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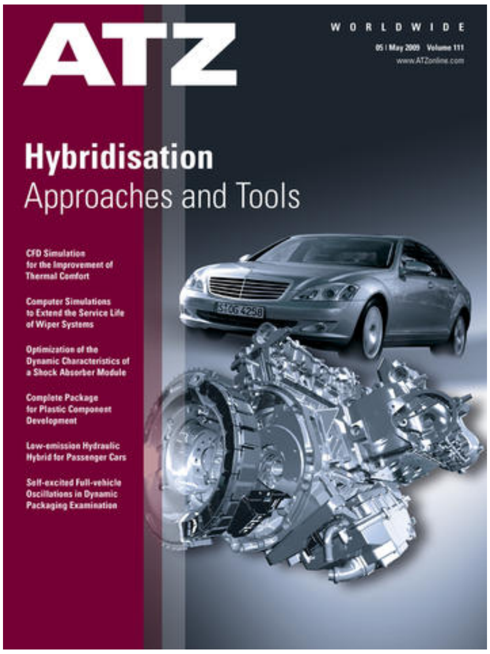

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Change in Values

Dear Reader,

The concluding panel discussion at the Stuttgart Symposium focused on the influence of the automotive industry on the world economy. Before the discussion began, some of the latest McKinsey data were presented. I found one of the charts personally very alarming. It showed that, in 2007, the entire international automotive industry generated earnings of $ 52 billion. In 2008, the figure had fallen to $ 17 billion and this year the industry even recorded a loss of $ 12 billion. This would not be too worrying in itself if it weren't for the prediction that, even on the basis of a very optimistic scenario, the profitability level of two years ago will not be achieved again until the year 2015.

One reason for this long period of recovery after the current downturn is the high level of investment required by new drive systems and the optimisation of vehicle CO_2 emissions. Against this background, there is a great temptation to postpone climate protection targets and research into new drive systems for a few years.

I can only warn against this. The reason is that customer awareness will continue to change and environmental protection will become even more important than ever. A Roland Berger study published a few weeks ago even predicts that customers would be prepared to pay more to protect the environment. It claims that as many as 20 % of German customers would be prepared to pay € 2,000 more for an environmentally friendly car.

And in spite of the financial crisis, there is still sufficient money available, for example in the classic oil-producing countries, to invest in new technologies. Abu Dhabi's investment in a share of the car maker Daimler was made explicitly against this background.

The automotive industry is reinventing itself – and ATZ is supporting you in this process, for example with the highly topical Cover Story in this issue.

Johannes Winterhagen
Frankfurt/Main, 5 April 2009

Johannes Winterhagen
Editor-in-Chief

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05|2009

Organ of the VDI-Gesellschaft Fahrzeug- und Verkehrstechnik (FVT)

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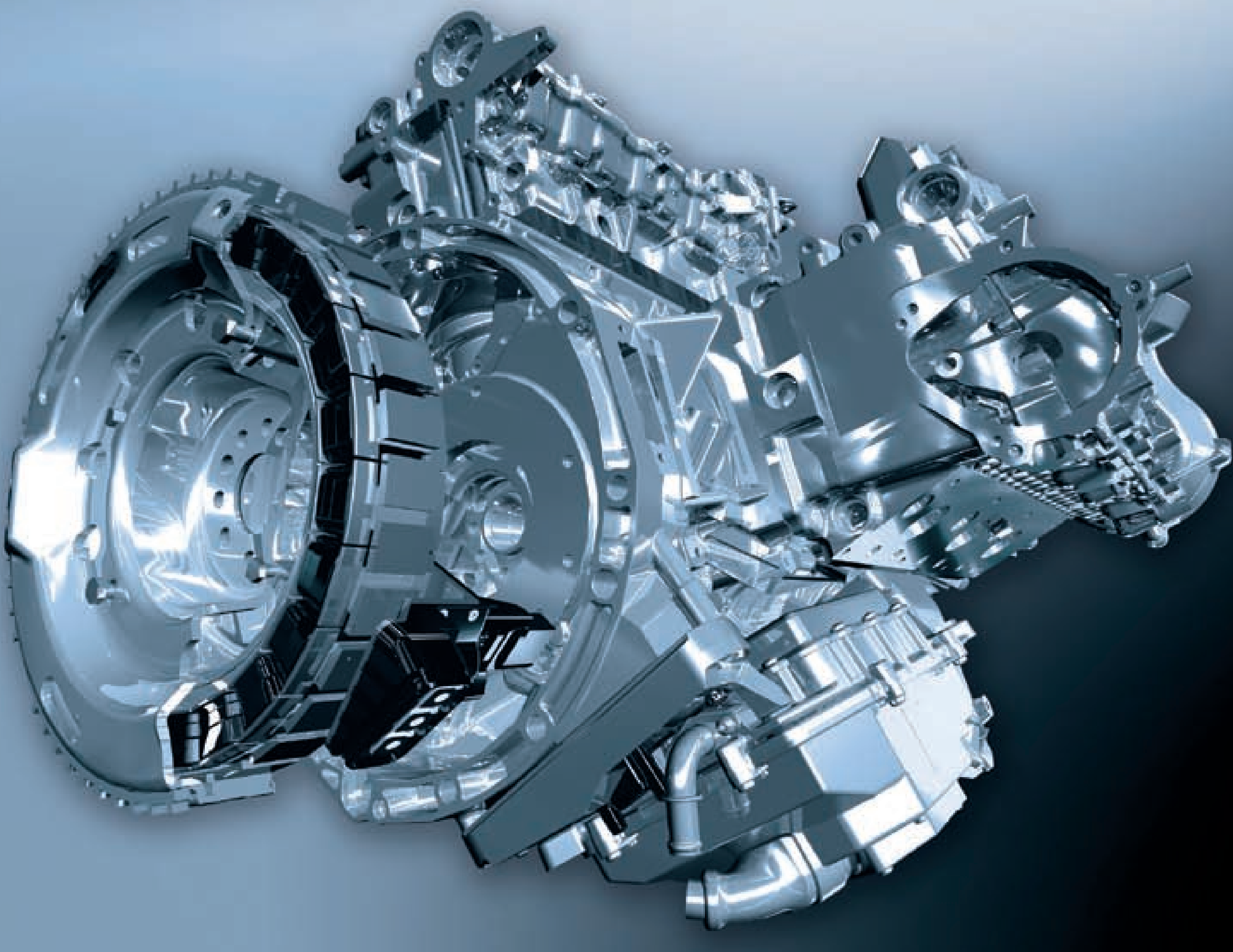
The Hybrid Solution by Mercedes-Benz in the S-Class



Mercedes-Benz is taking another step towards emissionfree mobility. The "S 400 Hybrid", available from this summer, is the premium brand's first hybrid vehicle. With its compact hybrid system and the world's first lithium ion battery in a production vehicle, Mercedes-Benz presents an innovative solution that offers the usual customer benefits of an S-Class while significantly improving fuel economy and CO_2 emissions.

1 Introduction

In order to achieve a general reduction in CO_2 emissions across all model lines, Mercedes-Benz is not only continuously improving the fuel consumption in its conventional drive systems. It is also consistently forging ahead with the electrification of the power train in all technology classes – from simple start/stop systems to various forms of hybridization right up to all-electrical systems such as battery-powered or fuel cell vehicles.

Following in the footsteps of the "BlueEfficiency" models already available in the A- and B-Class, which feature a start/stop system, the S 400 Hybrid will be the first vehicle to incorporate hybrid technology, and will be launched by Mercedes-Benz from mid-2009 on. In addition to the innovative Li-ion battery technology, this vehicle stands out in particular thanks to its use of a hybrid concept that is tailored to the needs of luxury class sedans. This therefore represents Mercedes-Benz's response to the growing demands for fuel efficiency that are being aimed also and in particular at the premium segment.

The most important objective for this vehicle was to fulfill without restriction all of the expectations that customers have of an S-Class and furthermore to reduce fuel consumption and CO_2 emissions to a minimum in the premium class.

In addition to retaining the agility and the familiar level of comfort, particular attention was devoted during the designing of the vehicle to ensure that there was no restriction of the effective load space and the load capacity avaiable to the customer compared with the base vehicle.

2 System Concept

In order to minimize the additional weight and space required to accommodate the hybrid system, a parallel concept incorporating just one electric motor, which is connected directly to the crankshaft, was chosen for the S 400 Hybrid, **Figure 1**.

Parallel hybrids are characterized by the fact that the electrical drive system is

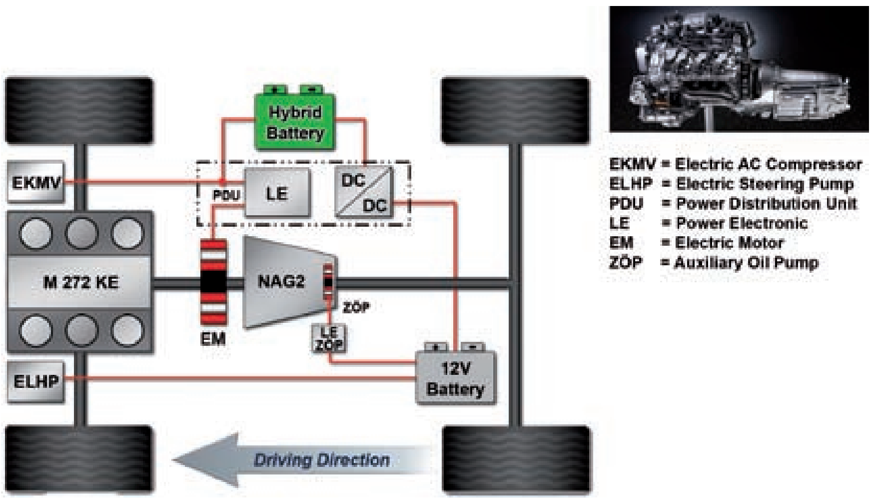


Figure 1: Hybrid concept for the S 400 BlueHybrid

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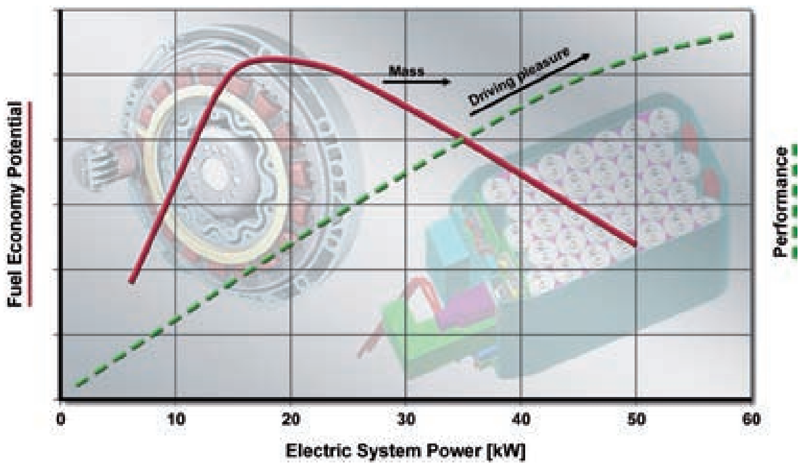

Figure 2: Fuel economy reduction potential

not combined with the conventional power train to create one integrated drive system, but is added on an additive basis. Both drive systems can therefore be either jointly or independently operated.

This allows a precise "metering" of the degree of hybridization tailored to demand, because – unlike serial or powersplit concepts – the power outputs of the two parts of the drive system do not have to be matched to one another.

An analysis of the fuel economy reduction potential in relation to the system output for the chosen vehicle/combustion engine combination shows that the maximum reduction in fuel consumption occurs in the range of 15 to 16 kW of electrical power, **Figure 2**.

At higher power outputs, the potential falls again, which is due on the one hand to the accompanying increase in system weight and, on the other, to the fact that the selected electric motor is operated increasingly in less favorable areas of the performance map as maximum requirements increase.

In order to achieve the desired saving potentials, an electrical design rating of approximately 15 kW is therefore stipulated. Along with the analysis of the system power taking into account the efficiency, additional measures aimed at reducing consumption have been implemented. For example the main accessories like power steering pump and A/C compressor are electrified to reduce the belt losses.

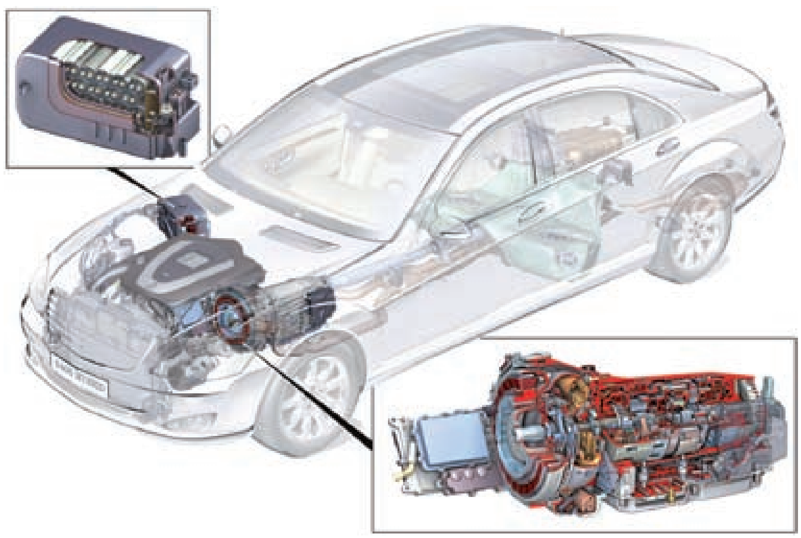

Figure 3: Packaging of the hybrid drive components

3 Hybrid Components

The core of the hybrid system is comprised of the high-voltage battery, the electric motor and the power electronics required to control and supply the electric motor with three-phase alternating current. Unlike previous hybrid designs, these and all the other components modified for hybrid operation are accommodated in the engine compartment of the S 400 Hybrid, **Figure 3**. This marks the first time that a production vehicle has been successfully hybridized without the need to intervene in the structure of the existing base vehicle bodyshell.

3.1 High-voltage Battery

The requirement to integrate the hybrid system as far as possible in the existing installation spaces, given a desired electrical system power of approximately 15 kW, puts the spotlight firmly on lithium-ion technology for the hybrid battery. Only in this way is it possible to install the high-voltage battery in place of the conventional starter battery, as the new technology is clearly superior to conventional nickel metal hydride technology with respect to energy content and power-to-size ratio.

Accordingly, an energy content of 0.8 kWh and a capacity of 6.5 Ah can be achieved in the amount of space formerly taken up by a 90 Ah starter battery.

3.1.1 Cooling System

Due to its installation in the engine compartment, the battery is exposed to substantial ambient temperatures. Since its service life is largely determined by the temperature profile to which it is subjected over its lifetime, the provision of a cooling system is essential.

To prevent battery cooling from dominating the air conditioning in the vehicle interior, cut-off valves are integrated into the system that allow the customer to switch off the air conditioning without interrupting the battery cooling, **Figure 4**.

When the engine is not running, the electrical A/C compressor not only provides the air conditioning but also guarantees that the battery's operating temperature limits are not exceeded in this state either. This ensures that the battery is not heated above 50 °C in any operating state, since it could be seriously damaged otherwise.

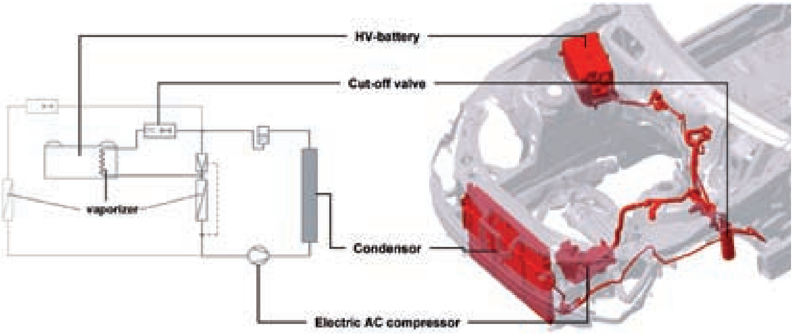

Figure 4: Cooling system for lithium-ion battery

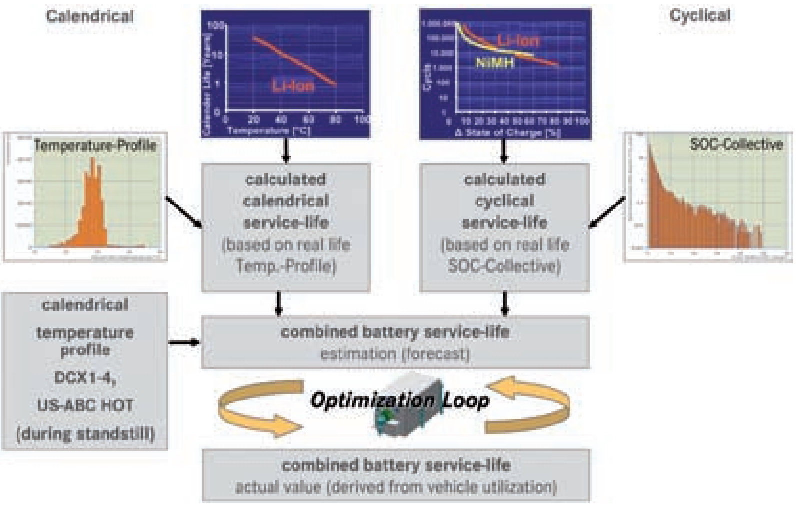

Figure 5: Lithium-ion battery service life

3.1.2 Durability Analysis

The battery can be negatively affected not only by transgressions above the operating temperature limits, which can damage the high-voltage battery directly, but also by the temperature profile during its lifespan, which can gradually reduce its service life over a long period. To ensure a long service life for the battery, it is therefore important to keep the average temperature as low as possible. This will have a decisive impact on the calendrical service life.

Factors of decisive importance in determining the real-life durability are not only the calendrical but also the cyclical service life, which in turn can also be determined by the number of charging and discharging cycles. Since this number cannot be influenced while the vehicle is running, efforts must be made to minimize the depth of discharge per cycle (<5 %).

These two factors must therefore be observed in real-life operation in order to obtain evidence of the average durability of the lithium-ion battery, **Figure 5**.

Data recorded in conventional and hybrid vehicles is used to create temperature profiles and SOC (state of charge) spectrums that enable an initial assessment of the service life. The assumptions are then continuously checked and optimized based on the most recent development statuses.

Current evaluations show that the battery temperature remains predominantly in the range between 10 °C and 30 °C and that it practically never exceeds 40 °C, **Figure 6**. With regard to the cycle strength, it can be seen that the focus of SOC differences (start class to target class) lies within the range up to 5 %. Values up to 10 % also occur at a very much lower level, while SOC cycles with a charge/discharge depth of more than 10 % are rarely observed, **Figure 7**.

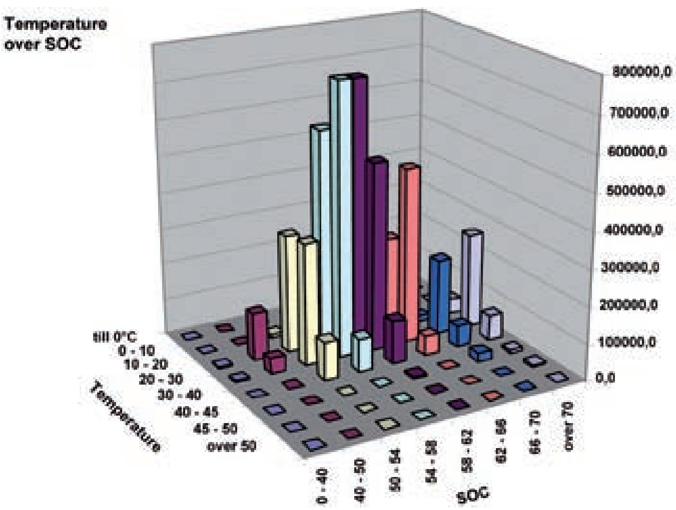

Figure 6: Real-life temperature profile for lithium-ion batteries

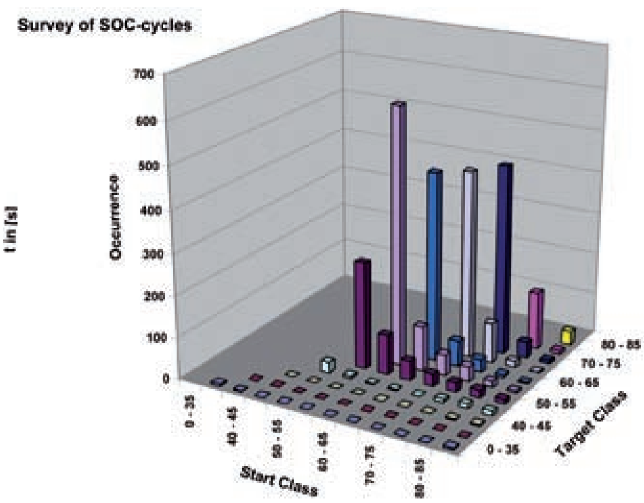

Figure 7: Real-life SOC spectrum for lithium-ion batteries

This evaluation makes it possible to confirm the expected service life of over ten years both on a calendrical and a cyclical basis.

3.2 Electric Motor

The electric motor in the S 400 Hybrid is a permanent magnetic synchronous motor featuring an external rotor design. It is connected directly to the crankshaft. Its layout allows for an extremely compact package that results in the drive train being lengthened by a mere 45 mm. Since the design of the S-Class had already made provision for an increase of 65 mm to accommodate possible future hybrid variants, the electric motor can be easily integrated into the existing drivetrain environment.

With a maximum starting torque of 210 Nm, the electric motor is capable of starting the combustion engine quickly and smoothly even at temperatures below -25 °C. The other key data such as rated torque of 160 Nm and maximum power of 15 kW mechanical/19 kW generator mode show that the electric motor can be used as an effective drive support.

Despite the impressive performance data, the electric motor does without air or liquid cooling. This also contributes to the straightforward integration of the electric motor into the drive train.

3.3 Power Electronics

The power electronics are a combination of the control unit for operating the electric motor and an inverter to generate the three-phase alternating current required to operate the electric motor on the high-voltage battery's direct current network.

The DC/DC converter required for switching the voltage between the 12 V on-board electrical system and the 120 V high-voltage network is not integrated as a component in the power electronics, but is installed separately for packaging reasons.

The power electronics are installed in the area of the conventional starter that the electric motor replaces. The maximum currents generated by the inverter are approximately 150 A continuous and 310 A short-time. The heat generated by the inversion of currents of this magnitude, as well as the not insignificant entry of heat into the area surrounding the manifold, is dissipated with the aid of an additional low-temperature water circuit.

The power electronics are directly connected to the electric motor by means of a bus bar. This not only reduces EMC problems but also allows for an extremely compact design here as well thanks to the elimination of cables and plug contacts.

3.4 Hybrid-specific Auxiliary Drives

Apart from the core components of the hybrid system, a range of auxiliary drives are also fitted. As mentioned above, these have been modified for hybrid application on comfort and fuel-economy reasons. The most important of these are the DC/DC converter referred to above for supporting the generator, the regenerative braking system, the electro-hydraulic steering, and the electric A/C compressor, **Figure 8**.

3.4.1 DC/DC Converter

The 12V/120V voltage converter installed in the right wheel arch is used primarily to provide fuel-saving support to the on-board electrical system without incurring belt losses. However, it features a bidirectional design that also allows the 12 V side to be used to aid cold starting or boosting. The performance data is 1.5 kW in the 120 V to 12 V direction, and 0.5 kW in the 12 V to 120 V direction.

3.4.2 Regenerative Braking System

The use of a regenerative braking system (RBS) is necessary to ensure an imperceptible combination of conventional mechanical braking and the electrical braking power provided by the electric motor operating in generator mode. To achieve this, the system uses a pedal travel simu-

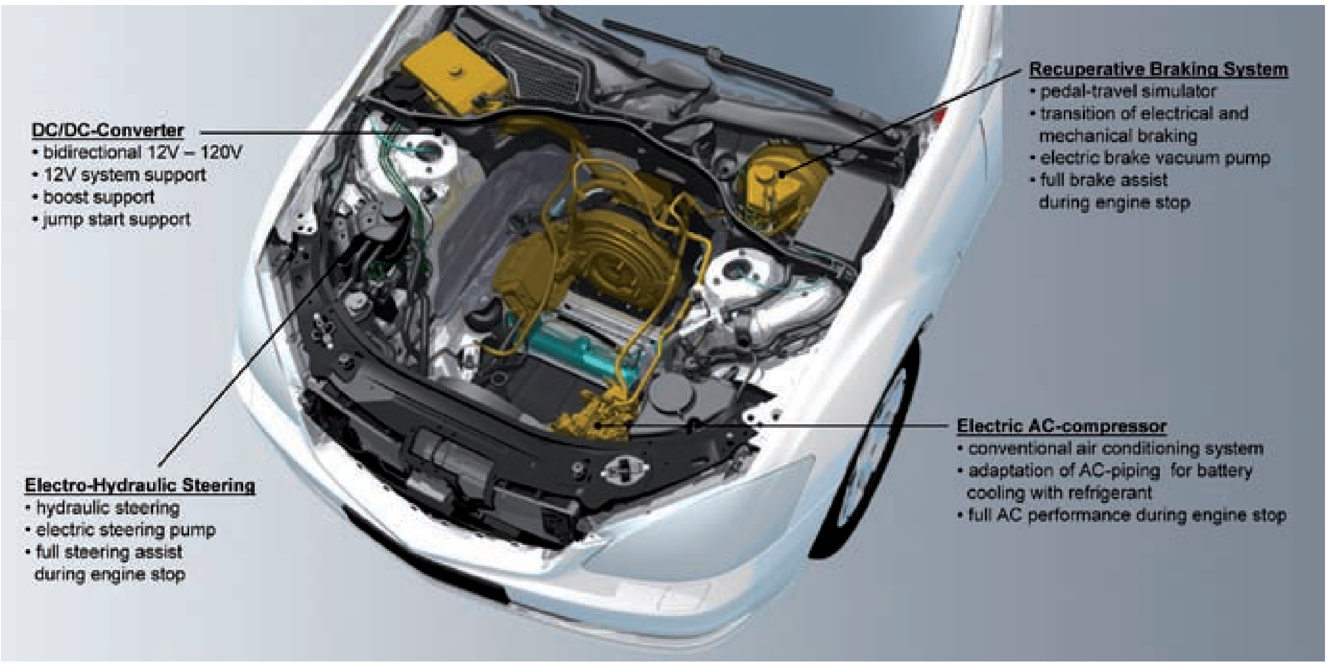

Figure 8: Hybrid-specific auxiliary drives

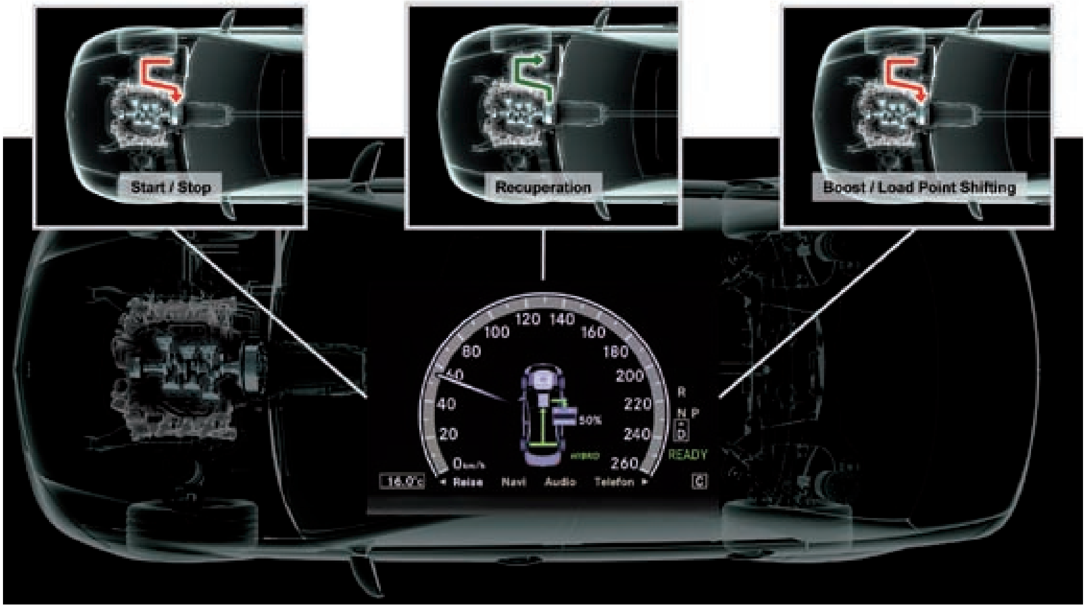

Figure 9: Hybrid functions

lator that provides the customer with pedal feedback that always corresponds to their braking torque requirement, regardless of whether the torque is mechanically or electrically generated.

This makes it possible to provide a higher regenerative performance – without an impact on comfort – than in non-regenerative systems in which electrical braking torque is simply added to the mechanical torque and where the customer receives different, non-reproducible overall braking torque levels for similar pedal positions depending on the battery charge state.

In addition to the adjustments for the regenerative braking system, a further modified component is the vacuum pump. Thanks to the electrification of this system, the customer has the full braking power of the system at their disposal even when the engine is not running. As a result, the brake pedal can also be subjected to repeated "pumping" when the car is standing still, without any loss in performance.

3.4.3 Electro-hydraulic Steering

The adaptations to the steering system relate primarily to the power steering pump, which was electrified for the hybrid application and relocated between the right wheel arch and the headlamp in the front end. The other components of the steering system remained largely untouched.

The electrification of the power steering pump has two main effects: on the one hand, the elimination of belt losses of approximately 0.2 l/100 km helps reduce fuel consumption. On the other hand, the electro-hydraulic steering system makes it possible to switch off the combustion engine before the vehicle reaches a standstill, which in turn saves fuel without any impairment of comfort.

3.4.4 Electric A/C Compressor

Similar to some of the auxiliary drives already mentioned, the move from a mechanical to an electric A/C compressor also delivers fuel-saving effects, although the main focus in this case is on retaining the climate comfort when the engine is not running. In virtually all customer operation scenarios, it is therefore possible to leave the combustion engine switched off, even at a standstill for longer periods, without any impairment to the accustomed level of climate comfort.

The compressor remains in the standard installation space provided for coolant compressors alongside the combustion engine. The power electronics required to operate the unit are integrated in the compressor and occupy the space that was freed up in the belt-drive system.

The remaining components of the conventional air conditioning system can largely be reused. Only the A/C lines and branch valves require modification to allow integration of the battery cooling system as described above.

4 Hybrid Functions

The previous sections have already clarified the importance of an operating strategy that is tailored to suit the requirements of the respective application. To fully exploit the fuel economy potential, for example, it is important to determine the correct degree of regenerative braking, but equally, to keep the battery's SOC range to a minimum for durability reasons.

The operating strategy must therefore strike the best conceivable balance between the sometimes contradictory requirements so that the vehicle can satisfy the customer's expectations as regards comfort, durability and fuel consumption (to name just a few areas) to the best of its ability.

The operating strategy of the S 400 Hybrid is based around the hybrid functions start/stop, regenerative braking, and boost (drive system support) or load point shifting provided by the concept. In other words, it involves moving the combustion engine's load point to less consumption-intensive areas of the performance map while at the same time offsetting the torque deficit with the aid of the electric motor, **Figure 9**.

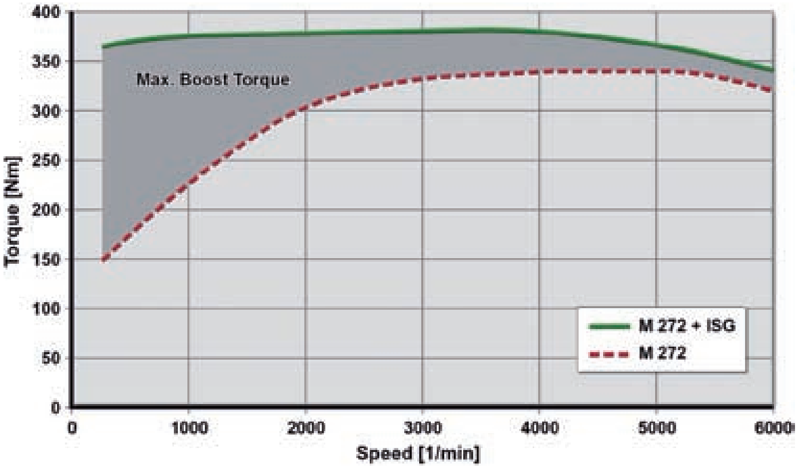

Figure 10: Combined torque curve

Due to the position of the electric motor directly on the crankshaft, driving using solely electrical power is not a viable option.

4.1 Start/Stop

The direct connection between the electric motor and the crankshaft allows exceptionally quick and smooth starting of the combustion engine from when it was not running. This represents one of the main advantages of this concept.

The startup procedure of just a few hundred milliseconds in duration also offers significantly improved comfort for cold starts. The ability to start the engine at high RPMs also releases fewer harmful emissions.

Furthermore, the use of the electrohydraulic steering system allows the combustion engine to be switched off even before the vehicle comes to a stop (as outlined above). Depending on when the signal to release the automatic transmission is given during braking, this is also possible at speeds of around 15 km/h. Engine stop times in traffic jams, for example, are significantly increased as a result.

4.2 Regenerative Braking

In order to fully recover the maximum power as determined by the size of the electric motor and the high-voltage battery without incurring any loss in comfort, the vehicle is fitted with the regenerative braking system described above.

However, the recovery of energy is not confined to braking maneuvers, it is already taking place during vehicle deceleration. The level of recovery must be carefully selected, though, as the electric motor operating in generating mode in this case exerts an amplifying effect on the engine brake, which can negatively impact customer comfort if the level of recovery intervention is too high.

Furthermore, the deceleration fuel cut-off phase in hybrids can also be extended to virtually the engine idling speed because the electric motor prevents the combustion engine from stalling even at low engine speeds. Taken together, these measures – during deceleration and braking – account for most of the reduction in fuel consumption.

4.3 Boost/Load Point Shifting

The term boost describes the electrical drive system support provided in addition to the conventional drive system powered by the combustion engine. Since the torque generated by the electric motor is at its highest at zero and low RPMs, the support for the vehicle's agility is particularly effective and noticeable when moving off after a stationary phase (pulling away from traffic signals) and when accelerating from low speeds.

The torque generated by the combustion engine in the low engine speed range is boosted by the electric motor in such a way as to create a combined torque curve that is much higher but also more harmonious across the entire engine speed range, **Figure 10**.

In order to prevent any negative influence on the cycle strength of the high-voltage battery, the operating strategy must be carefully configured to ensure that the discharge depth and frequency do not become too high on a permanent basis. Consequently, the electrical support should only be brought fully into play if actually requested by the customer.

Indicators of the customer request for swift power delivery, which are taken into account by means of the operating strategy, include a high accelerator pedal position and a large pedal value gradient.

The other aspect of electrical support not associated with agility is load point shifting. This involves deliberately reducing the load on the combustion engine and allowing the electric motor to supply the resulting difference to the customer's torque requirement. The discharge depths in this case are usually low, so that any impairment to the cycle strength of the battery is minimized.

Load point shifting can be used effectively especially during steady-state highway driving, which enables the hybrid to achieve consumption benefits here as well.

5 Summary

The objective for the S 400 Hybrid – to develop a hybrid vehicle that retains or even exceeds the existing level of customer benefit as regards comfort, effective load space and agility while also achieving significant reductions in fuel consumption – has been successfully implemented. The entire hybrid system is part of the front end; only existing installation spaces are used and an additional weight penalty of just 75 kg for the hybrid system means there is no need to restrict the payload. The targets were achieved above all thanks to the world's first use of a lithium-ion battery in a production vehicle.

The hybrid concept employed provides the vehicle with a noticeable improvement in agility when pulling away from traffic signals. Acceleration from 0 to 100 km/h is also improved by 0.1 s to 7.2 s. Fuel economy is 7.9 l/100 km in the NEDC. This represents a reduction of approximately 20 % compared with the conventional base vehicle. With a CO_2 output of 186 g/km, the S 400 Hybrid is even the "CO_2 world champion" of its class.

Taken all properties together, the S 400 Hybrid with its hybrid concept and the lithium-ion battery technology meets all expectations that a customer demands of a vehicle of this class. In addition new standards are set under environmental aspects.

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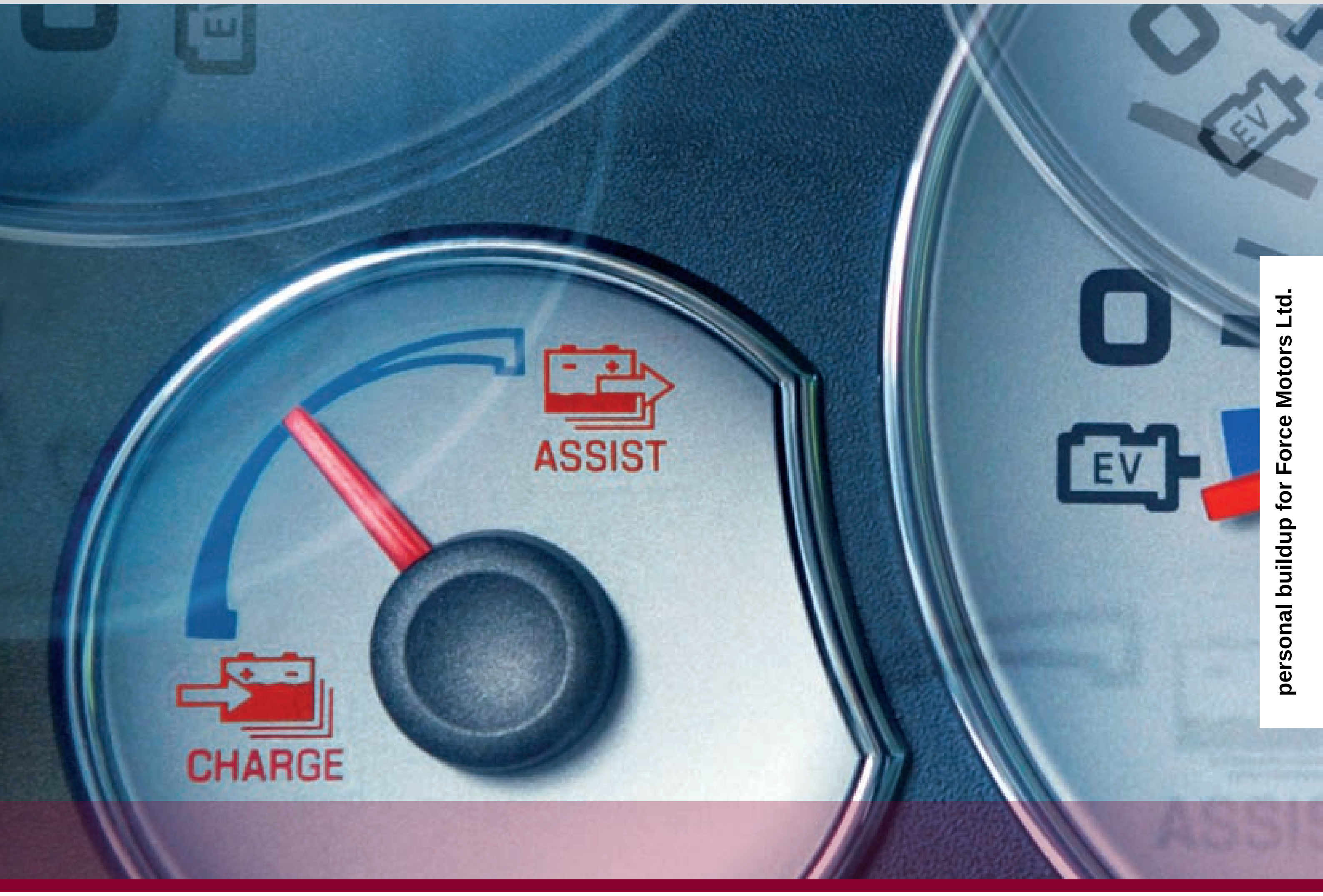

Simulation as a Solution for Designing Hybrid Powertrains

New drive concepts for cars are placing increasing demands on developers. A large number of specialists with extensive expertise are available to design conventional drive systems. However, for new types of powertrain, despite the availability of high levels of skill in individual disciplines, there is often no clear overview of the way in which the overall system behaves. The complexity of hybrid vehicles demands tools which allow developers to assess the consequences for the overall behaviour of the system of modifications to individual components and functions. IAV GmbH and Consulting4Drive GmbH (C4D) therefore recommend investing ten percent of the development budget in simulation.

1 A Matrix for Correct Product Decision-making

Development costs and times can be reduced significantly by systematically incorporating simulation into the product development strategy. In normal development practice, this takes place at the earliest when the main parameters for the product have already been established. However, this potential can be exploited to a far greater extent by using simulation to include the design-related differences of existing powertrain concepts as early on as the product decision phase which will help to ensure the financial success of the product.

With increasing competitive pressure, it is essential not only to satisfy the basic need for mobility with new concepts, but also to offer a product that is exactly tailored to the target consumer's requirements at the right time and in the right market. In this context, the complexity of the decision-making process increases with the number of parameters that need to be considered. The legal requirements are constantly growing on the homologation side and making new concepts necessary, while the requirements of car buyers vary widely, ranging from high performance to environmentally friendly solutions. New concepts frequently involve considerable investments from OEMs and subcontractors. When deciding on a product, therefore, economic success is largely a matter not of being committed to a promising powertrain concept from the very outset, but of employing the right criteria to assess the technological possibilities. One effective method of doing this comes in the form of the Product Strategy Decision Matrix (PSDM) which has been developed by Consulting4Drive GmbH in collaboration with IAV GmbH and validated in a wide range of projects, **Figure 1**.

An initial market and target consumer analysis establishes the customer-related parameters of the product being developed for the various markets, as well as the sales volumes anticipated over the product life cycle.

In addition to a specific, reliable description of the characteristics which the concept can deliver, such as driving dynamics and consumption, simulation also provides the initial design of the necessary components at a very early stage in the process. Concept variants can be produced in rapid iterations. This provides a sound basis for estimating the anticipated development and production costs and for identifying potential synergies within the product range. The evaluation is based on the resulting business models. In this context, relatively small conceptual differences can produce a significant change in the return on investment.

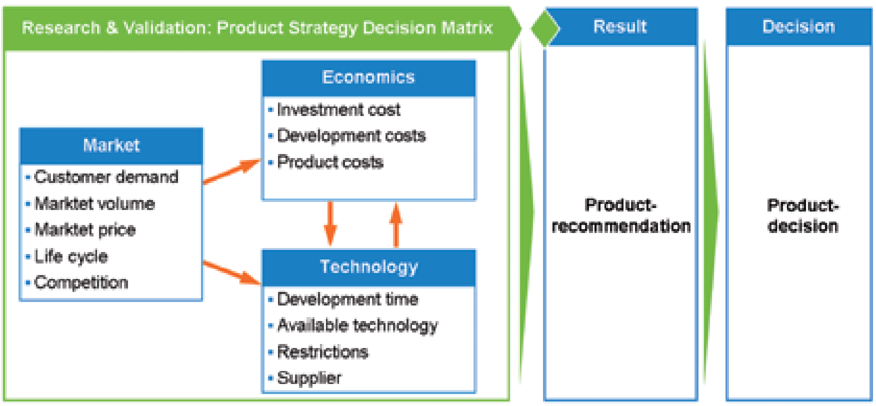

Figure 1: Interaction of market, profitability and technology factors in the product and investment decision-making process

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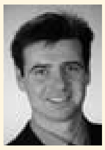

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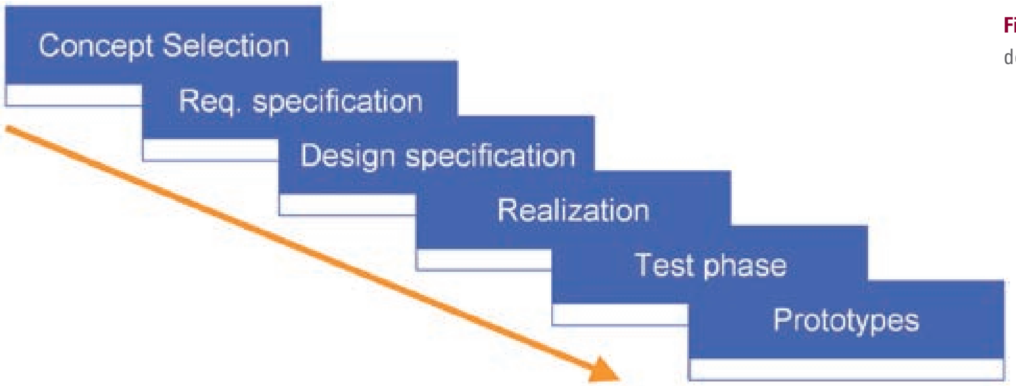

Figure 2: Phases of the development process

2 Simulation in the Development Process

The consistent use of simulation offers the possibility of managing the overall development process more effectively. Alongside the selection of the concept as part of product decision-making, **Figure 2** shows the main stages in the subsequent development process.

After the product decision-making concept has been successfully completed, countless constraints remain, which turn the process of producing a complete and coherent definition of the requirements into an unmanageable task. The example below illustrates the benefits which simulation can bring in the subsequent sub-processes when it is used consistently throughout the project.

The point of departure for this analysis is a concept with a starter generator and an electric drive system supplied by the same battery, which has a limited power supply. If the driver accelerates when the petrol engine is not running, the maximum power must be available. Starting the engine at lower power results in increased support from the electric motor. Although starting the petrol engine more quickly leads to its power being delivered sooner, this also reduces the electric motor's boost capacity.

The requirements specification should have defined the overload behaviour of the electric motor and of the starter generator that is applicable in a scenario of this kind. Conversely, the acceleration time defined in the requirements specification has repercussions on the power demands made on the electric motors. The situation described above and the two different requirements give rise to a typical conflict that can only be resolved using simulation.

A more far-reaching sensitivity analysis is able to reveal the correlation between the power output levels of the electric motor and the acceleration times. In the design specification, the component parameters normally diverge from the requirements. The sensitivity analysis allows conclusions to be drawn with regard to the vehicle's behaviour as well as its performance data.

In the implementation and testing phase, components from the overall system are in most cases available for testing functions and scenarios, as well as for examining the requirements in the specification. At this stage IAV GmbH uses component-in-the-loop simulations which involve incorporating real components into the overall simulated system. In the example given above, the engine management system, the petrol engine and the starter generator can be operated on the engine test bench, with the remainder of the vehicle, including the electric motor, being simulated. Full-load acceleration can then be investigated on the test bench together with the early influence of the operating strategy on parameters such as consumption and emissions [2]. From the other perspective, the real components provide parameters which allow the data in the simulation models to be optimised. If networked engine management functions and the interaction between the physical components are now also included in the simulation, models of a sufficient level of detail can be produced for use in algorithm development and more extensive tests.

Existing simulation models must be verified during the prototype phase. This enhanced basis can be used for conducting analyses of a more robust nature and for simulating model variants. Identifying faults in real systems often proves extremely complicated and costly. In addition, it is sometimes difficult to ascertain and measure the cause of faults. A model that has been kept up-to-date using the processes referred to above and that describes the principal physical and engine management relationships can make finding faults significantly easier.

As development progresses, increasing demands are placed on the simulation models. Only by consistently advancing and updating the models is it possible to guarantee added value in each phase. If the process of updating the model is interrupted during the course of the project, there is rarely an opportunity to make up for the interruption. For this reason, the simulation model must have a highly user-friendly design.

3 Requirements on the Simulation Concept

The factors described above can provide the basis for identifying the specific demands placed on a simulation process which supports product development:
- configurable, standardised simulation platform
- capability of simulating the entire vehicle
- modular simulation concept
- networking capability
- level of modelling detail adapted to suit the problem.

The basic simulation platform is intended to provide the user with the ability to simulate the overall vehicle's individual components from a functional and physical perspective. The models should be manageable in terms of configuration, parameters and simulation results.

It should be easy to combine the components provided to create an entire vehicle. There should also be a clear separation between physical models and engine management models, both from the perspective of components and signals.

In addition, it should be possible to simulate the function of each component individually as well as in the overall vehicle. This allows components to be used in hardware-in-the-loop and, in particular, in component-in-the-loop simulations. The simulated interfaces between the individual components should, as far as possible, reflect those in the real vehicle.

Component models which provide the corresponding level of detail often already exist on different simulation platforms. Modifications to the model can be investigated more easily and a more stable simulation of the overall vehicle can be produced if the models do not need to be exported.

Depending on the application of the simulation, models need to be produced with different levels of detail. In particular, cycle simulations and sensitivity analyses relating to global parameters tend to require simple models, whereas drivability evaluations, for example, require a significantly higher level of detail [1]. **Figure 3** shows the dependency between model detail and model benefit. The optimum solution takes the form of a specific ratio. This dependency can only be avoided if component expertise can be brought together using a simulation tool.

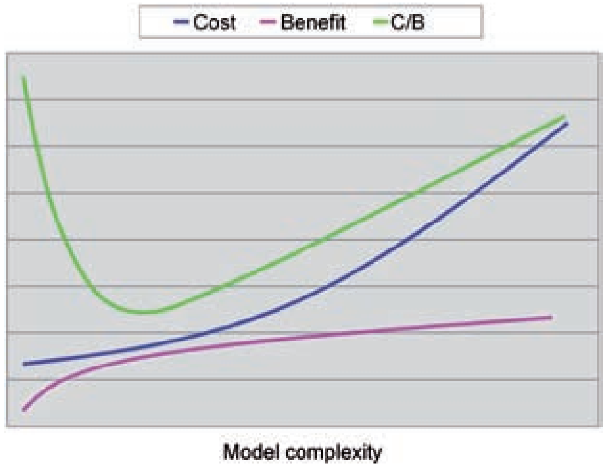

Figure 3: Cost-benefit ratio from using simulation

4 Incorporating Driving Dynamics

Whereas pure longitudinal dynamics and therefore consumption throughout the cycle can be thoroughly investigated using a high-quality simulation of the powertrain, additional vehicle characteristics must be included to produce results relating to lateral dynamics and therefore to vehicle safety. A networked simulation of the powertrain and chassis allows the following important influencing factors on driving dynamics and vehicle safety to be taken into account at the concept design stage:

- final drive concept (FWD/RWD/AWD)
- positioning of the hybrid components and therefore the mass distribution/centre of gravity
- safety aspects of recuperation and brake blending
- braking control systems (ESC, traction control/engine-drag torque control)
- electrical energy requirements and the reserves necessary for steering and braking.

The final drive concept (front, rear or all-wheel drive), mass distribution (particularly the weight and position of the battery) and the recuperation and gear change strategies have a major influence on the lateral dynamic behaviour and therefore on the safety of the vehicle.

In a conventional powertrain with an electric motor merely inserted between the petrol engine and the transmission, a simulation-based examination of differing lateral dynamic influences could focus simply on adapting engine-drag torque control and identifying a suitable safety concept for shutting down recuperation. In this powertrain the ESP could also act as safety supervisor of the driving dynamics without any fundamental modifications to the controller concept.

However, if a hybrid drive system takes the form of an independent, electrically powered rear axle added to a front-wheel drive vehicle, the result is an all-wheel drive system which has an all-wheel drive clutch consisting of software. In this case, the axles' drive torque and drag torque are not mechanically linked either in terms of their ratio to one other or their preceding sign.

Even the simulation of simple driving manoeuvres, such as braking or accelerating while cornering with additional torque on the rear axle, can help to determine the important parameter limits of the driving strategy in relation to safety. The simulation of the overall concept with the powertrain, driving strategy and chassis produces results which highlight and allow for further investigation of potential faults (for example, sudden drag torque on the rear axle) that may have repercussions on the design and selection of components or even call into question the choice of a specific concept.

Alternative drive concepts for hybrid or electric vehicles still feature wheel-hub motors or, as in IAV's proposal [3], individual electric motors for each wheel on an open differential, **Figure 4**. This provides the capability, which is highly effective in terms of driving dynamics, of allocating drive and recuperation torque to each individual wheel and allows braking, which would stabilise the vehicle but waste energy, to be avoided in many situations. In this case, the networked simulation of longitudinal and lateral dynamics provides decisive support in designing components to optimise driving dynamics and enhance safety while reducing consumption.

Regardless of the drive concept being investigated, detailed simulation of the influence of driver, road and chassis also produces important findings during the development process. It is particularly effective for analysing the interaction between the component controllers for the petrol engine, transmission and electric motor and the active control systems for the chassis.

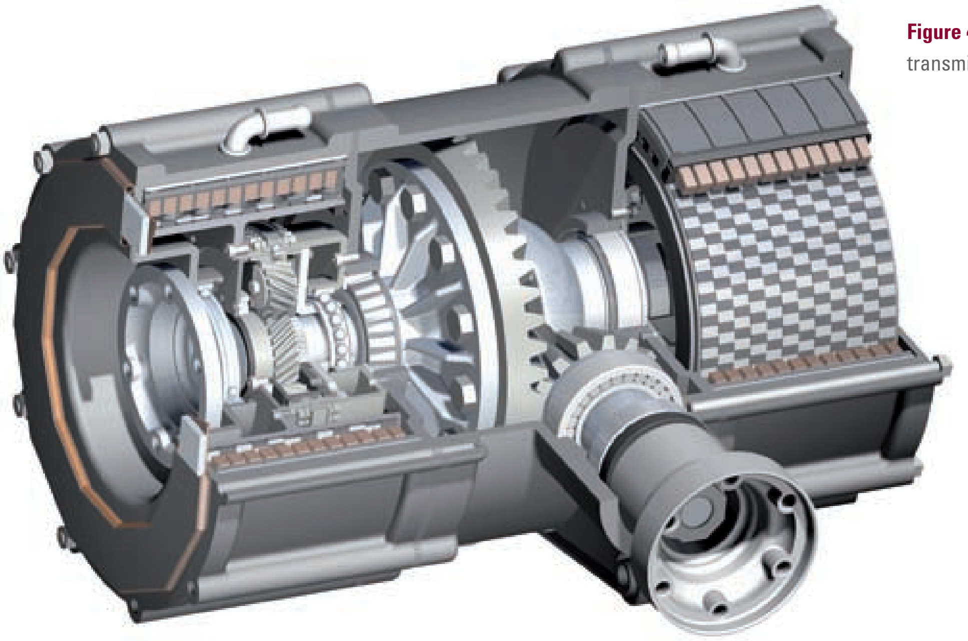

Figure 4: IAV's torque-vectoring transmission

5 Networked Simulation

Each different area of application for the modelling process places differing demands on the complexity and level of detail of simulation models. In order to meet these demands, various simulation tools have been developed for each level of detail and area of application. However, modern mechatronic systems involve close links between the different areas and this often means that individual simulation programs are not able to describe the overall system behaviour to a sufficient degree of accuracy.

The solution in this case can be to link the programs in the form of a co-simulation and to import partial models from other programs. Networking programs by means of co-simulation offers a number of advantages over the export/import option.

Interactive simulation allows the native interface to be used and all the internal states to be accessed. It also offers full debugging functionality. In each case the optimum solver can be used and parallelisation reduces the simulation time. Different tools, such as Matlab/Simulink, Dymola and TargetLink, can be linked together and the LAN/WAN capability offers the option of using non-local licences. The reuse of models allows for an integrated platform for SiL, MiL, HiL and CiL.

Figure 5 shows the structure of a modular, component-based simulation environment of the type used at IAV. When selecting the integration platform, it is

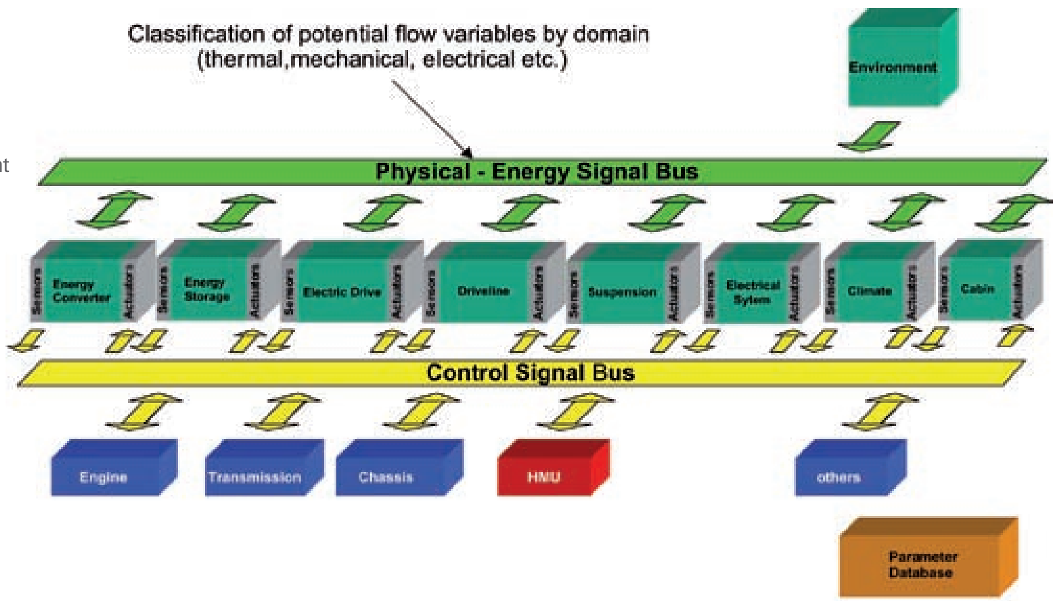

Figure 5: Simulation environment for hybrid drive concepts

important to ensure that it is both suitable for use as a standalone solution for concept studies and also provides open interfaces to other programs. For this reason, the Simulink-based "VeLoDyn" complete vehicle simulation program developed by IAV was used as the platform. Additional tools can be linked in using "EXITE ACE". The advantages to be gained from using a simulation environment of this type for various applications, including lateral dynamics, heat flows and developing hybrid control systems, have been verified in the company's own investigations [4, 5, 6].

In addition to Matlab/Simulink, which has become the standard tool in the field of simulation, there are a number of other tools, such as Dymola/Modelica, SimulationX and AMESim which, in particular, provide the capability of cross-domain modelling in one single program. Unlike Simulink, these are not based on signal-flow-oriented modelling, but take an object-oriented approach. Modelling takes place at the level of the components being simulated, with the relevant differential equations being generated by the program. The clear benefit lies in the shorter development times, as the user is relieved of the complex and error-prone task of producing differential equations. In the case of models with several degrees of freedom (for example, detailed transmission models), a time saving of several days is possible. This benefit, however, also comes at the price of a disadvantage that must not be underestimated. The generation of equations and real-time behaviour on HiL systems are in most cases no longer transparent to the user. This puts the above-mentioned time saving into perspective. In addition, the user must incorporate his own knowledge into the modelling process via the equation generator.

Alongside the practical benefits and drawbacks of different simulation tools, acceptance by the users in the relevant departments is also an important factor. Whereas Simulink/Stateflow, for example, has become the established tool for developing algorithms for engine management systems, GT-Power and SimulationX tend to be the preferred choice for design and computation. This is where co-simulation creates a bridge that enables users to continue using a familiar tool while at the same time linking the controller models from algorithm development with detailed component models from the design side. This allows any undesirable feedback between the controller and controlled system to be identified early on in the development process.

6 Summary

If specific process rules are observed, simulation can provide valuable support for product selection and for the process of developing a hybrid vehicle. It is important for the simulation and development processes to benefit from one other. In other words, simulations must answer the most important questions regarding component layout, strategy and target values and, conversely, data and algorithms from the individual stages of the development process must be fed into the simulation process. It is only on the basis of this interaction between the component-based simulation approach and networked simulation that benefits can be obtained in more complex areas, such as lateral dynamics, algorithm development and fault analysis. Around ten percent of the development budget should be allocated to simulation. If simulation is used consistently, the added value which it brings during the course of a project by far exceeds the investment and can also be used to advantage for subsequent projects.

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CFD Simulation for the Improvement of Thermal Comfort

Thermal comfort is an important criterion when choosing a car and, for this reason, it has assumed a high priority in vehicle design. In order to assess comfort not only qualitatively but also quantitatively, Ferrari, in close cooperation with the University of Pisa (Italy), has defined thermal comfort indices and made comprehensive evaluations of the general aspects of thermal comfort. For various reasons, an experimental approach is not practical, especially for the thermo-aerodynamic evaluations required. The calculations were therefore performed using the Computational Fluid Dynamics (CFD) software Fluent from Ansys.

1 Introduction

Thermal comfort is an individual perception and not a measurable value. Qualitative statements express the “physiological satisfaction related to the thermal environment” or a “situation in which thermal discomfort is not felt”. These descriptions do not really help car developers who want to measure the design quality and usefulness of design changes objectively and comparably.

The thermo-aerodynamic processes in the vehicle interior are important when it comes to making quantitative statements about thermal comfort. An experimental approach for determining the data required is virtually impossible because the flow velocities and the temperature changes are quite small and depend on the Reynolds number and geometric details. For this reason, it is impossible to use scale wind tunnel models; instead, measurements have to be taken on the real car. However, necessary modifications of the test scenario would be difficult or even impossible under these circumstances.

2 Virtual Testing with CFD

The CFD method has advantages for this scenario. Different variants of thermo-aerodynamic processes can be calculated and analysed quickly and with little effort, while taking into consideration the flow field and the thermal effects caused by thermal conduction, convection, radiation and solar irradiance. No complex physical models or testing configurations are required, and the design can already be tested, optimised and, if necessary, modified virtually at an early stage in the development. The CFD programme Fluent (Version 6.2.16) from the company Ansys was used for the evaluations. The programme has been in operation at Ferrari and the University of Pisa for several years. Before the project started, the performance of the software and the problems expected were examined. This allowed important information on the set-up required for the thermo fluid dynamic evaluations to be collected. The computer used was a Linux cluster with 32 AMD Opteron 280 Dual Core nodes each with 8 GB of RAM. A computing time of approximately 10 h was required for the complex calculations with more than 20,000 iterations.

3 Physiological Aspects

The temperature of the human body is kept constant by means of the equilibrium between the production and dispersion of heat. The energy balance of the human body can be deduced from the first law of thermodynamics. The energy (im)balance is expressed in Eq. (1):

$$\Delta = M - C_{res} - E_{res} - C - R - S - E_0 \qquad \text{Eq. (1)}$$

M = metabolic energy
C_{res} = heat dissipated in the respiration phase
E_{res} = energy dissipated by evaporation in the respiration phase
C = heat by convection on the body surface (with clothing)

Table 1: Mean skin temperature T_{SK} weighting factors

	Area weight	Sensitivity to warming	Sensitivity to cooling	T_{SK}
Face	0.07	0.21	0.19	0.200
Chest	0.09	0.10	0.08	0.090
Upper Back	0.09	0.11	0.09	0.100
Abdomen	0.18	0.17	0.12	0.145
Upper Legs	0.16	0.15	0.12	0.135
Lower Legs	0.16	0.08	0.15	0.115
Upper Arm	0.13	0.12	0.13	0.125
Lower Arm	0.12	0.06	0.12	0.090
Total	1	1	1	1

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R = heat by irradiation on the body surface (with clothing)
S = heat exchanged by means of transpiration
E_0 = energy dissipated by evaporation from the skin (diffusion).

Thermal sensors (cold and heat sensors) in the human skin form the interface between the human body und the environment. Their density differs according to the zones of the body. A sensitivity factor (T_{SK}) can be calculated for each part of the human body, although it is modified by clothing, **Table 1**.

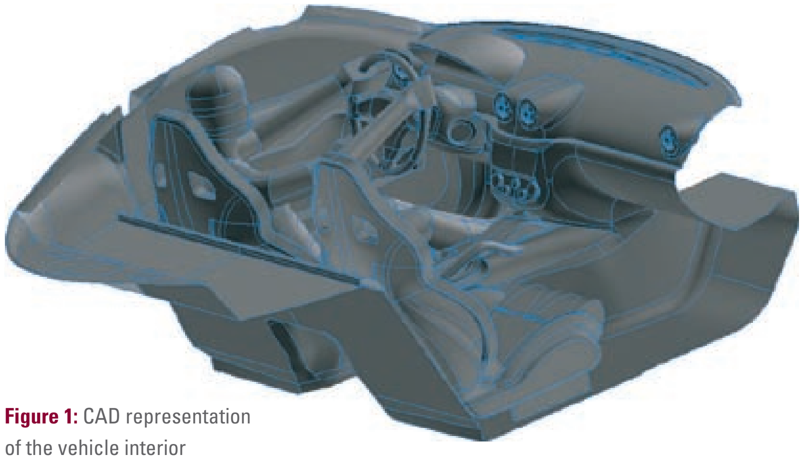

Figure 1: CAD representation of the vehicle interior

4 Aerodynamic Aspects

For the detailed analysis of the flow characteristics in the vehicle interior with respect to velocity and temperature, it has to be taken into account that some effects and processes are difficult to study:

- large reflective surfaces
- heat exchange with the environment
- direct solar irradiance
- small volume of the vehicle interior
- cold and warm air flow based on a limited number of small surfaces.

Furthermore, the thermo-aerodynamic effects are strongly influenced by the thermal characteristics of different elements with regard to their materials (glass, plastics, aluminium) and dimensions.

5 Definition of the Thermal Comfort Index

The global thermal comfort index is a derivative based on four local indices:

- the thermal equilibrium of the human body (I_E)
- discomfort caused by draughts (I_G)
- the vertical temperature gradient (I_{VT})
- the horizontal temperature gradient (I_{LT}).

The individual gradients were considered on a scale of zero to ten. For the present studies, two boundary conditions were set in Eq. (2a) and Eq. (2b):

$$I=10 \qquad \text{Eq. (2a)}$$

(optimum thermal comfort level)

and

$$I=6 \qquad \text{Eq. (2b)}$$

(sufficient thermal comfort level)

which had to be fulfilled to an equal extent by the global and the local indices. The global index for thermal comfort, in which both the boundary conditions and the weightings K_i derived from experience with existing cars can vary for different conditions and cars, is expressed in Eq. (3):

$$I_T = (K1\, I_E + K2\, I_G + K3\, I_{VT} + K4\, I_{LT})/4 \qquad \text{Eq. (3)}$$

5.1 The Global Thermal Equilibrium of the Human Body

On the basis of the assumption of the equilibrium Δ=0 for optimum conditions and an energy flux of Δ=15 W/m² for satisfactory conditions, an exponential law is obtained, as expressed in Eq. (4):

$$I_E = 10 \exp(-0.034065\,|\Delta|) \qquad \text{Eq. (4)}$$

5.2 Discomfort Caused by Draught

Discomfort caused by draught is the result of the temperature variation produced by the passengers. The loss of thermal energy depends on the temperature, the velocity, the turbulence level and the frequencies involved. With the same energy loss, a higher turbulence level produces higher discomfort.

The draught caused by an airflow is evaluated in relation to the more critical zones of the passengers, which are usually not protected by clothing. A zero number of dissatisfied people is assumed as the optimum condition. A figure of 15 % is assumed as a sufficient value. These values can be used to obtain the index I_G.

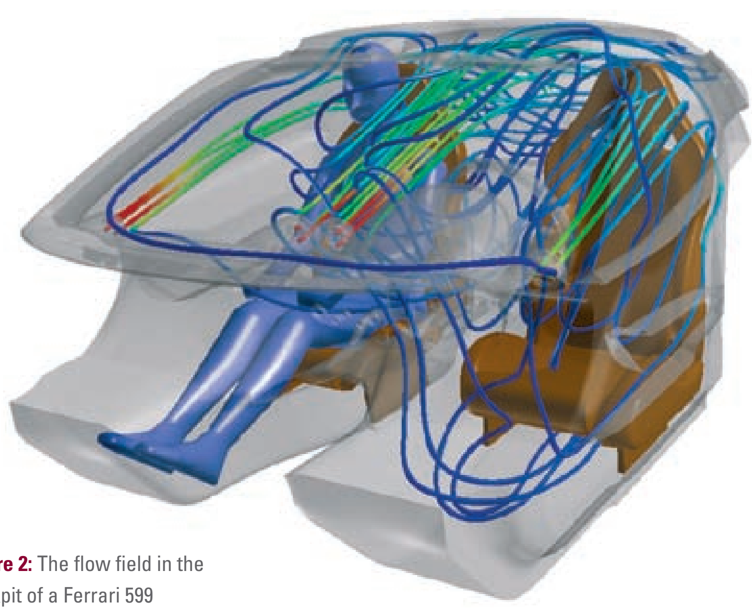

Figure 2: The flow field in the cockpit of a Ferrari 599

Table 2: CFD versus experimental temperatures (°K, warming condition)

Car element	CFD	Experiment	Driver	CFD	Experiment
Left fore window	285.6	286.7	Head	291.9	293.0
Right fore window	287.9	287.6	Right shoulder	290.5	294.8
Left rear window	286.3	286.6	Left shoulder	291.7	294.9
Right rear window	286.3	287.2	Right knee	289.1	290.6
Windshield (left side)	284.5	285.2	Left knee	288.9	290.3
Windshield (right side)	286.2	286.2	Right foot	289.2	290.5
Left door	287.0	293.3	Left foot	289.0	290.0
right door	288.0	294.0	Chest	290.0	292.0
Car top	290.5	293.3			

5.2.1 The Vertical Temperature Gradient (I_{VT})

The control points are assumed to be at the ankles and the face. In accordance with the European standard, 95 % of people feel comfortable when the temperature difference between these points is less than 3 °C. Therefore, a difference within 3 °C is assumed to be a satisfactory condition, while a zero difference is assumed to be the optimum condition. These values can be used to obtain the exponential law for I_{VT}.

5.2.2 The Horizontal Temperature Gradient (I_{LT})

The control points for the horizontal gradient of temperature are assumed to be at the seat edges. The same values as for the temperature differences for the vertical gradient can be used to obtain the horizontal gradient is I_{HT}.

6 The Numerical Procedure

The car under consideration is a Ferrari 599. In order to reduce the complexity of the numerical problem in this phase, the thermal effects related to the external flow were not considered. The non-structured grid contains 3.4 million tetrahedral elements. The mean skewness is 0.333 with a maximum of 0.874. For the definition of different conditions in terms of material characteristics and boundary conditions, the grid is divided into 18 elements, **Figure 1**.

A coupled-implicit approach was selected. Following a sensitivity analysis, the k-ε turbulence model with a non-equilibrium wall function treatment was chosen. For the conduction effects, the material properties were selected as aluminium for the car surfaces, glass for the windows, plastic for the dashboard and leather for the seats. For the radiation problem, the Discrete Ordinate (DO) Radiation Model coupled with DO Irradiation Solar Load Model for the solar load was selected.

"Mass flow outlets" and the temperature were defined for the inlet ducts of the air conditioning system. A pressure outlet condition was defined for the ventilation openings.

7 Experimental Verification

The physical tests were carried out in the Ferrari wind tunnel. Two conditions were analysed, **Figure 2**: a warming and a cooling condition. Because the velocities are low and are characterised by high gradients and a significant variation in their direction, measurements are correspondingly critical and not very precise. Therefore, it is difficult to compare the experimental and CFD results.

7.1 Warming Condition

The correlation between experimental and numerical data is satisfactory, especially when considering the fact that high gradients have a strong influence on the position of the measurement points. Significant differences can be found above all for the aluminium elements, **Table 2**. This is probably caused by the presence of thermal panelling in the experiment, which was not considered in the CFD evaluation, **Figure 3**.

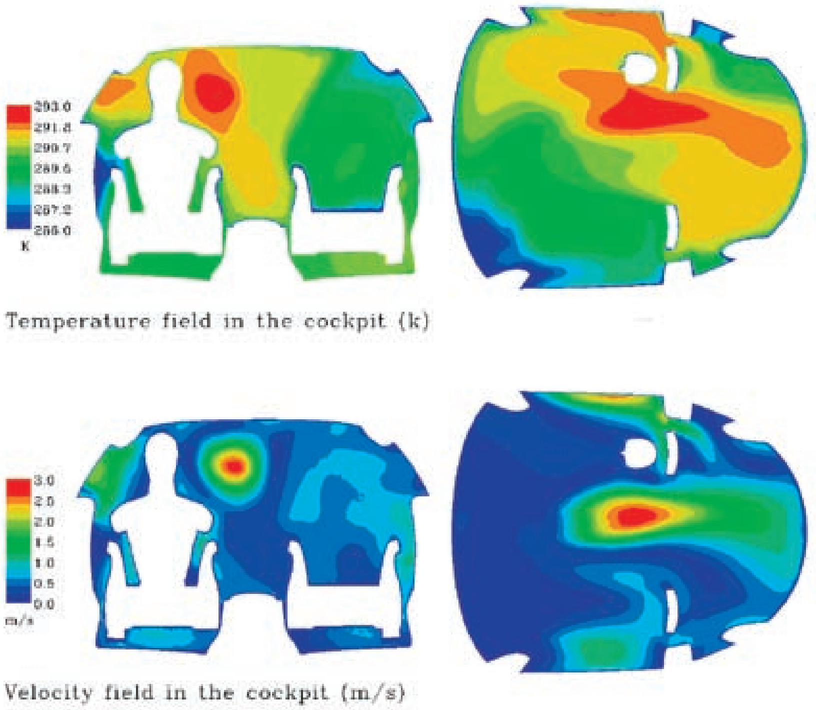

Figure 3: Temperature and velocity fields in the interior (warming condition)

Table 3: CFD versus experimental temperatures (°K, cooling condition)

Car element	CFD	Experiment	Driver	CFD	Experiment
Left fore window	296.5	296.0	Head	286.4	290.4
Right fore window	296.4	297.3	Right shoulder	285.6	287.5
Left rear window	297.0	296.0	Left shoulder	285.8	288.5
Right rear window	296.8	297.3	Right knee	288.5	290.0
Windshield (left side)	298.7	298.0	Left knee	288.8	290.2
Windshield (right side)	298.6	300.2	Right foot	292.2	293.7
Left door	287.4	291.2	Left foot	291.8	294.0
right door	286.8	290.5	Chest	287.0	289.0
Car top	287.0	294.8			

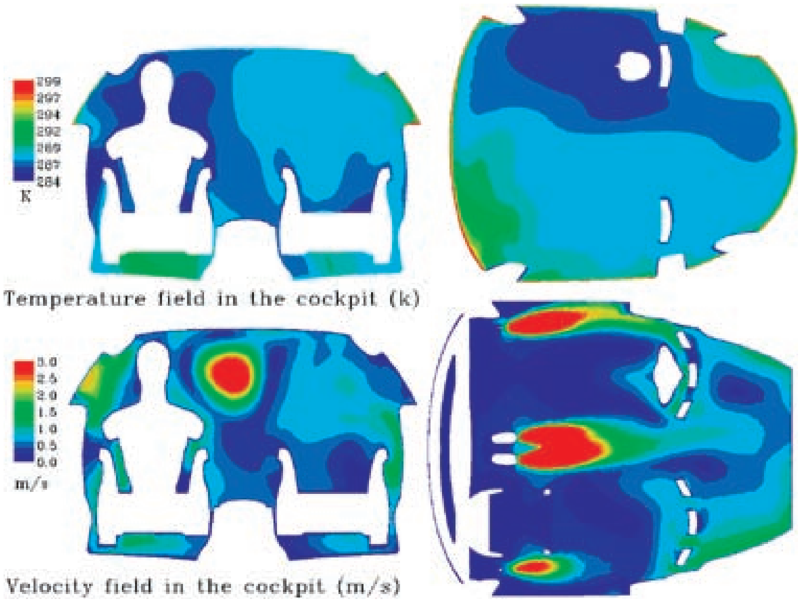

Figure 4: Temperature and velocity fields in the cockpit (cooling condition)

7.2 Cooling Condition

This is a particularly critical condition. The solar model must be considered in the CFD evaluation. The correlation between the experimental and the CFD analysis for this condition is less satisfactory. One reason could be the "glasshouse effect", which is represented in the experiment but is not well represented in the numerical model, **Table 3** and **Figure 4**.

8 Evaluation of the Thermal Comfort Index

The thermal comfort index for the car under examination can be determined from both the experimental and the numerical results. At this stage of the study, the contribution of the different terms in Eq. (5) is assumed to be:

$$K1= K2= K3= K4=1 \qquad \text{Eq. (5)}$$

In general, the correlation between experimental und numerical data appears satisfactory. However, significant differences are found for the local indices, which is due to the difficulty in determining the corresponding gradients from experimental data, **Table 4**.

From a design standpoint, the data generally indicate a satisfactory level of thermal comfort. Some effects seem to produce a certain degree of discomfort, which is related to the placement of the air inlets. Corresponding design modifications achieved a significant improvement in thermal comfort.

9 Conclusion

The definition of thermal comfort indices allows quantitative predictions of thermal comfort to be made for the first time. The fundamental aspects of the index definition are related to the aerodynamic situation in the vehicle interior. Without doubt, the accuracy of the predictions on the basis of comfort indices could be significantly improved if data for a large number of different cars and conditions were available.

The studies would not have been possible to this extent without the use of the CFD method and the Ansys software. A comparison between experiment and simulation shows a satisfactory correlation. The accuracy of the numerical analyses could be further improved if the external flow and the thermal characteristics of the insulation panels were considered and the "solar model" were better represented. ■

Table 4: Thermal comfort indices

		Warming		Cooling	
		Experiment	CFD	Experiment	CFD
Global Thermal Equilibrium of the Body	I_E	6.72	6.16	6.87	7.93
Discomfort Related to the Draught	I_G	4.46	6.08	3.96	3.87
Vertical Gradient of Temperature	I_{VT}	6.31	6.88	5.42	5.42
Horizontal Gradient of Temperature	I_{LT}	9.76	7.47	9.53	7.84
Global Thermal Comfort Index	I_T	6.81	6.64	6.26	6.44
Standard Deviation		1.90	0.57	1.67	1.96

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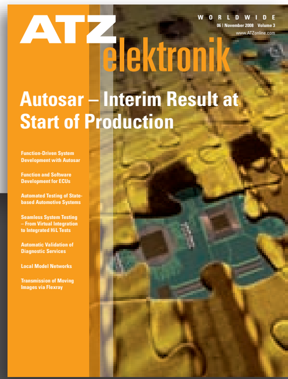

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Computer Simulations to Extend the Service Life of Wiper Systems

High-tech in the car: we are talking about high-performance engines, carbon bodywork and windshield wipers. The inconspicuous wipers have long since become a complex, state-of-the-art construction. Bosch, the German market leader in windshield wipers based in Bühlertal (Germany), meanwhile even uses computer simulations in order to test new designs. Calculations made by simulation engineers from Merkle and Partner seek to make the sensitive wipers more robust in the face of environmental conditions.

1 Introduction

If a car is parked outdoors in winter, the first major problem to solve before driving is to rid the frozen wipers of ice. Attempts are frequently made to scrape the wipers free, to loosen them with deicing spray or simply rip them off the windshield. However, the use of brute force can do considerable damage to this precisely matched and highly complicated structural component.

The correct functioning of the wiper system relies on a perfect interplay between a number of different factors, for example the kinematics, that is the drive system of the wiper arms, the design of the corpus, which affects the airflow, the electronics of the control system and the stability of material and design when confronted with vibration. This includes the rubber compound of the wiper blade. If it is too hard, the wiper will clatter. If it is too soft, it will be unable to wipe away dirt properly. In addition to these factors, which affect the performance of the windshield wiper, the wiper system itself is expected to operate without fail for years or even decades.

These modern high-tech devices no longer have much in common with the manually operated wiper seen at the start of the last century. The wipers of today are based on the latest research findings in the fields of physics and chemistry and are subject to a continual process of improvement. The varying chemical composition of the rubber within the wiper blade ensures, for instance, that the edge remains stable and sharp while the body of the blade is sufficiently flexible to allow the blade to turn over softly for the reverse movement without scraping the windshield. A spring-loaded rail rather than the older shackle design meanwhile optimises the aerodynamics of the wiper, reduces noise and boosts cleaning performance. Rain sensors independently determine the wiping speed depending on precipitation. Wipers are now precisely designed to match each new vehicle model to ensure optimal function. If changes are made to the body shell, geometry or structural design of the vehicle, the development of the wiper system is adapted accordingly.

2 True-to-Detail FEM Models

The engineers who develop windshield wiper systems, however, have to be constantly prepared to change their designs to suit new types of vehicle. Since the systems are intensively tested in order to minimise external impact despite their complexity, a prototype has to be constructed and tested before development engineers are able to make improvements and begin designing a new prototype.

In order to shorten this process and save costs, Bosch meanwhile makes broad use of computer simulations throughout the development process. They are easier and faster to adapt to modifications in preliminary designs than real testing facilities and thus reduce costs for the design engineers. Depending on the vehicle manufacturer, it takes between 18 months and four years from the initial concept to the standard serial production of a windshield wiper. Simulations account for about one-fifth of the scope of design for example dynamic calculations of movements or numerical simulations (CFD) of airflow and liquid.

The testing of design ideas and the interpreting of results are performed by an engineering company specialising in simulations for product development and design optimisation. For this purpose, experts at Merkle and Partner in Heidenheim (Germany) design the future windshield wiper with the help of computers using the so-called Finite Element Method (FEM) as a virtually lifelike model. The term refers to the fact that the body basically consisting of an infinite number of points is converted in the process into a finite number of calculatory nodes, **Figure 1**. The higher

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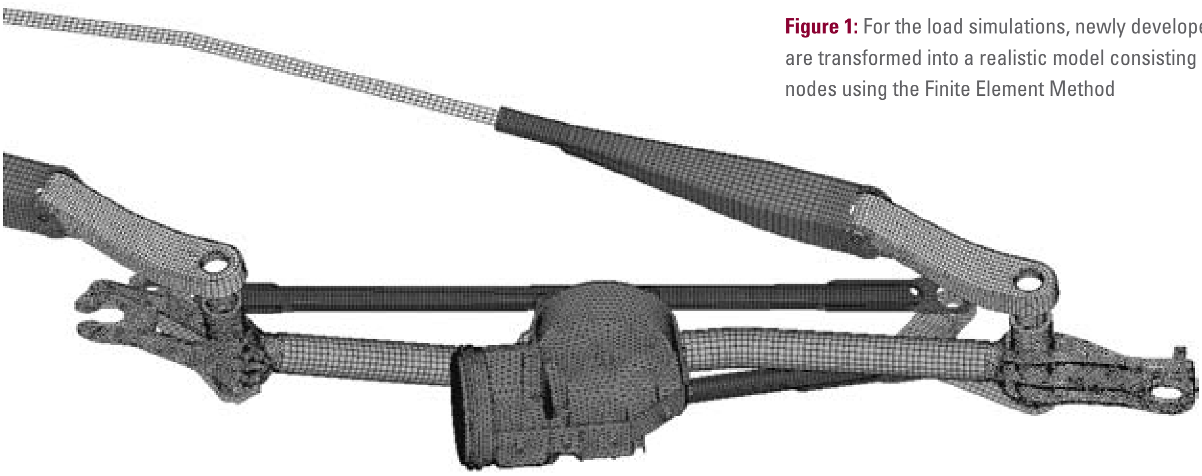

Figure 1: For the load simulations, newly developed wipers are transformed into a realistic model consisting of calculatory nodes using the Finite Element Method

the number of nodes and the lesser the distance between them, the more precise and also the more intensive the model will be to calculate. In order to create the physical behaviour of the model as realistic as possible, in addition to the preliminary designs the engineers utilise a great amount of various data. Amongst other things, the characteristics of the wiper depend on the material data and the composition of the wiper arm and rubber lip as well as the rigidity of the bearings. Moreover, the design of the windshield, which differs depending on vehicle, and the external temperature both affect the precise performance of the wiper system.

3 Load and Misuse Tests

Once completed, the model is then tested with the aim of optimising the durability of the system when subjected to various operating loads and incorrect use. The relevant load cases are determined in accordance with real life conditions and compared with those of the Bosch testing department. The windshield wiper is thus tested to assess its behaviour in five different virtual load and misuse tests.

Figure 2: In the blocking test, a routine scenario is simulated: windshield wipers that are put into operation in spite of frozen rubber lips; this results in ten times more strain on the motor

The blocking test simulates the holding back or freezing of the wiper. The blocking of the arm during operation presents the greatest challenge for the wiper motor, which is in danger of overheating in the process, **Figure 2**. The snow load test is of a similar nature. It simulates the formation of a ridge of snow in front of the wiper blade, **Figure 3**. The motor has to work harder to overcome the resistance in order to move the wiper and the arm is subjected to a far higher mechanical weight. The strain on the wiper system when blocked or under snow is approximately ten times greater than under normal wiping conditions. In contrast, the slam test is designed to ensure that the metal parts of the wiper arm do not come into contact with the windshield when falling with full force from a raised position onto the windshield. Raising the wiper blades is very common in winter during long parking to avoid freezing. Direct contact with the wiper arm during free fall can, however, damage the windshield. In contrast, the problems tested in the shaking test are those that can occur completely without human influence. Since the car vibrates constantly while the engine is running, imperceptible movements are also transferred to the wipers, which can cause damage to the system over a long period. The greatest stress on the wiper arm is simulated in the misuse test, that is incorrect use of the system. In this process,

Figure 3: Thick layers of snow on the windshield can damage the wiper system due to their excessive weight and resistance

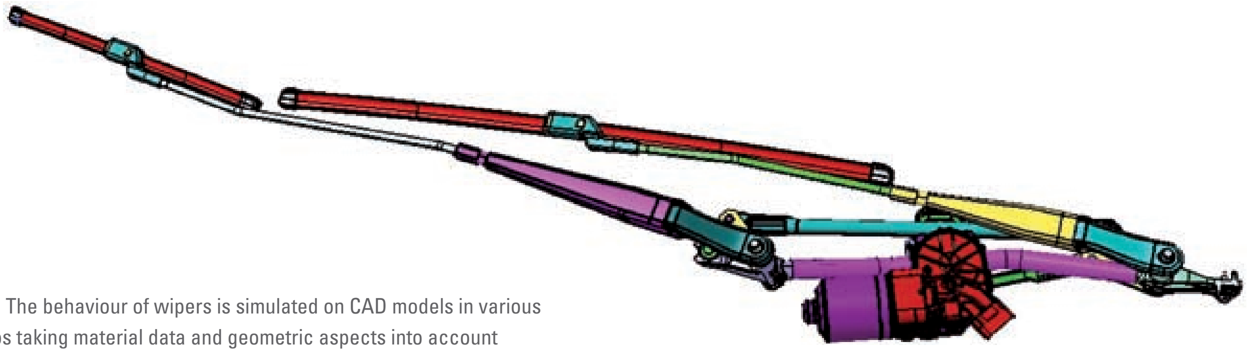

Figure 4: The behaviour of wipers is simulated on CAD models in various scenarios taking material data and geometric aspects into account

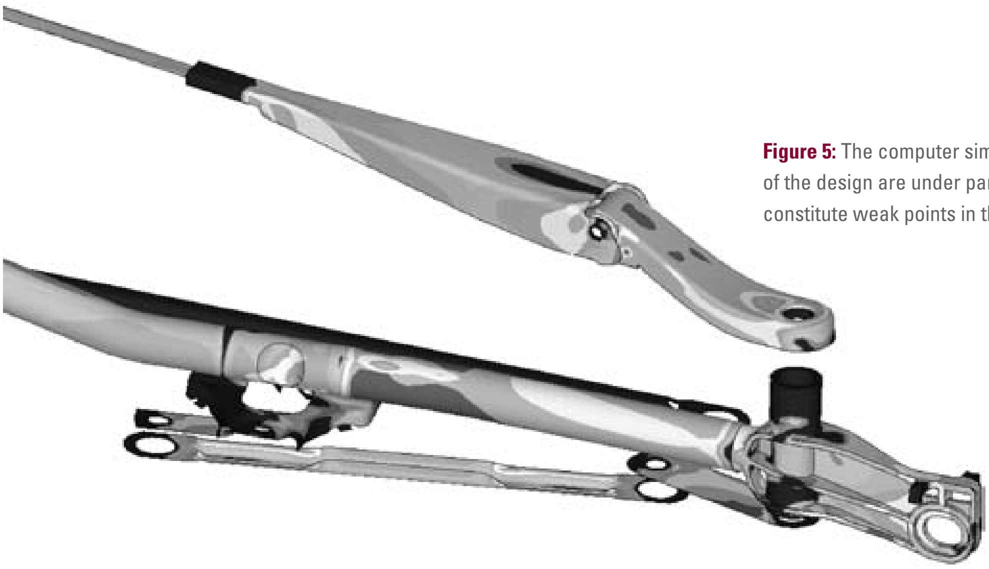

Figure 5: The computer simulation clearly shows which areas of the design are under particular strain and could possibly constitute weak points in the course of daily use

the exertion of lateral pressure on the raised arm is simulated, which usually occurs for example when the wiper arm is dismantled in order to replace the windshield. It may thus cause damage to the wiper arm or joint through the application of excessive force. Tests and experience show that this type of careless handling of the high-tech system is the most frequent reason for damage to the windshield wiper.

4 Cost Reduction and Improved Resilience

Multi-processor computers at Merkle and Partner work for three to four weeks collecting all relevant data from the extensive models and their simulated environment in order to test the various load cases in a given scenario before proceeding to the final real-life test, **Figure 4**. In the past, behaviour in various situations had to be determined through tests on real wipers. Both the tools required for the manufacture of new prototypes and the prototypes themselves had to be newly constructed for each step in development. Computer testing now makes it possible to go through the same situation over and over again with minute changes in parameter without having to change the existing test set-up each time. Above all, the development time for repeated and long-term tests is significantly shortened, making it possible to save approximately six weeks on each project.

The saving of time and costs, however, is not the essential factor. The main priority in the introduction of the modern testing method is to improve the wiper systems. They make it possible to detect weak points, particularly under worst-case conditions, and to design systems far more robustly, **Figure 5**. This increased durability also increases safety and customer satisfaction, since most drivers hardly ever change their windshield wipers. The complex high-tech systems are thus able to render their services for a great many years, regardless of rain, snow or sunshine. ■

Dynamic Characteristics of a Shock Absorber Module

Optimization of Vehicle Acoustics

The mutual interaction of the chassis components and with the vehicle body plays a decisive role for the buildup of noise in the high frequency range. The shock absorber itself represents only a part of the total system "Chassis – Body". An additional special challenge lies in the vehicle dependency of the vibro-acoustic quality of the shock absorber. The systematic NVH analysis integrated in the product development process at ZF Sachs AG and performed with the help of modern experimental and virtual methods allows for efficient product optimization.

1 Introduction

Within the context of chassis development the focus is usually on excitations in the lower frequency range and the thereto connected adaptation of shock absorber characteristics; here, characteristic features such as body behavior are monitored. However, behavior in the higher (acoustically relevant) frequency range is increasingly gaining in importance. Chassis components also, such as shock absorbers, must not develop or transmit acoustically noticeable vibrations.

The mutual interaction of the chassis components and with the vehicle body plays a decisive role for the buildup of noise in the high frequency range. The shock absorber itself represents only a part of the total "Chassis – Body" system. An additional special challenge lies in the vehicle dependency of the vibro-acoustic quality of the shock absorber. The systematic NVH analysis integrated in the product development process at ZF Sachs AG and performed with the help of modern experimental and virtual methods allows for efficient product optimization.

2 Methodical Approach

Vibrations excited by the road surface cause highly dynamic working cycles in the shock absorber; consequently, the corresponding characteristic damping force variations arise which lead to vibrations of the piston rod. These vibrations are transmitted to the body and perceived by the vehicle passengers as peculiar, striking noise.

The vibration characteristics of the shock absorber in these highly dynamic cycles are distinctly different from the "semi–static" responses which serve as a data basis during the design/engineering stage. It is in particular the dynamic behavior, such as the dynamic stiffness of the shock absorber itself and of the (shock-absorber) mounts, the self-resonance of the shock absorber structure together with the input impedance at the linking points, the characteristics of the body's structure, as well as interior spatial acoustics that constitute important parameters to be considered, analyzed, and, if necessary, optimized in the course of the shock absorber module development process.

The NVH analysis of the shock absorber module is performed by the methods described below. These were adjusted as well as further developed while taking into consideration the chassis-specific – stochastic, transient – stimuli. Experimental investigations are supported by virtual methods on a default basis. Characteristic-based and semi-physical shock absorber models are available for this purpose. Flow simulation (CFD) is used. For paucity of space, shock absorber simulation is not explained further in this article [4, 5].

Damper hydraulics (valve adjustment), damper structure (piston rod, pressure and reservoir tubes etc.), module components (top mount, spring, bump stop etc.) as well as transfer paths from the shock absorber to the driver's ear are used for systematic analysis and optimization.

3 Effect of the Shock Absorber on Vehicle Acoustics

When analyzing the chassis noise, it is important to differentiate between the transferred and generated structure-borne noise of the shock absorber module. In the case of generated structure-borne noise, noise which originates, for example, in the top mounts or in the shock absorber is meant. Among the various noise types, rattle always causes an increasing number of customer complaints, as the audible impression is interpreted as a technical fault, such as a loose screw fitting, for example. Processing these complaints involves the elimination of individual disturbing noises.

The vehicle interior noises, measured with an artificial head, were used to define the noises. The aurally accurate analysis of the recordings provides information about the frequency range and the time structure of the noise.

Rattle is a perceivable noise lasting about 0.2 to 0.3 seconds. It consists of several sets of impulses which follow at an interval of about 0.1 seconds. The main frequency content of this noise is in the range of 200 to 800 Hz and is composed of 8 to 10 resonances. Most frequently the noise occurs while traveling at a driving speed of 30 to 40 km/h on a raw, wavy country road.

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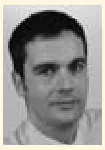

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4 Methods for the Characterization of Shock Absorber Characteristics

The vibration behavior of a shock absorber is described experimentally mainly through damping force characteristic measurements, dynamic stiffness measurements, and noise measurements effected at the damper.

4.1 Measurement of Damping Force Characteristics

At ZF Sachs, we use what are called Sachs-Performance-Testers, a world-wide standardized set of test rigs for the determination of characteristic curves. The test rig, testing procedure, and the evaluation of the data are defined in the corresponding process description. The test rig, usually an electromechanical one, generates a constant sinusoidal movement (e.g. +/-50 mm) at a frequency of about 0.01 to 6 Hz. The measured force and displacement signals are evaluated and displayed in the form of force-displacement- (F/s) and/or force-velocity charts (F/v). The standard F/s and/or F/v characteristics have the disadvantage of only describing the damping characteristics in the low frequency excitation range. Further, more comprehensive characteristics are needed.

4.2 Dynamic Stiffness Measurements

The frequency-related ratio of the complex damping force to the input-side complex vibration amplitude during simple harmonic vibrations is termed “dynamic stiffness”. This relationship is used to describe the dynamic response characteristics of a shock absorber. The dynamic stiffness of the shock absorber is determined in the frequency range up to hundred Hz by using the Fast Fourier Transform of displacement and force data. Normally, the amplitude (dynamic stiffness) and the phase (loss angle) as well as the real (stiffness share) and the imaginary part (damping share) of the transfer function are calculated.

4.3 Noise Measurement

A special metrology method was developed for the evaluation of striking noise characteristics of the shock absorber. Piston rod acceleration is measured in addition to the displacement and force signals. Measurements were effected on a servo-hydraulic test system. When measuring the noise, the shock absorber is installed on the test machine at the overall stroke center position, **Figure 1**. The shock absorber is fixed to the test rig at the bottom (reservoir tube) and the top (piston rod) with standard test mounts, the original vehicle mounts or is rigidly clamped. The acceleration sensor is attached at the upper end of the piston rod. A test cycle consists of blocks (sequences) which are run successively. The excitation frequency and amplitude are varied such that the total operating range of the shock absorber is tested. The piston rod acceleration and/or the second derivative of the damping force (peak/peak value for compression and rebound direction) are determined. The measurement signals are displayed as charts over time. These data characterize the rattle tendency of the shock absorber and make it possible to compare various valve adjustments and design variants by experts.

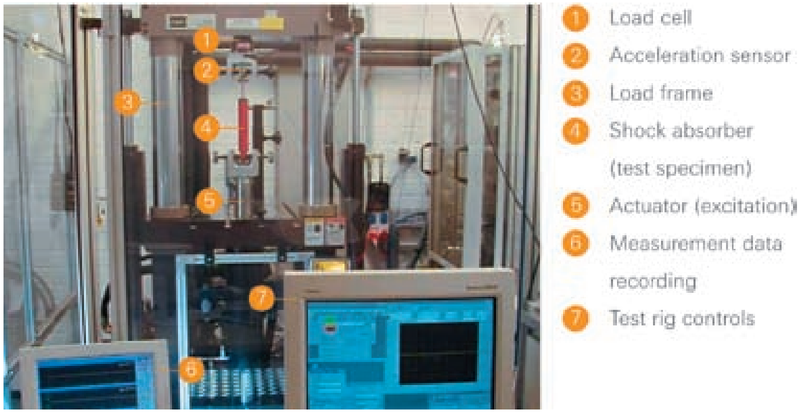

Figure 1: Test rig layout for shock absorber measurements

5 Damper Characteristics at High Dynamic Excitation

The dynamic F/v characteristics of a twin-tube shock absorber are shown in **Figure 2**. The black curve (1) shows the measurement with an excitation of 2 Hz, 0.5 m/s damper velocity in 0-displacement position; the red curve (2) shows the same shock absorber velocity but with an excitation frequency of 25 Hz. The following differences are noticeable:

- magnification of the hysteresis (delay in force build-up) in the pre-opening/bleed range (points 1R and 1C) which relates to higher spring shares in the damping force
- intersection of the rising and falling characteristic curves (points 2R and 2C)
- distinct force interferences (3R and 3C).

The measurement of the shock absorber’s dynamic stiffness provides for a more precise presentation of these effects. **Figure 3** shows the measurement results for the two excitation speeds: 0.1 m/s and 0.5 m/s with two shock absorbers that feature very similar characteristics. Quite obvious: At higher excitation frequencies, shock absorber B reveals a considerably higher stiffness share when compared to shock absorber A. The tendency of shock absorber B towards becoming stiffer is particularly strong at higher excitation speeds.

The different vibration characteristics of the shock absorber on a highly dynamic excitation depend on the structural resiliences and the compressibility of the hydraulic medium, as well as phase shifts of the valve switching cycles. Interior pressure measurements and valve stroke measurements are used for the investigation of these effects [1].

6 Optimization of Shock Absorber Hydraulics (Valve Adjustment)

Chassis tuning and setting with regard to optimal driving dynamics and good driving/riding comfort usually leads to digressive (i.e. to a certain extent discontinuous) shock absorber characteristics. Linear characteristics would be less critical for acoustics, however, they still prove disadvantageous for vehicle handling and driveability. Modifications of hydraulic components aim at a generally more continuous valve opening behavior. The emphasis in terms of the modifications made is on the interaction of the piston and the bottom valve. One of the disadvantageous consequences of this measure is the unavoidable change in damper characteristics. In terms of the driving dynamics, this can cause different vehicle response characteristics, even trade-offs when it comes to acoustics' targets set [2].

7 Optimization of the Interaction between Mounts and Shock Absorber

In the case of shock absorber module development, the interaction between upper und lower mounts and the shock absorber itself is at the center of attention. On the one hand, a rubber-to-metal component can, by itself, cause noise, such as the noise stemming from the unsuitable static design of characteristic curves (hard transition in progression), and on the other hand, wrong choice of material may contribute to the fact that the transfer of high frequency vibrations is not sufficiently prevented.

The lower shock-absorber mount should directly transmit the suspension movement in Z-direction into the shock absorber and thus effect a hydraulically damped motion. The stiffness/resetting forces in the cardanic and torsional motion of the mount should be minimal. Otherwise, axial movements initiate bending moments in the shock absorbers which increases the internal supporting forces at the shock absorber guide as well as at the valve piston, consequently contributing to increased friction. Since the upper shock-absorber mount is positioned directly on the chassis, insulation relating to vibrations originating from the piston rod, the supporting spring, and the buffer is considered particularly important.

The design of a so-called "multiple path" top mount proved successful thanks to its advantages in the NVH sector. In this case, buffer and spring forces are guided directly into the top mount housing while the piston rod forces are absorbed by an encapsulated rubber-to-metal component. This provides advantages in terms of the tunability of the damper mounts. Encapsulation – combined with soft and low-damping rubber mixtures including the corresponding calibration levels – offers good opportunities for supporting piston rod forces in line with the respective shock absorber movements with minor torque generation only, in order to live up to NVH and service life requirements.

Static and dynamic characteristic curves are recorded according to standardized specifications for the proper characterization of a rubber mount, **Figure 4**, dynamic top mount and shock absorber characteristics as well as the body stiffness in the top mount area are to be harmonized with each other. Low damping rubber mixtures have proven advantageous for the top mounts. The damping share of the rubber material is selected such that low-frequency vibrations subside in a controlled manner without any follow-up responses and high-frequency vibrations are damped with the smallest possible amplitudes. For the mount, a stiffening factor below 2 is envisaged (ratio dynamic vs. static stiffness).

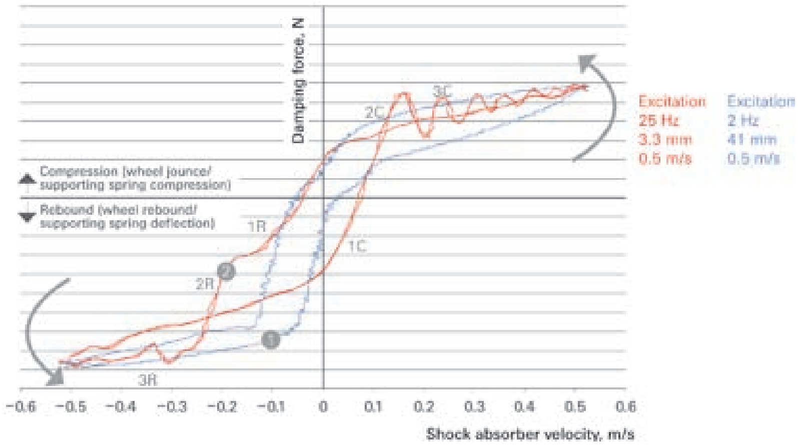

Figure 2: Dynamic F/v characteristics of a twin-tube shock absorber

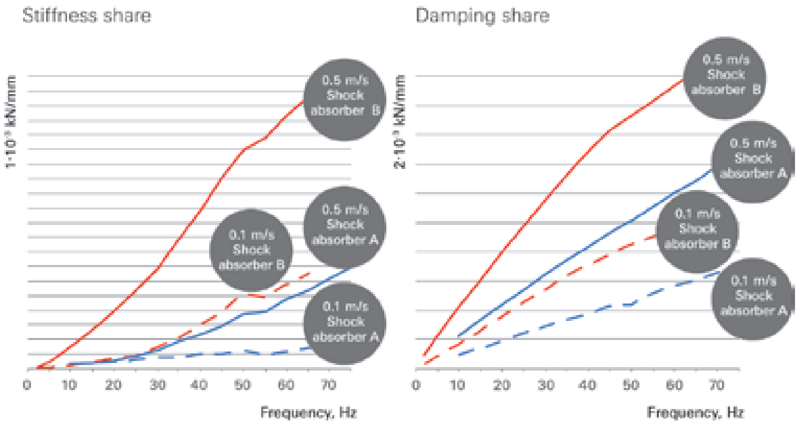

Figure 3: Measurement results of dynamic stiffness

8 Structural Optimization of the Damper Module

Experimental analyses (running mode analysis and modal analysis) as well as a computational analysis (FEM calculation) are used for optimizing the structure of the damper module. As an example, the results of a running mode analysis for a passenger car are explained in more detail. The vehicle measurements were made with real road excitation (about 40 km/h) on a four-post test rig system. The vibration response (accelerations) was measured with 3-axial acceleration sensors respectively, namely on the shock absorber (piston rod/reservoir tube), the knuckle, and the vehicle body (top mount area). Vibration shapes and their frequencies were determined on the basis of these recordings.

Twelve distinct vibration shapes were found in the frequency range up to 800 Hz. Four vibration shapes are in the fre-

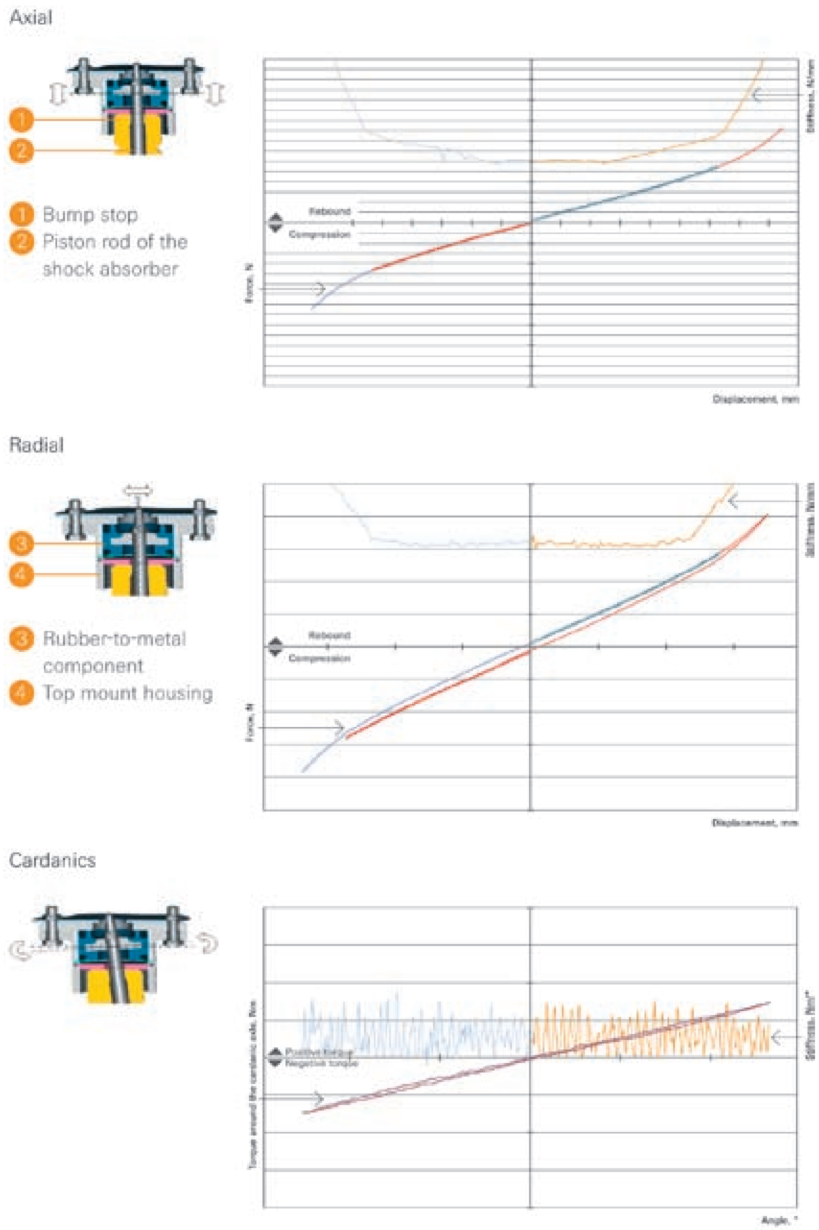

Figure 4: Characteristic curves for damper mount characterization

quency range below 100 Hz and are considered rigid body modes. Eight vibration forms are elastic modes and reside in the higher frequency range. The modes in the low frequency range (< 50 Hz) are attributed to the elastokinematics of the suspension and/or the individual movements of the module components, such as the shock absorber, spring and top mount.

The analysis shows that unrequested high secondary stiffenesses of the mounts in the case of cardanic and torsional deflections in combination with an unfavorable suspension layout cause high transverse tension/friction in the shock absorber. The result: Additional excitation in the high frequency range.

A modal analysis of the shock absorber is performed in order to determine the effect of the shock absorber structure's self-resonance on the vibration forms occurring during operation.

An FE model of a damper is created from the CAD structure. Here already – an at first glance – rough FE net and a low level of itemization have proven sufficient. The average degree-of-freedom amounts to approximately 15,000. Detailed mapping of the inner parts (e.g. piston rod guide) was not included in the model. Shell elements are used for the two ring mounts as well as substitute models for the elastomer bodies that were derived from the measurement data gained in the relevant frequency range; for the mounts, bar elements are applied as inner bushes. The calculation method used is "inverse power sweep". The final result: Vibration shapes and their frequencies, **Figure 5**.

For model validation, an experimental modal analysis (EMA) is performed at the shock absorber. The shock absorber is investigated on a special test frame which simulates the position and mounting in the vehicle in the best possible manner, **Figure 6** (left). The precise mapping of these boundary conditions in the FEM model has considerable influence on the significance of the modal analysis. Especially the coupling to the test rig requires precise knowledge about the possible degrees of freedom. A "free-free" condition analysis is not implemented.

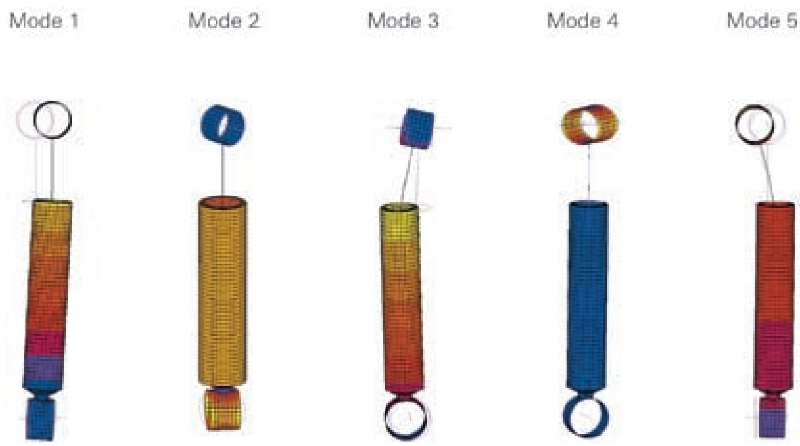

Figure 5: Examples for some shock absorber vibration modes with increasing frequency

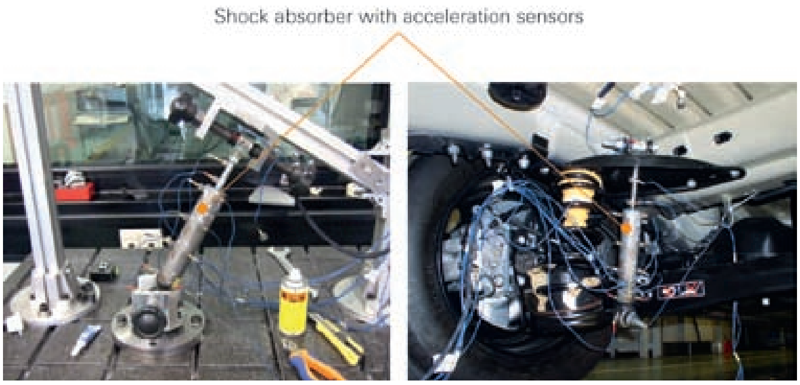

Figure 6: Measurement setup for experimental modal analysis at the test rig (a) and in the vehicle (b)

The shock absorber to be investigated is excited at one or several points in a sinusoidal or randomly (shaker) excitation, or by impulses (impulse hammer). The associated vibration responses are measured via acceleration sensors at one or more locations. The interdependencies of excitation and response are presented by means of transfer functions and the modal parameters determined on the basis of numerical methods.

Like any measurement on the test rig, the modal analysis performed respectively presents only an estimation of the vehicle's real characteristics. For comparison purposes, the same shock absorber was installed in the vehicle and the modal analysis was performed under real conditions, Figure 6 (right). The data analysis showed that almost all the modes occurring in the vehicle were found in the test rig measurements as well.

The correlation analysis of the measurement and calculation results shows the quality of conformity. In the process, grid points used in the FE model are identical to those used for the measurement. It is necessary that the coordinate directions in the FE model and in the measurement are the same. The calculated and measured modes are set side-by-side and are animated. Whether conformity suffices or not, i.e. whether the values are within the set limits, is decided on the basis of frequency deviations and MAC values (Modal Assurance Criterion). If that is not the case, adjustments are made in the FE model. The correlation analysis provides clues about the parameters which are to be adapted. Once the calculation model has the required quality, further parameter variations are continued on a virtual basis.

The optimization of the virtual shock absorber's structural elements shows how to fix a noise-relevant shock absorber resonance problem in the vehicle. For example, the results of the inspection performed with the rear shock absorber showed that rigid body modes (< 100 Hz) can be influenced through mounts' stiffness variations. The mount's cardanic stiffness as well as the optimization of the pressure and reservoir tubes changes torsional modes (100 to 200 Hz). Simple bending modes (200 to 800 Hz) and bending modes of a higher order (> 1800 Hz) can be influenced primarily through piston rod and damper tubes variants. Axial vibrations of the shock absorber (800 to 1800 Hz) are determined by the masses of the piston rod and the damper tubes as well as through the stiffness of the mounts in vibration direction. These findings provide sufficient leeway for avoiding critical modes and enable targeted shock absorber vibration optimization in terms of noise reduction in the vehicle.

9 Vehicle Sensitivity Analysis

Vehicle sensitivity data in relation to shock absorber excitation provides helpful information for NVH-related damper module optimization. This data is obtained by means of the direct or inverse vehicle measurement. In the case of the direct method, a known excitation (impulse hammer, shaker) is imposed on the connection points of the suspension to the vehicle body and the airborne noise response is measured by means of a microphone in the vehicle's interior. In the case of the inverse method, based on the principle of mechanical-acoustic reciprocity, a calibrated sound source is placed inside the vehicle. It generates a "white noise" signal that stimulates vehicle vibrations. These vibrations are recorded by means of acceleration sensors at the connection points of the suspension to the vehicle body. By means of both methods, the transfer functions (FRF) from the shock absorber to the driver's ear are calculated. They show the main transfer paths of structure-borne noise from the shock absorber to the interior and allow for an NVH comparison of different vehicles.

The analysis of measurements done with several vehicles showed that individual transfer paths can be clearly identified. For example, in the overall frequency range, noise sensitivity of the shock absorber's upper connection (top mount) in comparison with the lower connection points (trailing link, transversal control arm, chassis sub frame) is dominant, **Figure 7**. The comparison of the noise sensitivity of the top mount in three directions shows that, depending

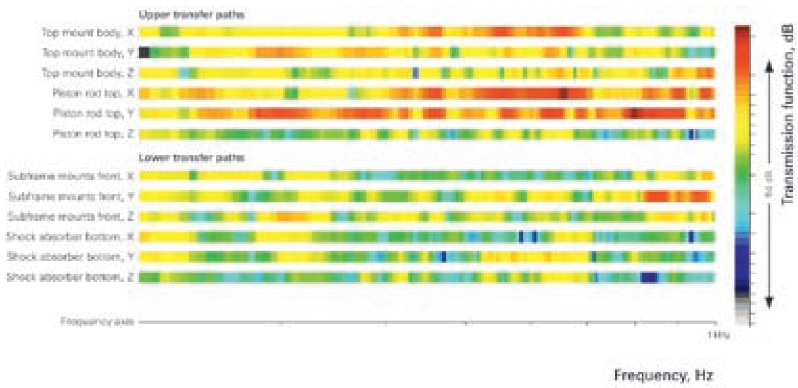

Figure 7: Measurement results for vehicle sensitivity with respect to shock absorber exication

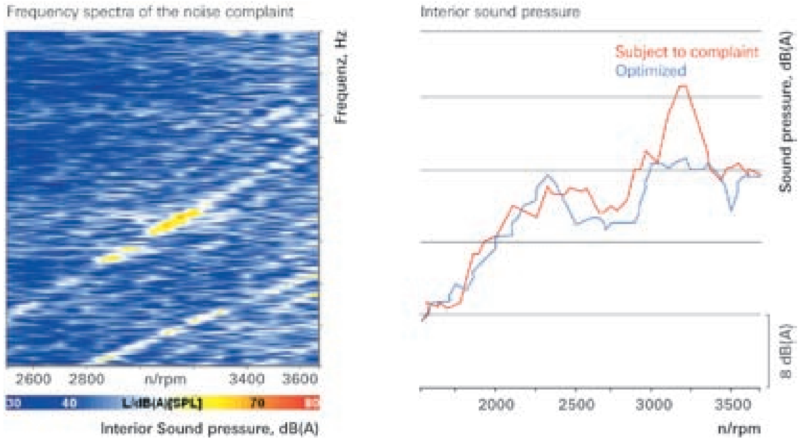

Figure 8: Reduction of vehicle noise

on the mount or the vehicle body design, the Y/Z directions are the most sensitive transfer paths. The sensitivity of the lower transfer path (chassis subframe – body – interior, suspension link – body – interior) also depends on the design of the suspension and the frequency range observed. Details about the application of the methods described for shock absorber development can be found in [3].

10 Examples of Noise Reductions Performed

10.1 Optimization of the Modular Structure

The vehicle (in this case an LCV) is equipped with a twin-tube shock absorber on the rear axle (driven beam axle with leaf springs). The top mounts are designed as ring joints. The rear axle differential is a freely suspended hypoid transmission. The transmission's meshing noise was transferred to the vehicle's interior during "coast condition" and customers complained about this "howling" noise. The transfer path analysis at the vehicle identified the rear axle shock absorber as one of the main transfer paths. The following measures were checked in terms of noise transfer reduction:

- optimization of the shock absorber's (valve setting)
- optimization of the shock absorber structure (piston rod)
- stiffness optimization of the top mount
- optimization of the cardanic and torsional stiffness of the lower mount
- optimization of the radial stiffness of the lower mount.

The verification of all measures showed: Noise transfer reduction was most efficient with the optimized lower mount's radial stiffness. The optimization of the bearing was achieved by using a different rubber mix (lower Shore hardness) and by adapting the rubber geometry. Thanks to these modifications, static radial stiffness was reduced by 45 %. In the process, it was possible to achieve a distinct reduction in the mount's dynamic stiffness; consequently, the stiffening factor of the mount was below the value 2 (compare with section 7). Durability requirements were met. The mount optimization led to a noise reduction in the vehicle's interior of roughly 8 dB(A), Figure 8; the vehicle status was rated as "acceptable".

10.2 Optimization of the Shock Absorber's Hydraulics

Noise-related complaints were filed in relation to the rear axle on a SUV. Since vehicle development was already in a later phase (near start of volume production), it was decided to address the noise issue by means of valve setting adjustments. When measuring the optimized shock absorber, the piston rod acceleration showed distinctly lower values, **Figure 9**. Vehicle measurements confirm shock absorber measurements made on the test rig and revealed significant improvements in terms of vehicle acoustics. The frequency spectra of the airborne noise in the vehicle during the road simula-

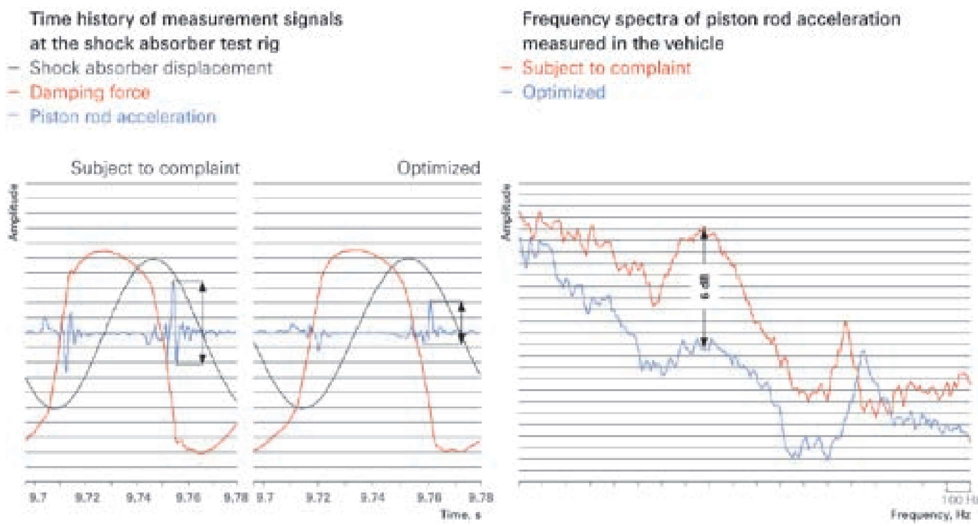

Figure 9: Reduction of piston rod acceleration by shock absorber optimization

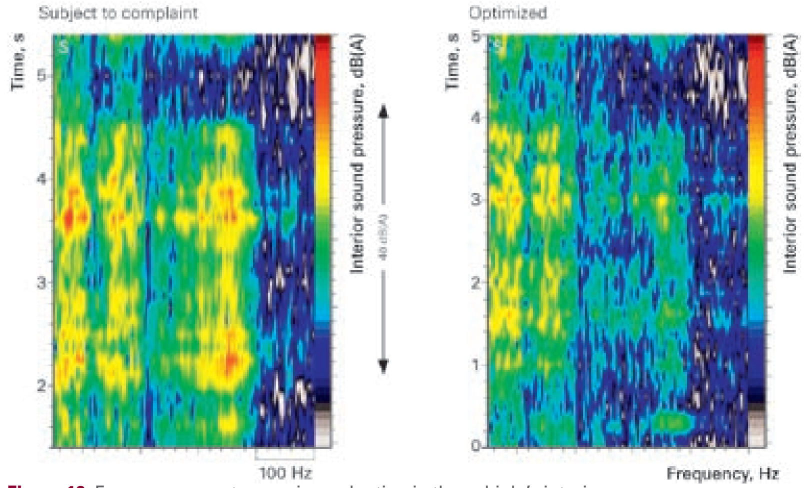

Figure 10: Frequency spectra – noise reduction in the vehicle's interior space

tion on the four-post test rig system are shown in **Figure 10**. The optimization of the shock absorber reduced the noise level by an average of 6 dB(A) in the frequency range of 300 to 600 Hz. The noise complained about is, subjectively, no longer perceptible. Test drives by the customer confirmed that the vehicle was "noise-free".

11 Summary and Outlook

In-depth understanding of the shock absorber's dynamic vibration behavior serves as the basis for a targeted intervention with regard to the NVH behavior of components and their respective environment.

In the product evolution process, acoustics was defined as a development objective and important milestones – quite essential for acoustic quality monitoring – were added. For example Acoustics feasibility study and risk analysis, acoustic targets in the functional requirements (SOR) and performance specifications and acoustic approval at each quality gate (checkpoint). Implementation is supported by well-drafted "Acoustics Checklists" and the NVH fault tree analyses.

ZF Sachs is envisaging even closer cooperation with the vehicle manufacturer in order to effectively and sustainably implement this strategy. This calls for very early involvement in OEM development activities. Thus, it would be desirable to have an influence on and/or the knowledge of NVH-related vehicle parameters such as input impedances at connecting points, vehicle sensitivity with respect to shock absorber excitation, installation position of the shock absorber, and suspension kinematics and compliances.

With such a holistic approach, it would be possible to achieve a high acoustic chassis quality, even with the existing conflict of objectives in the fields of riding/driving and handling behavior.

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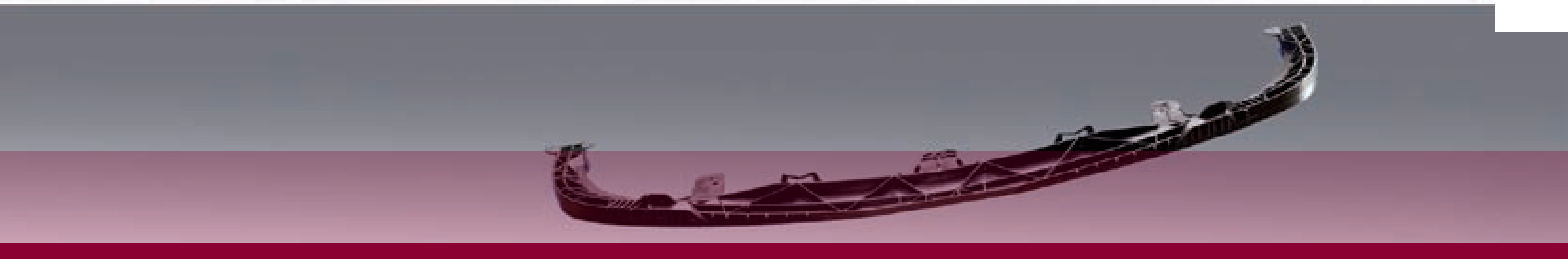

Simulation, Production, Testing
Complete Package for Plastic Component Development

Cost and weight savings together with the aspiration for design freedom coupled with optimum mechanical properties of a component are the drivers in the applications of engineering plastics in cars. Plastics manufacturers have long since ceased to be just suppliers of materials who produce a material merely in conformity with quality requirements and deliver it on time. BASF offers its application development services and know-how comprising demanding computer simulations, methodologically elaborate tests and process technology for parts production. With their aid a component can be supported from the concept phase to series deployment.

1 Process Technology – Producing Parts Efficiently

In the complex interactions in the development and manufacture of sophisticated components in automotive engineering both the nature of the material and the loading together with process-related influencing variables must be taken into account. In parallel with the choice or optimisation of a suitable plastic, important factors on the process side are correct mould design, optimum processing and assembly technology. In its pilot processing plant BASF has at its disposal injection moulding machines with clamping forces between 5 t and 1500 t. By means of the most diverse specimens and test pieces extensive material data sets can be built up.

New methods such as Water Injection Technique (WIT), **Figure 1**, two-component injection moulding and high-temperature plastics processing, **Figure 2**, are used in the development of series parts. At the same time the relevant process characteristics such as shrinkage and warpage, flowability, demoulding and the necessary screw design are evaluated and optimised. With this knowledge the team supports customers in the use of new products, directly in situ on their machines if required, or alternatively in emergency troubleshooting. Apart from this short run parts are produced in the pilot plant for special applications and investigations, for example behaviour in the event of a crash.

In order to offer automotive industry suppliers or OEMs a complete development environment, however, further building blocks are necessary which were often employed in the past as stand-alone services. These include demanding Computer Aided Engineering (CAE) methods involving laws for materials which take account of anisotropy, non-linearity and rate of loading, on the one hand, and the validation of first prototypes or (pilot) series parts using suitable test devices and experimental methods, on the other hand. In the course of an ab initio part development project they frame said technical possibilities in the process because before the first part is injection moulded there is part design and then thorough testing of the actual part, **Figure 3**.

Figure 1: The new hydrolysis-resistant Polyamide 66 grade Ultramid A3HG6 WIT is used to produce the cooling-water pipe for the new 2 l common rail diesel engine from the Volkswagen Group – in series production the mass back pressure process ensures the economically efficient manufacture of the 500 g component with minimum consumption of material

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Figure 2: A de-gassing screw developed by BASF is used during the injection moulding of high-temperature resistant polyether sulfone (PESU; Ultrason E) for the purpose of removing not only residual moisture but also trapped air; Ultrason E test part manufactured without de-gassing (left) and with de-gassing (right)

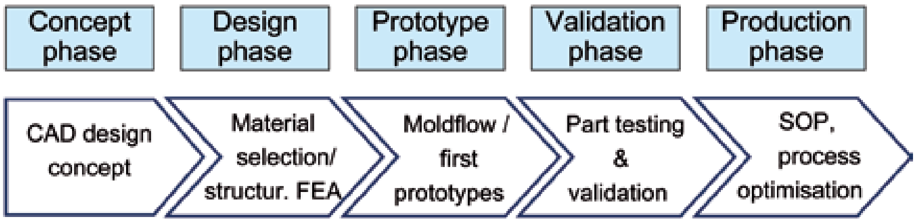

Figure 3: Finding customer-specific solutions: from the idea for the part to mass production

2 CAE Methods

Modern CAE methods have established themselves in the development of plastic parts. The evaluation of component concepts is done ever more commonly on a purely virtual basis.

When accurate virtual models of the material and component are not available many innovative applications remain closed to the plastic. It was possible to show how important this form of application support consultancy has become through the development of the method of Integrative Simulation. An example is the Lower Bumper Stiffener (LBS) for pedestrian protection which BASF together with Opel unveiled in March 2006. At that time it was used in the new Corsa. The latest applications include gear oil sumps, structural bodywork inserts, **Figure 4**, and the next LBS generation which has been developed with a further extension of BASF's simulation package, now called ULTRASIM [1-3], and installed as standard in the new Opel Insignia, **Cover Figure**.

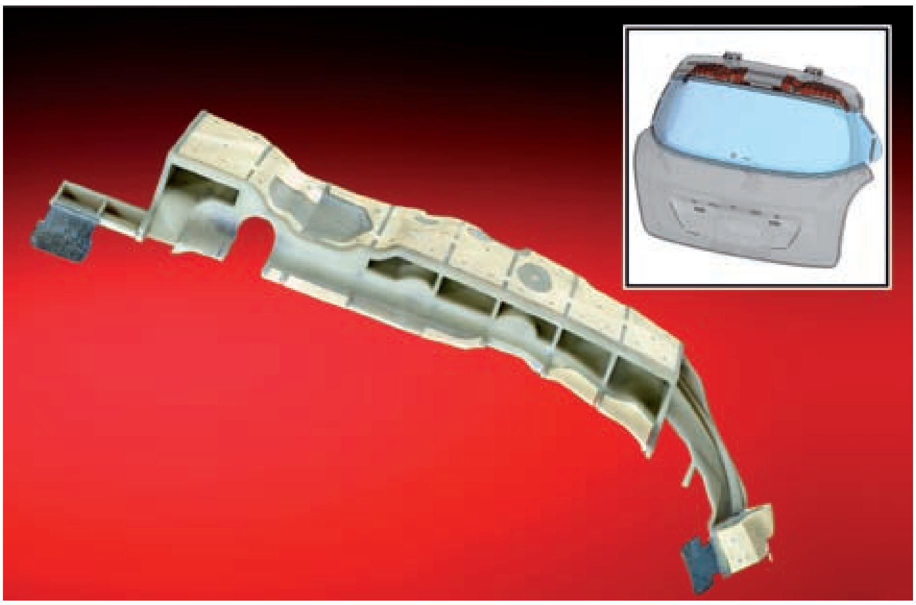

Figure 4: In the tailgate hinges of the new Peugeot 308 sw is concealed a high-strength hybrid component developed by Sika which consists of Ultramid A3WG10 CR (gray) and the structural foam Sika Reinforcer 911NT/2 (yellow)

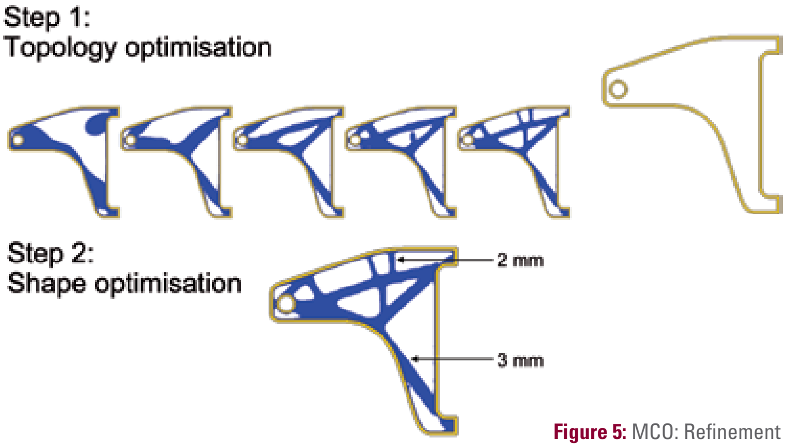

Figure 5: MCO: Refinement of the shape of the component on the computer

2.1 Mathematical Component Optimisation

Integrative Simulation starts its examination of component behaviour on a finished virtual component. If for this purpose, however, an unsuitably designed component is selected – the material is used wrongly – even the best plastic is useless because on the computer and also in reality the component will not be optimal. On the other hand, in many cases only a design adapted to the prevailing loads and force flows allows the optimal use of engineering thermoplastics and highly stressed components made of modern materials only fulfil their potential at all by means of the correct geometric shape. Accordingly, as part of CAE activities a method known as Mathematical Component Optimisation (MCO), now linked to Integrative Simulation, is gaining in importance. With its aid not only are optimum components designed on the computer but also it fills a hitherto apparent gap in virtual component development and represents its further logical development. While in its form to date Integrative Simulation serves the purpose of computing the part correctly, MCO points the way to the correctly designed part.

At the start of such a part development the coarse geometry of the part is first of all determined: At which points is material needed at all? How should any ribbed reinforcements be fitted? Ever since there have been designers, such questions are answered at the start of every part development. What is new, however, is that the intelligent application of software instruments that have now become available allows much more well-founded assertions about part geometry, about what is known as topology. In later phases the mathematical methods of shape optimisation then come into play. Here the precise details of the geometry and further improvements in the shape of the part are addressed.

2.2 Optimisation of Topology and Shape

Test case: A lever on which different forces are to act, **Figure 5**. The objective of topology optimisation is, for example, the requirement for maximum rigidity with utilisation of 20 % of the specified structural space. At the same time certain production boundary conditions are built into the optimisation so that in the end the part can be produced by injection moulding. A particular advantage of the method is that differing boundary conditions can be superimposed. Given simple geometric specifications and few constraints an experienced designer may still determine a suitable component shape in accordance with mechanical principles. This, however, is almost impossible when there are many different cases of loading and installation spaces of complex three-dimensional shape. Here, in view of the software available today, topology optimisation should not be dispensed with. It unfolds its greatest potential when it is employed at the start of a development project. Further refinement of the component then ensues by means of shape optimisation. In this step optimisation objectives such as minimum weight are mathematically combined with the constraints characteristic of the part. Depending on requirements these may be maximum deformation under different loads or a specified stress is not exceeded. The parameters to be determined in doing this are in the simplest case wall thicknesses in the different zones of the part to be optimised. The design engineer then translates the still abstract proposal in accordance with principles typical of plastic into an efficiently producible component.

2.3 Morphing

The shape of a part can be optimised particularly elegantly by means of morphing techniques. Morphing means that the fixed finite element model can be spatially distorted in accordance with certain rules. Like a structure composed of plasticine the numerical model of the part can now be virtually compressed, pressed, squeezed, bent and drawn out in length. In doing so each variable such as height, breadth or length is described in the programmes by a morphing parameter and can be continuously changed. In doing so the finite element network is automatically adjusted.

2.4 Optimisation for the Series

Mathematical Component Optimisation has now been successfully deployed in the development of the LBS II. This is the successor to the first LBS, the lower bumper stiffening system for pedestrian protection at Opel. As with the first LBS, strict pedestrian protection directives have to be complied with: bending angles in the knee region and lower leg acceleration of a human leg must not exceed fixed threshold values. The first

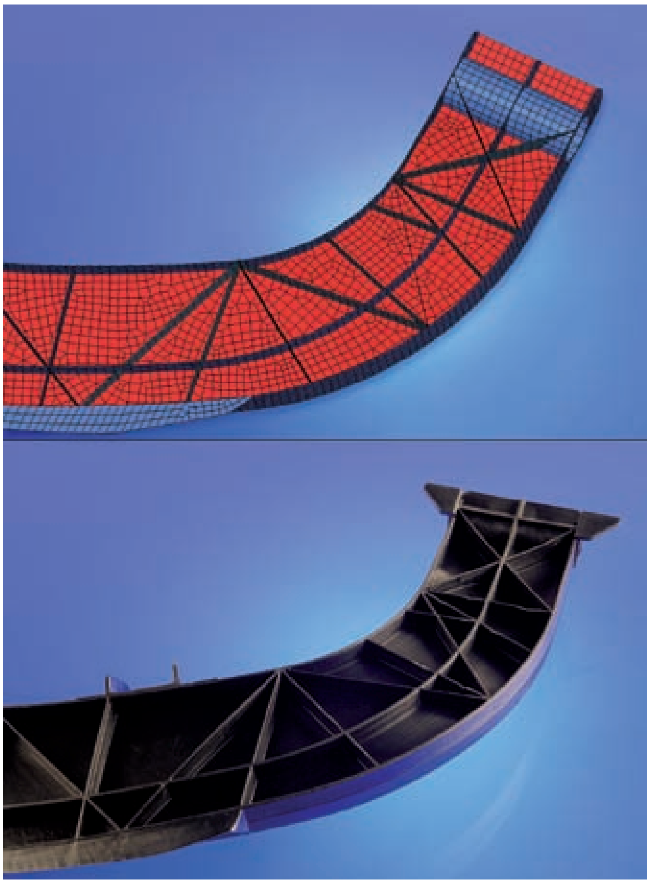

Figure 6: The prototype of the LBS II: top: FE model; bottom: real component

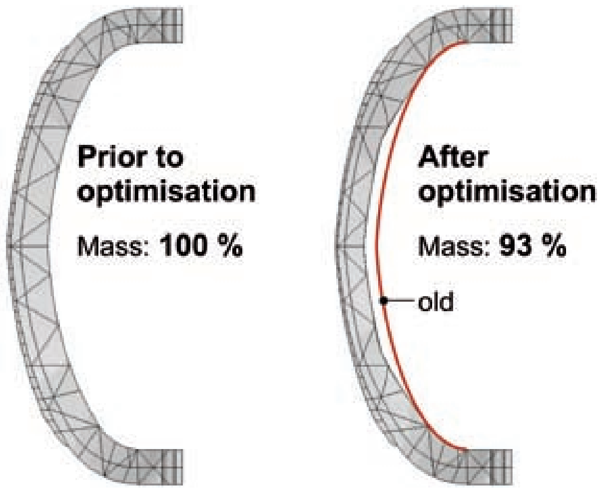

Figure 7: Shape optimisation by means of morphing: further 7 % weight saving

objective in the development of such a LBS is a high rigidity matched to the front of the vehicle in question. In doing this topology optimisation gives rise to a part which at all impact points selected on the leg simulator (lower leg impactor) ensures maximum stiffness. Once the results of topology optimisation are fed back in the form of CAD data into a virtual component of defined geometry, a genuine component can be injection moulded on the basis of this information. The real plastic part made from the material Ultramid B3WG6 CR specially optimised for crash applications exhibits, as expected, very good characteristics in the pedestrian protection tests, **Figure 6**.

In addition to the topology optimisation described, shape optimisation was also carried out with the aid of morphing techniques. The aim was to achieve a further reduction in weight. The contour pattern of the rear edge of the component was modelled by means of four morphing parameters. In this way the stiffness of the component along the longitudinal axis of the vehicle can be selectively varied over the width of the component. The basis of optimisation is provided by several dynamic collision analyses at various points on the leading edge of the vehicle. The optimisation objective is to minimise the mass under the constraint of a specified dynamic stiffness of the component in the direction of the vehicle resulting from the pedestrian protection requirements. In this way through shape optimisation the mass was reduced by a further 7 % and at the same time the part was better adapted to the impact requirements, **Figure 7**. These important insights from the work on the pilot run then flowed into the series-produced component.

3 Component Testing

Tests on the component such as the quasi-static investigations of fracture behaviour, **Figure 8**, were indispensable in the validation of the LBS prototypes. Despite the performance of modern CAE tools it is not possible to do without component testing and methods of experimental mechanics in the development and validation process, Figure 3. As described with regard to the LBS I and II the main effort in the development process once the first drawings have been prepared is devoted to CAE computations in order to clarify issues of both mechanics and mould design and filling and for carrying out initial virtual optimisation runs. After the first prototypes are available validation tests are then necessary to show whether the properties of the part meet the demands of the specification of requirements. For this purpose associated structures as close as possible to practice are needed so that real boundary conditions as well as geometric, physical and structural non-linearities can be taken precisely into account. On the basis of the results the quality of the CAE computation can be checked, **Figure 9**. If the requirements are not met a numerical or experimental optimisation loop is initiated.

Since engineering plastics are exposed to the most various stresses the require-

Figure 8: Apart from the high-dynamics trials on the entire system of the front of the vehicle at Opel, BASF carried out quasistatic tests on the part in its test laboratory – by means of these trials it was possible to verify the dynamic loading

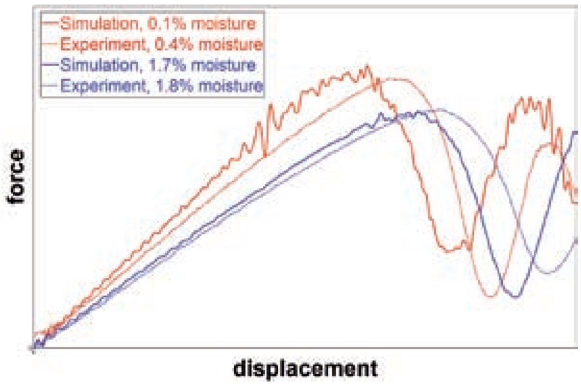

Figure 9: Comparison of force-displacement diagrams for the LBS II obtained from simulation and component testing; considering the differences in the moisture content of the component the agreement is very high; strain rate: 0.4/s

ments for component tests are highly diverse. Accordingly, a corresponding spectrum of possible tests, analyses and optimisation methods is needed. As a result of many years of project work the following experimental and test methods are at disposal which are recommended for investigations during the development process [4, 5]:

– analysis and optimisation of deformation and strain, for example speckle interferometry, static and transient stereophotogrammetry (Aramis)
– analysis and optimisation of oscillations, for example experimental modal analyse, shaker excitation, oscillatory form analysis under operating conditions
– analysis and optimisation of acoustic properties of components, for example sound intensity measurements, reflection levels, artificial head measurements, psychoacoustic parameters
– tensile, compression and bending tests, three-dimensional loads on components
– impact, crash, falling, head-on collision and flying stone tests
– high-temperature ageing, temperature and climatic tests
– diverse vibration tests, with superimposed temperature and climatic conditions inter alia
– flow tests
– static and dynamic bursting pressures at pressure change rates of up to 1 bar/ms
– transient pressure cycle tests
– documentation of transient events (crack formation, crash, stone impact) by means of high-speed cameras at up to 100,000 images/s.

4 Conclusion

Mathematical optimisation methods, comprising topology optimisation and shape optimisation including morphing, are useful supplements to virtual component development. They have been integrated into the already very extensive Integrative Simulation instrument which in this way has become the ULTRASIM package. The second generation lower bumper stiffener (LBS II) was developed on the computer with the aid of these methods. This contributed to the Opel Insignia achieving maximum points in the EuroNCAP lower leg test [6]. Validation of the LBS II likewise ensued with the assistance of the plastics manufacturer and in series production BASF was able to apply its knowledge to useful effect: after all the process-related challenges in the production of such a large component are not inconsiderable. All the service building blocks presented here, that is CAE, component testing and process technology, form part of the extensive but still flexible business model which can be adapted to the requirements of individual customers. In this way the development of highly demanding vehicle parts made from engineering plastics becomes very efficient so that reductions in weight, emissions and costs can be achieved more rapidly and reliably with maximum safety for pedestrians and occupants.

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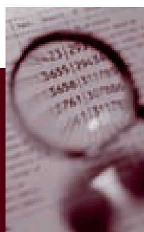

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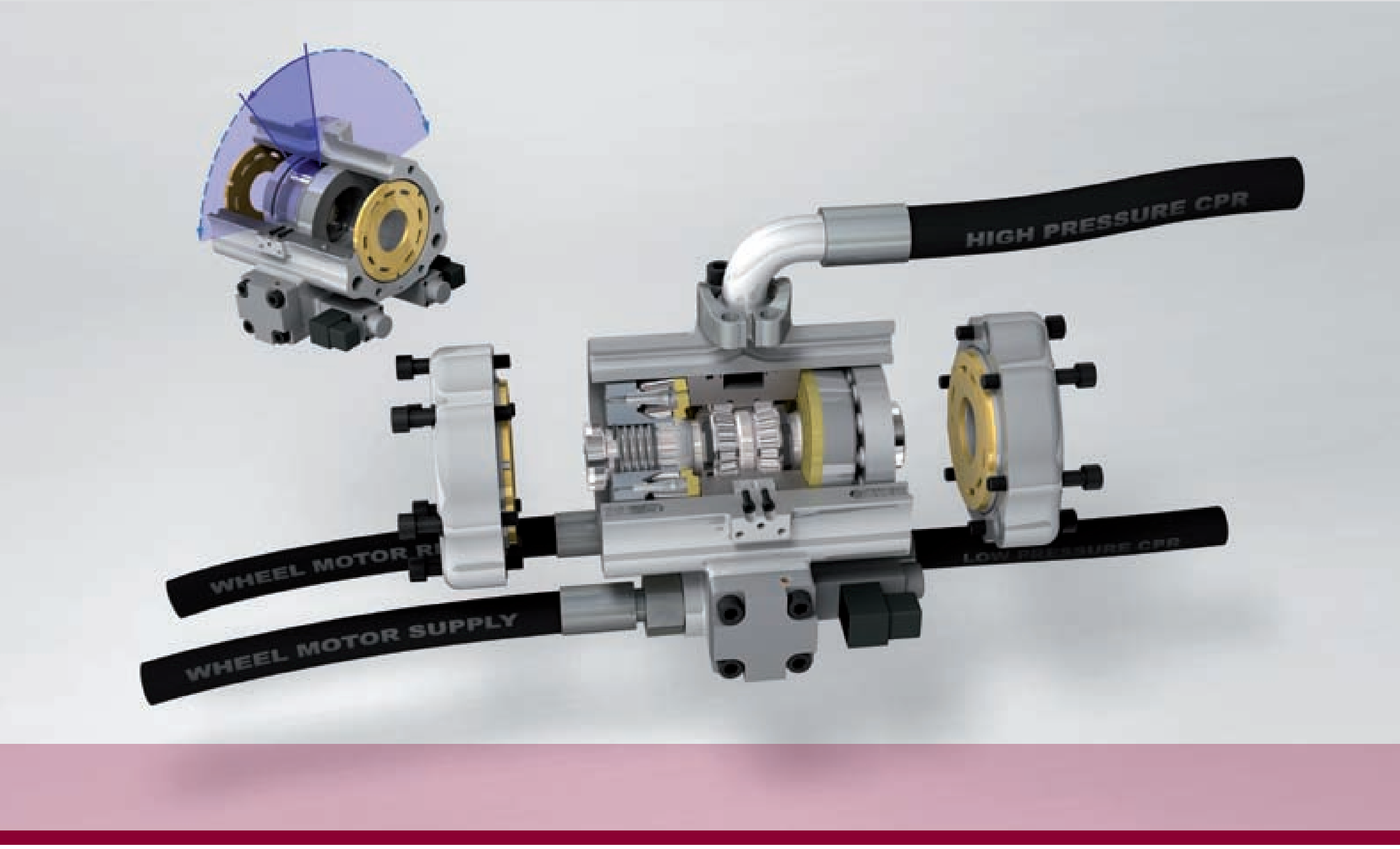

Low-emission Hydraulic Hybrid for Passenger Cars

Two new technologies, the „floating cup"-principle and a new hydraulic transformer, allow the complete substitution of the mechanical drive of a passenger car for a series hydraulic hybrid transmission. The new "Hydrid" power train from Innas and the RWTH Aachen University reduces the fuel consumption of a medium-sized car by 50 %. The CO_2 emissions are reduced to 82 g/km, a level far below the EU-limit for the year 2012 (120 g/km).

1 Introduction

Innovation is driven by changes in the economic environment. A change of market demands and requirements leaves the industry no other option: it has to innovate. The automotive industry is currently facing the most fundamental changes of its history: strong fluctuating fuel prices, new CO_2 limits and an unprecedented cost pressure. Given these circumstances the industry has to innovate and is indeed willing to do so. The new economic and legal demands seem to be both defying and conflicting. For the reduction of fuel consumption and CO_2 emissions, all hope is focussed on the (parallel) hybrid electric drive train, with the all-electric transmission on the horizon as the ultimate solution. The electric components however cause a strong increase of the manufacturing cost, resulting in a limited market acceptance. Recent studies showed a very limited potential for hybrid electric vehicles of less than 10 % of the total sales volume by the year 2035 [1, 2]. The hybrid electric transmission is also by far the most expensive option for CO_2 abatement [3].

The cost increase of hybrid electric transmissions is inevitable. Being a parallel hybrid solution, the electric system is an add-on to the mechanical transmission, and by definition increases complexity, weight and cost of the vehicle. Despite mass production, electric transmission components are still an order of a magnitude too expensive [4, 5]. Furthermore, the poor average cycle efficiency of the batteries and the electric motors result in a limited reduction of the fuel consumption.

A better solution would be to replace the mechanical transmission by a hydraulic transmission having accumulators for energy recuperation and power management. The result is a "Hydrid", a series hydraulic hybrid vehicle [6, 7], which has the same basic architecture as a series electric hybrid drive train, **Figure 1**. However, compared to electric batteries, motors and controllers, hydraulic transmission components are extremely robust, have a much higher power density and have much lower manufacturing cost. It is furthermore expected that a hydraulic transmission will have the same weight and manufacturing cost as the mechanical transmission it replaces. Key for the success of the Hydrid is the introduction of hydraulic transformers (depicted above) for power control. Furthermore a new multi-piston principle is applied which strongly reduces the noise, vibration and harshness issues related to conventional hydraulic motors. Simulations of the new drive train have indicated that the fuel consumption will be reduced by more than 50 %. Although the average cycle efficiency of the hydraulic transmission is not as high as of the mechanical transmission, this is more than compensated by the efficiency advantages of the hydraulic transmission due to its capabilities for power management and energy recuperation.

The new "Hydrid" transmission does not exclude the electric battery. On the contrary, it facilitates the introduction of a base load electric system which can now be much smaller than of current hybrid electric vehicles. But even without the help of batteries, small hydraulic accumulators with a volume of 10 to 20 l are sufficient to handle most of the brake energy recovery and all of the power and traction transients of the vehicle.

2 Maintaining the Performance

The drive train of a vehicle is dimensioned for peak performance, for instance for being able to climb a mountain pass with a trailer load or for fast acceleration from 0 to 100 km/h. These peak requirements differ very much from average, day-to-day driving conditions, **Figure 2**.

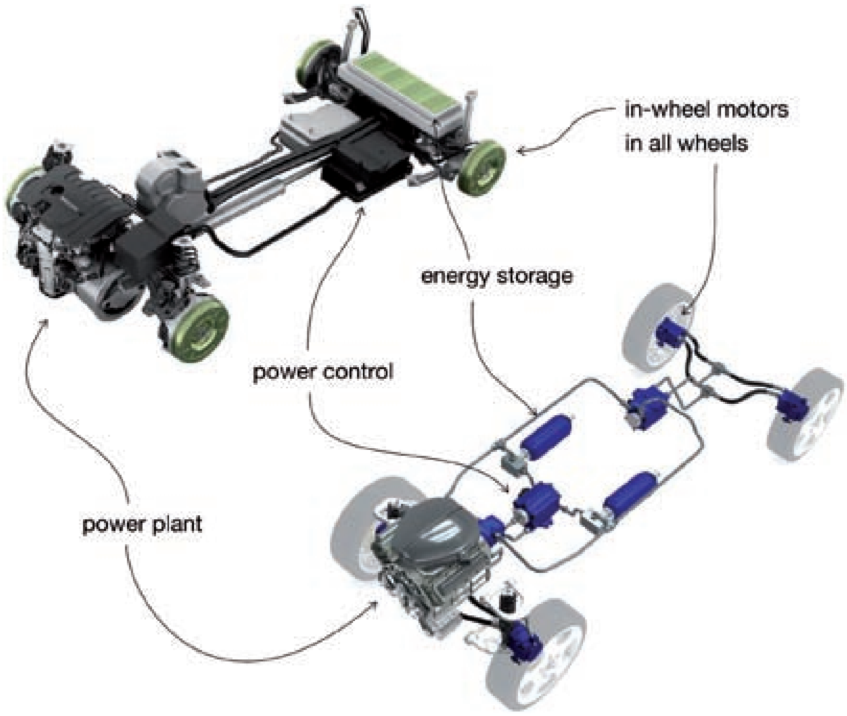

Figure 1: Comparison of a series electric hybrid (left) and a series hydraulic hybrid transmission (right)

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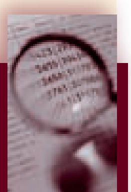

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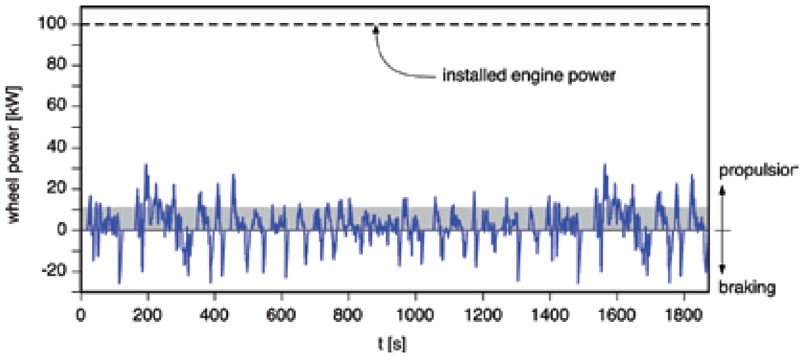

Figure 2: Required total wheel power for a mid-sized passenger car with a curb weight of 1450 kg driving the US FTP-cycle, in comparison to the installed engine power – during 80 % of the propulsion time, the required wheel power is less than 11 kW (indicated by the grey band in the diagram)

The engine operation has an enormous influence on the fuel consumption. When accelerating or cruising during the FTP-cycle, the maximum required wheel power is 32 kW, less than one third of the installed engine power. During 80 % of the time the power demand is less than 11 kW and for half the time even less than 5 kW. At these power levels the engine has a very poor efficiency, which is the main reason for the high fuel consumption of passenger cars, especially when driving in the city. Low load operation of the engine can however be completely avoided by applying an energy storage system in the drive train. The engine power does not need to be reduced and the vehicle performance is not compromised. The energy storage allows the engine to be operated at relatively high loads and power outputs, even when the vehicle only requires a low propulsion power. Whatever the engine produces in excess can now be supplied to the energy storage system.

There are several advantages for having an energy storage system integrated in the transmission of a passenger car. The engine operation is strongly improved, without compromising the maximum performance of the engine or the vehicle. Moreover, the storage system facilitates start-stop-operation of the engine, thereby eliminating idle losses of the engine. Furthermore, the energy storage system can also be used for recuperating the kinetic energy of the vehicle when braking. Contrary to most expectations, passenger cars do not need a large energy storage for energy recuperation. Most brake actions only require a storage capacity of about 30 Wh, much less than the capacity of an average starter battery. However, the power demands for such a storage system are quite high, Figure 2. Any energy storage system for a passenger car must therefore be capable of handling and managing high and strongly varying power levels: it foremost needs a high power density, not a high energy density.

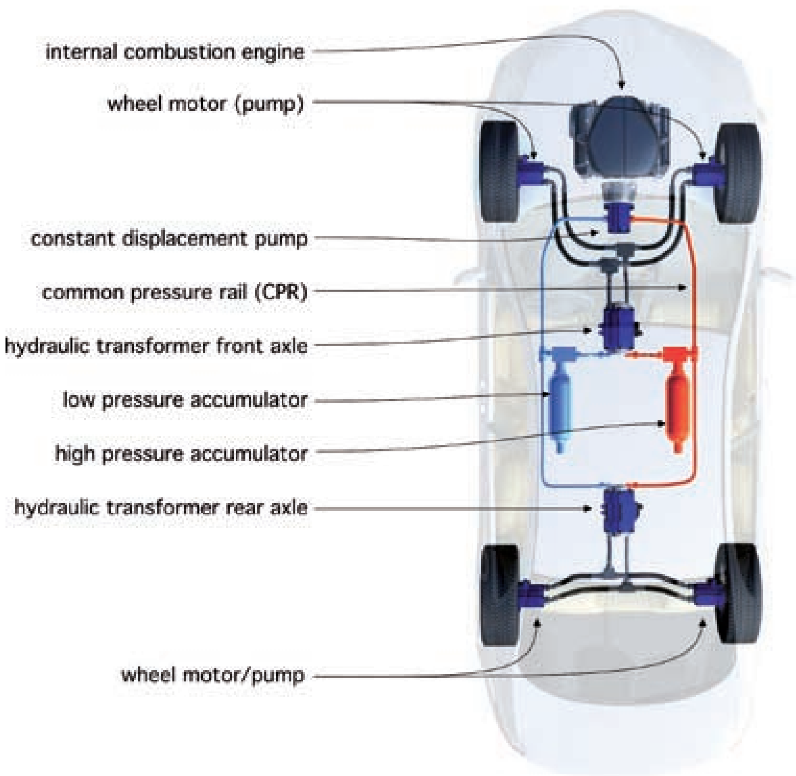

Figure 3: Layout of the Hydrid: a series hydraulic hybrid transmission

3 Replacing the Mechanical Transmission

Hydro-pneumatic accumulators fulfil these demands. An accumulator is basically a pressure vessel having an internal nitrogen volume. When oil is supplied to the vessel, the nitrogen is compressed thereby increasing the pressure of both the oil and the nitrogen. Accumulators are extremely robust, and although the energy content is much lower than that of batteries, they have an unparalleled power capacity of more than 20 kW per kg. Due to weight and cost constraints, the size of the accumulators have to be kept small, somewhere between 10 and 30 litres for an average passenger car.

The accumulators are connected to the common pressure rail or CPR, **Figure 3**, which is the power grid of the system. The internal combustion engine is no longer hard coupled to the wheels, but is instead only used to drive a hydraulic pump. The pump is a simple constant displacement pump for which the torque demand T is proportional to the pump pressure differential Δp:

$$T = \frac{\Delta p \cdot V}{20 \cdot \pi} \quad \text{Eq. (1)}$$

Hydraulic accumulators have a limited pressure range, for instance between 200 and 400 bar. Assuming a displacement volume V of the pump of 60 cc/rev, it can be calculated that the pump (and engine) torque T varies between 191 and 382 Nm. The engine can therefore only be operated at high loads. Strong part load operation is completely avoided. The hydraulic power plant is only operated if and when this is required for maintaining a certain pressure level in the high pressure accumulator. In case the accumulator does not need to be charged, the engine is shut down completely, thus avoiding any idling losses.

By means of the hydraulic transmission, the vehicle traction is directly created at the wheel shaft by the hydraulic motors. The traction is controlled by changing the pressure differential across the in- and output ports of these motors. When reversing the pressure differential, the hydraulic motors will act as pumps, thereby decelerating the vehicle while recuperating the brake energy and storing it in the accumulators. The hydraulic transmission can be an all-wheel drive, Figure 3, having a variable traction for both axles and allowing energy recuperation on all four wheels.

4 A New Hydrostatic Principle

In a conventional drive train the vehicle traction and speed is directly related to the engine torque and speed. The new Hydrid transmission separates the power supply from the load control and the wheel traction is directly created at the wheels. This makes the vehicle extremely responsive, but it also sets high demands for the wheel motors which now have to fulfil all extreme demands of the vehicle, and have to run in a wide speed and torque range. The maximum torque needs to be delivered at low vehicle speeds where conventional hydraulic motors have high friction losses. Furthermore the noise, vibration and harshness (NVH) is extremely critical for the design of a passenger car. The direct connection of the hydraulic motors to the wheels eliminates the opportunity of using dampers and flywheels to reduce torque ripples.

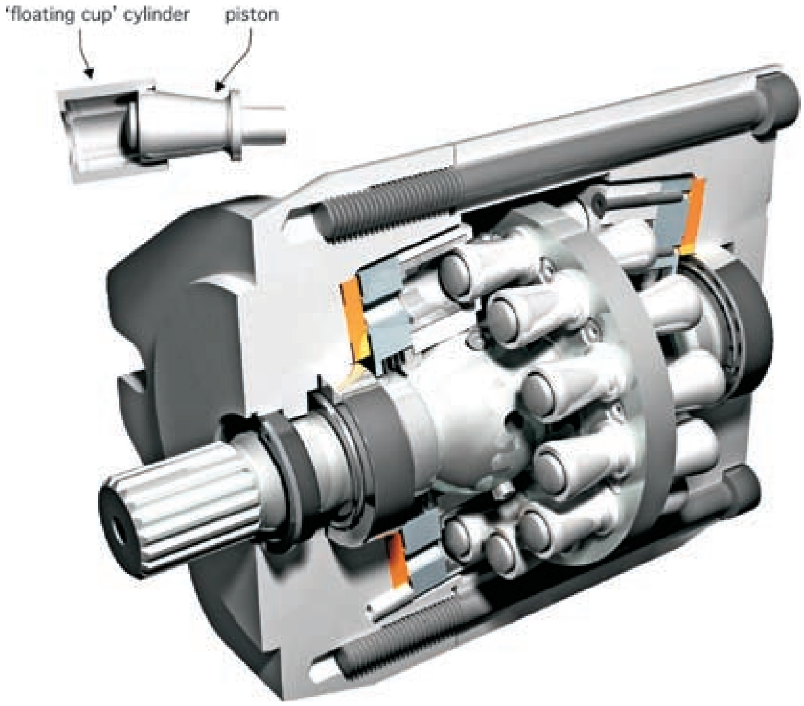

Figure 4: Cross section of a constant displacement floating cup motor/pump

The wheel motors therefore need to have an extremely constant torque output.

In the past few years a new hydrostatic principle has been designed to fulfil all these requirements [8]. The main characteristics of this "floating cup" principle, **Figure 4**, are:

- multi piston design typically having around 24 pistons
- mirrored configuration for reducing the hydrostatic load on the bearing
- a direct conversion of hydraulic pressure forces to torque (and vice versa) with very low friction losses
- fit for low-cost mass production technologies like deep drawing, sintering, fine blanking and sorting.

Figure 5 shows a comparison of the measured torque at near-to-zero speed conditions for three different hydraulic motors relative to the theoretical maximum torque. Only the floating cup motor shows

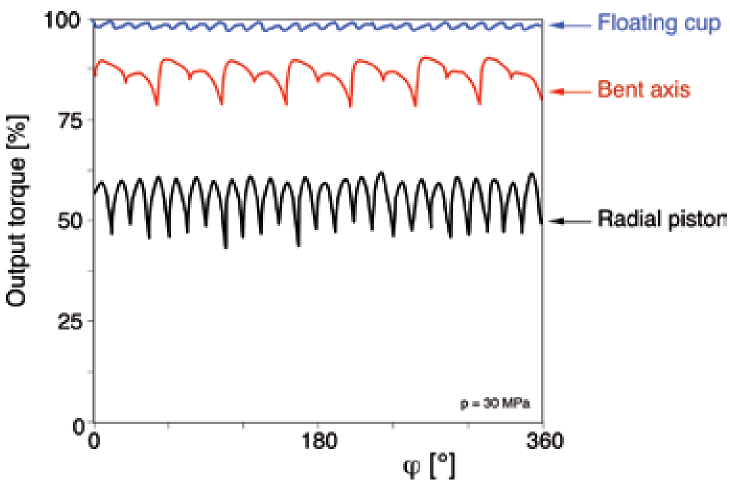

Figure 5: Measurement of the torque output of three different hydraulic motors, relative to the theoretical maximum torque output – the measurements are performed at 300 bar and rotational speeds below 1 rpm

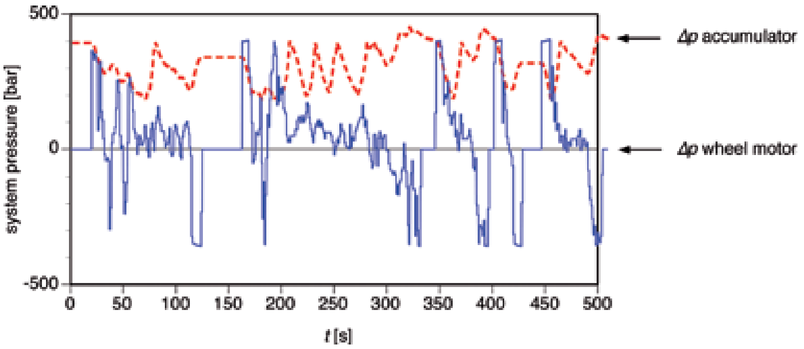

Figure 6: Pressure differential between the high and low pressure accumulators and the demanded pressure differential at the wheel motors during the first 500 seconds of the FTP75-cycle

almost no torque losses due to friction at these operating conditions. It also is apparent that the torque variations of conventional hydraulic motors are not acceptable for a direct wheel drive in a passenger car. Only by means of a substantial increase of the number of pistons, the torque variations can be reduced to acceptable levels.

The floating cup principle is capable of running in a wide range of operating conditions. As a motor it can directly drive the wheel without the need of a gear transmission. It can also be operated at high pressures, thereby creating a high wheel torque with a relatively small motor. At a pressure of 500 bar, a floating cup motor having a displacement of 56 cc delivers a wheel torque of 446 Nm. Having four-wheel motors, the total torque created is sufficient for a gradability of 44 %, or for accelerating the car from 0 to 100 km/h within 9 s.

5 The Hydraulic Transformer

The pressure level in the high pressure accumulator varies depending on the state of charge. The pressure level rises when recuperating the brake energy or when charging the accumulator with the engine-pump-combination. The pressure in the accumulator differs strongly from the pressure needed for creating the required traction at the wheels. **Figure 6** shows the calculated pressures for the first 500 s of the US FTP75 cycle. In the simulation the vehicle is only driven by two- wheel motors; the other two are disengaged. A positive pressure differential indicates a propulsion mode of the vehicle, whereas a negative pressure differential occurs when the vehicle is braking.

The gap between the two pressure differentials is bridged in the Hydrid by means of a hydraulic transformer. This transformer is the hydraulic equivalent of a CVT, converting pressure and flow instead of torque and speed:

$$(\Delta p \cdot Q)_{in} = (\Delta p \cdot Q)_{out} \qquad \text{Eq. (2)}$$

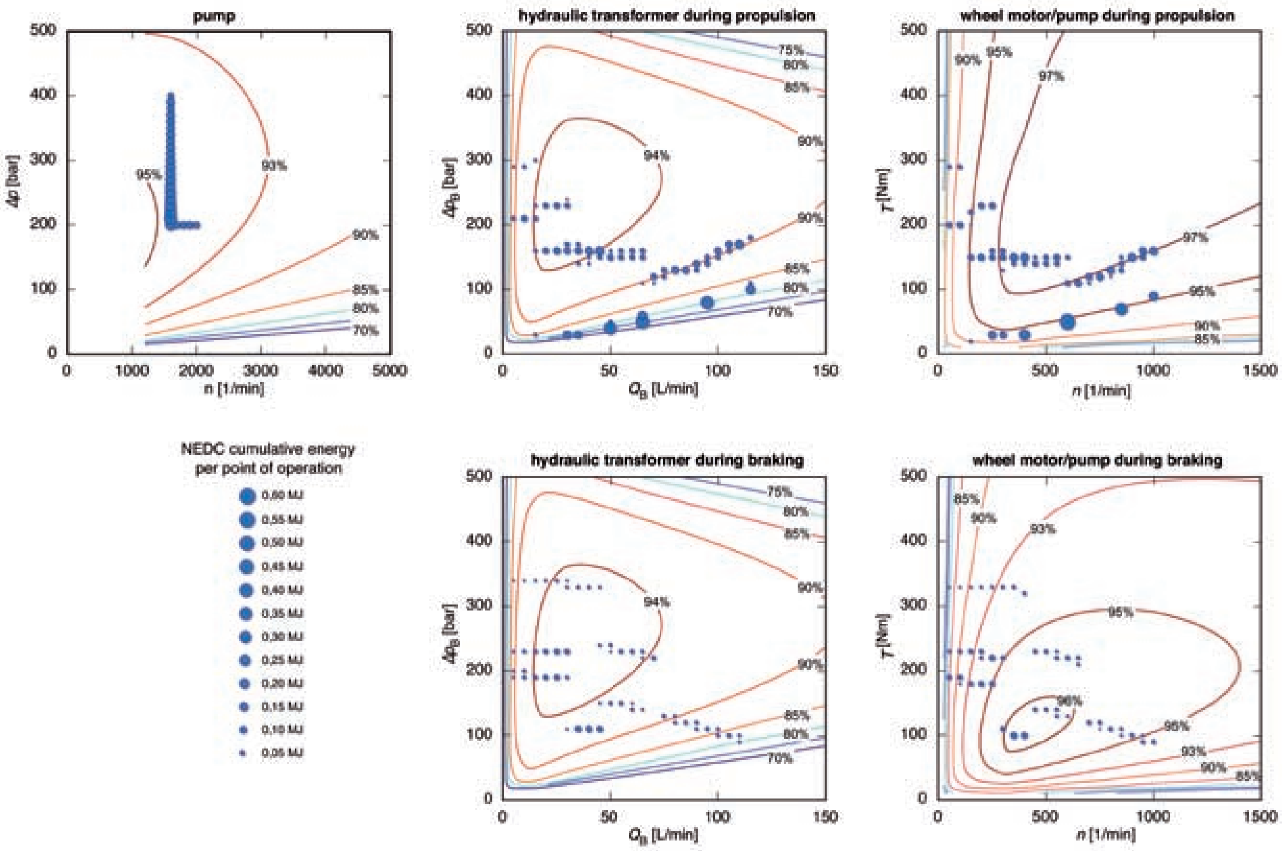

Figure 7: Efficiency maps of the main transmission components of the Hydrid, including the NEDC operating points

A new transformer has been designed on the basis of the floating cup principle to fulfil the requirements of automotive transmissions [9]. The transformer allows a full 4-quadrant operation of the vehicle (forward propulsion, forward braking, reverse propulsion and reverse braking). One of the most important features of the transformer is its ability to amplify pressures. This is especially needed for fulfilling demands concerning gradability, acceleration performance and elasticity, when the total required wheel torque is much higher than the torque needed for driving the NEDC or FTP-cycle. The hydraulic transformers allow the application of simple, small, robust and efficient constant displacement motors for driving the wheels. The motors can be kept small since the transformers can amplify the pressure level to a boost pressure of 500 bar, even when the pressure level in the accumulator is only 200 bar.

Without a transformer, variable displacement motors would be needed to meet the variable torque requirements at the wheels. However, the maximum torque of a variable displacement motor is determined by the pressure level in the high pressure accumulator. The advantage of pressure amplification is lost and the displacement of the variable motor needs to be increased to compensate for the lower pressure level. In order to create the same maximum torque, even when the accumulator pressure is only 200 bar, the motors would require more than double the displacement volume. Aside from the extra weight and cost of these larger variable displacement motors, the larger size also results in a stronger part load operation and a poor part load efficiency of the motors during normal operating conditions.

Table: Hydrid car parameters

empty curb weight	1450 kg
maximum traction	5700 N (4-wheel drive)
maximum vehicle speed	190 km/h
frontal area	2.26 m^2
drag coefficient	0.26
dynamic wheel diameter	0.63 m
rolling resistance coefficient	0.008
engine	100 kW diesel engine
size of the accumulators	20 Liter
pressure range of the accumulator	200-420 bar
pump displacement	56 cc/rev (constant displacement)
size hydraulic transformers	56 cc/rev (pump equivalent)
size of the wheel motors	56 cc/rev (constant displacement)
maximum Δp wheel motors	500 bar

6 Fuel Consumption and CO_2 Emissions

The Institute of Fluid Power Drives and Controls (IFAS) at RWTH Aachen University has performed an analysis of the specific fuel consumption and CO_2 emissions of the Hydrid. On the basis of efficiency tests, efficiency maps of all pumps, motors and transformers were derived, **Figure 7**. These maps were implemented in a DSHplus simulation model of a passenger car, including the most relevant valve losses, accumulator losses and losses in the connecting hydraulic lines. A mid-sized European passenger car has been taken as a benchmark, see the **Table** for the specifications. The fuel consumption has been calculated for both the NEDC and the US FTP75-cycle.

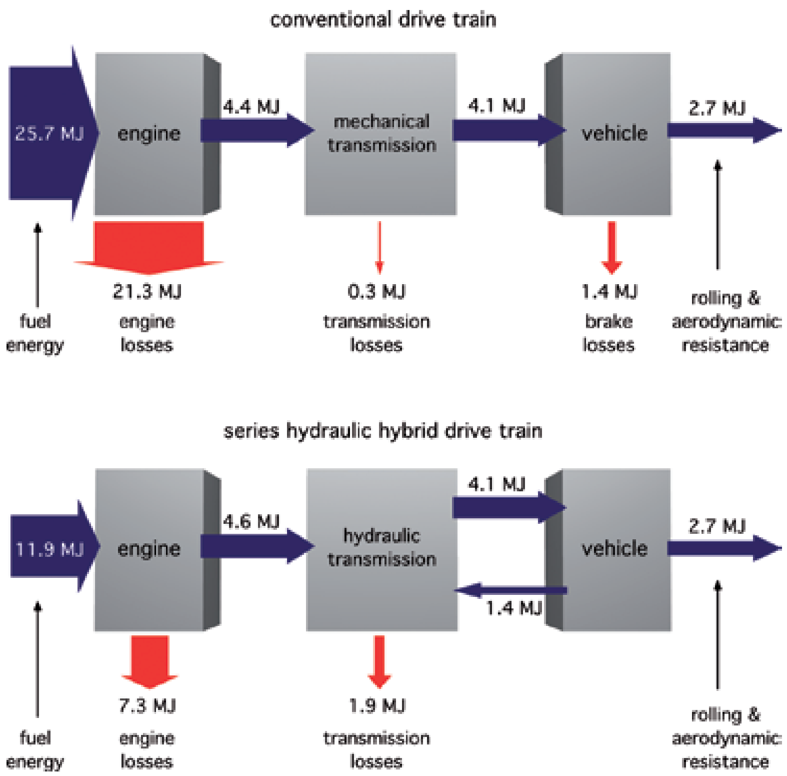

Figure 8: Calculated energy flows for a conventional mechanical transmission (six-speed, manual, AWD) and the new Hydrid drive train

The energy flow in a transmission with energy recuperation is always more complicated than in a regular transmission without recuperation. **Figure 8** shows the aggregated energy flows as calculated for the New European Driving Cycle (NEDC). The transmission losses of the mechanical drive train are almost negligible and are for certain much lower than of the hydraulic drive train. The energy loss of the hydraulic drive train is partly due to the limited energy storage capacity of the accumulator. With the hydraulic drive train, all brake actions are performed by the hydraulic wheel motors, which are then acting as hydraulic pumps. The brake energy is stored as much as possible in the hydraulic accumulator, but as soon as the high pressure accumulator has reached its maximum value, the rest of the brake energy is dissipated in a pressure relief valve. Of the 1.9 MJ calculated hydraulic transmission losses, 0.5 MJ is dissipated in the pressure relief valve. In the end 55 % (NEDC) to 76 % (FTP75) of the energy supplied back by the hydraulic transformers is effectively stored in the accumulator. All hydraulic losses are converted into heat. During the cycle the average required hydraulic cooling capacity for both cycles is around 1.6 kW. An advantage of hydraulic components is that the hydraulic oil transports the heat away from the transmission components. Unlike electric components there is no extra cooling system needed.

The amount of energy that needs to be supplied by the internal combustion engine is almost equal for both transmissions: the higher losses of the hydraulic transmission are offset by the brake losses of the mechanical drive train. Yet, there is a large difference in the fuel consumption of both drive trains, which is entirely due to the way the engine is operated and the effect this has on the engine efficiency. In the Hydrid configuration the engine is switched on and off and is only in operation during 10 % of the NEDC-time. But when it is in operation, it is always running at high loads, between 170 and 350 Nm, **Figure 9**. Operation at low load conditions, as is often the case with the mechanical transmission, is avoided completely. As a result, the specific fuel consumption is reduced from 6.6 l/100km for the reference vehicle to 3.1 l/100km for the Hydrid. The specific CO_2 emission is reduced from 174 to 82 g/km. The on-off

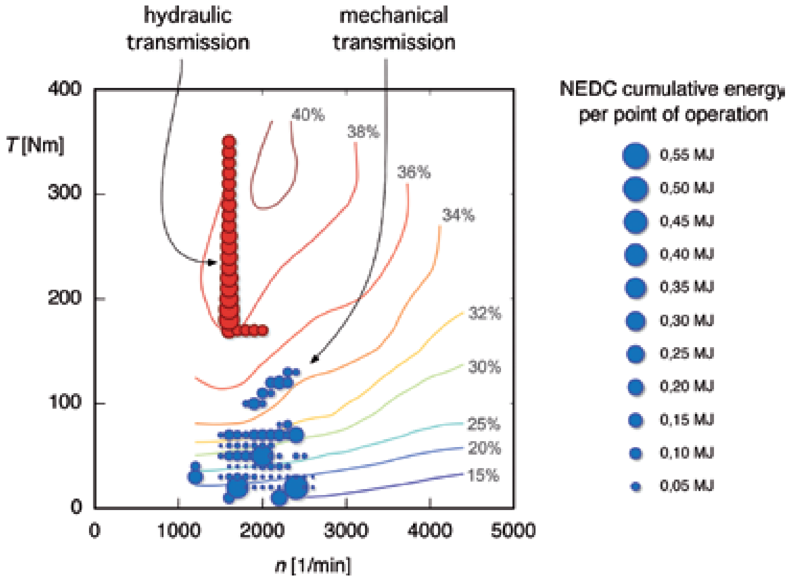

Figure 9: NEDC-operating points of the engine of a conventional vehicle with a mechanical transmission and of a vehicle with a Hydrid transmission, plotted on the efficiency map of the diesel engine – the size of the bubble indicates the amount of energy the engine produces at that point

operation of the engine and the shift of operating conditions will have an effect on the emission of particulates, NO_x and other legislated exhaust gas emissions. However, this is a concern for all hybrid vehicles in which the engine is shutdown whenever this is possible and the engine is operated in different operating points. Eventually it is to be expected that the engine control, and maybe even its design will be changed to adapt and optimise the new operating regime of the engine for the hybrid drive train.

7 Conclusion

Two new technologies, the floating cup principle and a new hydraulic transformer, enable the entire replacement of the mechanical transmission in passenger cars by a series hydraulic hybrid transmission. This new Hydrid transmission reduces the specific fuel consumption of a mid-sized sedan by more than 50 %, without compromising the cost, weight, acceleration performance or gradability of the vehicle. The CO_2 emission is reduced to 82 g/km, far below the limits set by the European legislation for the year 2012 (120 g/km). These reductions are mainly realised by forcing the engine to operate at high loads only and shutting it down when whenever the accumulator is charged. Essential for the low fuel consumption of the Hydrid are the high part load efficiencies of the new hydraulic components. The same series drive line layout may be realised with electric or conventional hydraulic units, but these lose a very large part of the reduction potential because of their rather poor part load efficiencies. The new Hydrid transmission also facilitates the application of electric batteries, which can be used for their strength (energy density) without letting their weakness (power density) lead to compromises regarding cost, weight or performance of the vehicle.

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Self-excited Full-vehicle Oscillations in Dynamic Packaging Examination

Dynamic clearance studies in test drives for vehicle approval include starting-off manoeuvres for ascertaining the maximum space required by the engine/gearbox unit in the longitudinal direction of the engine. Under certain circumstances, sharp moving-off procedures can result in what is referred to as the power hop phenomenon. This is an oscillation of the chassis, suspension and powertrain boosted by the tyres, which not only increases the amount of space required but also reduces driving comfort. In this paper, the Institute of Automotive and Powertrain Engineering (IFAS) of the Helmut Schmidt University (Federal Armed Forces University) Hamburg and Volkswagen AG not only discuss the measuring technology used and offer an interpretation of the effect, but also provide an outlook as to reproducibility and treatment of the phenomenon.

1 Introduction

For reasons of driving comfort, in vehicle manufacture the engine/gearbox unit is decoupled from the body by means of elastomeric mounts. This flexible mounting allows the engine to start moving during dynamic procedures. The package density in the engine compartment is increasing steadily; therefore it is becoming more and more important to guarantee the appropriate clearances around the combustion engine during driving.

The time sequences of the engine movement are determined in road tests at Volkswagen AG in order to ascertain the clearance conditions in the engine compartment. These are used for generating enveloping surfaces and providing them for the digital mock-up (DMU).

These road tests can only be performed at a later point in time when the subframe and/or prototypes are available. This means simulation involving multi-body system models represents a further opportunity for predicting the dynamic space requirement. Virtual techniques are being used increasingly in the car industry as a means to reduce the time required by the product creation process (PCP), which means the physical model can be assured virtually at a relatively early stage. Questions relating to component strength, driving dynamics, determining the collective load and technical control aspects are investigated. These different subdisciplines of virtual development are increasingly intertwined.

Increased use of simulation techniques is intended not only to shorten the creation process but also to cut the financing requirements because fewer subframes and prototypes are required.

The simulation process for dynamic space has been used at Volkswagen AG for quite some time now. This involves the driving manoeuvres of the approval process for the dynamic packaging examination being simulated in the multi body system (MBS) software program MSC.Adams.

The driving manoeuvres for calculating the maximum space requirement include driving on roads of varying unevenness, see [1] and [2], as well as various starting-off manoeuvres both forwards and in reverse. Driving on uneven roads excites the vertical movement of the engine whereas, because of the swinging mounting, starting-off manoeuvres lead to oscillating movements of the transversally installed engine in the longitudinal axis of the vehicle. The forecast quality of virtual clearance study is very high for the engine movement, as is described in [1] and [2].

2 Definition of a Project

Starting-off manoeuvres are also taken into account when calculating the required design space. These involve sharp starting-off manoeuvres ("jumping the clutch") in forwards and reverse on the flat and on gradients, involving the driver running the engine at about 3000 rpm with the vehicle stationary and then suddenly letting out the clutch. In vehicles with an automatic transmission, on the other hand, the driver attempts to accelerate as quickly as possible.

Figure 1 shows the movement sequences of a reference point in the longitudinal direction of the vehicle over time during a forwards starting-off manoeuvre. In general, the time sequence shown in grey can be seen, in which the engine moves against the stops of the mountings in the positive x direction.

Under certain circumstances, however, the overall system undergoes an increase in vibration, something which is shown in the approval tests primarily by increased engine vibration. A movement sequence of this kind is also shown in Figure 1 in black by way of example. In this, it is easy to see that the normal movement amplitudes are significantly exceeded and this means the engine needs more space.

This effect is referred to as the "power hop" and has been defined by Halbmann and Hölscher [3] as an oscillation of the chassis, suspension and powertrain excited by the tyre. Furthermore these oscillations can be caused by slipping in clutch systems [4]. This type of self-excited drivetrain vibration also occurs by using driver assistance systems like traction control system (TCS) which are available as standard nowadays.

As well as the problems with the space requirement already referred to, this ef-

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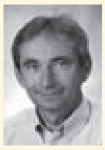

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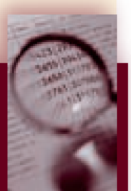

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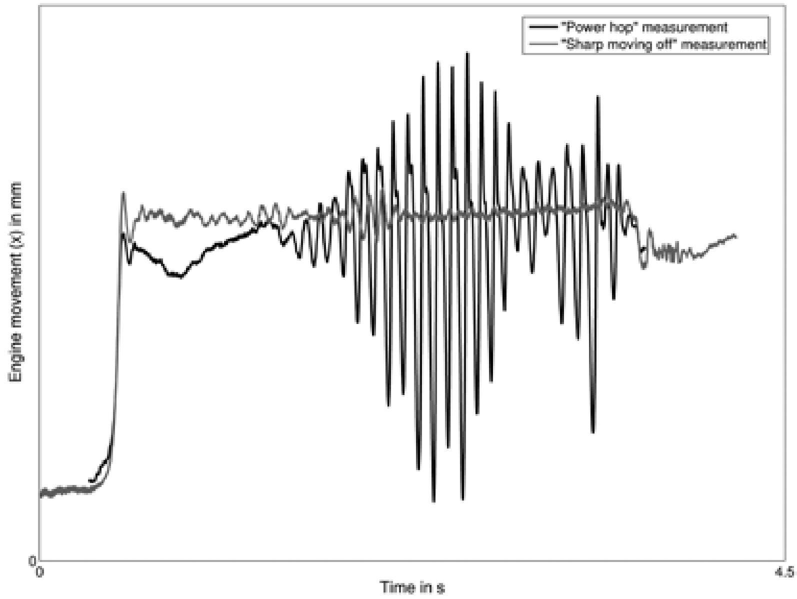

Figure 1: Comparison of measurement point movements on the engine in longitudinal direction

Table 1: Overview of measuring equipment

	Measuring technology	Anmerkungen
	Cable actuated position sensors	Measurement of engine movement
◇	Strain gauge bridges	Measurement of torques on driveshafts
○	Accelerometer	Measurement of: – Control arm acceleration – Body acceleration – Engine acceleration
✲	Via CAN bus	Measurement of: – Wheel rotation speeds – Accelerator pedal setting – Engine speed – TCS relevant data

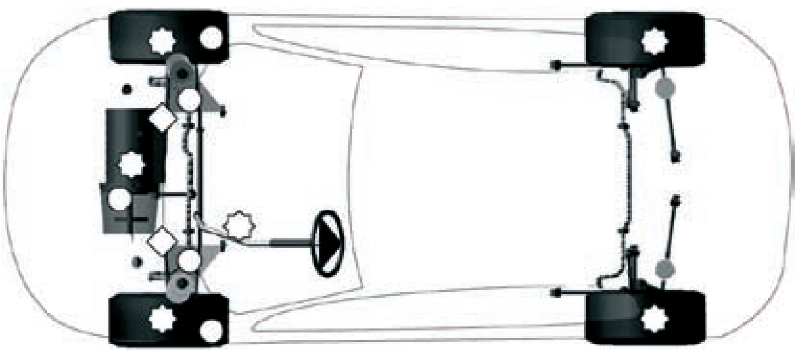

Figure 2: Block diagram of measuring equipment at the measuring vehicle

fect also has a significant influence on driving safety, driving comfort and the service life of chassis, suspension, engine mounting and body components. It is therefore on the basis of test results and observations of the Volkswagen AG, that the problem of the "power hop", also referred to as starting judder, is being investigated as part of a cooperative project involving the Institute of Automotive and Powertrain Engineering (IFAS) of the Helmut Schmidt University Hamburg especially investigating the influence of tyres. In addition to road tests the possibility of reproducing the "power-hop" within MBS-full-vehicle simulation will be analysed.

3 Measuring Vehicle

Consequently in the field of this this project, a front-wheel drive test vehicle was equipped with the standard measurement setup of approval tests for clearance studies [1]. Overall eight cable actuated position sensors were mounted on three different points to observe the engine movements. The pendulous front-transverse drive is linked to the subframe and static loads are taken by the engine and gear mountings. Furthermore additional measurement equipment, listed in **Table 1** was located at the measuring units positions shown in **Figure 2**.

The voltage is supplied to the measurement setup using a 12 V battery and a voltage transformer located in the luggage compartment. The measured data is sampled at a rate of 600 Hz and registered via a model MGCplus of HBM measuring booster and on a computer (laptop).

4 Measurement Results

A series of road tests had already been carried out using the vehicle equipped in accordance with Section 2. These tests correspond to the observations in [3] to a very large extent.

Figure 3 shows sample events in a power hop measurement. The text following presents selected, characteristic measurement results and explains the resulting loads on the powertrain, chassis, suspension and body. The typical engine movement in a power hop has already been shown in Figure 2, indicating a frequency of about 11 Hz. This frequency can also be seen in the rotational acceleration of the engine about its pitching axis, shown in Figure 3.1. This results in rotational acceleration values of up to 600 rad/s², which correspondingly lead

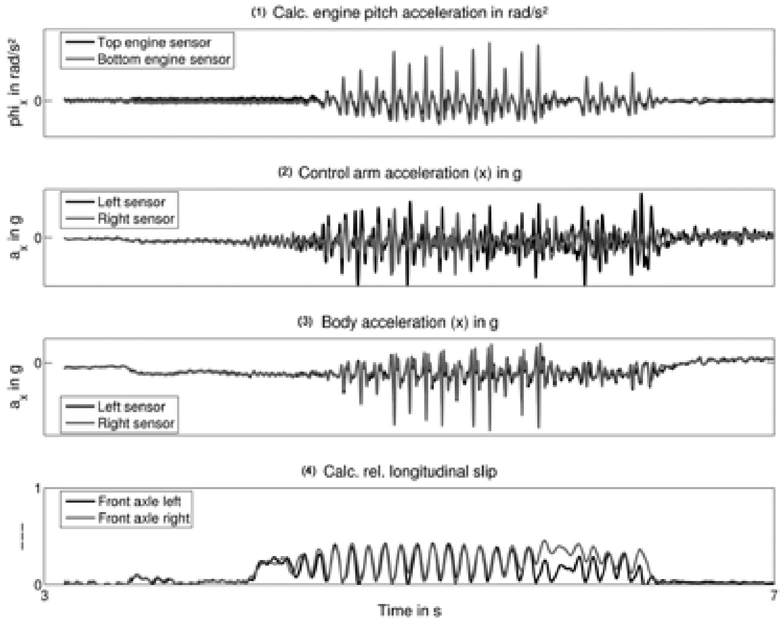

Figure 3: Sample events in a power hop measurement

to high forces in the engine mountings – above all the pendulum support.

Figure 3.2 shows the transverse link acceleration in multiples of the gravitational acceleration g in the longitudinal direction of the vehicle. This reveals amplitudes of up to 50 m/s² with a principal frequency component of about 23 Hz.

The profile of moments on the input shafts during a power hop manoeuvre also indicates a significant amplitude overshoot on both shafts when the power plant is in "resonance". In this condition, the shafts are subject to considerable dynamic load not only from the drive torque but also from the moment of inertia of the oscillating engine and the tyres. These oscillations have a principal frequency component of 11 Hz and also influence the engine's combustion process.

The low-frequency excitation at high amplitudes also results in significant body excitation that can be heard and felt. In this case too, amplitudes of up to 20 m/s² can be seen in Figure 3.3 in addition to the underlying acceleration from moving off.

Finally, when the slip values calculated from the wheel rotation speed signals carried on the CAN bus, Figure 3.4, are taken into account, it becomes apparent that starting judder is based on considerable oscillation of the slip from 0 up to 50 %. This is a sign of the stick-slip oscillations of the tyre and leads to the conclusion that what is involved is self-excited oscillation of the tyre [3]. This must be qualified by stating that the measuring equipment presented here does not afford any direct insight into the tyre behaviour, because the wheel rotation speed sensors on which the slip calculation are based are, to be quite specific, measuring the speeds at which the rims are rotating. Accordingly, the illustrated profile has a clearly apparent frequency component of about 11 Hz, because the rim is influenced by the engine oscillation to a considerable extent.

The fundamental considerations outlined in Section 2 can also be reproduced in the road test. It is revealed that, although intervention in the engine management system by the TCS does not suppress the effect, it is possible to influence the amplitudes and the duration of the increase in vibration. This influence depends on the prime mover of the system – the engine. Two system behaviours, which appeared sporadically, could be observed during the measurements (resulting in an irregular correlation in the course of the measurements).

In the following **Figure 4** the measurement results of a measurement without TCS intervention (grey) are compared to results obtained from the two-stage intervention (black) described here. On the one hand, it has been shown that "one-stage" intervention by the TCS in the engine governor makes scarcely any difference to the excitation of the engine. A "two-stage" intervention in the engine management system, on the other hand, dampens the oscillation of the engine. The term "two-stage" is used due to the intermediate step on which the invention rest before the second step of reduction is done.

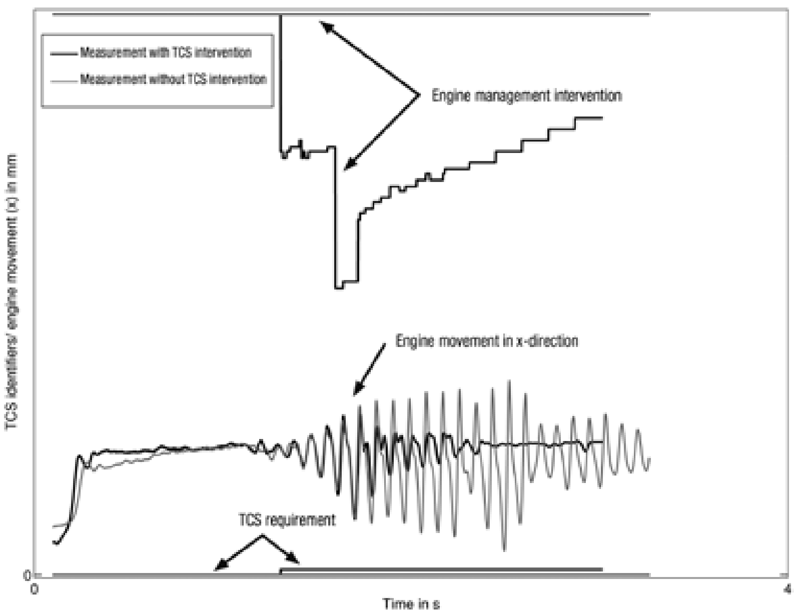

Figure 4: Comparison of measured engine oscillations without/with TCS intervention

Table 2: Calculated resonant frequencies of the equivalence model

Frequency in Hz	1.36	10.82	24.4
Assignment	Body	Engine	Wheels

Attenuation starts with immediate effect at this second reduction step of the engine torque. An initial consideration also based on this observation is that suitable processing of the sensor data produces specific closed-loop control of the engine torque build-up. However, this also means that the effect has to start first so that it can be detected reliably. As a result, there is still increased load on the powertrain at this time although the number of oscillation cycles and the magnitude of the amplitudes can be limited.

5 Interpretation of the Measurement Results

The spectral concentrations were calculated in order to determine the frequency contents. This produced the principal frequency components of about 11 Hz and about 23 Hz.

The absolute values of the frequency components vary according to the preload on the engine mountings that means depending on whether the vehicle is moving off on the flat or on a gradient, or what is the maximum longitudinal force that can be transmitted.

In order to allocate the frequencies, the model of a linearized rotational oscillator consisting of three rotational inertias and three stiffnesses was considered. The equivalent inertia of the engine, the mass moment of inertia of the wheel and the rotational equivalent inertia of the body mass were used as inertias for this purpose. The resonant frequencies and natural modes were calculated for this simple equivalence model. The calculated frequencies are assigned to the inertias as shown in **Table 2**.

As was to be expected, the equivalence system of the powertrain has a resonant frequency of about 11 Hz and the largest excursions result from the equivalent inertia of the engine. The resonant frequency to be assigned to the wheels correlates to the frequency component of about 23 Hz that is principally contained in the acceleration signals of the transverse links in the longitudinal direction of the vehicle.

Furthermore, simultaneous acquisition and recording of the measured data make it possible to observe the movement forms of the principal frequency components. As can be seen in **Figure 5**, the principal movement forms are characterised throughout by a considerable pitching movement of the engine/gearbox. At a frequency of about 11 Hz (left side), this pitching movement of the engine is accompanied by a vertical movement of the front axle whereas longitudinal oscillation of the front axle occurs at 23 Hz.

As has already been explained in Section 4 using Figure 3.4, the timing sequence of the slip indicates a stick-slip oscillation of the tyre that means this is friction-excited oscillation of the system. This assumption was supported by analysing high speed camera shots. As already described in [5], the creation of friction-excited oscillations of this kind is favoured by a large gradient in the falling branch of the μ-slip curve and by a high level of friction coefficient. Detailed fundamental considerations about this kind of friction oscillator are also contained in [6], for example.

A further mechanism of self-exciting mechanical systems was presented by [7] where this type of oscillations occurs as a matter of normal load variations even if the friction characteristic is defined as monotonically increasing. Those variations of normal force have been reported among others by [8]; in these, investigations into stick-slip procedures with a sample profile block have shown that the normal force and the longitudinal force profile behave comparably. It has been empirically demonstrated that the sample block undergoes lifting off together with longitudinal oscillation. Furthermore, it has been possible to show that the sample's tendency towards stick-slip oscillation is also strongly influenced by the elasticity and damping of the rubber blend. Therefore a longitudinal dynamic test rig is being set up at the IFAS, which enable to investigate the influence of wheel load characteristics on the longitudinal force and thus on the occurrence of self-excited oscillations. In doing so the complete drive train as well as the front suspension of the test vehicle is being mounted on this test rig.

6 Simulation

The MBS full-vehicle simulation is used for further consideration of the phenomenon and as the basis for investigating and evaluating remedial measures. At the beginning, however, it is necessary to ensure that the models can reflect the effect of friction-excited oscillation.

MSC.Adams is used as standard at Volkswagen AG as part of the packaging examination, whereas the FTire tyre model is used for simulating the tyre behaviour [9]. In order to simulate the available test vehicle in detail, not only the technical parameters of the full vehicle but also all significant individual components such as body springs and dampers as well as the chassis, suspension and en-

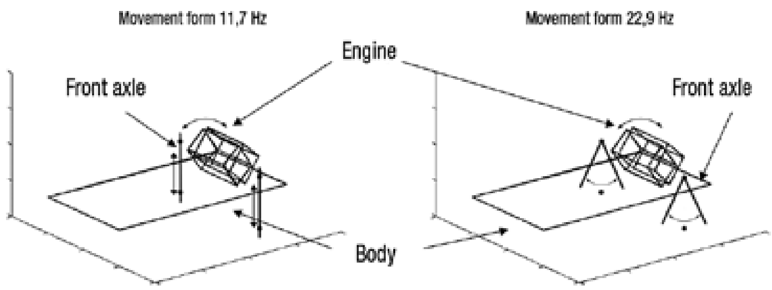

Figure 5: Movement forms of power-hop oscillations

gine/gearbox mountings are measured. The most important component measured in this case was the standard tyres on the test vehicle with dimension 225/40 R18 92 Y. This measurement was performed on the dynamic tyre test facility of the IFAS in accordance with the specifications for parameter settings in the FTire tyre model, and the corresponding FTire data sets were identified.

The virtual investigation into the ability to simulate the stick-slip phenomenon was based on the test programmes in [3]. **Figure 6** shows a comparison of measurement to simulation results for the stick-slip behaviour. At a test speed of 10 km/h, a tyre on a drum test rig was subjected to a torque jump that produced high slip values with regard to the boundary conditions. In this case, it was possible to reproduce the self-excited oscillations (see Figure 6, upper limit) due to the falling friction characteristic – even though the boundary conditions do exert a significant influence on the occurrence of the oscillation phenomenon.

The simulations were run on the virtual tyre test facility, a model of the tyre test facility at the IFAS, within MSC. Adams/Car. A drive torque jump was also applied in this case. However, this must be qualified by stating that in contrast to the test rig conditions in [3], no additional stiffness in the form of a drive shaft was simulated in this virtual test rig.

It was possible to reproduce the occurrence of the stick-slip oscillations in the simulation, as can be seen by way of example in Figure 6 (lower limit). Similarly to the experimental observations in [3], there was a high level of correlation between speed, wheel load, magnitude and rise period of the torque jump as well as the scaling function of the transmissible longitudinal forces in accordance with the carriageway modelling of the "Road Data File" format. This is described more in detail in the documentation of MSC.Adams/Tire [10].

Furthermore, the simulation also displayed an influence from the discretisation of the tyre treads and contact element distribution of the tyre model. So the investigation into the ability to simulate friction-excited oscillations by calculations with the tyre test rig model has shown that the tyre model used is basically suitable.

7 Summary and Outlook

At Volkswagen AG, both virtual and real road tests are performed for ascertaining the space requirement and to guarantee clearance. In real road tests, one of the driving manoeuvres such as moving off sharply forwards can result in a self-excited oscillation of the chassis, suspension and powertrain. This effect, referred to as the "power hop", not only increases the space required but also impairs driving safety, driving comfort and durability of a vehicle.

It has already been possible to reproduce the observations described in [3] in the same or a similar form in a series production car of the Volkswagen Group. One significant difference worked out in this paper, however, is represented by the availability of driver assistance systems such as a traction control system. In their current form, these are not yet able to prevent this effect in a controlled way. However, it became apparent during the measurement study that targeted intervention by the traction control system algorithm in the engine management system makes it possible to dampen the excitation of the powertrain. As a result, the load on the chassis, suspension, powertrain and body is reduced. This possibility of taking targeted influence of the drivetrain oscillations had been shown in [11].

The continuation of the work involves detailed considerations of the origin of the phenomenon, firstly on a longitudinal dynamic test rig that is being set up and secondly with the help of the MBS full-vehicle simulation to examine the phenomena more detailed.

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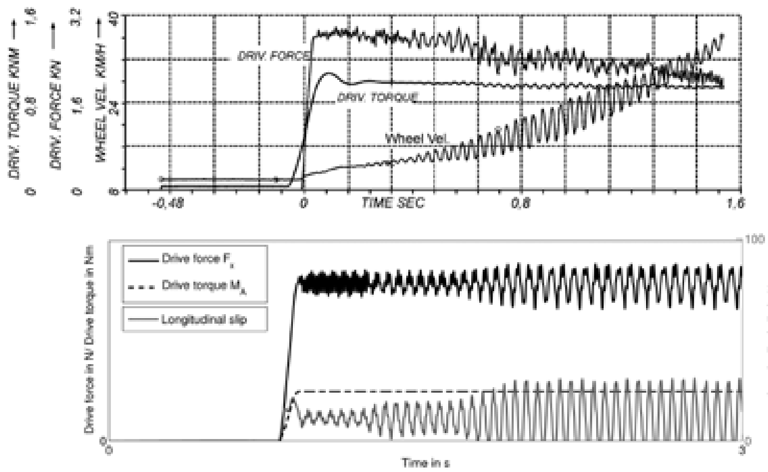

Figure 6: Comparison of measurement (according to [3], upper limit) to simulation (lower limit) results for the stick-slip behaviour