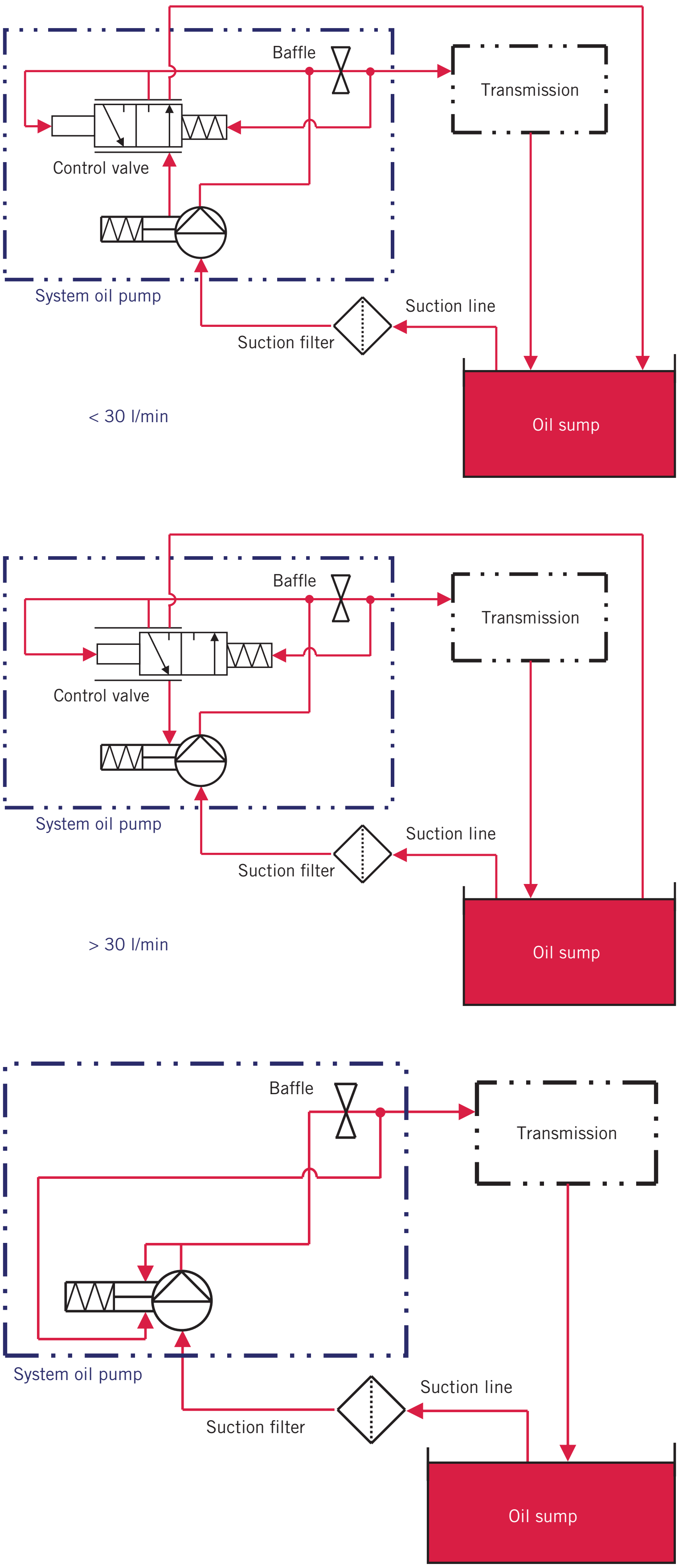
The increasing rate of automatic transmissions used in vehicles has driven demand for increased efficiency in the transmission itself. The known measures for reducing CO_2 emissions, such as improved torque converter lockup or the use of low-viscosity hydraulic oils, include optimisations throughout the transmission hydraulics. The use of controlled oil pumps for delivering oil for shifting, cooling, and lubricating transmission components is thus growing in relevance.

TYPES OF AUTOMATIC TRANSMISSIONS AND OIL PUMPS

Both in passenger cars and commercial vehicles the transmissions types
: converter stage automatic transmission (AT)
: wet dual clutch transmission (DCT)
can be found in various power classes. The oil pressure required for shifting typically ranges between 20 and 30 bar for these applications. In the passenger car market, volume flows of up to 30 l/min are generally sufficient, while a commercial vehicle transmission can require 150 l/min or more, particularly under critical thermal conditions.

One exception with respect to oil pressure for use in passenger cars is the continually variable transmission (CVT): In order to generate the necessary clamping forces on the variator, pressures of 60 to 70 bar are required. The mentioned other types of transmissions are used in conjunction with almost every known type of pump. Most of them are still not controlled.

The simple internal gear and gerotor pumps are preferably installed directly on the input shaft. The internal-gear pump cannot be controlled, due to the fixed sickle between the internal and external gears. The specific delivery volume of the



❶ Slider regulation with control valve – maximum (top) and reduced (center) pump volume; simplified slider regulation (bottom)

gerotor pump also cannot be changed geometrically. Shifting the outer ring, and thus changing the control times, allows limited control.

External-gear pumps must be driven in parallel relative to the input shaft, and driven by a chain or a spur gear. This type of pump is controlled by axially displacing the gears relative to each other, but is problematic with respect to leak tightness and hysteresis of the control device.

Vane pumps with double vanes, that means with two inlet and two outlet channels, are increasingly common in transmissions, but cannot be actively controlled, specifically with this design. Volume flow control can be achieved in these pump systems by installing a downstream control valve, which returns the excess oil back to the pump inlet. The excess quantity of oil is still pumped, however, which involves unnecessary power input.

The goal for transmission applications should be to use an actively controlled pump, which pumps only the quantity of oil that is actually required. The Mahle pendulum-slider pump is outstandingly suited for this requirement.

CONTROLLED PENDULUM-SLIDER PUMP

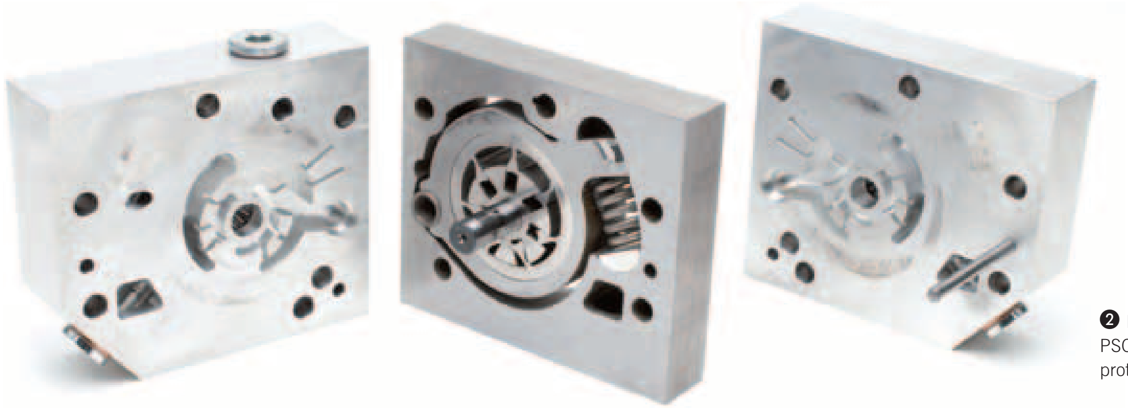
The patented Mahle pendulum-slider pump (PSC pump) has proven itself on the market for controlled oil pumps in the lubrication circuit of combustion engines [1]. The main advantages over other controlled oil pumps include:

: ability to withstand dirt (abrasive particles, soot, etc.)
: high overall efficiency over the entire service life
: depending on the size, suitable for rotational speeds of up to 14,000 rpm
: various control strategies can be used, short adjustment and response time.

The fast adjusting time and flexibility of controls also has advantages for vehicle transmission applications. As described before, the requirements for oil pressure and volume flow vary greatly depending on the transmission type and operating point.

PUMP CONTROL STRATEGIES IN TRANSMISSIONS

While the required oil pressure is the controlling variable in combustion engine



❷ Individual parts of the PSC pendulum-slider pump prototype

applications, the control of the required volume flow is often a greater priority in transmissions. The simplest method is the use of a fixed baffle at the pump outlet, which limits the volume flow to a prescribed set value for a specific operating point (temperature, viscosity).

❶ shows two pump's control strategies schematically – regarding a maximum (top) and a reduced (center) pump volume. A mechanical control valve is installed in parallel with the baffle, equalising the differential pressure before and after the baffle. As soon as the delivery rate exceeds the set value as the pump speed increases, back pressure builds up behind the baffle and causes the control valve to adjust. This opens the channel to the regulation-oil chamber in the pump, and thus allows the delivery volume to be adjusted without electronic controls.

The bottom part of ① shows a simplification of this control: the separate differential pressure valve is eliminated. Instead of that the control slider additionally fulfills the function of the valve. Eliminating the differential pressure valve has the advantage that reducing the number of spring-mass systems leads to shorter response times, a wider operating range available, as well as a simplified adjustment of the system. Should the volume flow vary with demand, then a variable throttle must be used in place of the fixed baffle. The control of this throttle is monitored by the transmission controller.

SAVINGS POTENTIAL WITH THE CONTROLLED PENDULUM-SLIDER PUMPS

In order to determine the savings potential, a comparative measurement of a prototype pendulum-slider pump and an uncontrolled double vane pump was performed. This vane pump, which is normally installed in series in a CVT, is fitted with a downstream volume flow control valve, which shunts the excess oil back to the pump inlet for delivery rates greater than approximately 30 l/min. The specific delivery volume is about 16 cm³ per revolution.

The PSC pump prototype, designed for the same specific delivery volume, is implemented as a functional prototype with a steel housing, without a separate differential pressure valve. In the functional prototype, the inner geometry of the pump is simply milled from an aluminium or steel block. Using such a prototype, the hydraulic characteristics of the pump are measured in advance, and compared to the values calculated from the design and simulation. The individual parts of the pump prototype are shown in ❷.

The PSC pump was run with the previously described volume flow control using a fixed baffle (limited to 30 l/min). Oil viscosity for the tests was 10.5 and 84 cSt, which corresponds to temperatures of 95 and 30 °C respectively for the test bench oil used. The results for power input are shown in ❸ and for efficiency in ❹ versus pump speed for pressure of 8, 30 and 50 bar.

At low oil temperatures and low pressures, both pump types are nearly identical with respect to power input, while the PSC pump shows significant power savings at speeds above 5000 rpm. At higher system pressures and above about 2000 rpm, the savings potential of the controlled PSC pump relative to the vane pump increases significantly. As the oil viscosity drops, the power input is further reduced, which can be attributed to the high volumetric efficiency, that means the high level of internal leak tightness of the PSC pump.

The characteristic curves derived for the power input are reflected in the curves of the overall efficiency of the vane pump and PSC pump. In nearly all operating ranges, the PSC pump has a greater overall efficiency relative to the vane pump (maximum 76 % at 50 bar).

An interesting aspect of both pumps is that the efficiency curves for low temperatures approach those measured at higher temperatures as the system pressure increases. This can be explained by the increasing proportion of volumetric efficiency in the overall efficiency at these operating points.

The test results for the power input, in particular, clearly demonstrate the savings potential from the use of the controlled pendulum-slider pump. For all types of automatic transmission, this allows for further potential increases in efficiency, and thus further CO_2 savings.

BI-DYNAMIC PRESSURE PUMP

The construction of a conventional pendulum-slider pump requires two delivery chambers: the main delivery chambers, between the pendulums and both rotors, and the smaller auxiliary chambers in the inner rotor, below the pendulums. Depending on the design, these can account for up to 25 % of the total delivery volume.

In the bi-dynamic pressure pump – an ongoing Mahle development of the PSC pump – these additional delivery chambers are put to use in that they are separated from the main chambers by a separate channel. This results in a second delivery stage that can be optimally used as an additional high-pressure stage, due to the low delivery volume. The main chambers, with greater volume flow, serve as the low-pressure stage. The pressure stages and delivery chambers of the rotor set are highlighted in color in ❺.

In order to reduce volumetric losses and thus achieve the required pressure build-up in the high-pressure chambers, additional sliders or pistons are attached to the feet of the pendulum, sealing off the delivery chambers more tightly than the conventionally shaped pendulum foot.

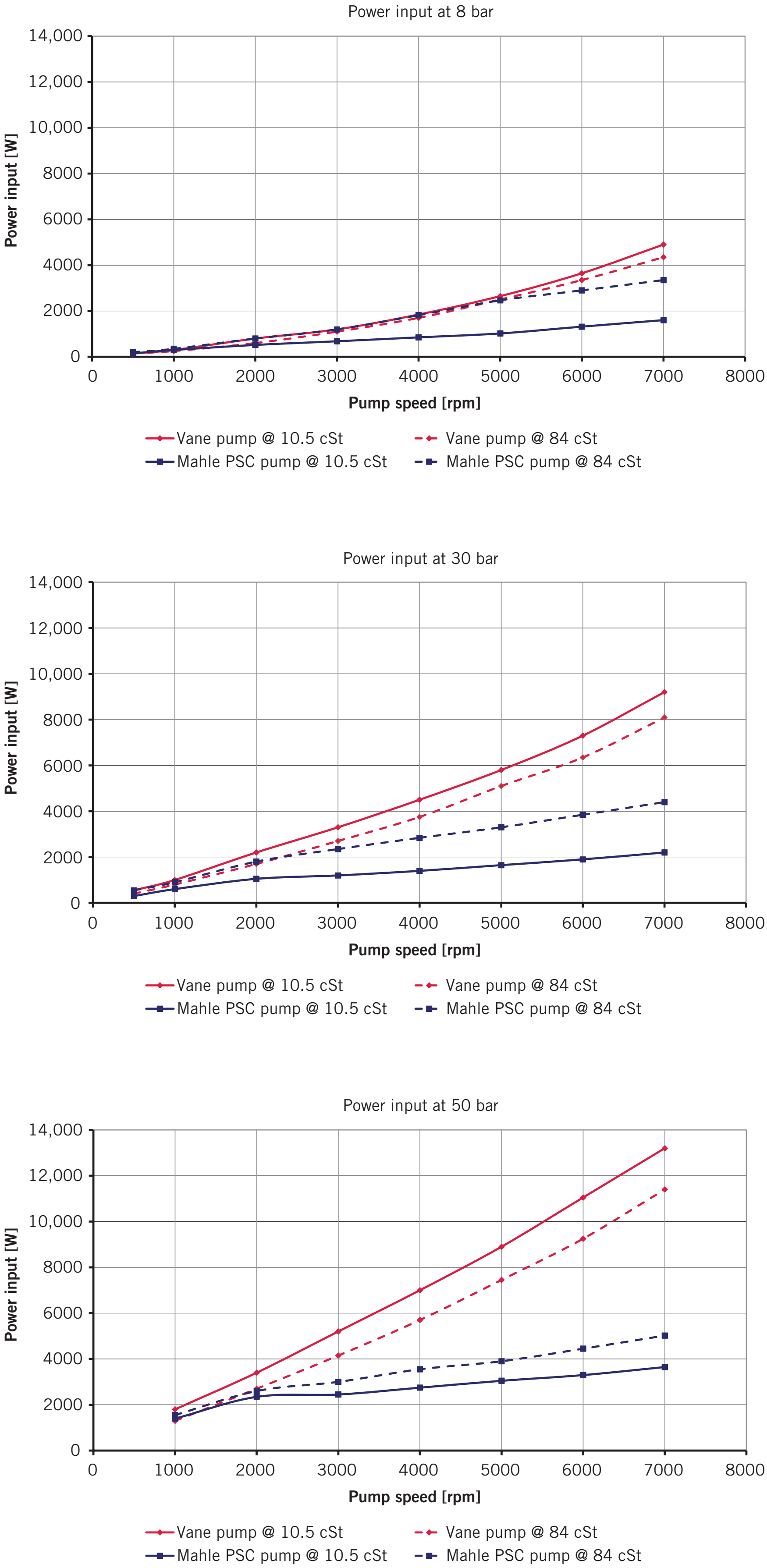
This type of rotor set still allows the eccentricity to be adjusted by means of a control slider; however, in principle, the two delivery stages cannot be controlled separately from each other. That is, a reduction in the volume flow in the low-pressure circuit inevitably means a reduction in the high-pressure circuit. This condition must be taken into consideration when designing the pump.

The prototype was designed with a ratio of specific delivery rates of about 1:10 (5.30 to 0.54 cm³). The resulting values for pressure and volumetric efficiency are promising: at a limiting pressure of 10 bar in the low-pressure stage, the delivery rate was around 20 l/min, and in the high-pressure stage, it was 2 l/min at 60 bar (at 3750 rpm, oil temperature 30 °C).

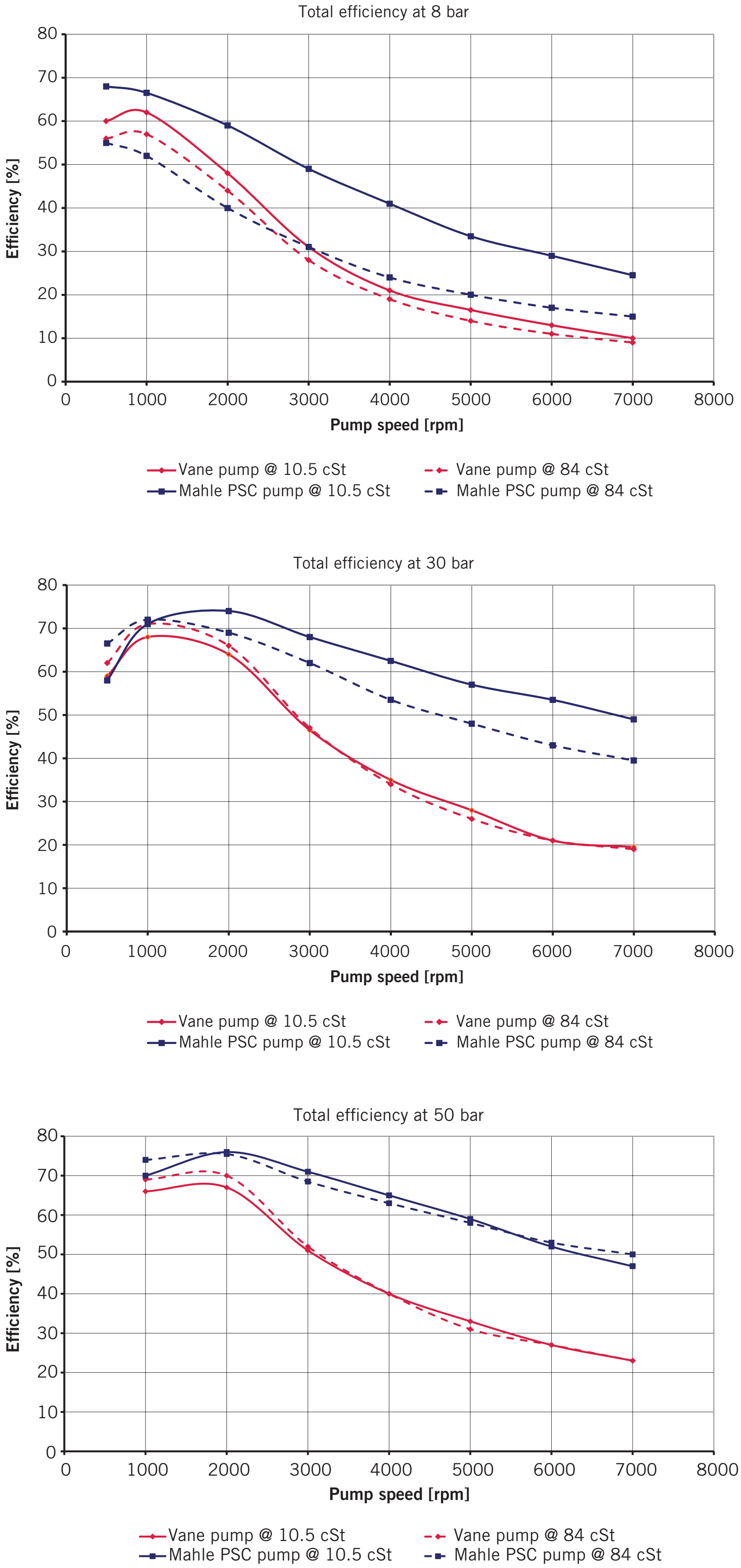
When used in automatic transmissions, it is thus possible to meet the different pressure requirements of the individual components appropriately with the two pressure circuits. The high-pressure stage can thus be used for shifting the clutches or brakes, and the low-pressure stage for cooling and lubricating the transmission components.

With additional control valves or throttles, various control strategies can be implemented. For example, as shown in ❻, the low-pressure stage can be used for preloading the high-pressure stage, in addition to supplying lubricating and cooling oil. A 2/2 directional control valve connected downstream of the low-pressure stage controls the volume flow to the transmission. If only a higher pressure and lower volume flow is required, then the valve closes the output to the transmission, and the low-pressure stages solely feeds the high-pressure stage. The excess oil is pumped to the oil reservoir via a pressure-relief valve.

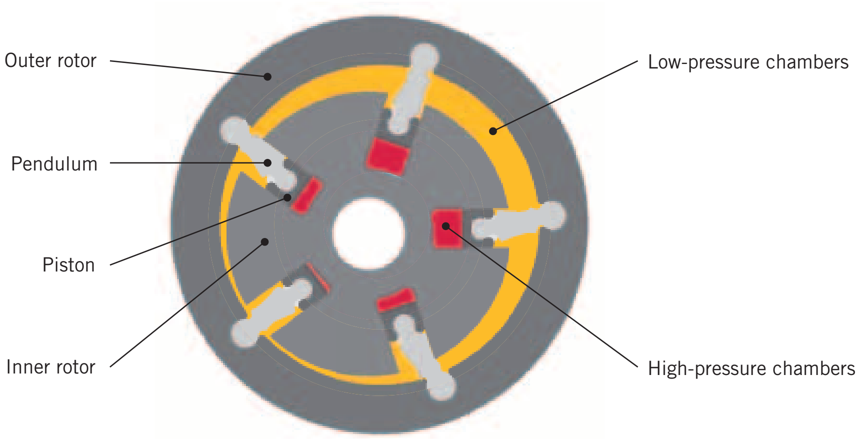
The bi-dynamic pressure pump can also be used advantageously with an electric drive, for instance as an auxiliary oil pump for hybrid applications. When using a fixed displacement pump, the two oppos-



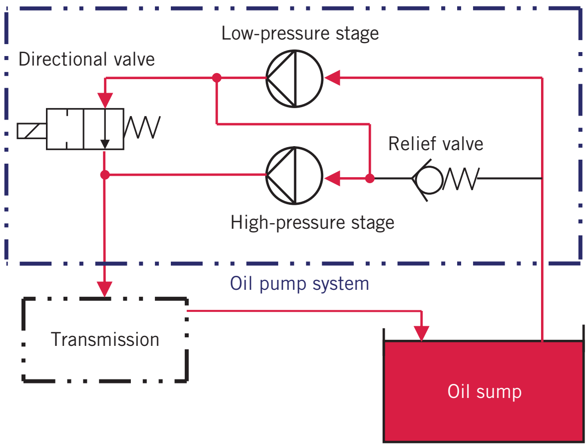
❸ Comparison of power input at 8, 30 and 50 bar



4 Comparison of total efficiency at 8, 30 and 50 bar



❺ Principle of the rotor set of the bi-dynamic pressure pump



❻ Regulation of the bi-dynamic pressure pump for high volume flow requirements – the low-pressure stage can be used for preloading the high-pressure stage

ing requirements for pressure and volume flow for shifting and cooling/lubricating are met by controlling the speed of the electric motor. This means that the motor must be run over a wide speed range, regardless of its efficiency curve. With the controlled bi-dynamic pressure pump, however, the electric motor can run at the operating range that has the greatest efficiency; the pump pressure stages need to be designed for the appropriate speed. This opens up possibilities for optimising the tuning of the pump and the electric motor, reducing the size of the pump drive.

SUMMARY AND OUTLOOK

Mahle's controlled pendulum-slider pump provides significant advantages when used with automatic transmissions, both with respect to further fuel savings, and by supporting variable control strategies. Its reliability and high efficiency over its entire service life has been demonstrated many times over in applications as a lubricating oil pump in combustion engines. The test findings confirm that, with an appropriate design and shape, savings of up to 50 % can easily be achieved. Further savings potential with thoroughly tuned control can be achieved by using mechatronic actuators, such as variable throttles that are actuated and monitored by the transmission controller.

For future applications the use of the bi-dynamic pressure pump opens the option of supplying both the pressurised oil and cooling oil circuits. This has advantages in the package and in the total drive power.

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SIMULATION OF THERMAL MANAGEMENT MEASURES

Thermal management offers many opportunities for reducing fuel consumption. Some of these options have not yet been used. Simulation can be used to reproduce the various measures and assess them without being affected by outside influences. In a cooperation between the BMW Group and Graz Technical University, a methodology has been developed that permits all the relevant processes to be modelled in conformity with the requirements for precision.

AUTHORS

AO. UNIV.-PROF. DIPL.-ING. DR. ANDREAS WIMMER is Professor at the Institute for Internal Combustion Engines and Thermodynamics at Graz Technical University (Austria).

DIPL.-ING. PETER UNTERGUGGENBERGER is Research Associate at the Institute for Internal Combustion Engines and Thermodynamics at Graz Technical University (Austria).

DR. ANDREAS EDER is Head of Predevelopment and Simulation Thermal Management at BMW AG in Munich (Germany).

DIPL.-ING (TU) FELIX SCHEDEL works in System Simulation Thermal Management Complete Vehicle at BMW AG in Munich (Germany).

MOTIVATION

The automotive industry needs to resolve the conflict of objectives between the continual tightening of statutory regulations governing CO_2 emissions and customer expectations such as comfort and safety. This can only be achieved by reducing all the individual losses and by efficient energy management. Apart from continual improvements in the combustion process and mechanical systems with reduced friction, optimisation of the engine heat flow and the use of waste heat recovery technologies make a significant contribution towards CO_2 reduction. Simulation is increasingly being used in this process, since experimental tests are time-consuming and cost-intensive for assessing the different savings measures and the test beds are very often not available at an early stage of the development process.

The BMW Group has therefore been cooperating with the Institute for Internal Combustion Engines and Thermodynamics at Graz Technical University for some time in order to develop complete vehicle models for modelling energy management simulating reality. The modelling is carried out in a uniform software environment based on numerous subsystems that are linked together. The mechanical power demand is calculated on the basis of simulating the longitudinal dynamics as a function of key vehicle data. The operating point thus determined provides the key input parameters for the thermal engine and gearbox model. The engine model is also linked with the passenger compartment sub-model through the heat exchanger used for heating in order to be able to assess the effects on driving comfort [1]. Other sub-models are the exhaust system and the onboard vehicle power supply.

MODEL DESCRIPTION

The model of the internal combustion engine is of central importance for the precise calculation of fuel consumption, as it offers the greatest potential for fuel efficiency improvements. These potential savings can also be expanded by new technologies that convert the (dissipated) heat energies into a form of energy that can be used efficiently [2].

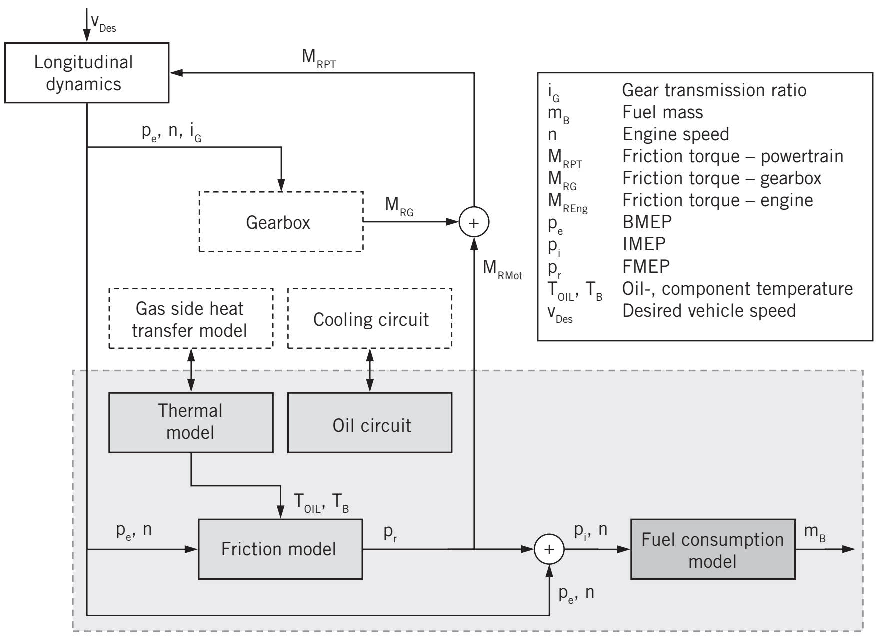
The structure of the engine model, ❶, comprises the following subsystems, which are linked together:
: gas side heat transfer model
: friction model
: fuel consumption model
: thermal model
: cooling and lubrication system.

The gas side heat transfer model describes the heat transfer of the working gas to the wall of the combustion chamber. This model is based on Newton's law. The flow of heat is determined using the heat transfer coefficient, the cross-sectional area and the gas temperature, at the same time taking account of the temperature of the component. The cyclically calculated mean gas temperatures and heat transfer data are used to calculate the warm-up of the engine. These temperatures are calculated from the cylinder-pressure curves measured on the test rig and a subsequent engine process calculation with appropriate heat transfer ratios [3, 4]. Changes in engine application which exert an influence on the heat transfer are taken into account by correction functions for the gas temperatures and heat transfer data.

The change in fuel consumption in a diesel engine is primarily based on a change in friction. A fuel consumption map showing the status at operating temperature based on the indicated mean pressure is therefore assumed for the consumption model. The indicated mean pressure can be determined from the effective mean pressure (initial parameter from the longitudinal dynamic) and the calculated friction mean effective pressure, ①. A change in friction during warm-up is incorporated directly into the fuel consumption calculation [5].

The approach of the friction model used describes all the friction components and effects of the real combustion engine. The friction mean effective pressure is calculated by totalling the frictional elements of the individual assemblies. The description is based on a combination of basic tribological principles and empirical results. It covers the entire engine-speed and temperature range through to very low temperatures [6].

In the thermal model, the component temperatures of the engine structure are calculated in this sub-model. Since some of the temperatures are relevant to friction and they are therefore input parameters in the friction model, this sub-system also influences the accuracy of the fuel consumption calculation. The method of concentrating the point masses is used for modelling. This involves material areas being grouped together to form a point mass with a heat capacity equivalent to the area. The temperature of the point mass is equivalent to the mean temperature of the area and hence to its energy content. Appropriate heat conduction relationships ensure correct heat distribution. On the basis of this method, the engine structure is modelled as a thermal network with a representative single cylinder. The number of substitute masses can be freely selected; however, a minimum of four masses is effective. A significantly higher number of substitute masses is required for heat thermal management measures which exert a significant influence on the relevant friction points.



❶ Basic model structure

ACCURACY REQUIREMENTS

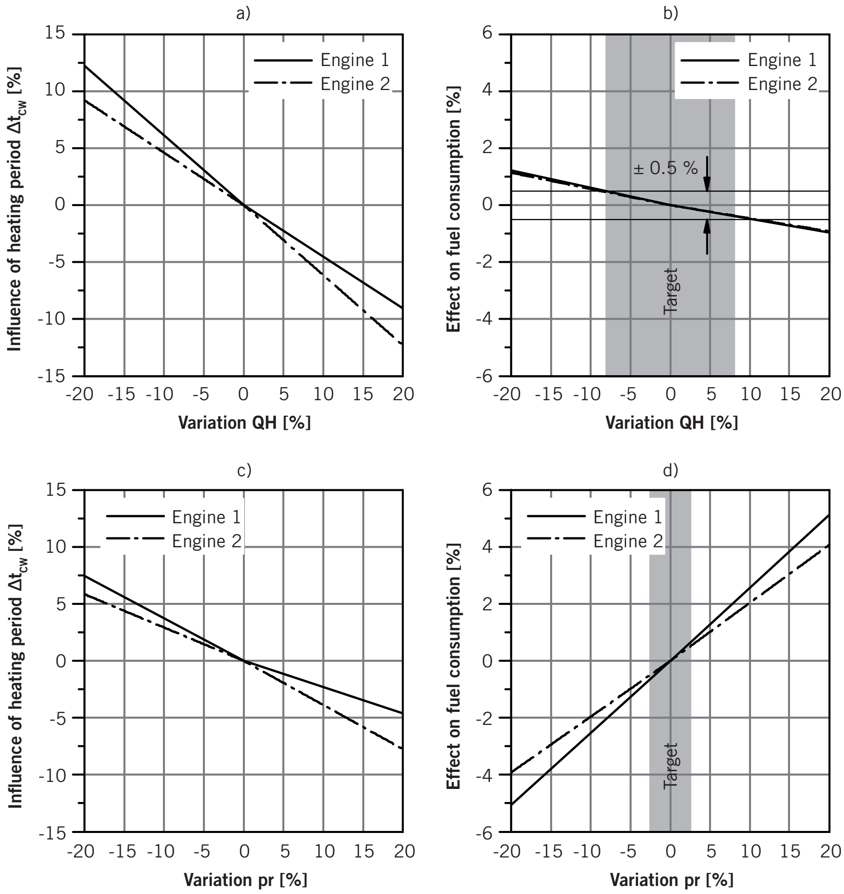
The main reason for using complete vehicle models is to provide a rapid assessment of the potential offered by different measures in varying test cycles. Since the fuel consumption effects are limited to a few percentage points depending on the measure being investigated and the underlying driving profile, high precision requirements are needed for the simulation. Sensitivity analyses were used to define the level of precision required for the modelling. Since combustion is the central heat source in the engine, it is responsible for the precise prediction of the thermal behaviour of the engine. Uncertainties of the gas side heat transfer of ± 20 % extend or shorten the engine warm-up in the New European Driving Cycle (NEDC) by some 10 %, ❷ (a). By contrast, the effects on fuel consumption are low at around 1 %, ② (b). The situation is different for friction mean effective pressure. Since the friction simulation results are included directly in the consumption calculation, it is particularly important to model the friction as accurately as possible. Errors in calculating the friction mean effective pressure of some 20 % generate deviations of some 4 to 5 % in the calculation of the consumption, ② (d). The thermal behaviour changes in the range between 6 and 8 %, ② (c), and therefore makes a decisive contribution towards engine warm-up, especially after a cold start.

The verification of the accurate simulation of the thermal behaviour and fuel consumption was carried out using the NEDC at different starting temperatures. The measured and simulated temperatures of the coolant and the main bearing are compared in ❸, and the accuracy achieved in calculating fuel consumption is also shown.

SAMPLE APPLICATION

The use of the simulation model for evaluating a thermal management measure is shown below using a concrete example. A possibility for shortening the engine warm-up time is provided by supplying energy with the assistance of an external energy store. In this case, a coolant heat store with a 5 l capacity and a temperature of 70 °C above the ambient temperature was investigated. This is equivalent to an energy store of 1.4 MJ.

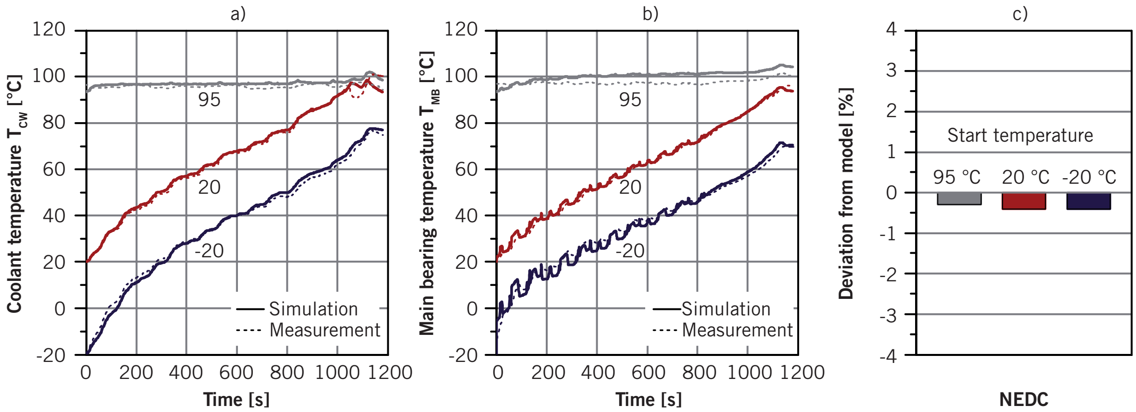
Numerous variants are simulated in order to establish the ideal integration of the heat store (HS) within the engine cooling system. The parameters of the appropriate version are also optimised. The most promising variant is then also subjected to experimental investigation. This is the variant where the coolant in the heat store is discharged into the "small engine circuit". Since the component and media temperatures differ in a cold start most markedly from the operating temperature immediately after the engine has started up, the fuel consumption potential improvement is greatest at the beginning of the warm-up.



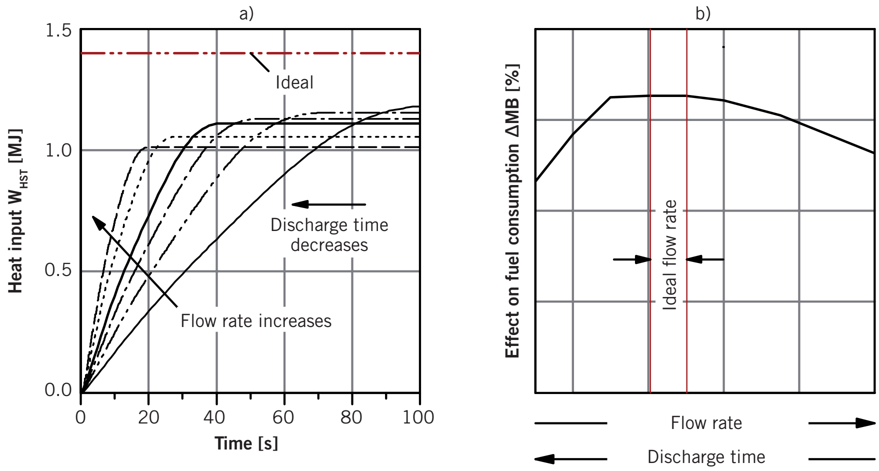
❷ Influence of the gas side heat transfer and the friction mean effective pressure on the thermal behaviour and fuel consumption

Optimum provision of energy therefore requires energy to be supplied as quickly as possible in order to achieve a positive effect in an early phase of the warm-up. This means starting immediately with a high power discharge of the heat store coolant. The discharge volume of the heat sink coolant is therefore defined as the parameter requiring optimisation in this version. ❹ (a) shows the effect of varying the discharge volume on the heat supplied to the engine. If discharge volumes are high, the time is too short to achieve extensive heat exchange between the coolant and the engine structure. After discharge, a relatively large amount of residual energy remains in the heat store. The amount of heat supplied to the engine can be increased by reducing the discharge volume. The disadvantage is that the focus of the heat transfer is later. A compromise must be found between heating the engine quickly and making the best possible use of the stored energy. The optimum discharge volume is that which maximises the reduction in fuel consumption, ④ (b).

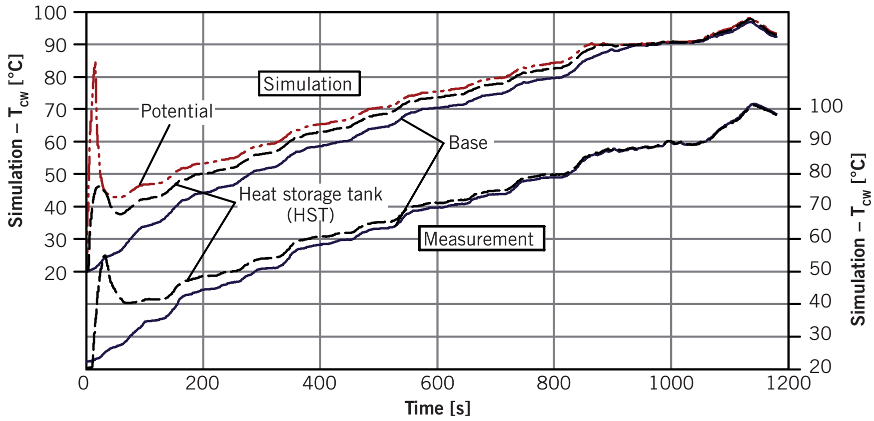
Experimental testing of the ideal variant with optimised parameters is carried out with a temperature-controlled full-engine test rig. The results of the measurements



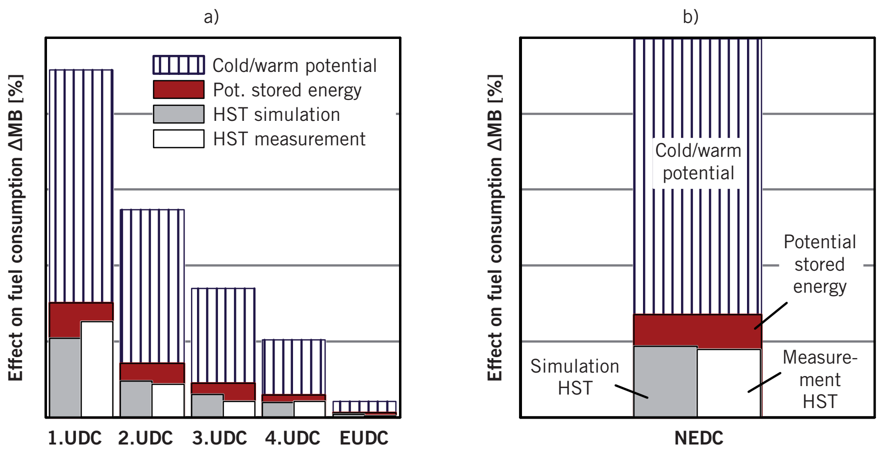
❸ Verification of the simulation of thermal behaviour and fuel consumption



❹ Parameter optimisation



❺ Thermal effects of heat storage on the coolant temperature in the NEDC



❻ Effect on fuel consumption of the heat sink in the sub-cycles and at the end of the NEDC

are compared with those of the simulation and compared with the theoretical potentials, ❺. In order to calculate the theoretical potential, an extra simulation is carried out in which the energy is introduced into the cooling system without loss via a heat source. The temperatures achieved with the assistance of the optimised discharge volume are close to those of an ideal implementation (theoretical potential).

The higher temperature level results in a reduction in fuel consumption that can be seen in ❻. At the end of the driving cycle, approximately three quarters of the storage energy of 1.4 MJ could be exploited to improve fuel consumption, and around one fifth of the potential fuel consumption improvement limit could be achieved.

CONCLUSION

This article presents a 1D calculation model that enables the thermal behaviour and the fuel consumption to be calculated in different test cycles. The good match between measurement and simulation therefore provides a methodology that can be used to answer questions about the effectiveness of a wide range of different energy management strategies at an early stage of development.

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BENCHMARKING OF THE ELECTRIC VEHICLE MITSUBISHI I-MIEV

Up to now, several concepts, prototypes and some series products of electric vehicles were developed with different powertrain concepts and body designs. For future developments the knowledge of the current state-of-the-art of these electric cars is of great importance. Therefore, the Forschungsgesellschaft Kraftfahrwesen mbH Aachen (fka) analysed one of the first series-production electric vehicles, the small passenger car Mitsubishi i-MiEV, within a design and functional benchmarking. The results were compared to these of a conventional powered small vehicle (VW Polo V).

AUTHORS

UNIV.-PROF. DR.-ING. LUTZ ECKSTEIN
is Head of the Institut für Kraftfahrzeuge (ika) at RWTH Aachen University (Germany).

DIPL.-ING. RENÉ GÖBBELS
is Scientific Assistant in the Body Department of the Institut für Kraftfahrzeuge (ika) at RWTH Aachen University (Germany).

DIPL.-ING. DIPL.-WIRT. ING. ROLAND WOHLECKER
is Manager Structural Analysis and Benchmarking of the Forschungsgesellschaft Kraftfahrwesen mbH Aachen (fka) in Aachen (Germany).

MOTIVATION

The conservation of resources and the associated efficient use of fossil energy resources are major challenges for the automotive industry. Beside the optimization of existing powertrain concepts regarding lightweight design and efficiency for the minimization of the CO_2 emissions, the electromobility gains more importance. Up to now, several electric vehicle concepts, prototypes and some series electric vehicle were developed with different powertrain concepts and body designs. The functional properties of the electric vehicles (for example regarding vehicle range and driving dynamics) are highly different to conventional cars. In terms of future developments of series applications of electric vehicles, the knowledge of the current state-of-the-art is of great importance.

The testing program, which was developed and executed by the Forschungsgesellschaft Kraftfahrwesen mbH Aachen (fka), is primary aimed at the analysis of the functional properties of electric vehicles (functional benchmarking). The basis for the analysis built the first real series electric vehicle of a notable automotive manufacturer, the Mitsubishi i-MiEV. During the functional benchmarking, several tests within the vehicle areas body, powertrain, chassis, electronics and acoustics were performed on in-house test benches. The functional properties of the Mitsubishi i-MiEV were compared to the performance of a conventional powered small vehicle (VW Polo V). Within the closing design benchmarking the vehicle was disassembled for the analysis and the weight investigation of all single parts.

BENCHMARKING SUBJECTS

The analysis of the Mitsubishi i-MiEV contains a design and a functional benchmarking. For the determination of the functional properties, the fka developed a test program for the vehicle areas body, powertrain, chassis, electronics and acoustics.

Within the design benchmarking the disassembly of the overall vehicle and the analysis of all single parts is executed. In doing so, the component weight, dimensions, materials, the fitting positions and the joining techniques are determined. The component design is documented by a detailed photo analysis. ❶ gives an overview about the process of the design benchmarking in five steps.

Within the functional benchmarking, destructive and non-destructive analysis are carried out. Regarding the vehicle area powertrain, the driving resistance, the energy consumption and the efficiency of the electric drive are determined and the high voltage battery is characterised. The analysis of the chassis contains the determination of the inertia and kinematics & compliance (k&c) parameters as well as the performance of relevant road tests.

Within the vehicle acoustics investigations, the interior and exterior noise is measured during different driving manoeuvres and the maximum sound pressure level during the accelerated passing is determined. Regarding the vehicle area electronics, the energetic main power supply and the energy consumption of the main electric components are analysed. The global and dynamic stiffnesses of the body-in-white are determined. ❷ gives an overview about the different topics, which are analysed within the functional benchmarking.

RESULTS OF THE DESIGN BENCHMARKING

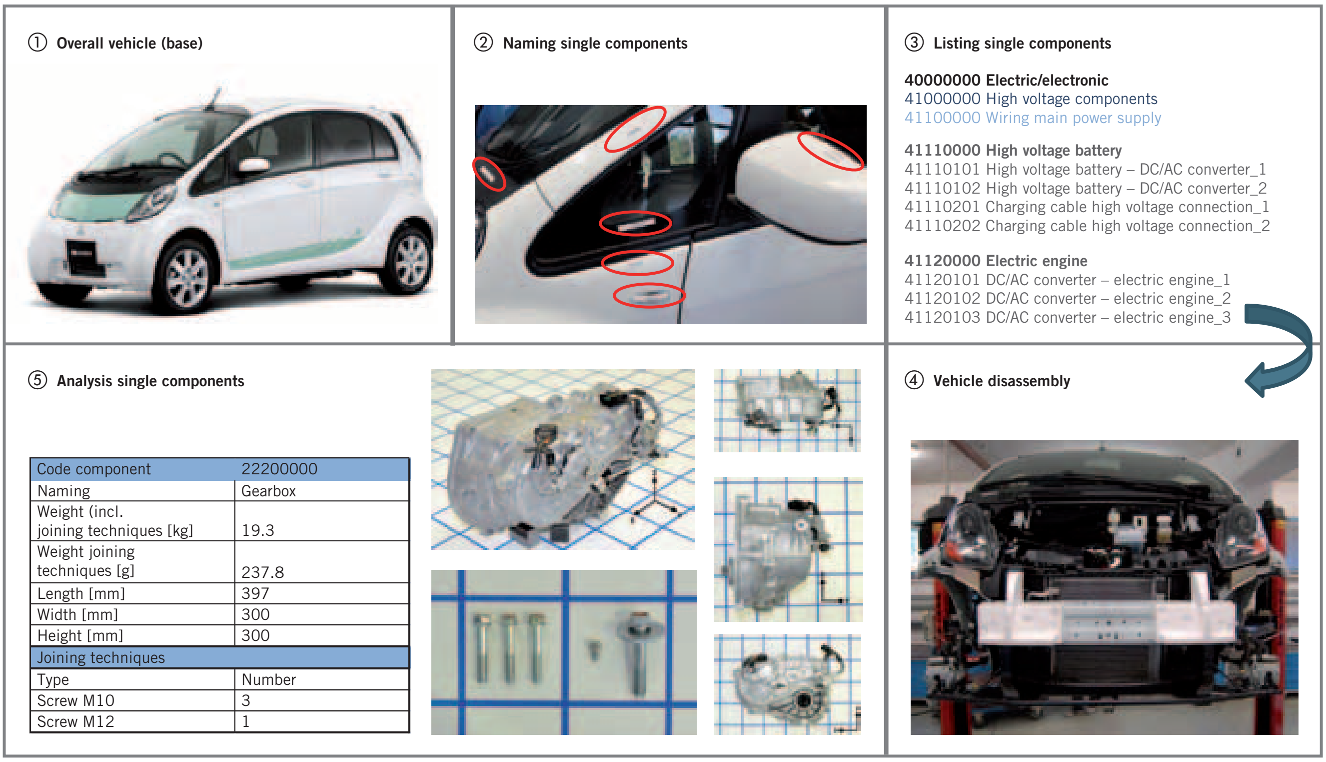
Within the design benchmarking, component weights, dimensions and materials as well as fitting positions and joining techniques of all single parts are determined. Therefor the single parts are clearly named by the use of a vehicle independent nomenclature (eight-digits number code). During the vehicle disassembly the corresponding data is collected. In addition to that, the part design is documented by a detailed photo analysis.

The vehicle curb weight of the Mitsubishi i-MiEV amounts to nearly 1100 kg (without driver). 383 kg are assigned to the vehicle area body and 322 kg to the vehicle area powertrain. The vehicle areas chassis, electronics and interior have an overall weight of 165, 110 and 119 kg, ❸. Thereby the overall weight of the vehicle area body is defined by 60 % of the body-in-white including fenders with 212.4 kg and by 16 % of doors and closures (61 kg).

Within the vehicle area powertrain, the weight proportion of the energy storages consisting of the high voltage (HV) battery and the 12-V battery is up to 72 %. The overall weight of the HV battery, consisting of 22 battery modules, with four battery cells each, accounts to 217 kg. The weight of the chassis is mainly determined by the front and the rear axle as well as the tyres and the rims (overall 72 % respectively 119 kg).

Within the electronics, high and low voltage components are differed. The comfort electronic (heater and air conditioning), the power electronics (DC/DC-converter/charge unit and DC/AC-converter) as well as the main power supply wiring of the HV battery, the electric motor, the DC/DC- and the DC/AC-converter belong to the high voltage components. The overall weight of the comfort electronics is up to 21 kg and the power electronics is up to 25 kg.

The comparison of the weight distribution of the Mitsubishi i-MiEV and the VW Polo V (vehicle curb weight approximately 1150 kg) shows the nearly identical weight proportions of the vehicle areas body, chassis and interior. The overall weight of these vehicle areas do largely not depend on the powertrain concept, but rather on functional properties like crash safety, driving dynamics and comfort. The weight distribution (weight proportion respectively amount of the weight) of the vehicle areas powertrain and electronics



Code component	22200000
Naming	Gearbox
Weight (incl. joining techniques [kg]	19.3
Weight joining techniques [g]	237.8
Length [mm]	397
Width [mm]	300
Height [mm]	300
Joining techniques	
Type	Number
Screw M10	3
Screw M12	1

❶ General approach of the design benchmarking in five steps

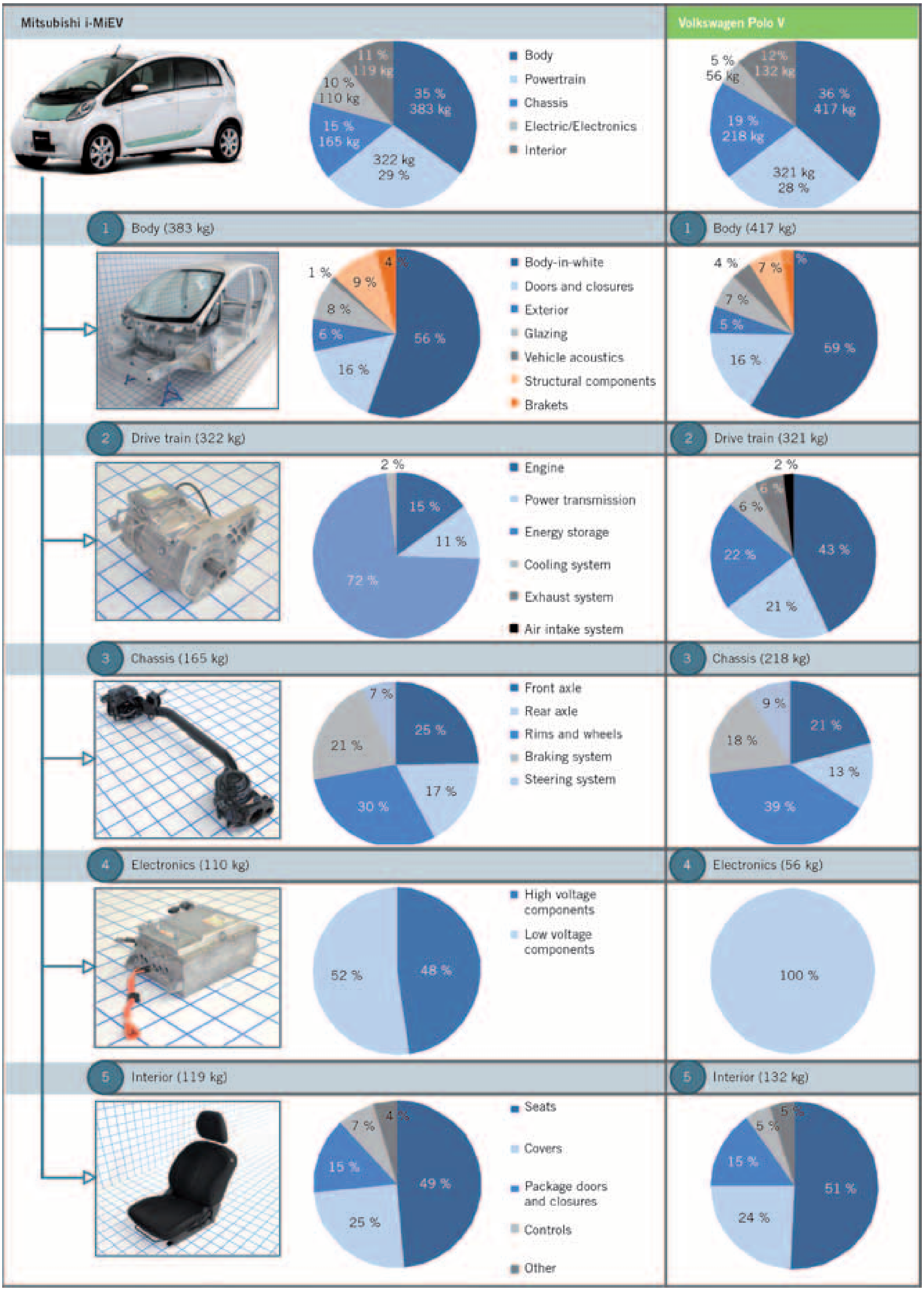
differ strongly, because of the different functional groups within these vehicle areas. It is recognisable, that the substitution of all conventional powertrain components by electrical powertrain components causes a redistribution of the weight proportions with a nearly identical vehicle curb weight. The overall weight of the conventional powertrain components is completely compensated by the high weight of the high voltage battery.

RESULTS OF BODY ANALYSIS

For the analysis of the static bending and torsion stiffness, the front strut towers and the rear spring supports of the body-in-white of the Mitsubishi i-MiEV (in combination with the stiffness increasing structural hang-on parts engine frame, cross car beam, windscreen and subframe front) are fixed on a special designed test bench. In order to determine the torsion stiffness, torsion moments of 1000, 2000 and 3000 Nm are introduced in the driver's side and in the passenger's side in the front strut towers of the body-in-white by a rocker. For the determination of the bending stiffness, the rocker is fixed in a horizontal position and the body-in-white is loaded in z-direction with test weights of 170 and 425 kg at the centre of gravity of the front and rear seats. In order to analyse the stiffness characteristics, 50 analogue dial gauges in nearly constant distances are used on the longitudinal beams and on the door sills.

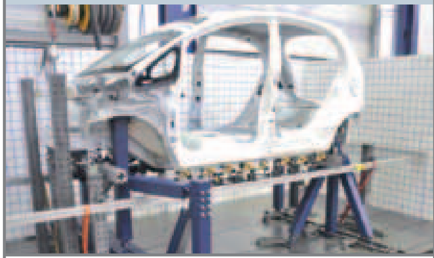
For the determination of the dynamic stiffnesses consisting of the first bending and the first torsion, the body-in-white (in combination with the stiffness increasing

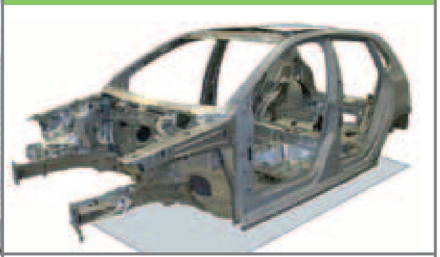
❷ Overview about the different testing procedures within the functional benchmarking



❸ Weight distribution of the Mitsubishi i-MiEV and the VW Polo V

Mitsubishi i-MiEV		Volkswagen Polo V [3]	





Body-in-white (without battery)		Body-in-white	
Torsion stiffness [Nm/°]	8337	Torsion stiffness [Nm/°]	18000
Lightweight quality [kg/(Nm/°*m²)]	9.31	Lightweight quality [kg/(Nm/°*m²)]	3.5
Body-in-white (including battery)			
Torsion stiffness [Nm/°]	9394		
Lightweight quality [kg/(Nm/°*m²)]	16.13		

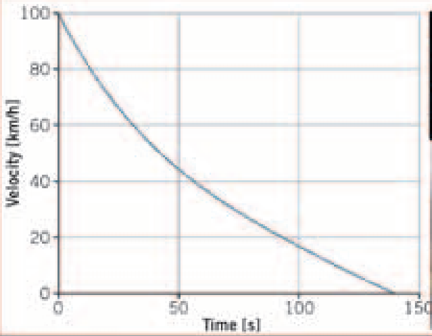
Mitsubishi i-MiEV		Volkswagen Polo V [3]	
Body-in-white (without battery)		**Body-in-white**	
1st torsion [Hz]	34	1st torsion [Hz]	43
1st bending [Hz]	57	1st bending [Hz]	46

❹ Results of the static and dynamic stiffness analysis

structural hang-on parts) is excited by a shaker. Therefore the body-in-white is mounted frictionless with rubber cords. The interpretation of the structure response is done by triaxial acceleration sensors. The resulting eigenfrequencies are shown simulative in a wire frame model.

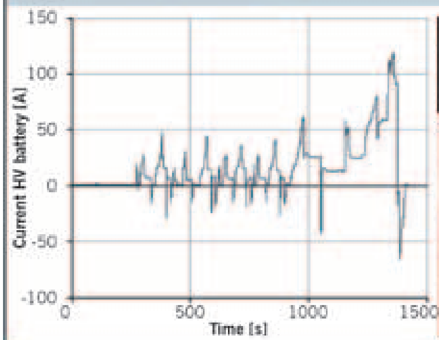
The results of the static and dynamic stiffness analysis are shown in ❹. Taking the high voltage battery into account, the static torsion stiffness of the body-in-white of the Mitsubishi i-MiEV increases up to 9394 Nm/°. The first bending and the first torsion reach a value of 34 and 57 Hz. In comparison to the VW Polo V, the Mitsubishi i-MiEV shows slowly decreased static and dynamic stiffnesses. During a general consideration of the vehicle class Small, it can be stated, that the stiffnesses of the body-in-white of the Mitsubishi i-MiEV are adequate to achieve typical driving properties of this vehicle class.

Analysis of the driving resistance Mitsubishi i-MiEV

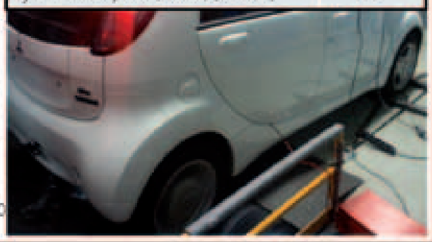


Driving resistance parameters	
c_w value [-]	0.33
Cross sectional area A [m²]	2.14
Rolling resistance value f_r [-]	0.011

Energy consumption Mitsubishi i-MiEV



Energy consumption	
Manufacturer information (NEDC) [Wh/km]	135
Cycle consumption (NEDC) [Wh/km]	165



Mitsubishi i-MiEV		Volkswagen Polo V [3]	
CO_2 emissions [g/km]		**CO_2 emissions [g/km]**	
Fuel mix (Germany 2010) [2] [g/kWh]	563	44 kW (gasoline engine)	128
CO_2 emissions [g/km]	93	77 kW TSI (gasoline engine)	124
CO_2 emissionen [g/km] (manufacturer information)	76	66 kW TDI (diesel engine)	109
		66 kW TDI BlueMotion (diesel engine)	96
		55 kW TDI BlueMotion (diesel engine)	87

❺ Driving resistance, energy consumption and CO_2 emissions of the Mitsubishi i-MiEV, compared to the VW Polo V

RESULTS OF POWERTRAIN ANALYSIS

As a basis for the analysis of the energy consumption of the Mitsubishi i-MiEV regarding the NEDC on a chassis dynamometer, the determination of the driving resistance parameters consisting of the roll resistance and the aerodynamic resistance is necessary by the use of coast load tests. Within the coast load tests, the vehicle is accelerated up to a velocity v_{car} of 120 km/h. After actuation of the circuit stage N of the automatic transmission, the velocity can be measured up to the standstill of the vehicle. As a result of the coast load tests, a regression curve for the driving resistance F_W is available, which can mathematically be described as follows:

EQ. 1

$$F_W = f_0[N] + f_1[N/(km/h)] \cdot v_{car} + f_2[N/(km/h)^2] \cdot v_{car}^2$$

In the following the coefficients f_0, f_1 and f_2 are used as input parameters for the determination of the energy consumption on the chassis dynamometer. Taking the NEDC into account, the energy consumption of the Mitsubishi i-MiEV can be analysed now and based on the actual electricity generation mix in Germany [2] the CO_2 emissions (in g/km) can be calculated.

At a measured NEDC consumption of the Mitsubishi i-MiEV of 165 Wh/km, the

CO_2 emissions reach a value of 93 g/km. In comparison to the VW Polo V it can be seen, that based on manufacturer information the BlueMotion model (55-kW diesel engine) reaches a CO_2 emission of 87 g/km in driving operation, ❺. In future a reduction of the CO_2 emissions during electricity generation could be reached by the intense use of regenerative energy resources.

RESULTS OF CHASSIS ANALYSIS

For the determination of the moments of inertia and the centre of gravity in x-, y- and z-direction, the vehicle is fixed on the "Vehicle Inertia Measuring Machine" (VIMM). By the rotation of the vehicle in all directions in space by servo hydraulic actuators, the measured accelerations and forces can be used to calculate the inertia parameters and the centre of gravity. The analysis shows, that the height of the centre of gravity of the Mitsubishi i-MiEV is in spite of a major overall vehicle height of 145 mm in comparison to the VW Polo V on a nearly identical level, ❻.

Within the analysis of the kinematic and elasto-kinematic parameters of the front and the rear axle the jounce and the rebound characteristics, the vehicle roll characteristics, the lateral and the longitudinal force characteristics and the steering behaviour in particular the steering kinematics and the steering friction are determined. During the road tests, the vehicle performance concerning steady state cornering, step steer, sinusoidal steer and the VDA lane change is analysed. As a result of these testing procedures it can be shown, that the Mitsubishi i-MiEV has adequate driving dynamics properties for his vehicle class.

RESULTS REGARDING ACOUSTICS

For the analysis of the vehicle acoustics, the interior and exterior noise measurement for different driving manoeuvres and the maximum sound pressure level for the accelerated passing based on the new testing method (ECE R51-3) and the old testing method (70/157/EWG) was determined. During the interior and exterior noise measurements, noise pressure levels are recorded for run-ups with different loads, coast loads, constant vehicle velocities and steady state respectively unsteady working points by the use of a binaural artificial head system and tri-axle and uni-axial accelerometers. The determination of the maximum sound pressure level regarding the accelerated passing occurs within a special defined working section. Therefore different microphones are positioned outside the vehicle.

The maximum sound pressure level for the Mitsubishi i-MiEV reaches a value of 68 dB(A) regarding the old testing method. Volkswagen claims a value of 72 dB(A) for its Polo. The resultant maximum sound pressure level of the Mitsubishi i-MiEV is primary caused by the take-off noise of the vehicle tyres. The noise level of the electric drive is therefore significant lower than the noise level of a comparable conventional driven vehicle.

SUMMARY

The testing program for the analysis of the functional properties of electric vehicles, which was developed by the Forschungsgesellschaft Kraftfahrwesen mbH Aachen (fka), was executed on the first series production electric vehicle of a notable manufacturer, the Mitsubishi i-MiEV. Within the functional benchmarking several aspects of the vehicle areas body, powertrain, chassis, electronics and acoustics were analysed in detail on in-house test benches. After the functional benchmarking a design benchmarking was performed for the analysis of the single parts and as a basis for the weight distribution.

According to this, the Mitsubishi i-MiEV causes a significant lower maximum sound pressure level than conventional vehicles. In terms of CO_2 emissions, the resulting values are on a nearly identical level with a VW Polo V BlueMotion with a 55-kW diesel engine, taking the German electricity generation mix into account. It can be estimated, that in the future the CO_2 emissions of the i-MiEV can further be reduced by the increased usage of regenerative energy resources. In addition to that the i-MiEV comes with adequate static and dynamic torsion and bending stiffnesses. In comparison to conventional vehicles of the vehicle class Small, the i-MiEV describes a well-balanced electric vehicle with good driving characteristics.

All benchmarking results were provided to an open customer consortium for a fixed unique contribution via an online database. The fka plans, to continue the benchmarking activities in this year (2011) and to analyse further electric vehicles in analogy to the Mitsubishi i-MiEV.

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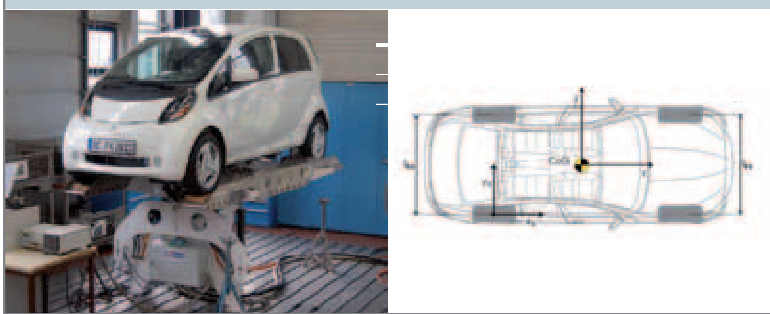
Mitsubishi i-MiEV



Mitsubishi i-MiEV

Vehicle dimensions	
x [mm]	3475
y [mm]	1475
z [mm]	1610
Centre of gravity	
x_s [mm]	1199
y_s [mm]	628
z_s [mm]	559

Volkswagen Polo V

Vehicle dimensions	
x [mm]	3970
y [mm]	1682
z [mm]	1465
Centre of gravity	
x_s [mm]	~ 1600
y_s [mm]	~ 730
z_s [mm]	~ 550

❻ Analysis of the centre of gravity of the Mitsubishi i-MiEV, compared to the VW Polo V

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VICE EDITOR-IN-CHIEF
Dipl.-Ing. Michael Reichenbach (rei)
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ulrich.knorra@springer.com

Chassis
Roland Schedel (rs)
phone +49 6128 85 37 58 · fax +49 6128 85 37 59
ATZautotechnology@text-com.de

Electrics, Electronics
Markus Schöttle (scho)
phone +49 611 7878-257 · fax +49 611 7878-462
markus.schoettle@springer.com

Engine
Ruben Danisch (rd)
phone +49 611 7878-393 · fax +49 611 7878-462
ruben.danisch@springer.com

Online
Katrin Pudenz M. A. (pu)
phone +49 6172 301-288 · fax +49 6172 301-299
redaktion@kpz-publishing.com

Production, Materials
Stefan Schlott (hlo)
phone +49 8726 9675972
Redaktion_Schlott@gmx.net

Research
Johannes Winterhagen (win)
phone +49 611 7878-342 · fax +49 611 7878-462
johannes.winterhagen@springer.com

Service, Event Calendar
Martina Schraad (mas)
phone +49 611 7878-276 · fax +49 611 7878-462
martina.schraad@springer.com

Transmission
Dipl.-Ing. Michael Reichenbach (rei)
phone +49 611 7878-341 · fax +49 611 7878-462
michael.reichenbach@springer.com

Dipl.-Ing. (FH) Moritz-York von Hohenthal (mvh)
phone +49 611 7878-278 · fax +49 611 7878-462
moritz.von.hohenthal@springer.com

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Markus Bereszewski (mb)
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THE SEAL OF APPROVAL FOR SCIENTIFIC ARTICLES IN ATZ.
ATZ PEER REVIEW
REVIEWED BY EXPERTS FROM RESEARCH AND INDUSTRY.

METHODS TO ESTIMATE THE TIRE ROAD FRICTION FOR ADVANCED DRIVER ASSISTANCE SYSTEMS

The maximum available coefficient of friction between road and tire is one of the major variables in the interaction of driver, vehicle and environment, since all forces acting on the vehicle have to be transmitted by the small contact patch and thereby significantly affect driving safety. The present paper gives a systematic overview of the existing approaches. The focus lies on methods that evaluate measurement data related to the dynamic state of the vehicle and regarding the application in Advanced Driver Assistance Systems.

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AUTHORS

DIPL.-ING. CORNELIA LEX is Research Associate at the Institute of Automotive Engineering, Graz University of Technology (Austria).

DR. ARNO EICHBERGER is Lecturer at the Institute of Automotive Engineering, Graz University of Technology (Austria).

PROF. DR. WOLFGANG HIRSCHBERG is Head of the Institute of Automotive Engineering, Graz University of Technology (Austria).

1 MOTIVATION

The determination of the maximum friction and its application for Vehicle Dynamics Controls (VDC) and Advanced Driver Assistance Systems (ADAS) in terms of vehicle stabilisation, accident prevention and accident severity reduction was studied intensively for decades; still the practical application is limited. Most ADAS cannot use their full potential because they lack the knowledge of the maximum friction.

The two exemplary ADAS Collision Avoidance Systems (CWS) and Predictive Brake Assistant (PBA) have proven to provide significant potential for collision avoidance and collision severity reduction. The potential to avoid fatal accidents was mentioned to be 4.2 to 37.9 % for CWS and 13.0 to 21.5 % for PBA systems in dependence of the driver reaction and the control strategy, [1]. However, the functionality of CWS and PBA will be effectively improved upon a reliable estimate of the time-to-collision (TTC) and therefore the knowledge of the friction potential. The two influencing factors to estimate the vehicle trajectory which is necessary for TTC are firstly the driver reaction and secondly the maximum friction [2].

2 METHOD

2.1 CLASSIFICATION OF EXISTING METHODS

Although the amount of existing methods is quite impressing, no universally valid solution exists. The focus of the present overview lies on the basic ideas of the methods that evaluate measurement data related to the dynamic state of the vehicle which mainly consist of methods that are slip-based or that measure the effects on other vehicle dynamics quantities.

Classical theory states that for small slips, the shape of the slip curve, ❶, is only depending on the properties of the tire carcass. In contradiction to that theory, Dieckmann experimentally showed a linear relation between the slip slope k and the maximum μ_M friction for small values of the slip by the friction demand μ_D, [3]. In Eq. 1, the variable s denotes the longitudinal slip and the variable δ is an offset factor that is necessary in practice to deal with model uncertainties like the changes in the effective tire radius.

EQ. 1	$\mu_D = k\,(s-\delta)$

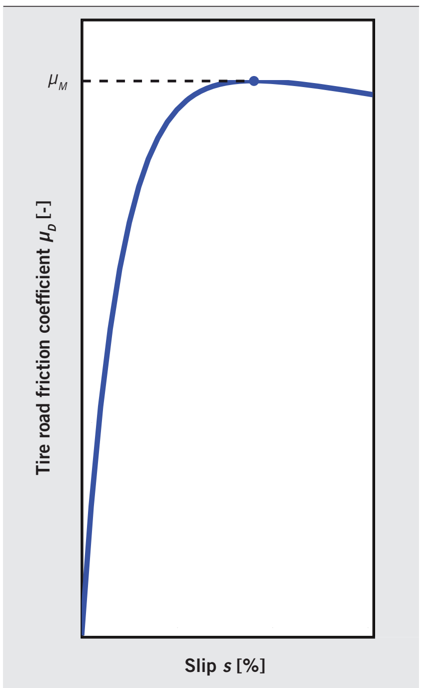
However, the quite simple relation in Eq. 1 raises new challenges. One disadvantage of estimating the slip slope k instead of using another physical quantity is that a mathematical relation between k and μ_M is necessary like additional knowledge on the tire characteristics. Many different mathematical approaches have been investigated to solve the problem of estimating the maximum friction, ❷. Besides the common classifications based on physical characteristics, a distinction in algebraic, statistical, observer-based and optimization-based methods is introduced.

Another challenge is the calculation of the longitudinal slip s and the demanded friction μ_D which cannot be measured without great effort. So processing the quantities that are necessary from the data of the typically available low cost sensors for further model-based processing is a key point. The mathematical classification in ② only investigates the algorithms used to estimate μ_M directly and not the methods that are used to treat the necessary quantities before they are processed in further steps. This distinction is very important because dealing with friction estimation often results in trying to find accurate observers or optimization methods to give evidence on the real vehicle state or other necessary quantities like the tire forces. Analogous to the estimation of the maximum friction in longitudinal direction, methods exist that use the lateral force characteristics and its dependence on the lateral slip.

Most of the methods that investigate other quantities than longitudinal slip or the side slip angle need lateral excitation of the vehicle dynamics to estimate μ_M. Those methods for example evaluate a correlation between the steer angle and the lateral acceleration [4], the tire aligning torque [5] or the yaw rate and lateral acceleration [6].

2.2 EVALUATION OF EXISTING METHODS

Regarding the application of an algorithm to estimate the maximum friction in CWS and PBA, the aforementioned classification is not of practical use. To derive the most promising methods, other characteristics have to be considered. Dealing with multiple criteria, a benefit analysis was chosen as proposed in [7]. Criteria and weighting factors are determined regarding the applicability in the proposed ADAS, ❸. Some of the methods need excitation of more than 50 % of the maximum possible acceleration a_M in either longitudinal or lateral direction to deliver reliable estimates. This limits the applicability because the most difficult state to be identified is driving on a highway with little dynamic change, so



❶ Slip versus coefficients of tire-road-friction μ_M and μ_D

	ALGEBRAIC	STATISTICAL	OBSERVER BASED	OPTIMIZATION METHODS			
				DETERMINISTIC		STOCHASTIC	
				LINEAR	QUADRATIC	FUZZY LOGIC	NEURAL NETWORKS
SLIP BASED (LONG. SLIP s, SIDE SLIP ANGLE α)	Holzinger [4]	Ray [19-21]: Bayesian Probability	Gerard [24]	Uchanski [25]	Baffet [26]: Quasi-Newton		Pasterkamp, Pacejka [27]
	Villagra et al. [9]	Gustafsson [15-18]: Kalman Filter		Svendenius [28]: Least-Square/Gauss-Newton			
	Witte, Zuurbier [12]	Boßdorf-Zimmer et al. [29]: Extended Kalman Filter		Li, Hedrick, Yi [30]: RLS + FF[1]			
	Rajamani et al. [31]		Nishira et al. [32]	Germann et al. [33]: RLS + FF[1]			
	Liu, Peng [10]		Han et al. [34]				
			Faraji et al. [14]				
OTHER PHYSICAL QUANTITIES	Holzinger [4]: Steer angle		Ahn et al. [35]	Ding, Taheri [6]: RLS + FF[1], yaw rate and lat. acceleration		Ivanov et al. [36]: On- & Offboard parameters	
				Umeno [37]: RLS[2]			

[1] Recursive Least Square Method with Forgetting Factor
[2] Recursive Least Square Method

❷ Mathematical classification of existing methods

the criterion maximum friction has been chosen. Also based on these considerations, the criterion direction was introduced to give preference to those methods that provide reliable estimates when the vehicle is only excited in longitudinal direction as in a typical highway situation. Predictive determination of the maximum friction is desirable for ADAS considered in the criterion actuality.

The criterion applicability in ADAS evaluates for which number of the selected FAS the method can be used. In the present study the applicability on the following systems has been assessed: CWS, PBA and airbag prefiring. The necessary requirements are a sufficiently fast and robust detection of the maximum friction, and in particular its change. The essential difference between the airbag prefiring refers to the higher demands on the accuracy of the actual value. In contrast, the other applications require a fast TTC prediction. The approach should not require any additional sensors or an active intervention in the driving state, has to be capable to deliver online estimates and has to be investigated regarding the everyday limits, as an example the detection of gravel has been used.

All together, 32 methods have been rated. For individual results and detailed explication of the criteria, see [8]. Those methods that had a rating of 99 points or more in the first benefit

MAXIMUM FRICTION	8
AVAILABILITY (LONG./LAT./COMBINED)	3
ACTUALITY (YES/POSSIBLY/NO)	2
ACTIVE INTERVENTION	5
APPLICABILITY IN ADAS	4
ONLINE CAPABILITY	7
NO ADDITIONAL SENSORS NEEDED	6
DETECTION OF GRAVEL	1

❸ Evaluation criteria and weighting factors

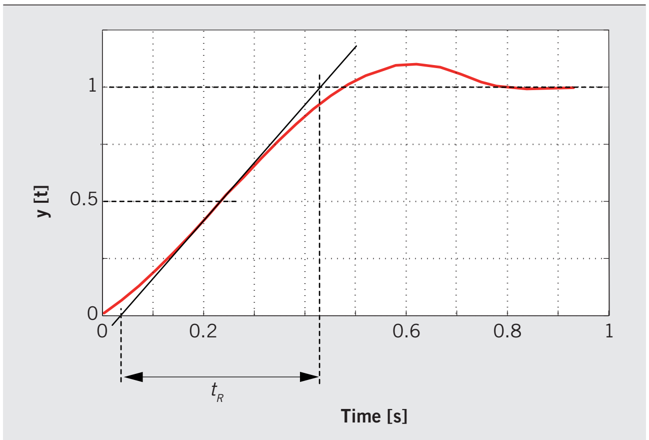
analysis have been further investigated regarding the information in the literature. The eight left methods were well described in the literature that was available; still some more details in [9-14] would have been necessary to rebuild them without further assumptions. The remaining approaches by Gustafsson [15-18] and Ray [19-21] have been rebuild. Gustafsson uses the relation in Eq. 1 in a linear Kalman Filter whereas Ray compares estimated longitudinal forces with forces stored in lookup tables and derives the most probable maximum friction using Bayesian Probability Theory.

The maneuvers have been simulated using the academic simulation platform Moves-2 (Modular Vehicle Simulation System) [22, 23] which is based on a modular multi body system of a full vehicle. The used vehicle model was an Opel Combo 1.6 CNG which was validated with driving tests. Normal distributed noise was added to the output of this simulation. Further it is assumed that the vehicle is not exposed to high dynamical excitation when detecting a possible critical situation. Therefore a longitudinal manoeuvre with constant acceleration with comparably low slip (maximum 5 % wheel slip) has been chosen. After 10 s, the maximum friction suddenly decreases from $\mu_M = 0.9$ to $\mu_M = 0.6$.

The parameters Response Time and Response Threshold are examined in this second evaluation. The first parameter is estimated using the rise time, ❹, which describes the speed of the system response to a change in the input signal. The response threshold is the smallest change in the input signal that can be detected and almost only on the signal noise of the slip estimate.

3 RESULTS

Both methods detect the change of the road condition very fast. The disadvantage of Gustafsson's methods is that from the measured slip slope k it is not automatically possible to extrapolate to μ_M. See ❺ for the estimation of the slip slope k before and after the decrease



❹ Rise time t_R

of friction. It has to be mentioned that the slip slope is known in this case, ❻, which cannot be considered for everyday use.

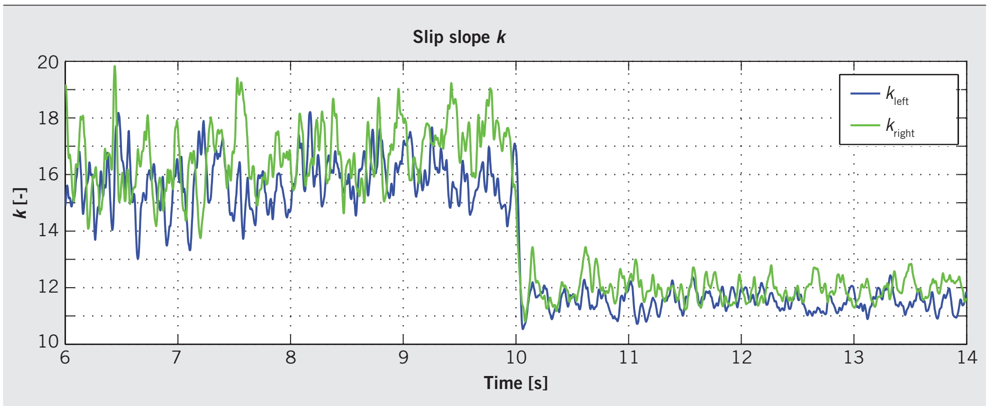
Ray decoupled the estimation of tire forces and the estimation of μ_M and only calculated one robust estimate of μ_M for every second. This time step can be reduced, but the higher change in tire force estimates results in a higher change of the estimated μ_M. For a comparison of the estimates for two different time steps Δt=1 s and Δt=0.02 s, ❼. This method enables time adaptive application, meaning that when a potentially dangerous situation arises, the algorithm compares the known robust estimate from Δt=1 s with an event triggered estimate of Δt=0.02 s to indicate a possible change of the road condition that suddenly occurred.

Concerning the response time, the method proposed by Ray is better, because a smaller time step can be used. In regard of the response threshold, Gustafsson proved to be the more reliable method. To calculate the TTC as fast as possible, the response time is of higher importance than the response threshold, so the method of Ray is chosen for application. Still, experience showed that parameterization of both the Kalman Filter Parameters and the covariance matrix for the Bayesian Probability used in the both methods are sensitive to parameter variations and also require a lot of knowledge on the signal noise and the model uncertainties. Another factor that has to be considered is that the used signals have been simulated and hence only proof the application to more or less laboratory conditions, so further investigation is necessary.

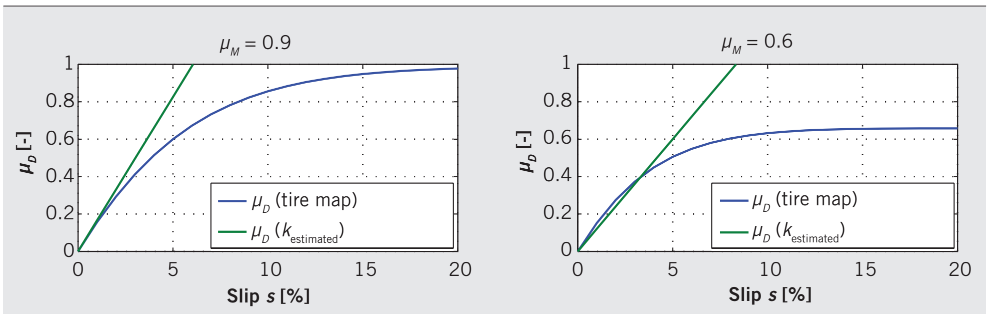
4 CONCLUSION

The high amount of the existing research on estimation of the maximum tire road friction is an indicator for two things. First of all, the maximum friction is a very important quantity for vehicle safety and vehicle stability. The knowledge of both demanded friction and maximum available friction are of high interest in ADAS applications as collision warning systems or the predictive brake assist.

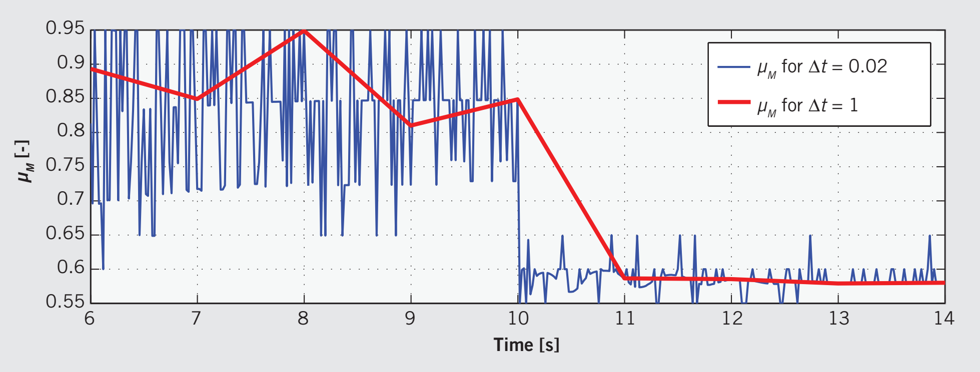
The second conclusion that can be drawn is that no general method exists that is able to estimate both demanded and maximum friction in every possible driving state and that is suitable for every type of application. One reason is that the maximum friction and the physical quantities that are necessary to calculate it, cannot be measured directly during driving without considerable effort. The quality of these estimated quantities depend very much on all the simplifications that have to be made for certain driving manoeuvres.



❺ Slip slope *k* according to Gustafsson



❻ Normalized tire characteristics before and after change of μ_M



❼ Friction potential according to Ray before and after time of change at t=6 s

To design a system that works in most driving states, an adaptive method switching between different approaches using different input signals depending on the most reliable quantities that are available in the current driving state seems to be the most promising method. Other environmental low cost sensors, that are already used in mass production vehicles like rain or outside temperature sensors, can improve those algorithms by helping to avoid bad estimates.

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AUTHORS

DIPL.-ING. (FH) CHRISTIAN WILK
is Project Engineer at Euro Engineering AG in Stuttgart (Germany).

DIPL.-ING. PHAN-LAM HUYNH
is Scientific Assistant at the Research Institute of Automotive Engineering and Vehicle Engines in Stuttgart (Germany).

DR.-ING. MICHAEL GRIMM
is Head of the Department Automotive Mechatronics/Electronics at the Research Institute of Automotive Engineering and Vehicle Engines in Stuttgart (Germany).

PROF. DR.-ING. HANS-CHRISTIAN REUSS
is Chairman of Automotive Mechatronics at the Institute of Internal Combustion Engines and Automotive Engineering at the University of Stuttgart (Germany).

RETROFIT ELECTRIC DRIVE KIT FOR CONVENTIONAL VANS

Small and medium-sized companies located in Baden-Württemberg banded together in the promoted Elena project to develop a retrofit electric drive kit for hybridization of conventional diesel vans. ATZ presents as an example the upgrade of a Mercedes-Benz Sprinter.

1 INTRODUCTION

To achieve the goal of the German government of one million electric vehicles operating in Germany until 2020, the federal Ministry of Transport, Building and Urban Development (BMVBS) promotes the electric mobility in eight pilot regions with a total of 130 million euros from the second economic stimulus package. The program is managed by the National Organization for Hydrogen and Fuel Cell Technology (NOW).

Within the Electric Mobility Pilot Region Stuttgart twelve associates have banded together to start the project Elena (retrofit electric drive kit) in order to promote electric mobility. The goal of the joint research project is the development of a retrofit electric drive kit for conventional vans with diesel engine and the assembling of a prototype for testing and demonstration purposes. Besides the development of the retrofit kit the holistic development approach included also the build-up of battery-charging stations as well as training and equipping of technical personnel.

The particular in this project is the fact that most of the project partners are small and medium-sized businesses supported by universities and research institutes from Baden-Württemberg. The Elena associates as well as their duties and responsibilities are pictured in ❶.

2 DEVELOPMENT STAGES

In the first stage of the project a hybridization concept had to be developed. Therefore the concepts and ideas which are available in the market were gathered and evaluated. After the analysis of possible hybridization concepts two hybrid types emerged as possible solutions for the retrofit electric drive kit: the serial and the parallel hybrid drive. In order to support the decision process for one of the preferred hybrid types a simulation model was generated that provided important information about vehicle characteristics such as acceleration, elasticity and range considering operational profile, recuperation and boost mode for both hybrid modes. Also the saving potential regarding diesel fuel and CO_2 emission for both hybrid modes and all possible operating strategies were calculated in the simulation and allowed an objective comparison. After analyzing the simulation results and evaluation of known advantages and disadvantages of both hybrid types, the Elena associates decided in favor of the parallel hybrid drive, ❷. It has the decisive advantage that the original driving characteristic is preserved and in addition pure electric and therefore emission-free driving is possible. The simulated acceleration of a loaded van with parallel hybrid drive is shown in ❸. For the further development process the decision for the parallel hybrid and the possibility of pure electric drive played an important role for the modification and control of the auxiliaries, which also have to be available in pure electric drive mode.

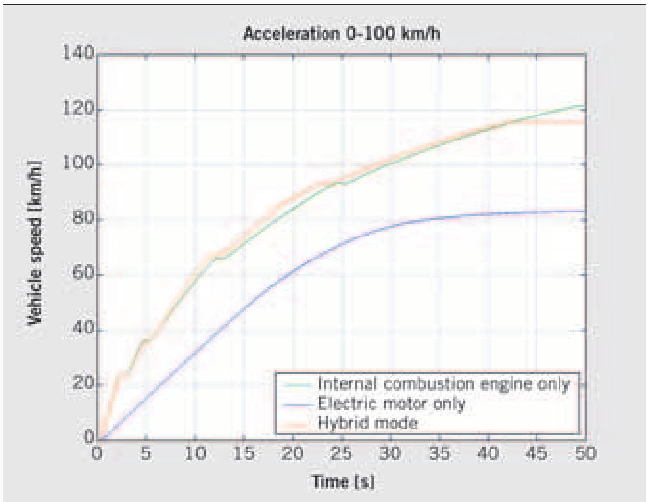
The first step of the development process was to search for capable vans for retrofitting on the market. Besides the interest of the associates to choose a van from a German manufacturer the sales figures and the capability to integrate a retrofit kit were important criteria for selecting a vehicle. As a result the Elena associates chose the Mercedes-Benz Sprinter 313 in the best selling

ARADEX AG	Electrical drive, design of electric motor
FRAUNHOFER IPA	Production concept, follow-up action, business plan, project management
FORSCHUNGSINSTITUT FÜR KRAFTFAHRWESEN UND FAHRZEUGMOTOREN STUTTGART	Simulation of driving characteristics
HELDELE GMBH	Battery charging station
HUBER AUTOMOTIVE AG	Drive control
IBZ, HOCHSCHULE ESSLINGEN	System architecture, requirement specification, test plan, coordination of integration
J. EBERSPÄCHER GMBH & CO. KG	Heating system
KOMPETENZNETZWERK MECHATRONIK BW E. V.	Communication and public relations
LAUER & WEISS GMBH	Mechanical design for system integration, cooling of battery and electrical drive
TELEMOTIVE AG	Communication interfaces
TÜV SÜD AUTOMOTIVE GMBH	Safety concept
WS ENGINEERING GMBH	Garage equipment, training

❶ Elena-associates and their duties and responsibilities

VEHICLE	Mercedes-Benz Sprinter 313 CDI
	High-top van
INTERNAL COMBUSTION ENGINE	OM 651 DE 22 LA
	P_{nom} = 95 kW
	M_{max} = 305 Nm
	n_{max} = 4200 rpm
ELECTRIC MOTOR	Motor technology: asynchronous
	Type of cooling: water cooling
	P_{nom} = 60 kW
	$P_{short\text{-}term\ overload}$ = 120 kW
	M_{nom} = 227 Nm
	$M_{max,\ short\text{-}term\ overload}$ = 458 Nm
	n_{max} = 5910 rpm
	$n_{max,\ dragged}$ = 8500 rpm
	l = 395 mm
	d = 268 mm
	m = 100 kg
TRACTION BATTERY CONSISTING OF 3 BATTERY PACKS	Q = 48 Ah
	E = 17.4 kWh
	U_{nom} = 360 V
EACH BATTERY PACK	l = 1080 mm
	b = 145 mm
	h = 200 mm
	m = 43 kg
GEARBOX – GEAR RATIO HYBRID STRATEGY	Hybrid mode: 1:1.423 (electric motor) 1:1 (internal combustion engine)
	Electric motor only: 1:1.423
	Internal combustion engine only: 1:1

❷ Technical data sheet



❸ Acceleration with loaded vehicle

version with the medium wheelbase and diesel engine with an engine output of 95 kW.

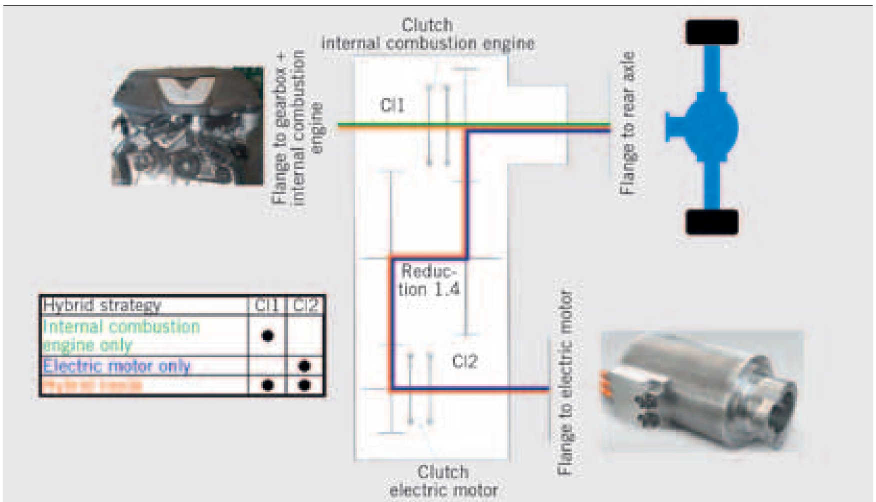
The next step was to improve the simulation in order to define basic technical data such as torque, power and battery capacity depending on the available space. The electric drive was designed based on the extensive simulation results gained especially in the pure electric drive mode with focus on gradability, vehicle speed and range. In the process the design of the electric drive caused a trade-off with the available space. The simulation was performed with a complex model in Matlab/Simulink whereas the design of components took place in a CAD model wherein the data gained in the simulation was adapted to the available space. The result mainly affects the design of the electric motor since the motor is attached to the cardan shaft of the vehicle over a gearbox and the space is limited.

Besides the electric drive which was developed especially for the retrofit kit several other components which are required for the electric drive mode had to be integrated in the vehicle. Among these components there are new developed components as well as components available on the market. The asynchronous motor is controlled by an inverter. Both components are matched accurately to each other and provide a compact and efficient system. The complete electrical drive system has been tested in detail on the test bench prior to the installation in the vehicle. The water-cooled electric motor has a continuous output of 60 kW and a torque of 227 Nm. The top performance output is 120 kW at a temporary maximum torque of 458 Nm. The electric motor including the housing weighs about 100 kg. An additional gearbox is necessary to transmit the power over a cardan shaft to the rear axle. Since the design of a completely new gearbox would have gone beyond the scope of this project regarding time and costs a cheaper and faster solution had to be found. A transfer case of a four-wheel drive vehicle was found as a feasible solution after a comprehensive market research and contacting the manufacturer of the transfer case. The two-part cardan shaft was substituted by a three-part shaft so that the electric motor could be connected to cardan shaft via the gearbox. The installed gearbox comes with two internal clutches that enable the connection of the electric motor and the internal combustion engine as well as the disconnection of the internal combustion engine from the rear axle as shown in ❹. As a result three possible driving modes are shiftable: internal combustion engine only, electric motor only and the combination of both.

To install the electric drive consisting of electric motor and inverter as well as gearbox and traction batteries under the vehicle special mounting brackets and frames were designed as shown in ❺. These parts have to withstand forces and protect the components from mechanical hazards and need to be retrofit on the vehicle. The design of these mounting brackets and frames was supported by stress analysis performed in finite elements models for different operating conditions.

A second cooling circuit (with an extra cooler) was installed to keep the drive components within a safe operating temperature range. Additionally, this circuit has additional water pumps to help dissipate heat while in both electric and hybrid mode. A flow simulation was performed on this cooling circuit in order to determine the necessary amount of coolant to keep the system within a safe thermal envelope.

Given the extensive investigation of the undercarriage space at the beginning of the project, it was possible to integrate the retro-



❹ Schematic diagram of the gearbox

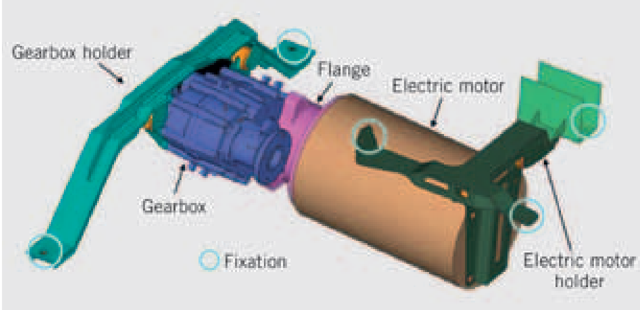
fit kit without extensive changes to the vehicle as well as maintaining the original storage space. Of those notable changes, the exhaust system had to be rearranged as well as the cardan shaft, ❻, and great packaging in the underbody helped to maintain storage space.

Aside the "normal" driving components, other components were necessary for the electric mode such that driving characteristics and comfort were not compromised. For instance, an additional heater was installed for driver comfort. This heater consisted of a heating module, a small ethanol tank as fuel for the heater, and necessary regulatory circuits. The heated air reaches the vehicle's interior through the original blower. The driver is able to regulate the desired temperature via an additional heating control element. Another peripheral component installed was the electrical steering pump to support the driver steer while in electric mode. This part is currently available in the market and was adapted to meet this special case.

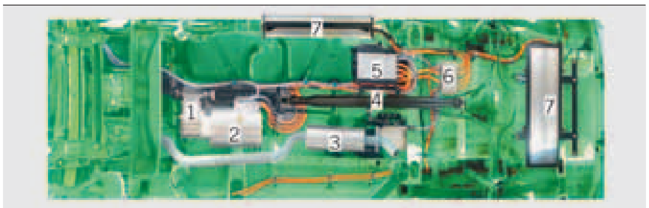
In addition to a low-voltage wire harness (to connect the additional low-voltage components), a high-voltage wire harness was also installed (to connect high-voltage components such as electric motor, inverter, traction battery, HV connector box, DC/DC converter and charger). The traction battery consists of three modules which are integrated in the vehicle's underbody with a nominal voltage of 360 V, 17 kWh capacity at a weight of 130 kg. The charger was installed under the co-driver's seat and supports loading single-phase at 220 V, as well as special three-phase charging stations. Based on the data, a range of 50 km can be attained in electric mode.

To control the entire system, a vehicle control unit was developed to control the communication between the new components. Furthermore, the new control unit is able to read data from the original vehicle control unit and includes them in its' work process. This is important for security related functions, such as ESP and ABS, so they are available in electric mode. The new control unit also monitors the functionality of the retrofit components, ❼. The newly developed vehicle control unit has been tested, verified, and validated in the simulation model without jeopardizing functionality and safety prior to installation in the vehicle. The driver does not notice this new control because the only interface the driver has is the newly developed touch screen enabled HMI display, ❽. The HMI display allows the driver to switch between driving modes and view important data, such as battery charging state.

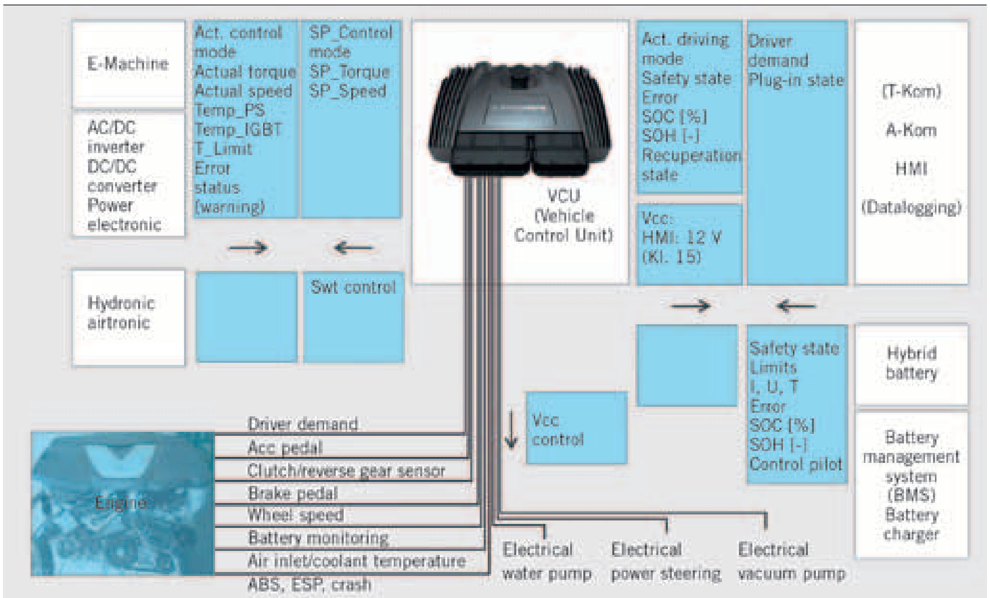
Apart from the conventional acceleration and breaking pedal, a retarder switch was installed in the cockpit. This switch allows the



❺ Motor/gearbox unit and fixation



❻ Components in the underbody of the Elena vehicle:
1 Gearbox
2 Electric motor
3 Rearranged exhaust system
4 Cardan shaft
5 Inverter
6 HV connector box battery charger
7 Traction battery



❼ System architecture control unit

driver to set the strength of recuperation in incremental stages while in electric mode, ❾. This gives the driver the option to load the traction battery while driving in hybrid mode. Additionally, this design gives control to the driver instead of being patronized by an electric control system. Thus, on long distance drives, it is better to use the internal combustion engine (where the internal combustion engine is considered more efficient) and leave the electric motor disconnected so it is not dragging the engine. When the traction battery is discharged, it can be loaded by connecting the electric motor to the drive and using the electric motor as a generator. This allows the driver to not be bound to a charging station. When the driver needs more power, such as going uphill or drive with more aggressiveness, he can use hybrid mode (thus harnessing the power of drive systems). In environmentally sensitive areas, he can drive in electric mode and not worry about emissions. This electric mode is good up to a regulated designed speed of 90 km/h. Via a steering column switch, the driver can brake the vehicle; thus, less wear on the conventional brakes. Since both the internal combustion engine and electric motor transmit mechanical force over the rear axle onto the road, the original driving dynamics of the rear-wheel drive does not change in all three driving modes.

❽ HMI display

❾ Cockpit
1 Recuperation lever
2 HMI touch screen
3 Heating control element

3 HOLISTIC DEVELOPMENT APPROACH

Essential elements for a successful development process within 18 months were the creation of comprehensive functional requirements with specifications for each component and the interfaces between those components as well as the very good coordination between the participating associates.

Besides the development of the retrofit kit the project included further important topics of electric mobility. A battery charging station was build-up including the necessary communication capabilities of the charging station and the vehicle. Three-phase loading of the Elena vehicle is possible with this charging station. Methods and training materials were designed to train and prepare technical personnel for the electrification of vehicles with regard to quality, cost effectiveness and safety at work. In addition necessary tools and devices for integrating the retrofit electric drive kit were developed and provided to garages. During the modification of the prototype each step was documented in an assembling instruction.

To guarantee the safety of the developed retrofit kit, the project was always executed considering current norms and standards. Part of this was the creation of an integrated safety concept. This included the implementation of an extensive hazard and risk analysis for all three available driving modes, a system analysis based on the specifications and the functional and electrical safety of the retrofit kit. All of the originally available safety systems are still available without restrictions in any of the three driving modes and guarantee a high-level safety after the modification.

The integration of the developed components into the Elena vehicle has been finished without any problems and the first test drives were successful. The next steps will be detailed inspection according to the test plan and an individual registration.

4 SUMMARY AND OUTLOOK

Within the Electric Mobility Pilot Region Stuttgart twelve associates have developed a retrofit electric drive kit for conventional vans with diesel engine and build-up a vehicle with this kit for demonstration purposes. The assembled and roadworthy prototype can be driven with internal combustion engine only, with electric motor only and in hybrid mode. The driver is able to select the driving mode via HMI touch screen. The screen displays all important data like battery state, recuperation and boost availability to the driver. The benefit of this hybrid concept is the possibility of driving the vehicle as originally with internal combustion engine on the highway without limitations. In downtown locations purely electric drive or hybrid mode is possible. The vehicle is equipped with regenerative brakes and therefore able to charge the battery while driving. For this purpose a recuperation lever with several stages was installed in the cockpit. Besides the development of the retrofit electric drive kit and the assembling of a vehicle for demonstration purposes a three-phase loading battery-charging station was build up. Training and equipping of technical personnel was also part of the project.

The next project phase is aiming at equipping more vehicles with the retrofit kit and testing under real conditions to demonstrate the suitability of the retrofit kit for daily use. The Elena associates are aiming in the long term for a homologation for a small batch series of the retrofit electric drive kit.