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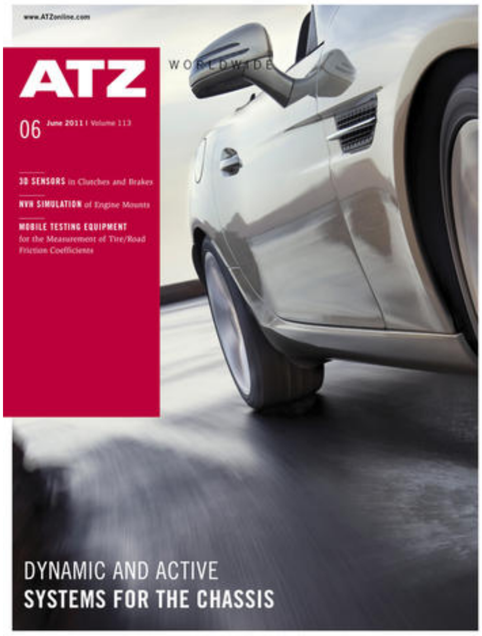

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06 June 2011 I Volume 113

3D SENSORS in Clutches and Brakes

NVH SIMULATION of Engine Mounts

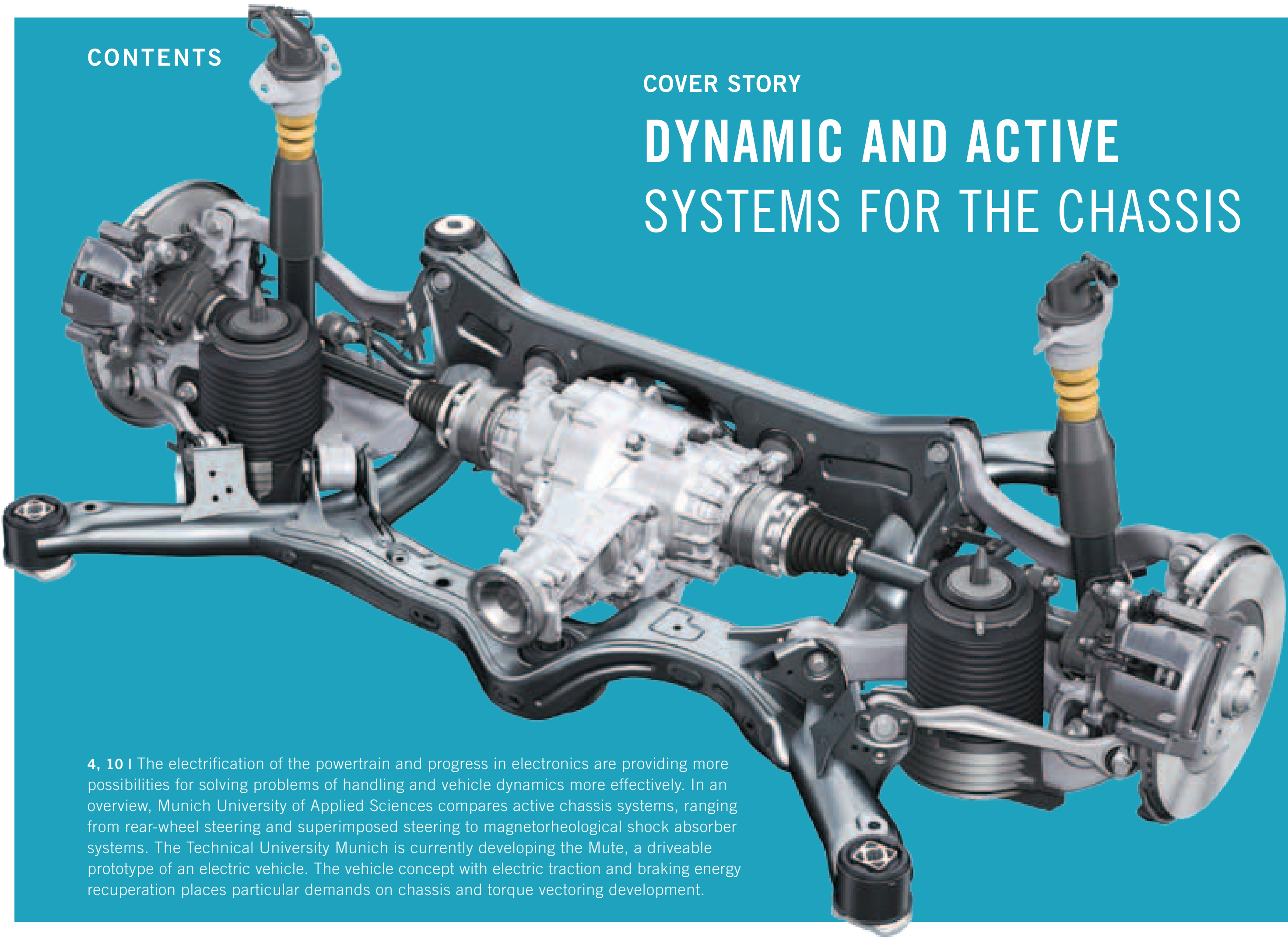
MOBILE TESTING EQUIPMENT for the Measurement of Tire/Road Friction Coefficients

DYNAMIC AND ACTIVE **SYSTEMS FOR THE CHASSIS**

CONTENTS

COVER STORY

DYNAMIC AND ACTIVE SYSTEMS FOR THE CHASSIS



4, 10 I The electrification of the powertrain and progress in electronics are providing more possibilities for solving problems of handling and vehicle dynamics more effectively. In an overview, Munich University of Applied Sciences compares active chassis systems, ranging from rear-wheel steering and superimposed steering to magnetorheological shock absorber systems. The Technical University Munich is currently developing the Mute, a driveable prototype of an electric vehicle. The vehicle concept with electric traction and braking energy recuperation places particular demands on chassis and torque vectoring development.

COVER FIGURE Daimler **FIGURE ABOVE** Audi

BACHELORS

Dear Reader,

Germany has brought on its troubles itself: in the fall of 2011, many large Federal states such as Bavaria, Baden-Württemberg, Lower Saxony and Hessen will be bidding farewell to pupils from two years. This is due to the turbo high school qualification, the so-called G8. The discontinuation of national service and community service is additionally exacerbating the run on technical colleges and universities. According to WeltOnline, 46 percent of one year's intake now go on to higher education; in 2005, this number was just 37 percent. Adolescents are heeding the call to prepare themselves well for vocational life – and politicians now appear to be completely surprised. How can this onslaught be withstood?

The government is pleading to reduce the length of time required for studying and instead to be content with a Bachelor degree. However, 6000 often extremely specialized Bachelor degree courses are spoiling these recently qualified pupils for choice. Each university conducts its own entry level tests. Harmonization would be a good thing in this regard. Also to prevent freshmen from being forced to simultaneously apply to a number of universities in parallel.

This and much more besides is being discussed today by universities, student associations and politicians at the second national Bologna conference in Berlin. "We are all agreed that the Bologna process offers students and universities major opportunities", emphasized Germany's Federal Minister for Education Annette Schavan. "Over the next five years, we will establish 335,000 new places at university". And what about those students who wish to matriculate this fall? The Federal Minister for Education claimed that graduates with Bachelor degrees would have good vocational prospects. This was revealed by new studies. If you take a look at our careers4engineers job market in the German print edition, however, it can be seen that this brave new world has not yet materialized in industry. None of the companies listed there utters the word "Bachelor". They are therefore demonstrating that there is a world of difference between political wishful thinking and industrial reality. This form of degree does not appear to be so popular after all in Germany. It only remains to be hoped that the Bachelor does not lead to eternal "singles".

Yours,

Michael Reichenbach

DIPL.-ING. MICHAEL REICHENBACH,
Vice Editor-in-Chief
Wiesbaden, 6 May 2011

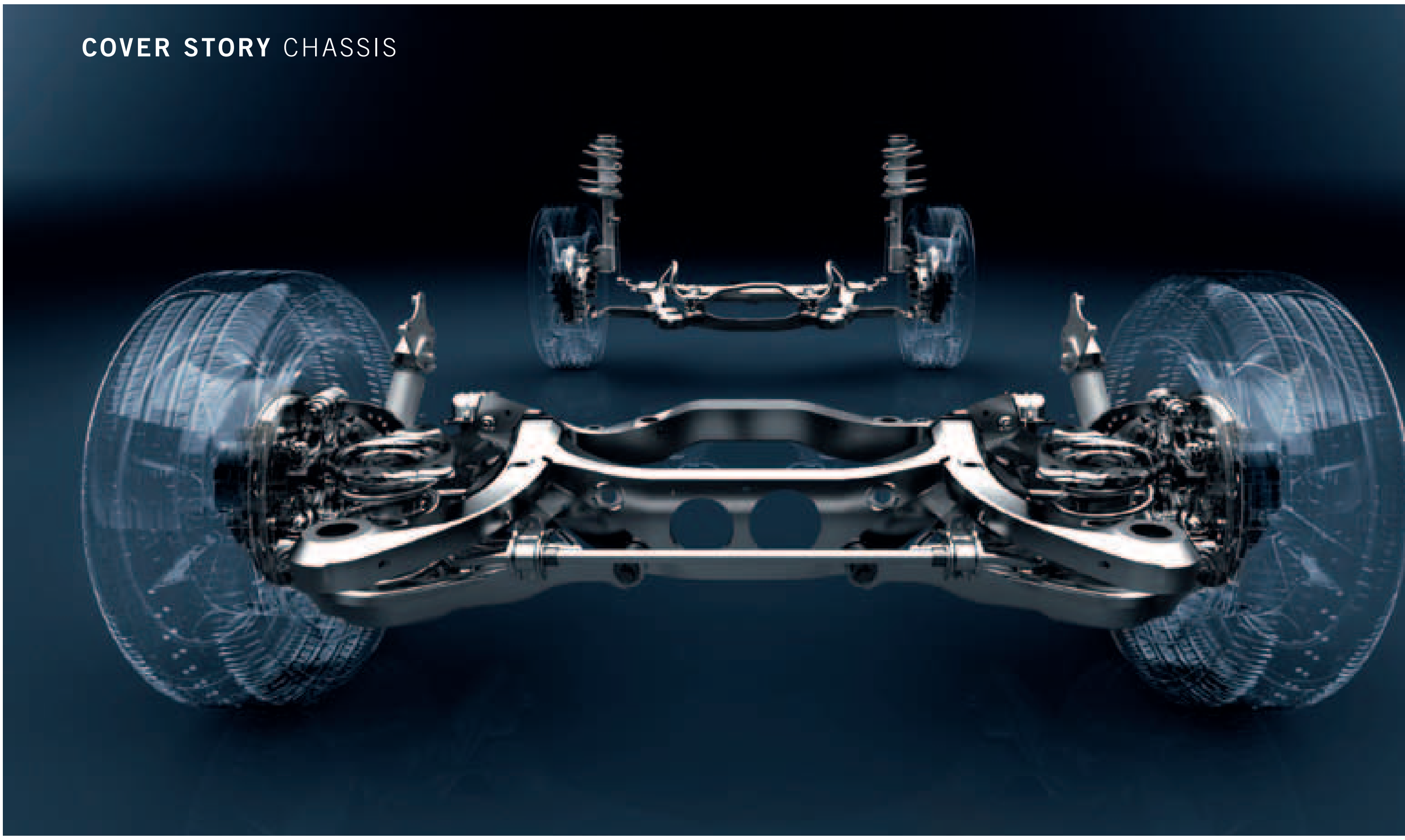


FIGURE © Lexus

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ELECTRONICS AS A DRIVING FORCE

At a very early stage in the 125-year history of the automobile, efforts were made to positively influence vehicle handling by using passively reacting systems. For example, straight-line driving was optimised by the castor angle, kingpin angle and toe-in. Braking stability was decisively improved by a combination of a diagonal separation of the brake circuit (dual-circuit braking) and negative kingpin offset.

Only with the availability of low-cost electronic systems did the widespread application of active chassis systems in the vehicle begin. Outstanding examples of this development are Anti-locking Brake Systems (ABS) or active systems that improve vehicle stability, such as Electronic Stability Controls (ESC). The following report presents and describes current developments in the field of active chassis systems.

The term active chassis system is used to describe all chassis control systems in the vehicle. According to their function, these can be classified into "vehicle dynamics control systems", "vehicle comfort control systems" and "driver assistance systems". Vehicle dynamics control systems are used to enhance vehicle stability and/or vehicle agility. Vehicle comfort control systems are ones that increase ergonomics or improve operating convenience and ride comfort [1]. Driver assistance systems that support the driver in driving the vehicle will not be considered here. The operating principle of active chassis systems can itself be subdivided into the areas of longitudinal, lateral and vertical dynamics. Most active chassis systems operate across all of these areas. ❶ shows some of the currently available active chassis systems and their influence on tyre forces; the motion of the vehicle is derived from these. It is clear from ① that the different active chassis systems have the same control path. If the systems were installed independently of one another, the individual control paths might influence each other and, in the worst case, endanger vehicle stability and safety.

SYSTEM NETWORKING – INTEGRATED VEHICLE DYNAMICS CONTROL

The aim of system networking, in other words the system-overriding coordination of the individual active chassis systems, is to achieve the required driving characteristics in an optimum manner. Furthermore,

ACTIVE CHASSIS SYSTEMS

Innovations and new functions are the lifeblood of the automotive industry. Most of these new developments relate to active chassis systems. These systems enhance the driving experience and improve safety. The Competence Centre of Vehicle Dynamics at Munich University of Applied Sciences is working on the further development of these systems. This report presents an overview and a current assessment of the development in active chassis systems.

integrated vehicle dynamics control can provide a significant increase in functions, such as active steering intervention at the front and/or rear axle to support ESC.

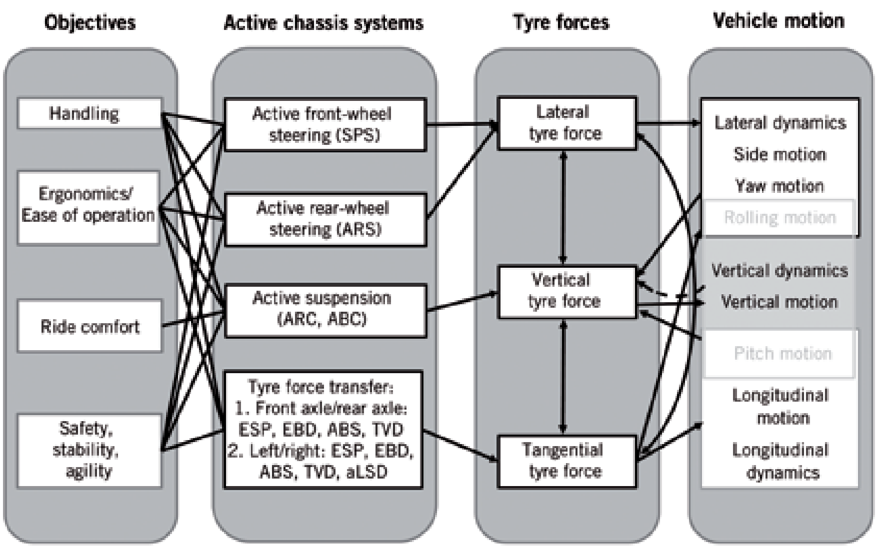
Almost all OEMs and many suppliers are working intensively on such system networking. On the one hand, system architecture approaches aim at achieving a state of peaceful coexistence. This ensures that the systems do not work against each other. On the other hand, integrated vehicle dynamics control systems use a central microcontroller that influences all active chassis control systems in order to achieve the target state in an optimum manner. This principle has already been implemented in premium vehicles [3].

A compromise between the two sides is an approach known as moding. In this concept, a microcontroller specifies the algorithms or parameters coming from other microcontrollers that are to be implemented [4]. Generally, the trend is towards increased system networking. All these networking approaches are aimed at achieving optimised vehicle handling and increasing the scope of available functions by intelligent integration of the active chassis systems. The integration of systems enables new functions to be provided as software solutions using available hardware.

LATERAL DYNAMICS – CONTROLLING YAW MOMENT

The primary task of active chassis systems that are relevant to lateral dynamics is to maintain vehicle stability and to increase the lateral dynamics potential. This can be achieved if yaw motion remains within the specified limits. To achieve this aim, measures can be applied that are summarised under the term "yaw moment control". The active chassis systems listed in ❷ are particularly suitable for yaw moment control [5] and are already in series production.

These active chassis systems differ very strongly in their potential for influencing



❶ Objectives and control paths of active chassis systems (according to [2])

STEERING	SUSPENSION	BRAKES	DRIVETRAIN
Superimposed Steering (SPS) •	Active Anti-Roll Bar (ARC) •	Electronic Stability Program (ESP) •••	Active Limited Slip Differential (aLSD) •
Active Rear Steering, (ARS) ••	Full Active Suspension (ABC) •	Torque Vectoring by Braking (TVbB) ••	Torque Vectoring Differential (TVD) •
	Active Geometry Control Suspension (AGCS) •		

❷ Active chassis systems for yaw moment control – potentials for influencing yaw moment, with points for level of effectiveness (••• high, •• medium, • low)

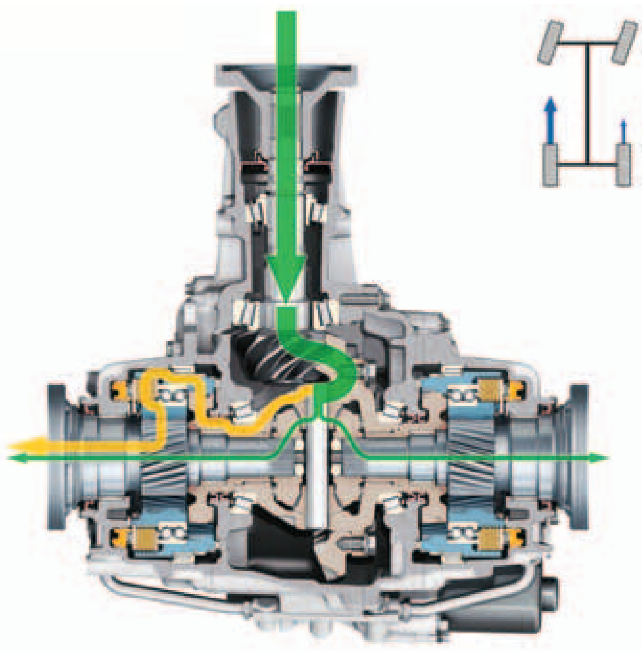
yaw moment. ② quantifies this potential for effectiveness of the different systems on a points scale. Yaw moment can be influenced most effectively by ESC, followed by active steering at the front and rear axle.

As the most effective system for influencing yaw moment, the Electronic Stability Program (ESP) has been proven to reduce the frequency and severity of accidents. In the meantime, approximately 80 % of new vehicles registered in Germany are fitted with ESC [6]. By 2012, all new cars in the USA and, by 2014, all new cars in the EU must be equipped with ESC. To an increasing extent, ESC or a central control unit are taking over the coordination of all yaw moment-relevant systems.

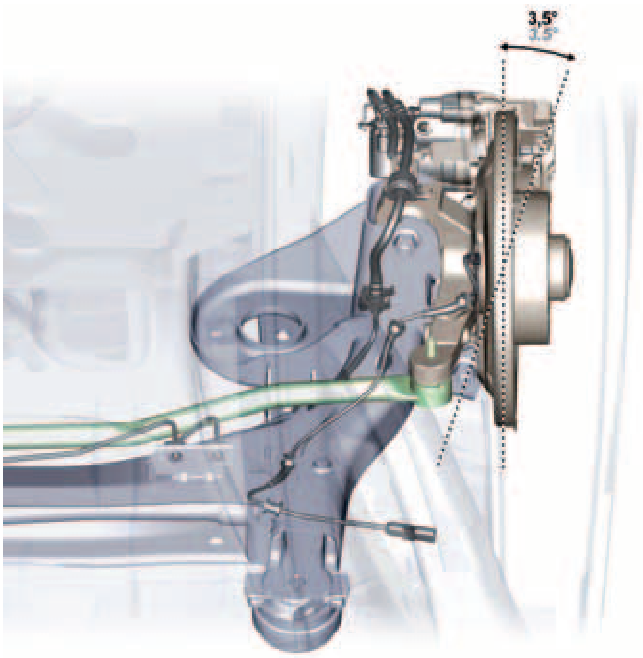
The following additional functions are currently covered by ESC or a central control unit. Emergency Brake Assist (EBA) recognises an emergency braking system by the rapid movement of the driver's foot from the accelerator pedal to the brake pedal. The brake system is primed to reduce its response time and, if necessary, the brake pressure is increased during braking. Roll Stability Control (RSC) prevents a vehicle rollover by targeted braking of individual wheels.

To assist the driver of vehicles fitted with electromechanical power steering, the "driver steering recommendation" function for µ-split braking controls a steering impulse that indicates the direction of the necessary steering angle correction to the driver. In vehicles with superimposed steering, a steering angle is directly superimposed on the steering angle applied by the driver during µ-split braking or in the case of oversteering or understeering in order to maintain the desired driving line [7]. Stabilisation is more "gentle" than with ESC intervention and therefore results in less loss of speed.

If the Hold Assist function is used, after the car has been braked to a halt, braking pressure is built up and the vehicle is held in position until it moves away again. If the available braking pressure is not sufficient to keep the vehicle stationary, the electromechanical parking brake is additionally applied. Trailer Stability Assist (TSA) stabilises a vehicle-trailer combination by targeted braking of individual wheels of the towing vehicle. Other possible functions are automatic application of the brakes to dry the disks in wet weather, fading compensation, "soft stop" and engine drag moment control.



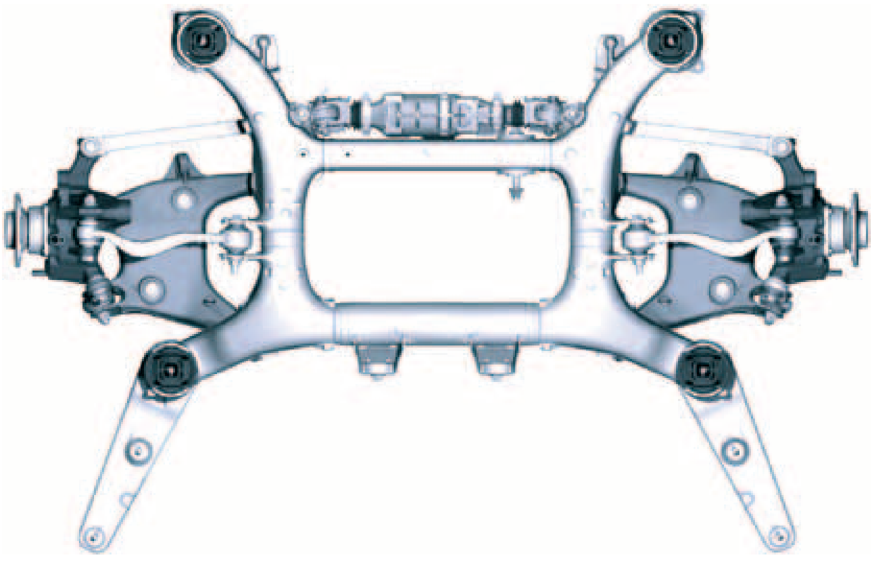
❸ Driving torque flow at the rear axle differential with torque vectoring function (Figure © Audi)



❹ All-wheel steering in the Renault Laguna (Figure © Renault)

An important system is Torque Vectoring in its various versions. In the Torque Vectoring by Brake (TVbB) function, which involves wheel-selective torque control, the inner wheel of the driven axle is braked when the vehicle is cornering (electronic differential lock). In four-wheel drive vehicles, the longitudinal distribution of the driving torque is also influenced. This achieves more neutral self-steering characteristics and improves the agility of the vehicle.

The Torque Vectoring Differential (TVD) works in a different way. The torque distribution between the front and rear axle and between the wheels is actively influenced by an overriding drive at the differential, ❸. This improves the vehicle's agility. Such TVD systems are currently used by Honda and in the BMW X6 as well as by Audi in several model series.



❺ Integral V rear axle with actuator for rear-wheel steering in the BMW 7 Series (Figure © BMW)

ELECTRIC POWER STEERING AND SUPERIMPOSED STEERING

Due to the energy savings required by CO_2 legislation, hydraulic power steering systems are being successively replaced by Electric Power Steering (EPS) systems. In addition, these systems offer extended functionality, such as active return-to-centre or speed-dependent steering wheel torque [8]. The current focus of development is on improving efficiency to enable vehicles with high front axle loads and therefore with high rack forces to use EPS. In order to compensate for the reduced steering feedback of EPS systems, new control concepts are being developed [9].

In Superimposed Steering Systems (SPS), the steering angle (front wheel turning angle) set by the driver is superimposed by an additional steering angle. At lower driving speeds, the steering angle is increased in order to generate more direct steering characteristics and therefore greater agility. Conversely, the steering angle is reduced at higher speeds in order to improve stability. For cost reasons, the application of this system is currently restricted to optional equipment for premium vehicles of the Audi, BMW, Nissan and Toyota brands.

Steer-by-wire systems have still not been implemented in series production because the functions have largely been covered by superimposed steering. The additional functional benefits do not justify currently the extra costs that result from the necessary redundancy.

ACTIVE REAR STEERING, ACTIVE GEOMETRY CONTROL SUSPENSION AND ACTIVE SUSPENSION

Active Rear Steering (ARS) pursues a similar objective to superimposed steering. At lower driving speeds, the wheels on the rear axle are steered in the opposite direction to those at the front; at higher speeds, the wheels are steered in the same direction. In the systems currently implemented, the maximum steering angle at the rear wheels is only a few degrees, thus achieving a moderate reduction in the turning circle. Active rear steering is currently used by BMW, Nissan and Renault, ❹. In its 7 Series, BMW uses an integral V rear axle with an actuator in order to implement rear-wheel steering, ❺. As active rear steering offers advantages in vehicle stability compared to superimposed steering and allows a faster build-up of lateral acceleration, an increased use of the system is expected in the future.

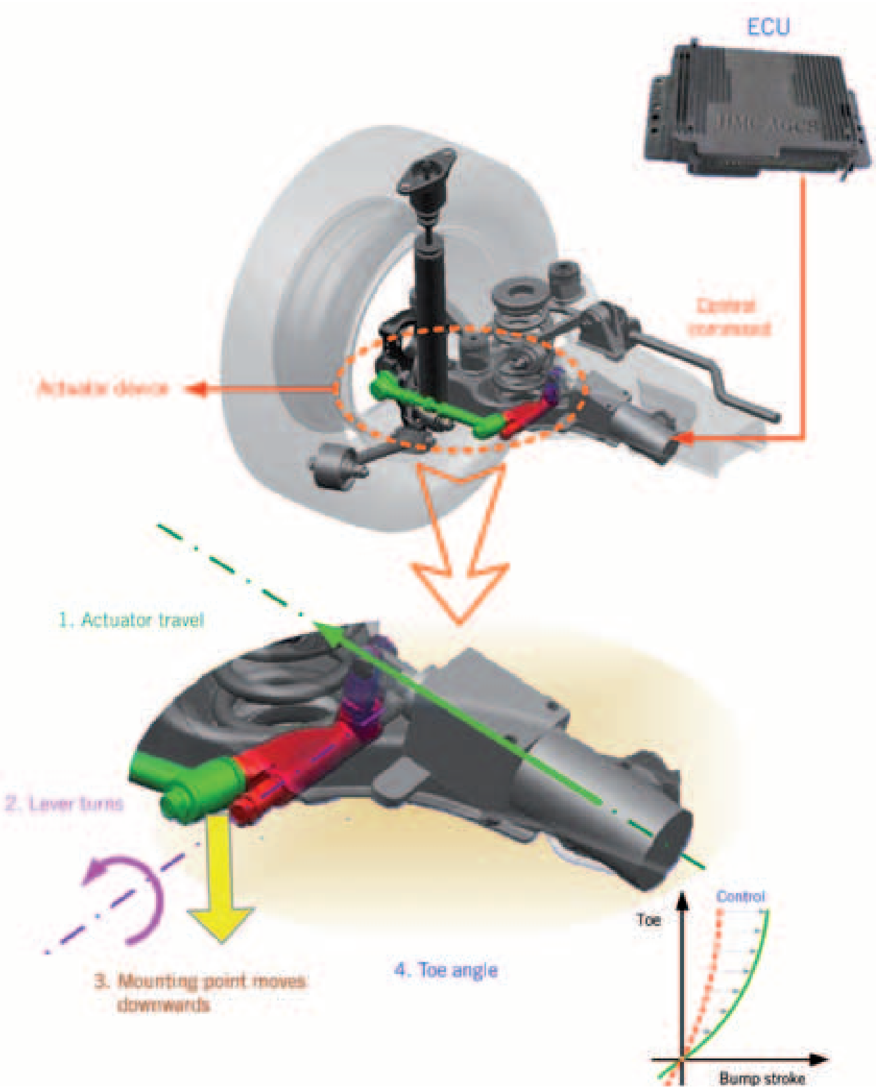
Active rear axle kinematics can be used to actively influence the toe and camber characteristics of the rear wheel and therefore the vehicle's self-steering properties. Such a system, ❻, is implemented by Hyundai at the rear axle of its Sonata model, and is called Active Geometry Control Suspension (AGCS) [12]. In this system, a change in kinematics is achieved by varying the inboard mounting position of the suspension link in the vertical direction.

Active suspension makes use of the effect that, when the load on a wheel changes, a change in lateral force occurs due to the change in toe angle and camber angle. If the wheel load is varied by an active suspension or by active anti-roll bars, a speed-dependent yaw moment can be generated. Depending on the driving speed, this yaw moment can replace a steering wheel angle of approximately 10° [10]. Active suspension is implemented in series production by Mercedes-Benz as the Crosswind Stabilisation function of its Active Body Control (ABC) system [11].

LONGITUDINAL DYNAMICS AND BRAKING

Currently, most passenger cars are equipped with hydraulic brakes, a vacuum brake booster, an ABS and Electronic Brake Force Distribution (EBD). The electrification of the drive system is now resulting in new requirements. Due to the possibility of braking energy recuperation, the division of the braking force between the friction brake and drivetrain braking must be designed to be variable (blending).

What is more, new solutions must be found for brake boosting, as the generation of a vacuum is energy- and cost-intensive in electrified powertrains. A possible solution is the electromechanical or



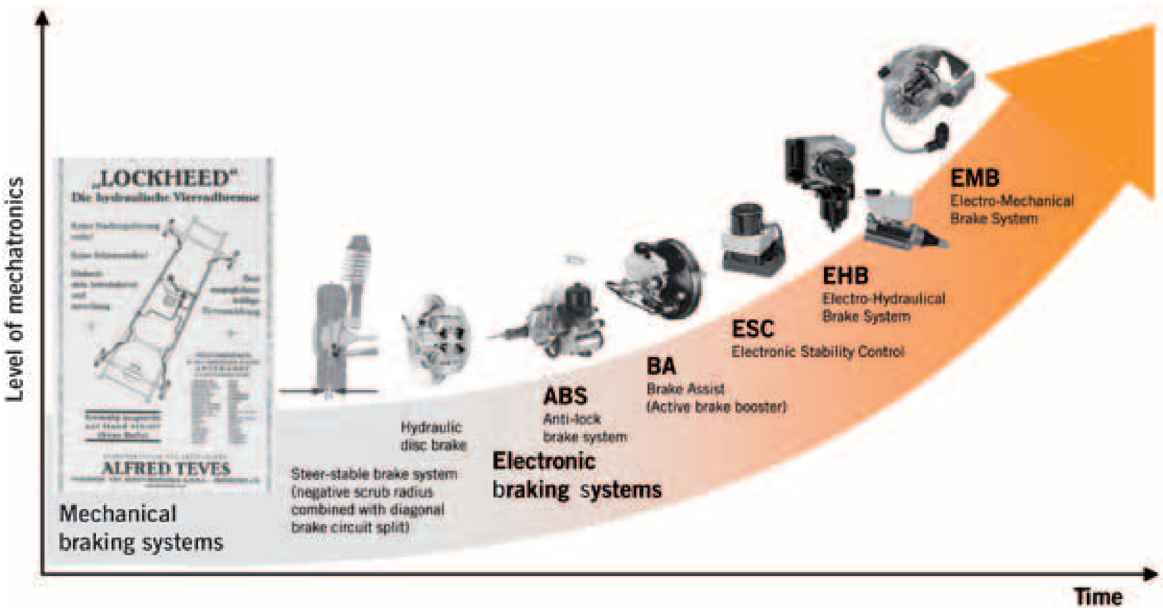
❻ Active rear axle kinematics in the Hyundai Sonata (Figure © Hyundai)

electrohydraulic brake [13, 14], ❼. In a conventional on-board electrical power system, the power required for an electromechanical brake is too high. An intermediate solution might be to equip only the rear axle with an electromechanical brake. Increasing use is being made of complex brake pedal actuators in order to ensure a consistent braking feel independent of recuperation. The electronic wedge brake that has been announced has not yet been implemented in series production.

VERTICAL DYNAMICS

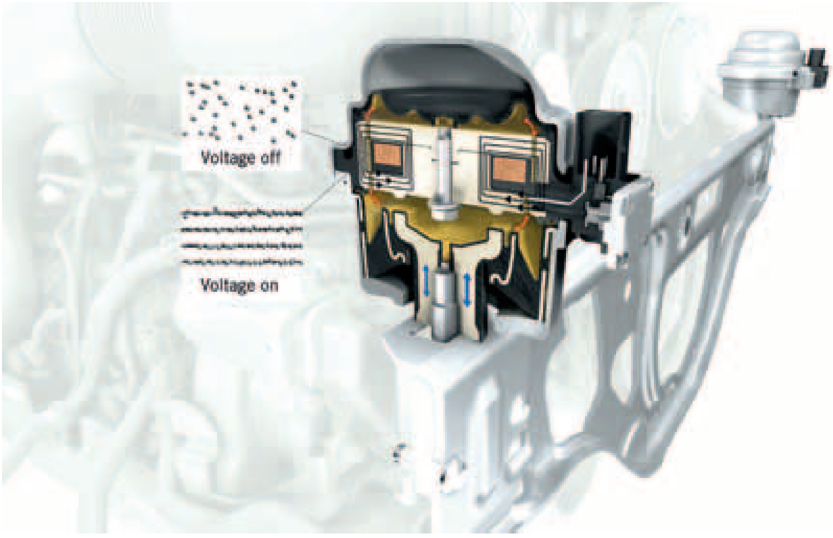
Passive chassis systems remain the standard equipment for passenger cars. From the compact vehicle segment upwards, many car makers offer semi-active shock absorbers as optional equipment. These are usually controlled according to the “sky hook” principle. In most cases, they are also superimposed by algorithms that minimise the vehicle’s roll and pitch motion. The body and wheel motions are measured by acceleration and position sensors.

Various continuously variable shock absorber systems are available on the market. These often use a controlled valve. In the BMW 5 Series and 7 Series however, the compression and rebound strokes each use a separate valve [3]. Sports cars like the Chevrolet Corvette, the Audi TT and R8 or Ferrari models use magnetorheological shock absorbers, ❽. An electromagnetic field is applied to change the viscosity of



❼ Development of passenger car brake technology over time (Figure © Continental)

❽ Magnetorheological shock absorbers in the Cadillac SRX (Figure © Delphi)



❾ Dynamic magnetorheological drivetrain mount with variable stiffness (Figure © Porsche)

the damping fluid, thus varying the damping force [15].

Fully active suspensions have not yet made a breakthrough. The most widespread systems are active roll stabilisation systems, although even these are only available in some premium vehicles as optional equipment. They were introduced for the first time in the Citroën Xantia Activa in 1995 [16]. German premium car makers use hydraulic systems [17]. The advantage of hydraulic systems is their high adjustment dynamics with high adjustment forces, which, in the common open-centre systems, is at the expense of very high energy consumption. Japanese manufacturers also use electromechanical systems, which can also be employed for fully electric driving. Mercedes-Benz has launched its Active Body Control (ABC) anti-roll system, which comes very close to a fully active chassis.

In order to optimise NVH characteristics, car makers are increasingly focusing on the mounting system for the internal combustion engine. Hydraulic engine mounts and hydraulic bearings in the suspension are now standard equipment in passenger cars. For vibration-critical engines, hydraulic engine mounts with variable stiffness are used to optimise engine idling comfort. Porsche [17] uses dynamic, magnetorheological drivetrain mounts with variable stiffness, ❾, to optimise idling comfort and vehicle dynamics. Actively controlled engine mounts are used only in a few cases.

VALIDATION AND OUTLOOK

The development of active chassis systems is much more complex than that of passive systems because several additional areas of development need to be covered and validated:

: software and hardware development and integration
: system safety (ASIL, ISO 26262, …)
: extended functionality and combinatorics
: energy management
: NVH and EMC.

In order to consider these development areas, development processes that are specifically adapted to active chassis systems are required. Furthermore, additional virtual methods, such as software-in-the-loop and hardware-in-the-loop (SiL and HiL) tests, need to be used.

Active chassis systems are increasingly being used to resolve the conflict of objectives between ride comfort and vehicle dynamics. Current efforts to reduce the CO_2 emission of vehicles will result in a sharp increase in the use of those active chassis systems in particular that have a low energy demand.

Once the optimisation of passive safety has reached its limits, the focus will turn towards active safety. The major trend and emphasis in vehicle development in the future will therefore be on partially autonomous or fully autonomous driving. Active chassis systems form the basis for this.

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VEHICLE DYNAMICS DESIGN OF THE ELECTRIC CAR MUTE

The Technical University Munich is currently developing the Mute, a prototype of a drivable and production-capable electric vehicle. It will be presented to the public at the IAA 2011 in Frankfurt. The vehicle concept with electric traction and braking energy recuperation places particular demands on the development of the suspension and torque vectoring. The following report describes the vehicle dynamics design of the Mute.

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MUTE: COOPERATIVE PROJECT OF TECHNICAL UNIVERSITY MUNICH

Currently the Technical University Munich is building a production-capable electric vehicle prototype called Mute. More than 20 chairs cooperate in this one year and a half long project. In 2011 the prototype will be presented at the IAA in Frankfurt. The vehicle concept is mainly aiming at urban areas, where usually less than 100 km per day are driven, often with only one to two persons travelling and the maximum speed rarely exceeding 120 km/h.

In order to verify the characteristics and driving dynamics as well as the energy management system of the prototype prior to completion, a test vehicle with the same driving dynamics as the final prototype is built. Modern design methods of vehicle dynamics require a suitable infrastructure. The multi-body model with the characteristics of the vehicle is built up in the MSC.Adams environment and is embedded in a co-simulation with Matlab Simulink. Therefore, the development of the torque vectoring control system is also possible with this model. The measurement data of real driving tests with the test vehicle is used to validate the vehicle simulation model and for suspension and chassis design improvements.

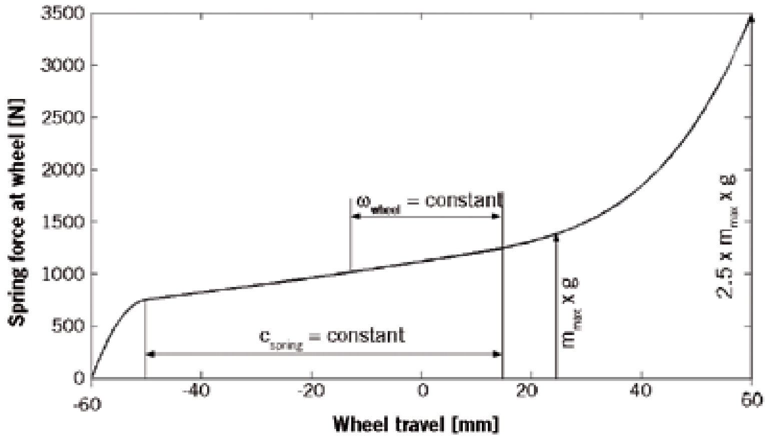
To maximize the range of electric cars the developers tend to minimize rolling and air resistance. Due to the rolling resistance reduction by very narrow tires (115/70R16) the priority of the lateral dynamics has to be increased in order to use the whole potential of the tires during cornering. A compromise between driving comfort and minimization of body movement has to be found during the definition process of the suspension springs. The comfort plays a minor role for the development of the damper characteristics. In this case the minimization of the dynamic fluctuations in the vertical forces and the utilization of the lateral dynamic potential of the tire over a wide frequency band are important.

The design of the torque vectoring control is based on the layout of the passive suspension. Among other things torque vectoring is used to adapt the vehicle behaviour to the behaviour with ideal brake force distribution whilst braking only the rear axle. Thereby, the level of recuperation in curves can be raised significantly, providing an increase in total energy feedback to the same extent. An active differential allows the almost loss-free distribution of great torques with low actuator power [1].

SUSPENSION SPRING DESIGN

Due to space limitations the suspension in the Mute prototype is realized with four McPherson struts on front and rear axle. Although the cornering potential of the narrow tires is very low, the ride comfort has to be prioritized over safety in the definition of the body natural frequency for road vehicles [2]. The spring stiffness of the suspension springs at front and rear axle is designed according to the defined natural frequency of 1.33 Hz to 1.5 Hz [3]. To avoid subjectively disturbing pitch oscillations of the chassis due to stochastic road excitations, the body natural frequency at the rear axle is selected to be about 12 % higher than in the front [4], so that the body is mainly following lifting movements. With a body natural frequency of more than 2 Hz the dynamic fluctuations of the vertical forces would be lower, but the ride comfort would decrease significantly as well [4]. With the natural frequency of 1 Hz the chassis movement, its reaction forces and, consequently, the fluctuations of the wheel loads are higher. Moreover, the steady-static ride height and thus the overall height of the centre of gravity have to be increased for a vehicle with softer springs in order to achieve the same dynamic ride height.

The ratio between the spring force and the resulting contact patch load is chosen slightly progressive. This way, the body natural frequency remains constant over a wide range of the wheel travel while the original spring characteristic can be linear, ❶. Therefore, the comfort is independent of the driver's weight. With additional load the spring force in the tire contact patch increases progressively due to the bump stop. As a consequence, the natural frequency of the chassis increases. The result is a poorer ride comfort. The bump stops are needed to provide enough suspension travel at maximum vehicle payload.



❶ Front suspension spring characteristics of the Mute prototype (ω_{wheel}: chassis eigenfrequency, c_{spring}: spring stiffness, m_{max}: maximum body mass)

DAMPER DEVELOPMENT

The dampers on front and rear axle are not linearly independent. Therefore, a change of wheel load on one axle is affected by the shocks of both axles. Thus, the vehicle needs to be handled holistically and not divided into two separate vertical vibration systems for the damper design process.

The damper development is based on a series of simulations with the multi-body vehicle model with various damper levels in bump and rebound on the front and rear axle. To keep the average ride height (dynamic ride height of the vehicle) constant while driving, the ratio between bump and rebound cannot be changed arbitrarily. A vehicle with a very hard bump and a softer rebound would rebound increasingly with each obstacle.

For the simulation series the vehicle is excited with a sinusoidal oscillation in the frequency band of 0 to 25 Hz in the four wheel contact patches. Thus, the vehicle chassis performs a synchronous vertical movement. The maximum vertical velocity (at zero crossing) is kept constant for every frequency. It is set to ± 50 mm/s for the definition of the low speed damper curve and to ± 150 mm/s for high speeds. The definition of the speed at zero crossing results in a decreasing amplitude with increasing frequency.

As described above, in the design process of the damping characteristics the driving safety and the lateral dynamics have a higher priority than the comfort. With regard to this a quality criterion is defined containing the standard deviation of the vertical forces on front and rear axle as well as the standard deviation of body acceleration, ❷. The priorities chosen in this work are shown numerically in Eq. 1:

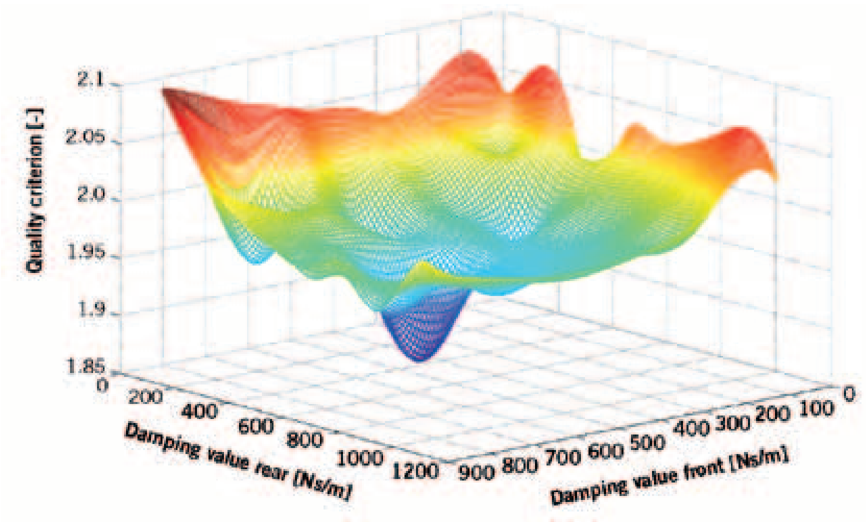
EQ. 1	$J = \sigma_{FR,VA} \cdot 0.4 + \sigma_{FR,VA} \cdot 0.4 + \sigma_{a\_chassis} \cdot 0.2$

Several multi-body simulation series have shown that the values in ② are not changed as long as the difference of the damping capacity of bump and rebound remains the same, illustrated in ❸ as areas A_{Z_LS} and A_{D_LS} for low and A_{Z_HS} and A_{D_HS} for high shock speeds. Thus, a mean attenuation on front and rear axle (slope k in ③) can be determined before the ratio of bump and rebound.

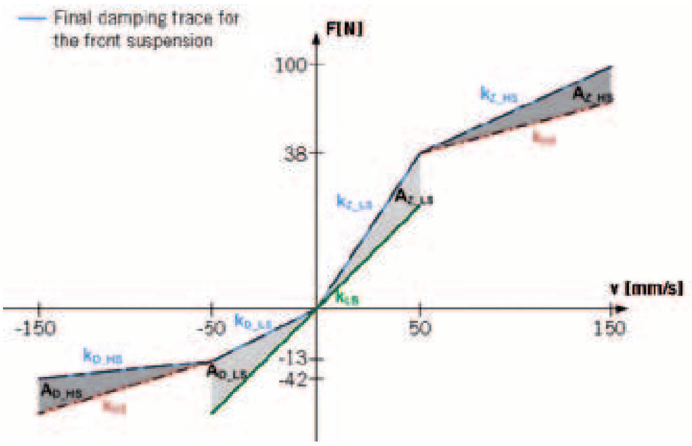
The ratio between bump and rebound, and thus between the slopes of k_{Z_LS} (damping constant of the rebound in the low speed range) and k_{D_LS} (compression) in ③ is according to [3] set to 5. It needs to be ensured for the final design that the areas A_{Z_LS} and A_{D_LS} are equal. Therefore, the total damping capacity of the system remains constant and the dynamic ride height converges to a constant final value while driving over bumps.

For the development of the damper characteristics on the front axle in the lower speed range the relationships described result in the following procedure:

: definition of the quality criterion
: determination of the global minimum of the 3D curve of the quality criterion through a series of simulations with different combinations of the damping values at the front and rear axle



❷ Quality criterion as a function of damping specifications on front and rear axle



❸ Strategy for the damper development in the Mute project

- definition of the relationship between bump and rebound force
- determination of the slope of the damping constant as a function of speed to match the areas A_{Z_LS} and A_{D_LS} in ③.

The damping characteristics in the higher speed range and on the rear axle can be determined similar to the procedure described above.

MOTIVATION AND POSSIBILITIES FOR AN INCREASE IN RECUPERATION CAPABILITY

The drag-recuperation in the Mute, which is controlled through the accelerator pedal, has several limits. This work focuses on the increase of the limits in vehicle-dynamics when electrically braking the rear axle. As explained below, drag-recuperation in straightforward drive cannot be higher than in curves. Therefore, an increase of the recuperation possible in curves directly influences the total energy-feedback.

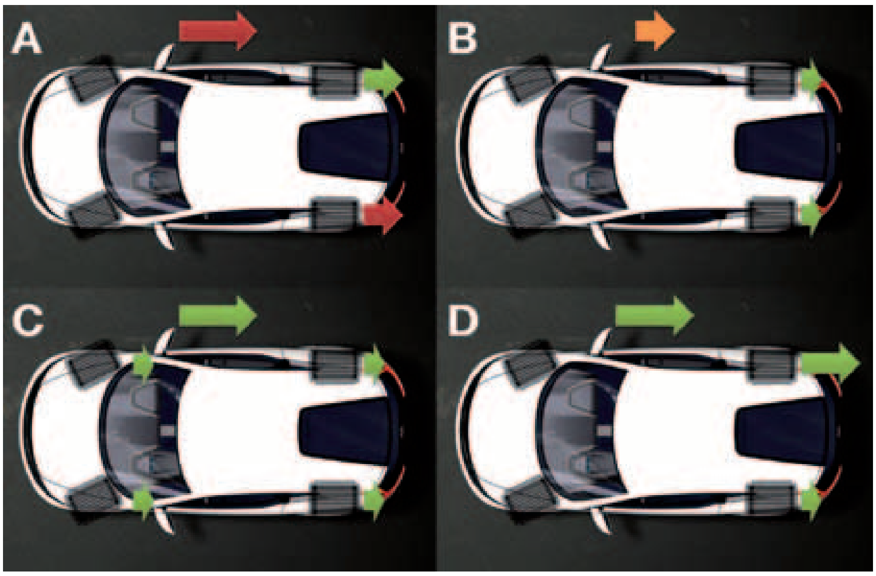
❹ shows the challenge and possible solutions for rear axle only braking. As shown in ④ (A), in a curve the inner wheel can loose traction. This can lead to undesired yaw reactions or even to loss of control. The conventional approach to limit deceleration when the vehicle becomes instable, ④ (B), would lead to an unanticipated deceleration depending on the driving situation. Therefore, the drag-recuperation should be (speed-dependent) deterministic. This could be achieved with a shift of braking torque to the front axle shown in ④ (C). However, the possibility to build up hydraulic pressure on the front axle through the ESP unit is discarded. The noticeable pressure build up for recuperation in combination with high lateral acceleration would suggest an unsafe situation to the driver. The torque shift to the outer wheel shown in ④ (D) makes use of two effects that add up:

- The cornering stiffness of the rear axle is increased as longitudinal forces are distributed in line with the contact patch load.
- The resulting yaw moment M_z acts against the turn-in of the vehicle.

POTENTIAL ASSESSMENT

After the exclusion of B and C described above, the potential of torque vectoring for the increase of recuperation capability is estimated. The aim is to master all normal driving situations only through the use of torque vectoring. The driver behaviour explained in [5] and [6] is considered normal with regard to the combination of lateral and longitudinal acceleration. The relation is approximated by a hyperbolic function. Thus, in high lateral acceleration only little deceleration is applied. The shift of brake torque to the front axle mentioned above and in ④ (C) can be used to handle extreme situations or very low friction coefficients before the conventional ESP is used.

The three cases (ideal brake force distribution, uncontrolled and controlled braking of the rear axle) are evaluated according to ISO 7975 braking in a turn (r = 100 m,



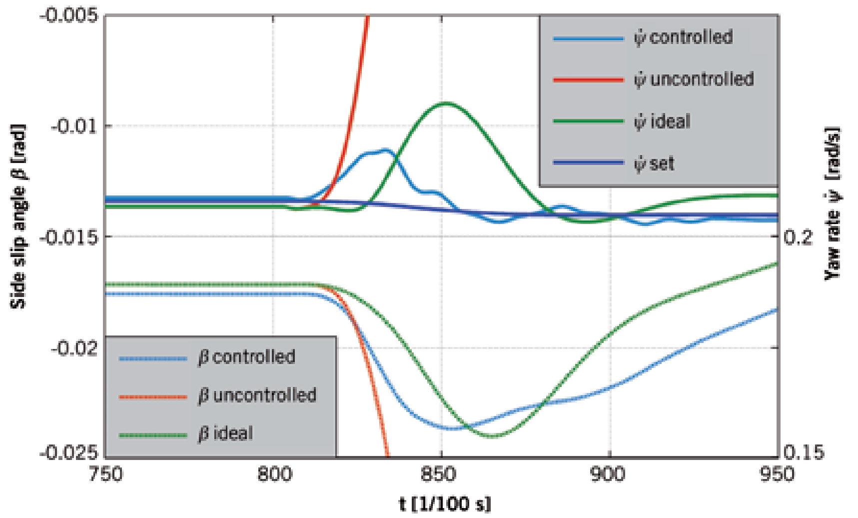
❹ Braking of the rear-axle: uncontrolled with loss of traction of the inner wheel (A), reduction of braking (B), shift of braking torque to the front axle (C), shift of braking torque to the outer wheel (D)

$v = 72$ km/h $a_y = 4$ m/s²) with decreasing friction coefficients ($1.0 < \mu < 0.5$). Deceleration was increased within 1 m/s² < $a_{x\text{-set}}$ < 5 m/s². It should be noted, that according to [5] normal drivers do not reach this lateral acceleration and only under 3 m/s² combine it with significant deceleration.

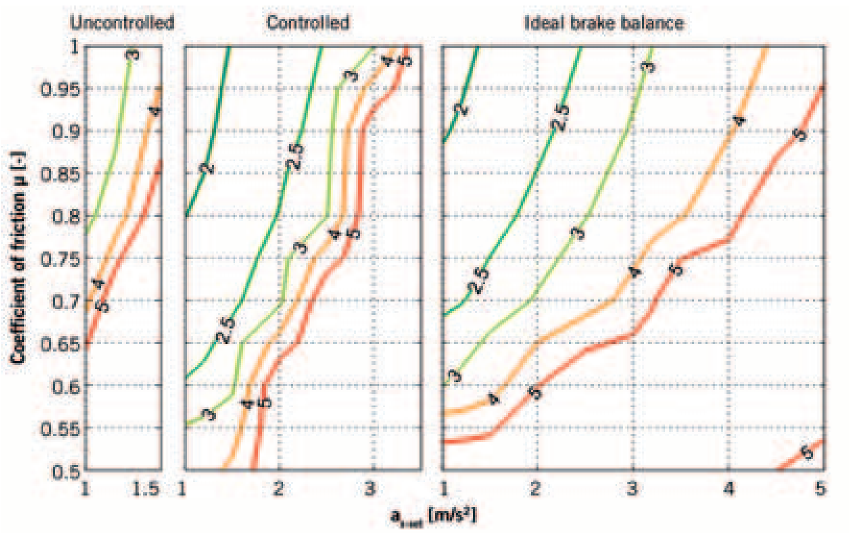
❺ illustrates the course of the yaw rates and sideslip angles for all three analysed cases as well as the set yaw rate for controlled braking of a sample manoeuvre. Uncontrolled braking of the rear axle is not possible in this case. However, the controlled vehicle follows the set yaw rate sufficiently accurate and its yaw and sideslip reaction is lower than with ideal brake balance.

For the evaluation of vehicle stability the maximum absolute sideslip angle β_{max} and the existence of a local maximum is used. If the absolute sideslip angle exhibits no local maximum, the vehicle will not stabilize without interference of the driver.

❻ shows the maximum sideslip angles as contour lines as a function of the set deceleration and coefficient of friction. Sideslip angles above 0.05 rad are not depicted, because stable behaviour was not observed anymore. Already with moderate coefficients of friction uncontrolled braking is only possible up to 1 m/s². With high coefficients of friction the controlled braking shows similar behaviour to the ideal brake force distribution up to decelerations of 3 m/s². Overall a doubling of the deceleration compared to uncontrolled braking is possible.



❺ Yaw rate and sideslip angle for uncontrolled, controlled recuperation and braking with ideal brake balance, set yaw rate of the controller ($a_y = 4$ m/s, $\mu = 0.9$, $a_{x\text{-set}} = 0.2$ m/s)



❻ Vehicle sideslip angle β [1/100 rad] as a function of coefficient of friction and deceleration

CONCLUSION AND SUMMARY

In light vehicles the ratio of driver mass to total vehicle mass is much higher than in common vehicles. To guarantee equal driving comfort for heavy and light passengers a constant eigenfrequency over a wide range of wheel travel is necessary. With the developed kinematics this is possible with linear springs. A quality criterion to develop the basic damper trace is defined. Due to the same damping power difference of bump and rebound, the car has a constant dynamic ride height.

Rear wheel drive electric vehicles and highly electrified hybrids can greatly benefit from torque vectoring as a means to increase their recuperation capability. Further development including a flatness-based feed forward control will make it possible to handle all normal driving situations with torque vectoring aided recuperation. Significant benefits are possible especially in combination with low rolling resistance tires as used in the concept vehicle Mute.

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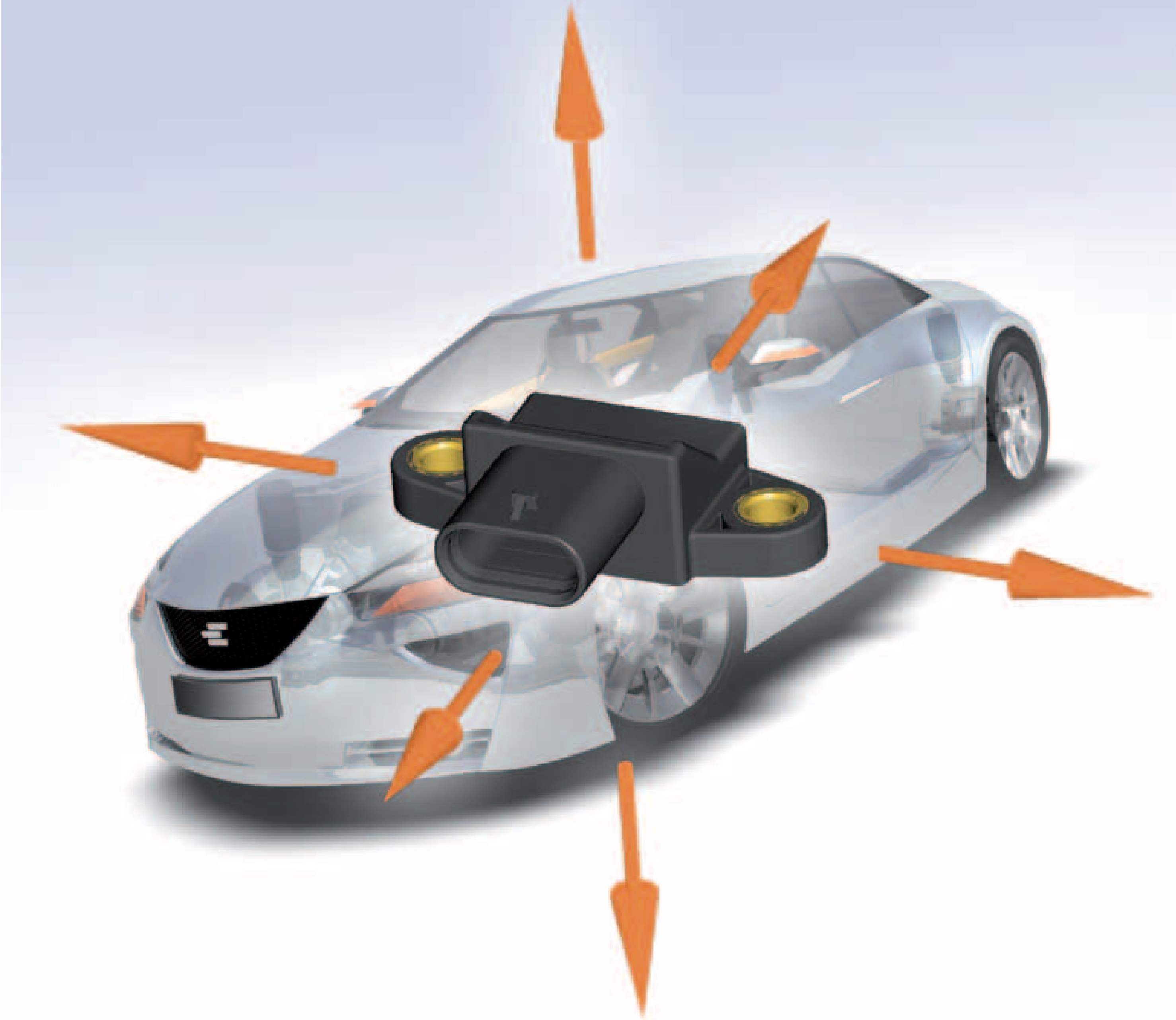

3D SENSOR TECHNOLOGY IN CLUTCH AND BRAKE APPLICATIONS

To meet rising demands for measurement data of high reliability and precision, non-contact position sensors are increasingly used for clutch and brake applications. TE Connectivity has developed a new sensor design which is compatible with vehicle on-board electrics systems and is able to compensate the tolerance effects typical of Hall sensor technology.

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MOTIVATION

To meet the needs of future generations of vehicle requirements in terms of safety, comfort, economy and environmental compatibility, new, innovative, smart electronic systems are being developed. New applications are, for example, automatic stop & go systems, regenerative and electro-hydraulic braking and automatic clutch and gearbox systems. At the interfaces between the mechanical functional elements and the electronic systems, contactless position sensors play a crucial and often safety-relevant role. The need for highly accurate measuring signals from the sensors requires the sensors themselves to be designed in a holistic manner which can stand up to the ambient conditions of the application and shifts in tolerances resulting from the installation environment and working conditions. At the same time, it is important to be able to detect faults and interference and guarantee the integrity of the signals.

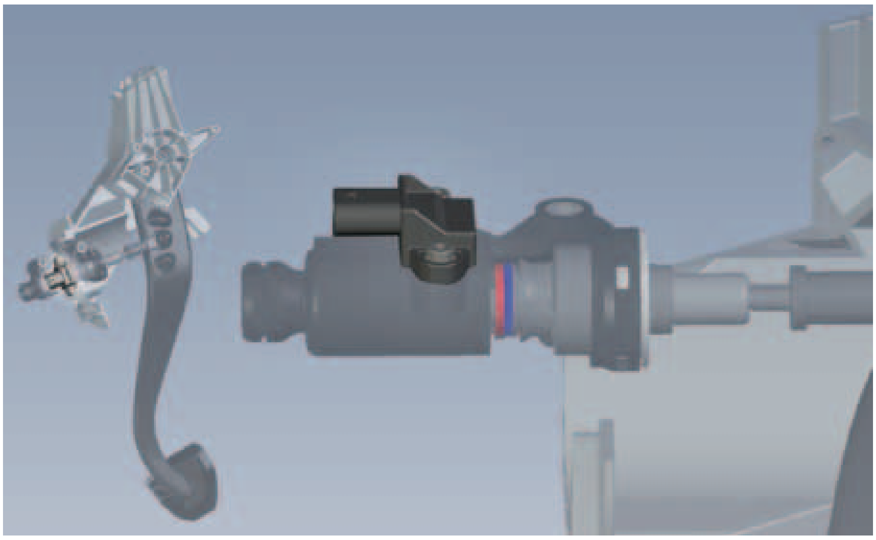
Today‘s solutions rely on stringent adherence to component specifications and compensation strategies to minimise factors affecting tolerances. With magnetic sensors, it is necessary to employ correspondingly powerful control magnets, which assumes that they are of an appropriate size. The new sensor design developed by TE Connectivity (TE) which is compatible with vehicle on-board electrics systems is able to remove and compensate the tolerance effects typical of Hall sensor technology. The sensors make use of three-dimensional magnetic field measurement by means of Hall semiconductor structures combined with integrated digital signal calculation. This enables a non-linear magnetic trigger field to be mapped to a strictly linear, application-specific output signal. This new concept means that smaller magnets can be employed wherever conventional systems generate a linear magnetic field which is determined by the physical size of the magnet. The result is a direct reduction in the application‘s system costs.

These benefits can be directly applied for the position sensors used on hydraulic clutch and brake master cylinders. Despite the annular magnet being small, the full stroke, sometimes more than 40 mm, can be detected and mapped, with high measuring accuracy, to a linear signal that has a very low linearity error, ❶.

MEASURING PERFORMANCE AND ADAPTATION

A feature of the monitoring of piston travel is the fact that the integration of hydraulic actuators in brake and clutch applications is much restricted by both the functional requirements and the physical lack of space for the trigger magnet on the piston rod. As a result of the design requirements, the amount of space available for contactless sensor technology means that only slim magnets can be used. Due to the short distances between the poles, the linear magnetic field available for measurement purposes is very restricted and cannot be exploited for measuring the full length of the piston stroke. In a simple, linear Hall effect solution, the distance between the poles of the trigger magnet should correspond roughly to the desired piston stroke.

Although it is possible to extend the measuring range by using a number of Hall effect sensors end-to-end, the individual sensor



❶ Sensor application clutch cylinder with ring magnet

signals then have to be evaluated and the overall curve calculated. Limited linearity performance and excessive sensitivity to fluctuations in the magnetic field have to be either accepted as given or compensated by means of additional complicated measures. This makes it necessary to incorporate component and system compensation technologies in the vehicle for each individual sensor used. That approach necessitates a large-scale installation of electronic devices and soon develops into an uneconomical and, as a result, less than attractive way of measuring travel. A further problem area in measuring piston travel is the effective signal resolution. If the walls of the actuator are thick and pressure resistant, this results in greater distances between the magnets and the sensors and corresponding reductions in the strength of the trigger magnetic field and increases in distance sensitivity. In particular, when the piston is at the extremes of its travel, the signal-to-noise ratio is reduced to such a low level that it is no longer possible to determine the exact position. As a result, the number of faulty measurements increases considerably.

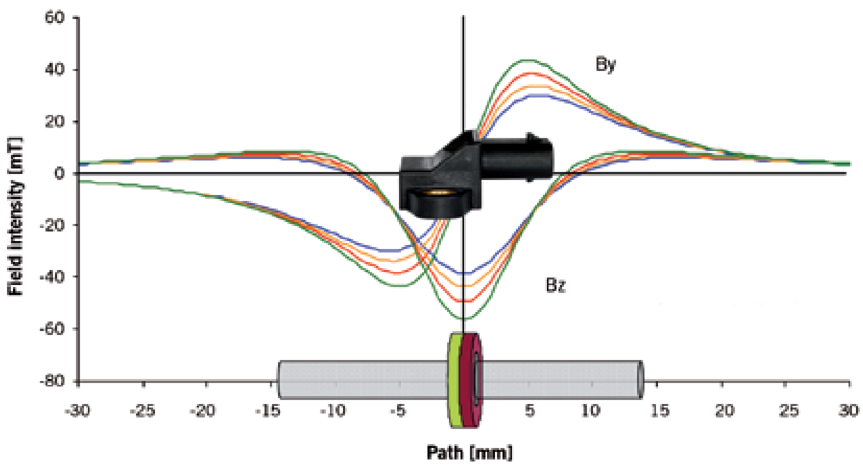
A state-of-the-art approach to using sensors to detect the full piston travel is one which takes account of the part of the magnetic trigger field that is non-linear. Since evaluating only one field vector component of the magnetic field can lead to equivocal results, several field vector components have to be detected and made available for evaluation. In principle, multidimensional 2D and 3D Hall effect sensors can meet this requirement, ❷.

A new kind of sensor developed by TE builds on and makes full use of these facts. To guarantee high measuring performance from the base up with absolutely linear data and, at the same time, accurate resolution, exactly calibrated, lateral and vertical Hall sensors are used. The new 3D Hall elements are based on the HallinOne technology developed by the German Fraunhofer Institute for Integrated Circuits (IIS).

By using two 3D pixel cells in a differential measurement approach, this sensor system guarantees robustness against external influences like stray magnetic fields. The two 3D Hall elements are mounted together on a single chip with digital signal processing and calculation. In addition, the sensor contains an EEPROM (Electrically Erasable Programmable Read-Only Memory) to provide maximum flexibility when programming for the application. A specially developed arithmetical algorithm evaluates the 3D Hall signals and enables, by means of optimised calculations, absolute linear position measurement of the piston despite the trigger magnet being extremely slim.

To achieve a high degree of path measurement accuracy using 3D Hall technology, a complex linearising process is applied which, under nominal conditions, will generate linearity errors of well below 1 %. The high degree of accuracy is achieved by using up to 32 linearisation points within the measuring range. The accurately linearised path information can be made available for further processing within a response time of less than 1 ms thanks to the powerful, fast signal processor core.

Fluctuations in the magnetic field which always occur during normal operation due to temperature drift or distance changes, for example, will generally impair the quality of the measurement signals in Hall sensors. As a result, the system offers optional correction functionality. The adaptive signal calculation module corrects changes in sensitivity fully automatically by using magnetic field memory during sequential application of current, which compensates for unwanted magnetic field fluctuations.



❷ Magnetic field strength of a target magnet from a moving piston with a variation in distance of +/– 1 mm

COMPATIBILITY WITH ON-BOARD ELECTRICS SYSTEMS

A key advantage in using this new sensor philosophy is its ability to use the same range of voltage as the vehicle electrics. The basis for this is a 0.35 µ high-voltage semiconductor technology. The sensor interface complies with typical EMC requirements for on-board electrics applications. The output signal is either from surge-proof pulse-width modulated drivers or discrete switching outputs. As a result, the frequently required 5 V supply, which has to be exactly regulated via an electronic control unit (ECU), is not required. In many cases, the supply cable harness can be simplified. The internal power loss is so low that the sensor can be used, in conjunction with appropriate enclosure technology, in the engine space at operating temperatures of up to 125 °C. If used on the master brake cylinder, a further feature, in addition to its on-board system compatibility, can be of great advantage. The implemented wake-up feature means that the sensor can even be active when the vehicle has been parked.

WAKE-UP FUNCTIONALITY

As a result of the increasing number of convenience and safety functions that are required to respond extremely quickly even when the engine is not running, the electronic systems on-board of a car are always active. No customer would consider a car to be acceptable that needs a relatively long response time after the remote key has been actuated to wake up the electronic circuitry that opens the doors. In addition to their conventional functions, systems like this require a further function: a very low-power sleep mode. Only when an external triggering event is registered will the system change from the sleep mode to its normal active mode at the full performance level. The versatile sensor design developed by TE already includes an additional wake-up feature which transmits a reliable signal to the controller to change it to its normal mode if an event, such as a brake pedal being actuated, occurs. The sensor itself requires no more than a few microamperes in this state and can, as a result, be left permanently connected to the vehicle battery. To enable various controllers to process the signals, there are a number of ways of configuring the sensor. It can, for example, simply transmit a change of state triggered by a signal change when an event occurs or it can transmit a measured value.

A key example of the use of a wake-up function is the actuation of the brake lights in a vehicle. It is a legal requirement for brake lights to light up when the brake pedal is actuated, even if the vehicle ignition system is off. Where, in the past, this function was performed by a mechanical power switch on the pedal, in today‘s vehicles it is a contactless sensor that wakes the corresponding controller and instructs it to switch on the brake lights. The sensor and the controller must, therefore, be permanently connected to the vehicle battery and, as a consequence, must have a sleep mode.

FUNCTIONAL SAFETY ACCORDING ASIL NORM

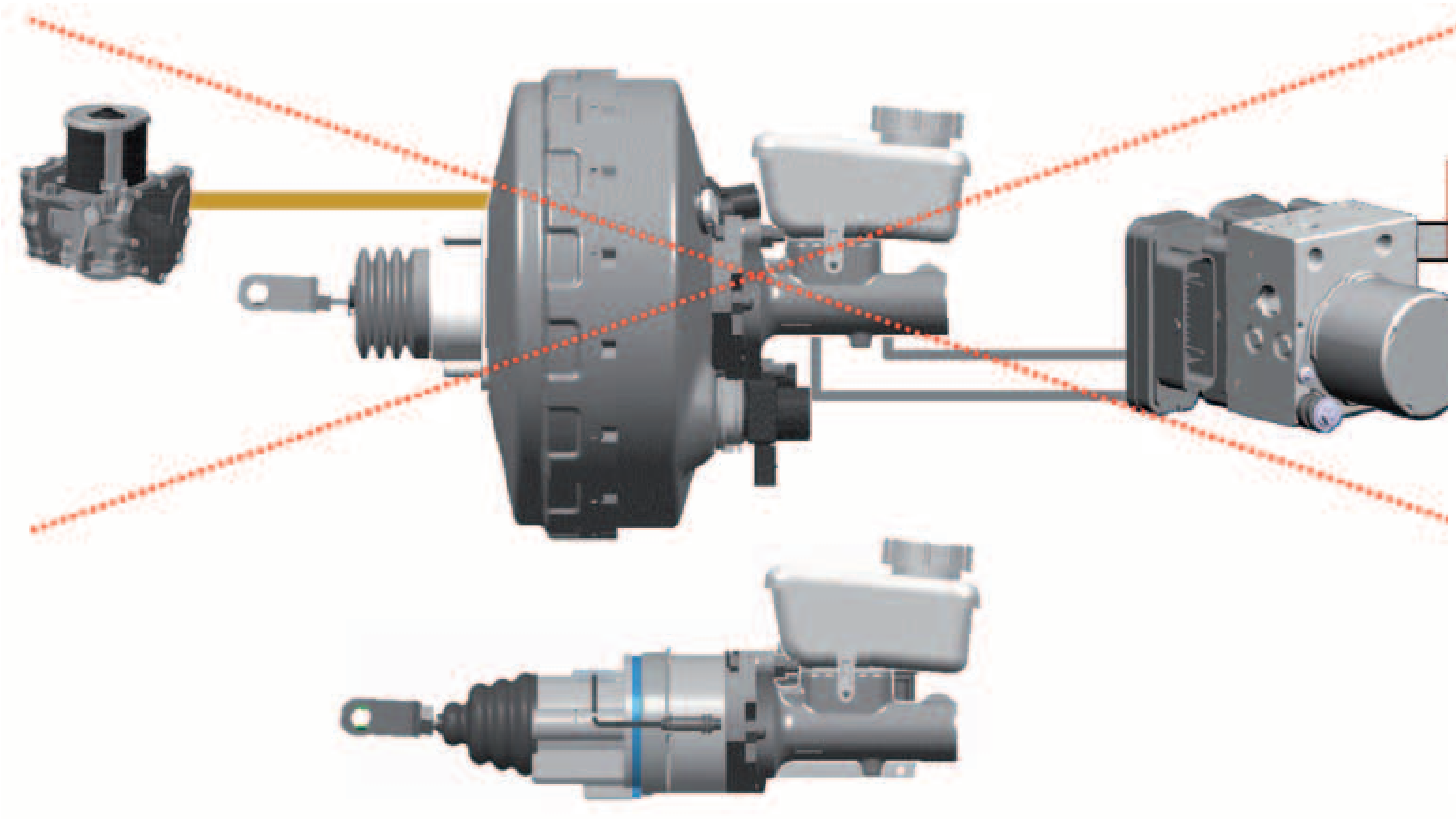
In today‘s automotive world, nearly every vehicle manufacturer provides smart systems that contribute to considerable reductions in emissions. The automatic stop & go systems for engines have taken a foothold in recent years and is partly offered for all models of a vehicle platform. Since the vehicle restarts the engine automatically under certain conditions, high demands are made on the stop & go systems‘ functioning perfectly every time. The basic idea of a the ASIL norm ISO/FDIS 26262 is the requirement that a single defect in an electronic component to a probable plausibility must not lead to a unwanted response (e.g. cause the engine to restart inadvertently). In some system solutions, the clutch master cylinder sensor is given a crucial monitoring function.

To meet safety requirements, this is sometimes achieved by incorporating several sensors and comparative circuitry in the vehicle but this results in complex, costly electronics systems which are not attractive to vehicle manufacturers. A more cost-effective method is to install a single sensor with a high safety standard (ASIL B). By using a single sensor of a high safety standard, it is possible to retrofit vehicles with existing E/E architecture more easily with an automatic stop & go system.

Thanks to sophisticated, effective self-monitoring functions, the new sensor concept achieves this safety standard, thus obviating the need for a further sensor and additional plausibility analysis within the controller. Typical monitoring functions such as internal short-circuit and interruption detection have been completely reworked to create a comprehensive safety concept that incorporates all the internal function components. Before the sensor outputs an updated measurement, it first checks the entire signal path from the transducer to the evaluation point. If a value is not plausible, an error signal is output to the electronic control unit. Furthermore, the entire memory is checked for consistency at regular intervals and any deviating memory contents will cause the sensor to change immediately to the passive mode. Finally, a large number of the internal calculation functions have been implemented as redundant systems to enable the results of calculations to be compared and, if they differ, to switch the sensor to a safe mode.

SUMMARY

Thanks to large-scale European research activities in recent years and intensive further development of existing 3D Hall effect technologies, it is hard to envisage anything preventing much increased application of 3D Hall sensors in cars. These sensors will expand into ever higher temperature ranges where they will be able to take on key functions in the engine space and even in exhaust gas recirculation (EGR) systems. Even now, the manufacturers of 3D Hall technologies are falling over themselves to offer larger measuring ranges with the same trigger magnets. This development is far from over, due to ever more sophisticated algorithms and enhanced computational capacities in the sensors. TE is committed to this technology and now includes a range of sensors with 3D Hall technology in its range of products.



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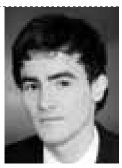

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HISTORY

Already in 1936, Bosch has a patent [1] on a device to prevent a locking of the wheels of an automobile, the first anti-lock braking system (ABS). The ABS which has been invented in 1965 is used worldwide today and extended to the vehicle dynamics control system ESC in 1995. Valve block, electronically commutated (EC) motor, brake fluid pump, 8 to 12 magnetic valves, accumulator and pressure sensor (PS) are assembled in one unit together with the electronic control unit (ECU) and together with the brake booster mounted in the engine compartment. Concepts in which the complete hydraulic function including the booster is integrated were on the market in the 1980s but were for cost reasons not successful [2, 3].

Because of the optimization of the combustion engines not only the diesel engine of today requires a vacuum pump for the brake booster [4]. Also in hybrid-drive vehicles the pump must be electrically driven – this gives sufficient motivation for a new integrated booster concept with ABS/ESC functionality, as it was realized by the company LSP Innovative Automotive Systems with the new integrated modular brake system (IBS). The cover figure shows a comparison of the IBS with the state-of-the-art system with substantial advantages in package, weight and assembly effort combined with outstanding functionality.

FUNCTIONAL PRINCIPLE

❶ shows, that for conventional systems, and also for EHB, an inlet valve (IV) and an outlet valve (OV) are required for pressure modulation for each wheel brake, ① (top). Brake fluid is dumped to the accumulator chamber (AcC) to reduce the brake pressure in the wheel cylinder (WC). The recirculation (RC) pump transports the dumped fluid back to the master cylinder (MC) which leads to the well known pulsating pedal.

For the pressure control a complex time control of the valves with limited accuracy is used as explained in the following:

: Control valves with throttle define the pressure gradient (vehicle adaptation).

MODULAR BRAKE SYSTEM WITH INTEGRATED FUNCTIONALITIES

Since the end of the 1970s, anti-lock braking systems and since the 1990s ESC systems are on the market to optimize braking and vehicle dynamics of cars. Using the new integrated modular brake system (IBS) from LSP Innovative Automotive Systems, it is possible to build such systems more compact, lighter and simpler. A very accurate pressure control is feasible.

- The pressure change Δp results from the time control Δt of the magnet valves (MV).
- Complex simulation models for PWM control. This is non-linear and depends on coil temperature (current of coils), pressure change Δp, viscosity (temperature) und tolerances in the MV as well as on pressure level.

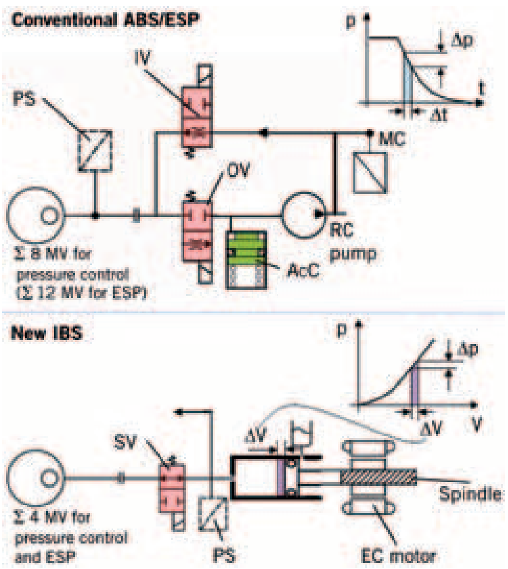
The new IBS, ① (bottom), uses a spindle with an EC motor to move the MC piston which transports the fluid volume back and forth to the WC for pressure decrease and increase. A switching valve (SV) between the MC and WC is used for pressure hold. Thus the WC pressures can be controlled sequentially, but also simultaneously. The IBS has the following features:

- A switching valve without throttle: the EC motor with piston control determines the pressure change and the pressure gradient.
- Large pressure decrease and increase gradients are possible.
- Individual vehicle adaptation is carried out by software.

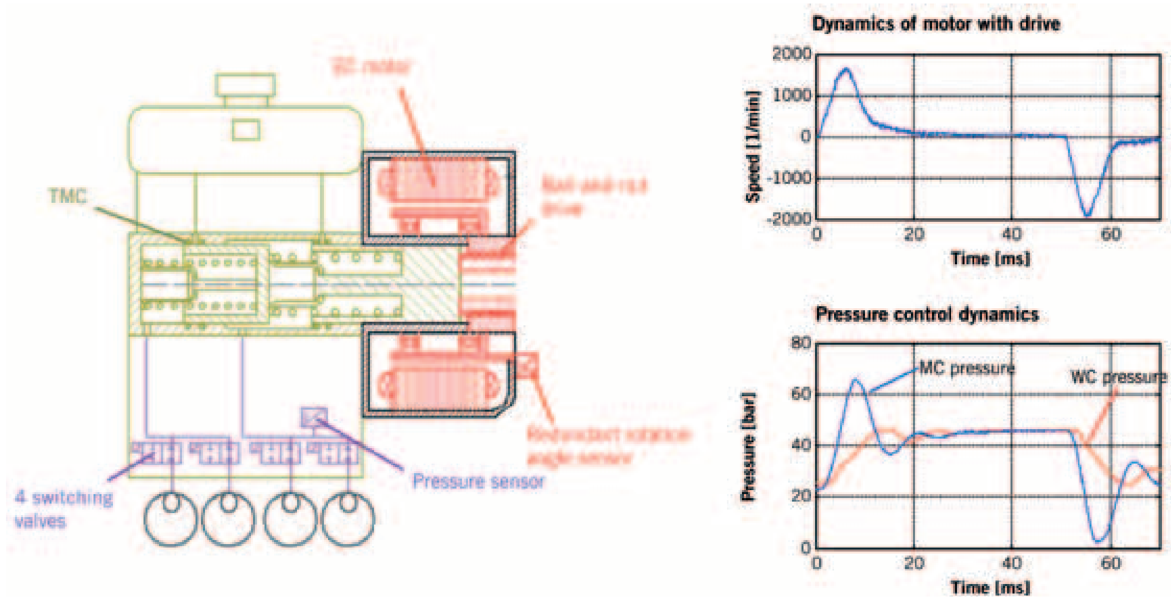
The big difference of IBS to a conventional system is that a pressure change as required by the ABS control is realized by a position control of the MC piston using the pressure-volume characteristic of the wheel brake and not by a valve control. Thus a very accurate pressure control is possible with the additional possibility to choose the pressure gradient to be by factors larger than that of conventional systems.

CONTROLLING SEVERAL WHEEL BRAKES

❷ shows schematically the system design for the pressure control of the wheel brakes of the IBS. A high dynamic EC motor moves the MC piston for the sequential wheel individual pressure



❶ Pressure control of the integrated modular brake system IBS (bottom) compared with a state-of-the-art ABS/ESP system (top)



❷ Fundamentals of the high dynamic pressure control (HDS)

control. This so-called multiplex procedure has been investigated in the 1980s with hydraulic and pneumatic boosters. Problems with noise, insufficient system dynamics and thus the inability of simultaneous pressure decrease made both approaches fail.

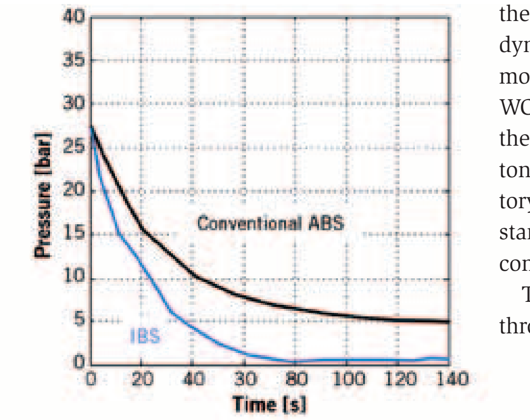
In the upper right part of ② the system dynamics is shown where in less than 4 ms the motor reaches 2000/min revolutions and in only a few milliseconds the rotational direction is reversed. The pressure curves of the MC and the WC show the corresponding high pressure control dynamics. From the ABS control the motor receives a nominal value for the WC pressure. Using a hydraulics model the motor controls the corresponding piston position with a prescribed time trajectory. Thus the pressure oscillation is substantially reduced as compared with the conventional system.

The valves IV and OV are designed with throttles which determine the pressure gradients in dependence of the vehicle. In the IBS, the valves have no throttle. Thus the piston velocity determines the pressure gradient which can be freely chosen for every pressure change.

MEASUREMENT RESULTS

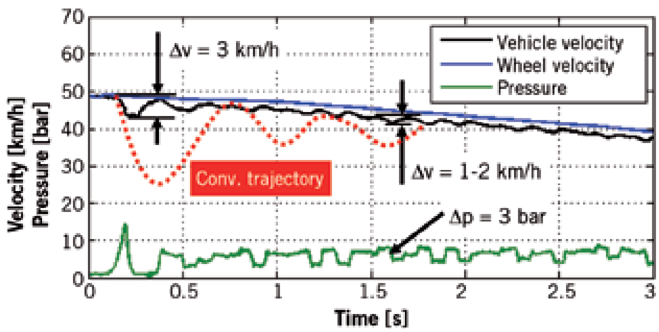
❸ (a) shows a comparison in pressure decrease which is representative for straight line braking on ice in which IBS decreases the pressure from 10 to 5 bar four times faster without an increase in the noise level than conventional systems which need 60 ms. This improves also the performance at fast brake apply on ice.

③ (b) shows an IBS-control with pure sequential pressure control at hard braking on ice with an average pressure level

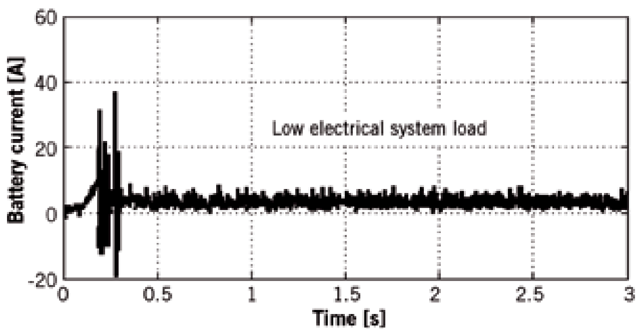


a) Pressure reduction is with IBS significantly faster

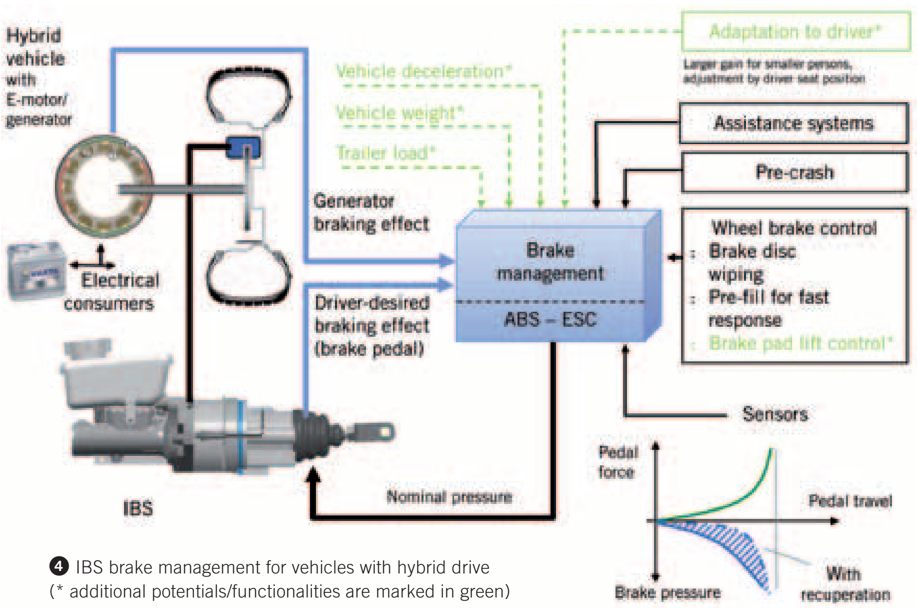
❸ Full straight line braking on ice with IBS



b) Control on ice



c) Battery current of control on ice (U_{DC} = 12 V)



❹ IBS brake management for vehicles with hybrid drive
(* additional potentials/functionalities are marked in green)

of 6 bar. Because of the higher pressure decrease rate the initial Δv is only 3 km/h. Later in the trajectory Δv is only 1.3 km/h with correspondingly small pressure changes of 3 bar. For comparison a typical trajectory of a conventional ABS is shown by the dashed curve.

Reductions in braking distance amounted up to 20% with an additionally improved vehicle stability as compared with conventional systems. Noise and pedal pulsations are virtually not noticeable. The comparison of nominal and measured pressure values shows that the highly dynamical pressure control follows the nominal pressure very fast and accurate which is also shown by the small wheel acceleration values. Better results of the electro-mechanical brake (EMB) [5] or the full electric brake ("wedge brake") are not known.

③ (c) shows the load of the electrical system which is independent of the pedal force. With increasing pedal forces the conventional ABS current increases also and up to more than five-fold.

BRAKE MANAGEMENT REQUIREMENTS

❹ shows the brake management of the IBS as it can be characterized for hybrid drive vehicles [6] and driver assistance systems. For the recuperation of brake energy in these vehicles the generator generates a brake torque which is limited and varies strongly while the brake management computes a complimentary nominal brake torque for IBS to comply with the driver desire. The desired retardation is derived from the pedal travel and pedal force. Additional potentials/functionalities are shown in ❹ in green, for example vehicle deceleration, vehicle weight and trailer load can be used in the evaluation so that the driver no longer has to adapt the pedal force and pedal travel to for example brake fading. Furthermore, the gain can be adjusted to suit smaller people with smaller foot forces.

It is well known, that brake pads normally do not fully separate from the brake disc except after severe cornering maneuvers. The residual braking torque increases the CO_2 emission. IBS can control the MC piston and the valves in a way which lifts the brake pad a little and thus reduces the residual braking torque substantially.

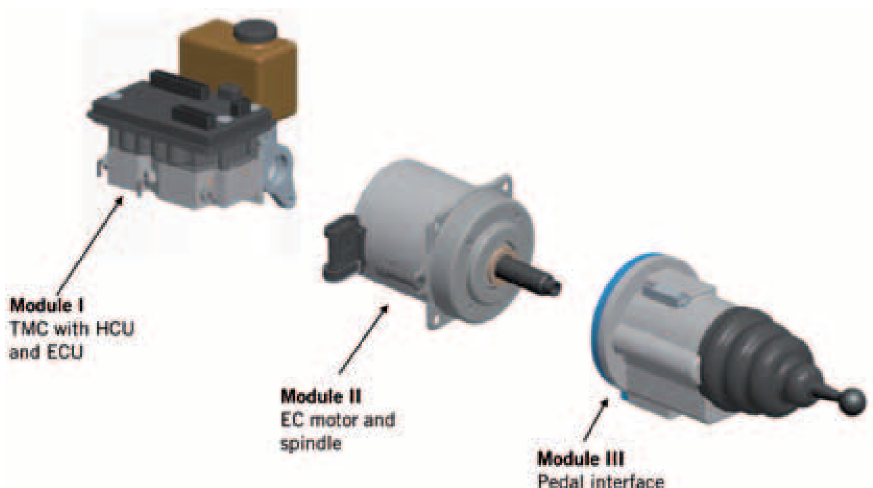
MODULARITY AND PACKAGING

❺ shows the modularity of the IBS in a feasible fundamental structure. The IBS comprises of:

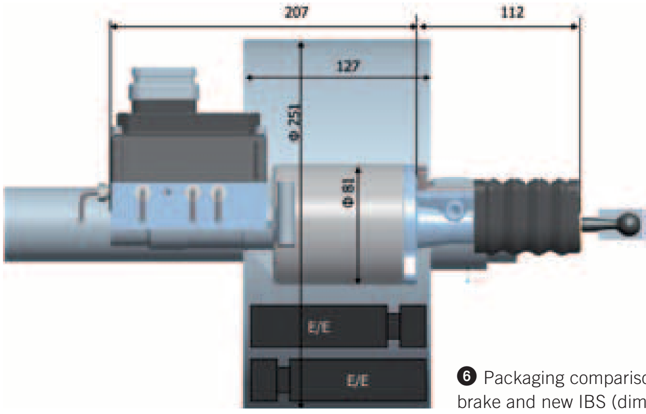
: Module I: Tandem master cylinder (TMC) with valve block (HCU) and ECU
: Module II: EC motor and spindle specifically with ball-and-nut-drive
: Module III: Pedal interface with transmission to the pedal travel simulator and sensors. Also located here is a tappet which connects the pedal push rod with the MC piston in case of a booster failure, and which is a basis for a fall back function.

At present a two-module-structure is developed. The ECU also monitors the fluid level of the brake fluid reservoir so that beside the cable set to the electric system (voltage supply, bus and wheel speed sensors) no further external cable sets are required.

❻ shows a packaging comparison of the vacuum booster with 8/9 inch diameter in relation to the IBS. The difference is obvious and enabled by the coaxial design of IBS. The freed space in the engine compartment, which is very valuable, can be used for mounting E/E components. It is imaginable that ECUs in the baggage area are preferably positioned here.



❺ Design variants of the three modules I, II and III of the IBS



❻ Packaging comparison of conventional brake and new IBS (dimensions in mm)

PEDAL CHARACTERISTICS

❼ (a) shows various pedal characteristics of modern vehicles which in general depend on the vehicle weight. The reason is that the fluid volume to actuate the WC increases with the vehicle weight since the actuation force of the WC-piston increases also with the vehicle weight. Given the pedal stroke, pedal gain and volume the dimension of the TMC can be evaluated. In case of a booster failure the UN/ECE regulation Nr. 13-H requires that with a brake pedal force of 500 N a vehicle deceleration of at least 2,44 m/s² must be reached. Because of this requirement the pedal stroke of the heavy car C cannot be chosen to be smaller. Today all designs comply with this regulation so that vehicles larger than class B reach a deceleration of about 0.3 *g*.

Already on the market are vehicles with substantially higher rigidity of the wheel brakes and correspondingly with shorter pedal strokes which can ideally be exploited for example for improved utilization of the passenger compartment or vehicle length. The bold curve shows the typical pedal characteristic of the IBS. It is well known, that with a pedal travel simulator an ideal pedal characteristic can be implemented which is independent of the vehicle weight.

COMPARISON OF BOOSTER CONCEPTS

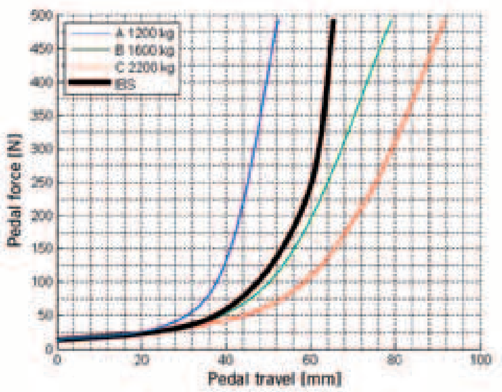
⑦ (b) shows a comparison of a conventional booster with so-called servo control (marked with “A”) and a brake-by-wire booster with pedal travel simulator (“B”). Facts which are well known to the expert are also mentioned here. For booster A the dimensions of the TMC and of the booster are constrained by the regulations mentioned before.

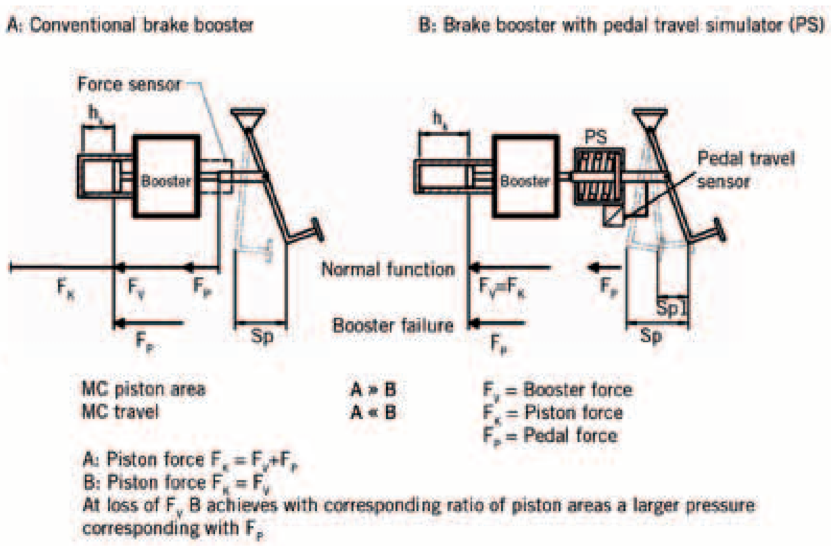
The design of booster B with the pedal travel simulator can exploit the big advantage of the separation of pedal travel and MC-piston travel. Substantially reduced diameters of the TMC piston can be used. The fluid volume needed can be made available by a longer piston stroke or by a replenishment assembly. Thus in the fall back situation with the specified pedal force of 500 N much larger decelerations beyond 0.5 *g* (> factor 2) can be reached (UN/ECE: ≥ 2.44 m/s²), which is very much appreciated the driver.

Moreover, booster B lacks the additional springs and the friction of the conventional vacuum booster A, so that in the fall back situation the pedal rod force actuates the MC unhindered. For normal and for fall back the force-travel characteristic of type B is similar which is decisive for the driver. The only difference is the relation between the pedal force and the vehicle deceleration to which the driver can easily adapt himself. He will be notified anyway by a message in the display that a service is outstanding.

Up to now brake-by-wire systems with pedal travel simulator partially show a synthetic pedal feeling, which is undesirable, but which can certainly be improved during the intensive continued development since for example the highly dynamical EC-motor offers additional potential.

It is known that boosters using an EC motor as a type A with servo control are on the market today or are under development in order to be independent of the availability of vacuum. For the fall back





❼ Comparison of a conventional brake booster with a brake booster with pedal travel simulator – left: pedal characteristics for three vehicles and an IBS vehicle; right: design and piston force solution the ESC pump can be used for the boosting gain. However, the usual pedal characteristic cannot be maintained. This booster gain is not available if the voltage supply fails as is the case among others if the car is towed. The following comparison shows the advantages of both booster types.

Advantages of the type booster A:

- usual good pedal feel with booster operation
- experienced drivers recognize the brake state from the pedal travel

Advantages of booster B:

- shorter pedal stroke, faster pressure increase
- variable, adaptive pressure jump at brake apply (optimization for each model family)
- improved fall back in case of booster failure (usual pedal characteristic maintained and smaller pedal forces)
- pedal characteristic independent of fading
- ideal for hybrid drive vehicles with recuperation
- no increased pedal travel in case of brake circuit failure
- automatic diagnosis of the bleeding state.

IBS with its fast, accurate and low noise pressure modulation can also be used for a pressure based electrical parking brake. As already mentioned, the brake pads can be lifted a little by a short vacuum control.

SUMMARY

The development of the integrated modular brake system (IBS) by LSP Innovative Automotive Systems started in 2004. Well known pioneers (Heinz Leiber, Anton van Zanten et al.) of the ABS and ESC development community supported an effective and interdisciplinary team. Modern development methods and tools were used. Three winter tests in Lapland showed the potential and the capability for realization of series production. The supreme features of IBS are:

- high functionality also for all known driver assistance systems and with performance at least equal to EHB and EMB
- reduction of weight and mounting space
- relatively low complexity using known standard components
- high safety in case of failure and outstanding fall back performance. Possibility to diagnose all safety relevant functions
- potential for additional functions like simplified electrical parking brake and control of brake pad lift

These features together with the test results in the car and in simulations on test beds should provide the OEM a basis for the decision to introduce the IBS on the market.

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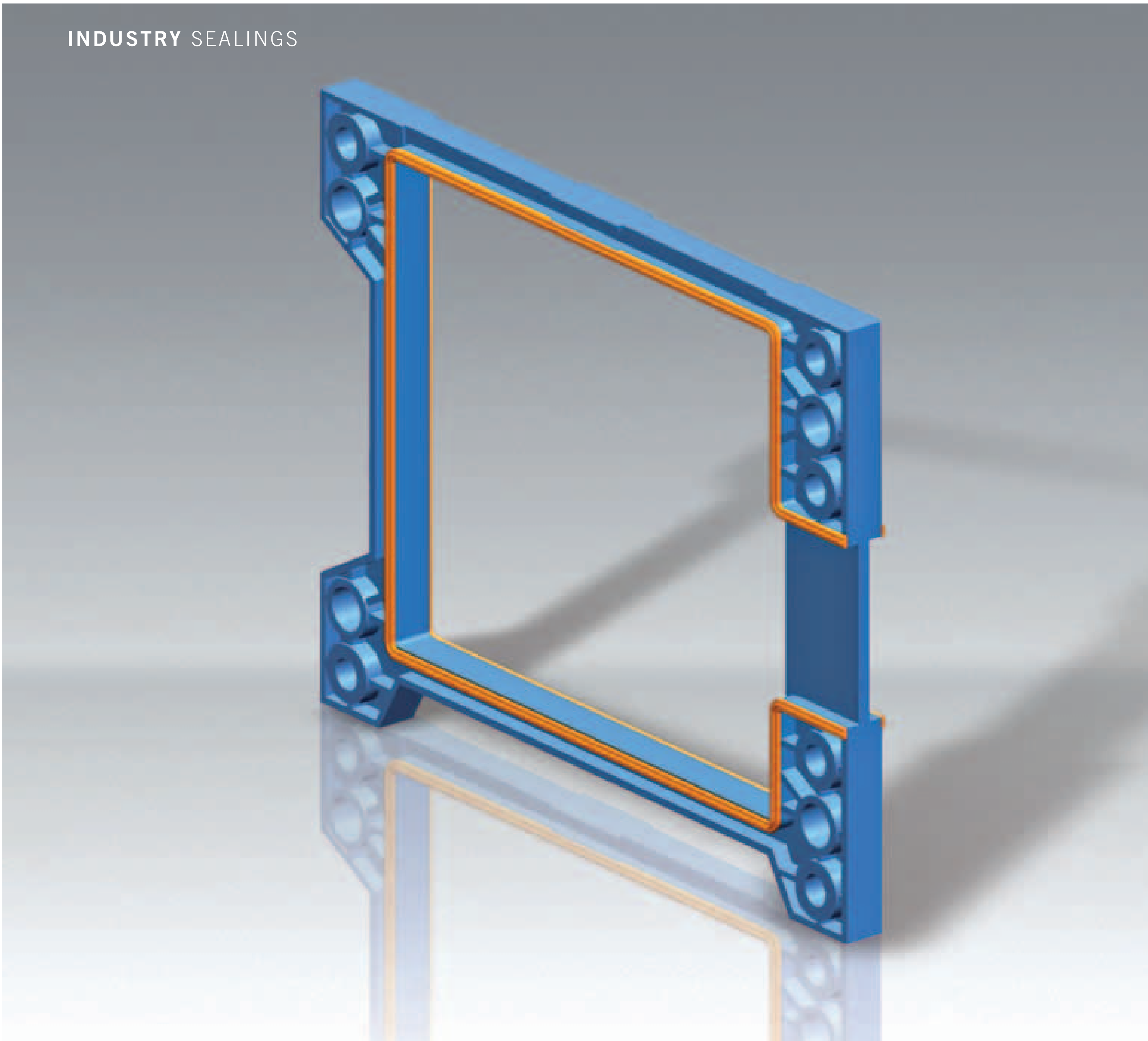

SEALING AND FIXING ELEMENTS FOR FLEXIBLE CELLS IN LARGE-SCALE LITHIUM BATTERIES

High-capacity batteries for electric vehicles raise safety issues in many respects. Freudenberg overcomes system-specific disadvantages of the battery design by embedding the cells into a cell frame seal. The concept can also be used to transport the cells safely and facilitates subsequent battery assembly.

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TASK

Pouch cells, also referred to as "coffee bag cells", have major advantages due to their low specific mass, their scalability in production and their low manufacturing costs. This makes them particularly suitable for electric vehicles with high-capacity batteries. However, the flexible housing that is currently in use has a number of system-specific disadvantages. For example, the sealing seam of the cells has to be intact during the entire service life of the battery – in the case of automotive batteries for at least 10 years. This is made even more complicated by the fact that the cells "breathe" during cycling, i.e. their thickness varies depending on the battery's current state of charge. Furthermore, there is a fluctuating pressure difference between the inside of the cell and the environment, caused by changes in atmospheric pressure, and the sealing seam has to bridge a "gap" at the feedthrough of the current collector.

What is more, the fixing of the cells inside the battery housing and their electrical connection is difficult, as is their thermal integration. Finally, in the event of a fault resulting in pressure overload inside the cell, it is possible that the cell's sealing seams open up irregularly. In a worst-case scenario, flammable gases such as electrolyte or degradation products can come into contact with current-carrying components, which can lead to fire and/or explosions. Cells open up at relatively small pressure overloads.

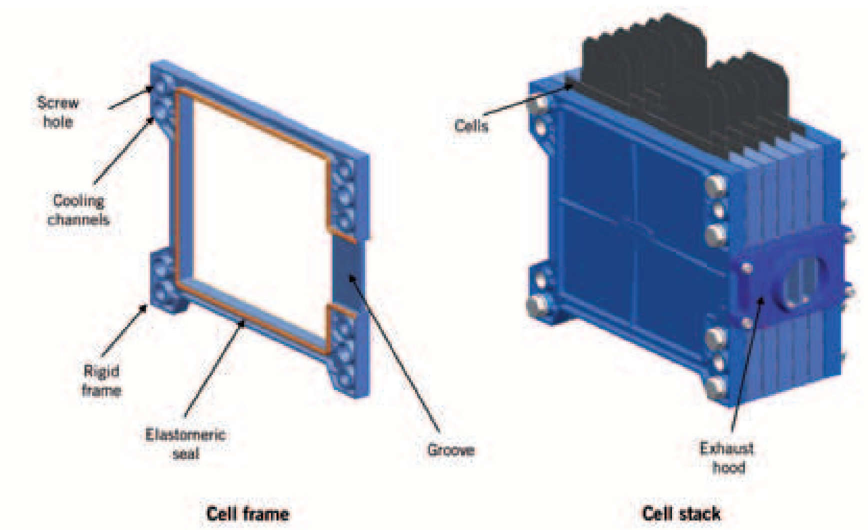
All of these problems can be solved by embedding the cells into an innovative cell frame seal, ❶. Each individual cell is placed between two rigid frames that carry a circumferential, elastic seal either on one side or on both sides. This seal applies pressure along the entire circumference of the cell's sealing seam.

MECHANICAL ADVANTAGES

Due to the integration of this flexible element, the cells can be sufficiently sealed and can also be fixed in a flexible and elastic way (PCC – pouch couch concept). This fixing method levels the tolerances, thus improving the assembly of the cell stack inside the battery housing, extends the service life through a reduction in mechanical strain and a resulting improvement of the reliability of electrical contacts, and increases safety by buffering mechanical impacts. The additional seal makes the sealing seam particularly tight in those places where the electrode plates are fed through.

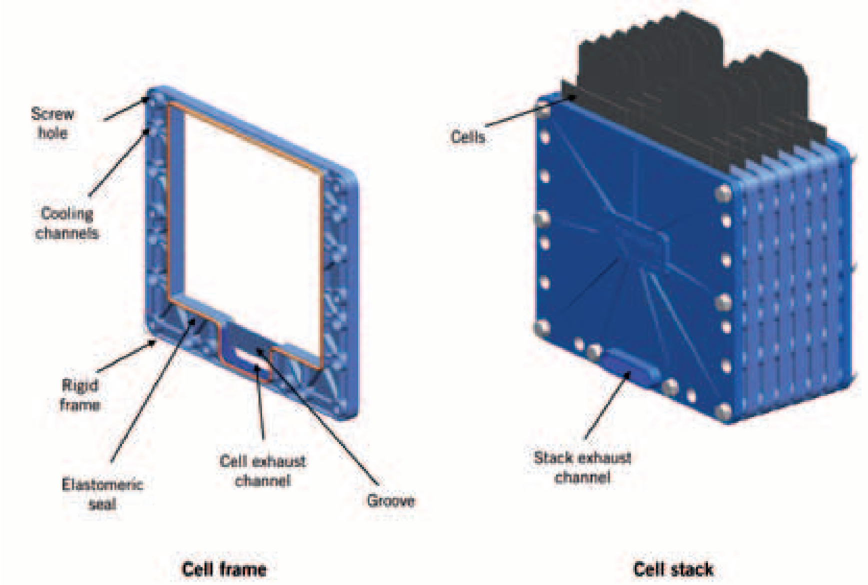
The circumferential seal has been designed with a recess at which point it applies no pressure, or very reduced pressure, to the sealing seam. The recess has to suit the internal design of the cell (electrode separator stack) and acts as an intended interruption of the sealing seam through which any uncontrolled pressure build-up can escape from the cell in a controlled manner.

The recesses of neighbouring cells can thus be covered with a blow-off hood that



❶ Concept of a cell frame seal for pouch cells, featuring a sealing seam and an integrated predetermined leakage point in the area of the recess channels harmful gas emissions safely to the outside. An additional valve in the opening of the hood can prevent ingress of dirt and moisture into the battery housing under normal operating conditions.

In another design, ❷, the blow-off channel is integrated into the individual frame. The channel is therefore situated orthogonally to the cell plane. This arrangement has further advantages in addition to the versions described above. The integration of the blow-off channel into the frame reduces the number of components. It allows improved scalability of the battery system. It facilitates the addition or removal of individual cells in order to adjust the capacity of the overall system. Sealing between a blow-off hood and the cell frame is no longer required and, by locating the blow-off channel opposite the deflecting plates, it is possible to have an axially symmetrical construction of frames and lids.



❷ Concept of a cell frame seal for pouch cells, featuring an integrated blow-off channel

If the blow-off channel is located on the side opposite to the electrodes, it is also possible to have an axially symmetrical design of the two lids which, in turn, reduces the complexity of the components.

The mass of the frames can be optimised by a suitable mechanical design. In the case of the example shown in ①, it was possible to reduce the mass of a frame by more than 60 %, compared to solid material.

THERMAL ADVANTAGES AND ADDITIONAL BENEFITS

At present, pouch cells are typically cooled by the current collector. This can lead to an accumulation of condensation liquid at the current-carrying parts, which can result in short circuits.

An important additional benefit of the cell frame seal is the fact that it allows the integration of thermal management elements. In order to do this, cooling channels that carry cooling pipes can be integrated into the frame. The pipes can be flushed with water-based media or can be directly linked to an air conditioning system.

The cells are thermally connected via the sealing seam. The surrounding horseshoe-shaped connection creates a very homogenous cell temperature, a fact that is vital for a long service life of the cells.

In this case, the frame material has to be thermally conductive. Suitable materials are, for example, thermally conductive thermoplastics whose conductivity lies well above 2 W/[m*K]. Another possible alternative for the material of the frame could be metal, bearing in mind that there must be no electrical contact between the frame material and the discharge plates.

A suitable method for integrating cooling elements is the provision of cooling circulation through pipes, which can be run vertically through the cell frame seal. The heat transfer to the cells can take place via the frames. The design of the cooling system has to suit the relevant application.

In large-scale battery systems for electric vehicles, it makes sense also to integrate heating elements. In temperatures below 0 °C, the electrical properties of batteries are not only considerably impaired, the cells can also be irreversibly damaged through the formation of lithium dendrites, especially during charging. The purpose of heating elements is therefore to either prevent a battery from cooling down at operating temperature, or to slowly condition it prior to charging or operating.

The Freudenberg concept makes it possible to place sheet-type heating elements, such as those already being used for heated wing mirrors, in the space between the cells. The heating elements are connected via cables or flexible circuit boards, which are insulated and channelled through the frame.

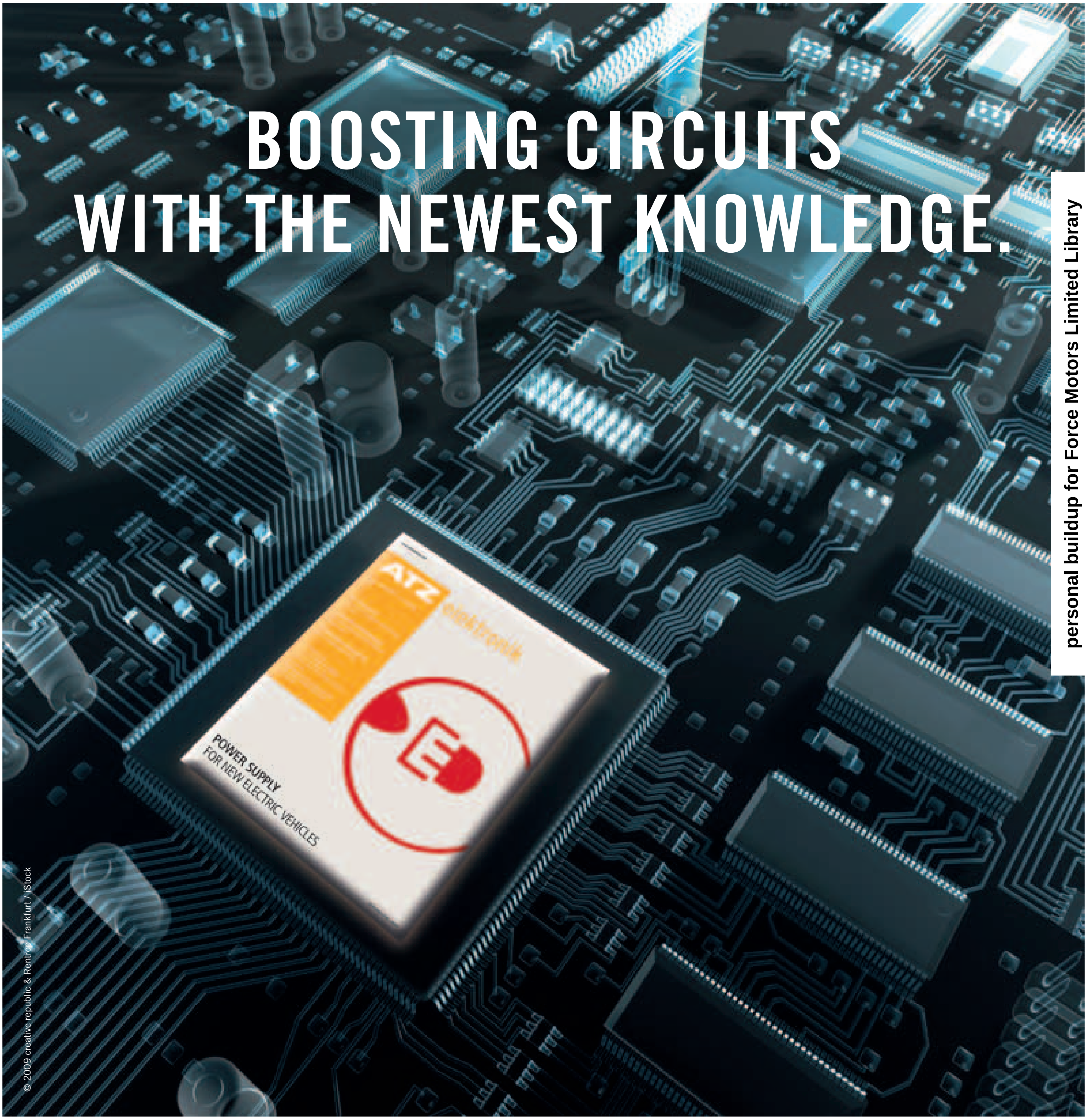
The thermal connection can be further improved through additional integration of thermally conductive, compressible elements, such as nonwovens, in the spaces between the cells. They are then not only thermally connected via the sealing seam but also via the considerably larger cell surface. This improves the effectiveness as well as the homogeneity of the temperature. The compressible elements follow the movement of the cell surface during cycling. By doing so, they provide a tight fit between the thermally conductive compensation element and the cell surface.

Porous media, such as thermally conductive nonwovens or foams, are suitable materials for these elements. Placing phase transfer materials inside the compressible intermediate layer can also help to level localised temperature peaks on the cell surface. Additionally, these elements help to protect the cells against mechanical impact and support the changing thickness of the cells during the charging/discharging cycles.

INTEGRATION INTO PRODUCTION

The present concept can also contribute to the safe transport of cells and to a fast, resource-saving production method for battery systems. With that aspect in mind, the cell frame seals are already used as transport aids during the manufacture of the cells.

At a later point in the production process of the battery system, it is possible to quickly and securely assemble the cells due to the fact that the frames are already in place. The elimination of one type of disposable packaging also means a reduction in waste material.


BOOSTING CIRCUITS
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PROGRESS IN CFD VALIDATION IN AERODYNAMICS DEVELOPMENT

In 2009, Audi, Volkswagen and Seat presented a method of utilising numerical flow simulation for aerodynamic applications based on the OpenFoam open source CFD code. Since then, it has been possible to gather experience from the series-production development of various model lines. Air forces were determined and compared with wind tunnel results. An overview of the validation and successful implementation of the method in the aerodynamics development process shows the advantages of the process.

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INITIAL SITUATION

Computational Fluid Dynamics (CFD) is already one of the vehicle aerodynamics engineer's firmly established development tools. The suitability of CFD for the reliable prediction of target values such as aerodynamic drag and lift for various basic vehicle patterns has already been confirmed in numerous publications and is also applicable to demanding development tasks, for example brake cooling and side-wind investigations [1, 2]. The state of the art has so far always consisted of the use of commercial software packages with either the Reynolds-averaged Navier-Stokes equations (RANS) or the Lattice-Boltzmann equation used in their flow generators. In such cases, the aerodynamics developer is increasingly dependent on fast CFD processes that are free from interruptions, particularly since the increasing number of model versions limits the availability of the wind tunnel as a development tool.

In 2009, Audi, Volkswagen and Seat introduced a complete aerodynamic computing process based on the OpenFoam open-source software [3]. It was found that the results, in this case using the Detached Eddy Simulation (DES) method, correlated well with wind tunnel tests. The following paper describes the results of further validation work

NUMERICAL METHOD, MESH GENERATION AND COMPUTATIONAL PROCESS

Large Eddy Simulation (LES) means that large coherent structures are computed directly by way of the Navier-Stokes equations, with the influence of small-scale turbulence on the main flow described by a turbulence model. The large scales of motion convey the most energy and are mainly determined by peripheral conditions and the extent of the flow body. Using the LES method, flow values are interpreted as the sum of a large-scale constituent and a small-scale constituent. After this, the Navier-Stokes equations have a filter applied to them over a space of width Δ. By means of this filtration, the large-scale constituent can be defined according to width Δ [4]. Additional terms occur on the right side of the filtered Navier-Stokes equations. These are grouped together and defined as the Sub-grid Scale stress term (SGS), which has to be modelled when the LES method is used:

EQ. 1

$$\frac{\partial \tilde{u}_i}{\partial t} + \tilde{u}_j \frac{\partial \tilde{u}_i}{\partial x_j} = -\frac{1}{\rho}\frac{\partial \tilde{p}}{\partial x_i} + \nu \nabla^2 \tilde{u}_i - \frac{1}{\rho}\frac{\partial \tau_{i,j,SGS}}{\partial x_j}$$

In the RANS process, the flow values are interpreted as the sum of a mean time value and a fluctuation constituent. After this, the Navier-Stokes equations are averaged over a sufficiently long time interval. The non-linear terms on the left side then yield the additional links referred to as Reynolds stresses $\tau'_{i,j}$, which are also modelled (for example, according to [5] for aerodynamic flow):

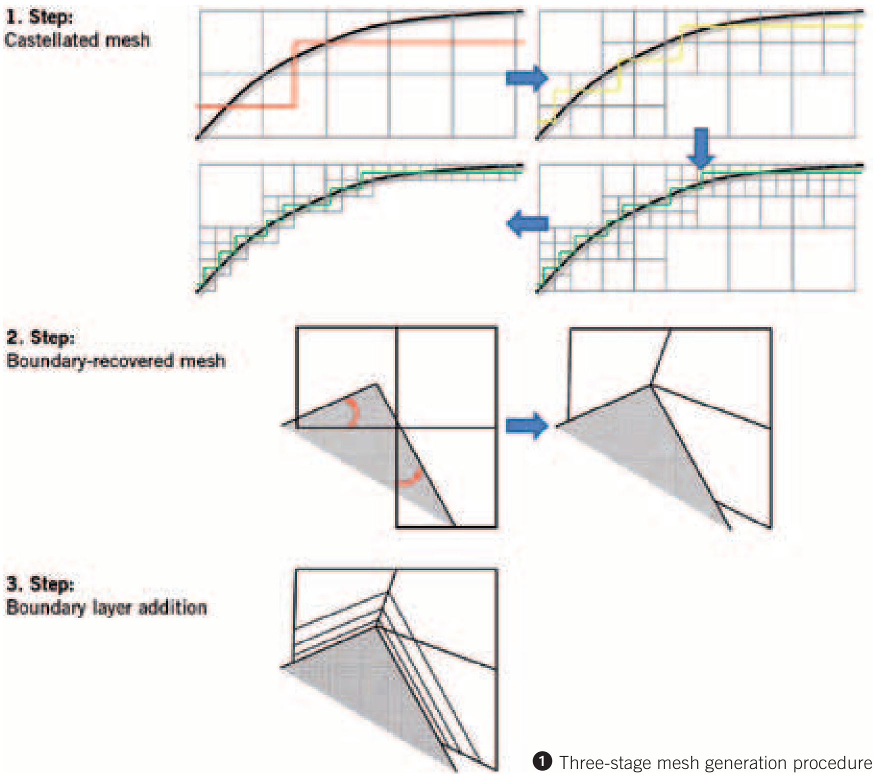
EQ. 2

$$\frac{\partial \bar{u}_i}{\partial t} + \bar{u}_j \frac{\partial \bar{u}_i}{\partial x_j} = -\frac{1}{\rho}\frac{\partial \bar{p}}{\partial x_i} + \nu \nabla^2 \bar{u}_i - \frac{1}{\rho}\frac{\partial \tau'_{i,j}}{\partial x_j}, \text{ with}$$

$$\tau'_{i,j} = \rho \cdot \begin{pmatrix} \overline{u'^2} & \overline{u'v'} & \overline{u'w'} \\ \overline{u'v'} & \overline{v'^2} & \overline{v'w'} \\ \overline{u'w'} & \overline{v'w'} & \overline{w'^2} \end{pmatrix}$$

Compared to the Reynolds stresses, the SGS terms relate to a significantly smaller part of the turbulent energy spectrum, which means that comparatively lower demands can be imposed on the turbulence models. For a filter width Δ, LES uses the mesh width of the numerical grid. DES uses the LES only in the outer flow and in wake areas. In areas close to the wall, the transient RANS equations are used. Provided that the turbulent longitudinal scales in the boundary layer are of a distinctly smaller order of magnitude than in the outer flow, this procedure can be regarded as a good approximation.

Mesh generation takes place in three stages, ❶. In the first stage, an orthogonal hexahedral mesh is generated with the aid of the user-defined refinement levels. This operation results in the vehicle geometry having a shingle-like surface ("castellated mesh"). High mesh resolution is chosen in particular close to walls or in suspected wake areas. The section of the mesh located inside the geometry around which the flow passes is cut away. In the second stage, the cell surfaces of the mesh boundaries are projected onto the CAD surface, with particular care being taken to retain the principal feature edges. In the third stage, prismatic cell layers are generated on the basis of the surface mesh that has now been obtained, and inserted between the surface mesh and the volumetric mesh, with the effect of compressing the latter.

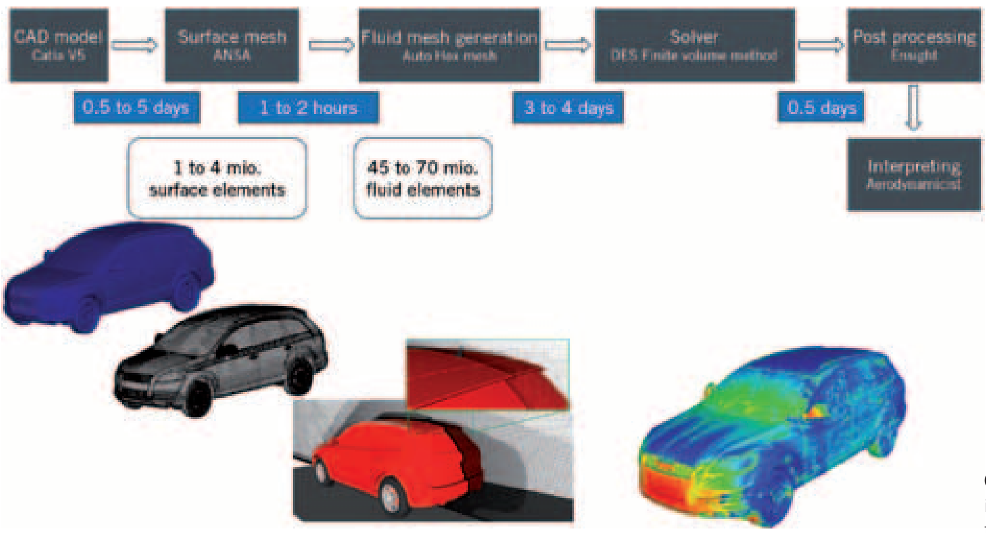


❶ Three-stage mesh generation procedure

❷ shows the complete Audi aerodynamic simulation process chain. The data, usually obtained from Catia V5, are discretised with Ansa within five days in order to obtain a triangulated surface, after which the fluid volume mesh is produced automatically in one to two hours. Depending on the scope of the computing work, two to four days have to be allowed for the CFD solution, using a 2.66 GHz Intel Xeon Cluster with 128 CPUs. Finally, post-processing (Ensight) takes half a day.

RESULTS

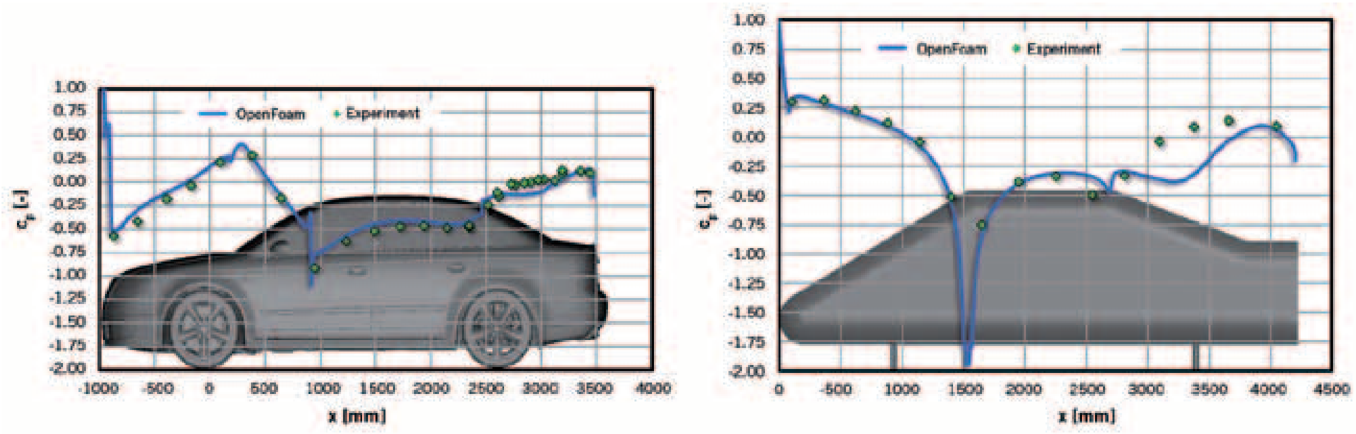
In the following, the CFD results are compared with wind tunnel measurements. The vehicles that have been investigated can be divided into squareback (Audi Q7, Q5), fastback SUV concept and notchback (Audi RS4, 2005 model and SAE body). These vehicles, which differ considerably in shape and size, emphasise the robust



❷ Process chain for the use of CFD in aerodynamic development at Audi, from CAD model to CFD solution

VEHICLE	Δc_D [–]	$\Delta c_{L,f}$ [–]	$\Delta c_{L,r}$ [–]
Audi Q7	–0.002	+0.012	+0.050
Audi Q5	+0.006	+0.009	–0.024
Audi SUV Concept	+0.002	–0.002	+0.020
Audi RS4	–0.001	–0.033	+0.026
SAE notchback	+0.009	–0.016	+0.012

❸ Air force coefficients for various vehicles and evaluation of the deviation between wind tunnel measurement and CFD



❹ Static pressure distribution on the upper surface of the vehicle at the central y-section; comparison between wind tunnel measurement and CFD result

nature and quality of the CFD method for aerodynamic applications. The computing zone is a rectangular block of sufficient size to avoid blocking effects. In the vehicle area, the base causes friction, so that a boundary layer comparable to that obtained in the experiments flows against the vehicle. Apart from this, the base, upper surface and side walls are free from friction. A fixed inflow velocity and turbulence are chosen at the inlet, and a constant pressure at the outlet. The computations are conducted at $v_\infty = 140$ km/h and occupy 1 s of real time. The width of the time steps is $1 \cdot 10^{-4}$ s, with the results averaged over the final 0.75 s. A potential flow problem solution is used as the starting criterion.

❸ is a summary of the air force coefficients. Maximum drag deviation is $|\Delta c_D| = 0.009$, maximum front-axle lift coefficient deviation $|\Delta c_{L,f}| = 0.033$ and maximum rear-axle lift coefficient $|\Delta c_{L,r}| = 0.050$, in other words the drag coefficients in particular deviate by under 3 %. At first sight, the deviations recorded for Audi Q7 rear axle lift appear to be rather high, but it should be noted in this connection that comparison of time-averaged values is insufficient for the validation of force coefficients. The wind tunnel experiment is also subject to the transient character of the vehicle's wake. In this case, a force coefficient value averaged during 1 min is supplied. The transient behaviour pattern is especially obvious on the Audi Q7; this may be due to non-stable wind breakaway in the diffuser area.

Both lift coefficients for the two notchback bodies RS 4 and SAE notchback exhibit noticeable deviations. Comparisons with measured pressure distributions on the surface of the body indicate that CFD predicts detached flow in the rear window area, but that these cannot be identified in the experiment, ❹. If pressures on the inclined rear surface are too low, an upward force component is generated and causes higher rear axle and lower front axle lift. This explains the deviations in ③.

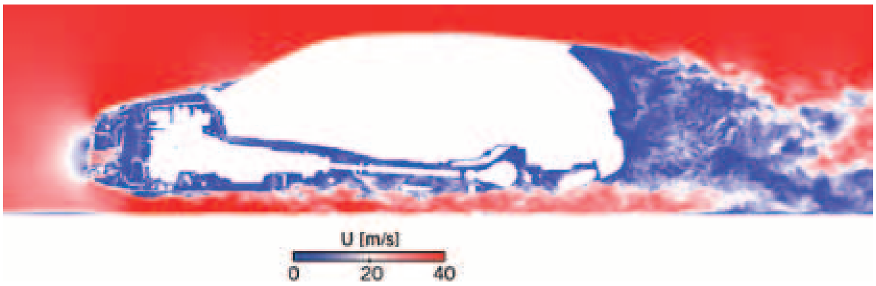
The cause of this unrealistic flow from the rear of the vehicle is to be found in the function of the wall law in the RANS section of the computing area. At the transition between the roof and the rear window, the standard set-up used generates computational cells that prove to have a mean distance from the wall of $5 < y^+ < 25$. Optimal modelling of flow close to the wall is not possible using the wall law. A change in the fineness of the mesh consequently has a clear influence on the formation of this kind of recirculation. Breakaway can be prevented in this area by means of body-adapted mesh parameters, which also improves the force coefficients. This applies to SAE bodies in particular.

Until now, the mass airflow through the heat exchangers at the front of the vehicle was disregarded in flow calculations. This additional simulated mass airflow imposes much more severe demands on the computing process. For an Audi Q7, for example, the number of fluid cells increases from about 60 million to 105 million. In a typical situation, airflow through the front end of the vehicle results in a change in the air forces; this is normally referred to as "cooling air delta", and can account for up to 10 % of the aerodynamic drag. The cooling air delta value from drag and lift was determined for two vehicles with differing rear-end styling, namely the Q7 (squareback) and RS 4 (notchback).

❺ shows the deviation between measured and computed cooling air drag. Agreement can be regarded as good for both vehicles: all changes proceed in the

VEHICLE	$\Delta\Delta c_D$ [–]	$\Delta\Delta c_{L,f}$ [–]	$\Delta\Delta c_{L,r}$ [–]
Audi Q7	0.007	0.047	–0.009
Audi RS4	0.003	0.041	–0.005

❺ Changes to the air force coefficients of various vehicles when the cooling air is taken into account; evaluation of the deviation between wind tunnel measurement and CFD



❻ Calculated distribution of the flow velocity increment at the y centre section of the Audi Q7

correct direction and are always underestimated. For the Q7, the deterioration in drag was predicted as 77.3 % and for the RS 4 as 91.4 %. The figures for front axle lift were 40.2 % and 60.6 %, and for rear axle lift 80.0 % and 83.9 %, respectively.

An impression of the flow field calculated by the DES method, and which is transient in extensive areas, is shown in ❻ for the Audi Q7 with the airflow through the engine compartment as shown. The turbulent wake structure and the additional turbulence of the flow under the floor as the flow path increases can clearly be seen. These effects are decisive factors for good predictability from the CFD solution.

OUTLOOK

Evaluation of the force coefficients shows that the CFD method as presented here operates in a robust manner and, thanks to its high prediction accuracy, is suitable for the vehicle development process. However, a further improvement in the conformity of the rear lift values would nonetheless be desirable. Further comparisons between wind tunnel measurements and CFD computations will comprise a detailed examination of the influence of the cooling airflow on the vehicles. Other important aspects are the simulation of vehicle wheel rotation and thus of realistic road travel conditions and the representation of wind tunnel geometry, as the flow field boundary is of interest for comparisons with wind tunnel measurements.

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TIRE ECOPIA WITH LOW ROLLING RESISTANCE FOR A BETTER ENVIRONMENTAL COMPATIBILITY

With Ecopia, Bridgestone developed a eco tire for everyday life cars with modified polymer chain ends. The newly selected materials and the innovative pattern technology allowed the tire weight to be reduced by up to 9 %. Depending on the tire model, it was possible to decrease the rolling resistance by up to 14 % which corresponds with a lowering in fuel consumption of approximately 1.7 % – without sacrificing any safety performance.

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ENVIRONMENTAL FRIENDLINESS IS THE TARGET

For the Japanese tire manufacturer Bridgestone [1], founded 1931 by Shojiro Ishibashi, its mission for the environment lies in its promise to help to provide a healthy environment for present and future generations. The motto of the company founder ("Serving Society with Superior Quality") thus still applies today.

There was – and is – a whole series of eco or ecological projects at Bridgestone. Examples for this are the children's Eco-Art Contest, environmental education in schools as the initiative "Make Cars Green" with the international automobile federation FIA and a whole range of other projects. In addition, Bridgestone has made, and continues to make, various contributions to environmental education on all continents, either alone or with other organizations. These include seminars on water conservation in Spain, the cleaning of beaches in Venezuela or plantations of indigenous trees in the USA [2].

With regard to the principal product, tires, the entire lifecycle was subjected to an optimization process with regard to environmental aspects like:

- Tire manufacturing with
  - planting of natural caoutchouc
  - tire designs adapted to environmental concerns
  - optimized manufacturing and logistics
- Use of tires to reduce CO_2 emission with
  - low rolling resistance
  - weight reduction
  - longer period of use
- Process at the end of lifetime with
  - complete overhaul to extend lifetime
  - recycling
  - energy production.

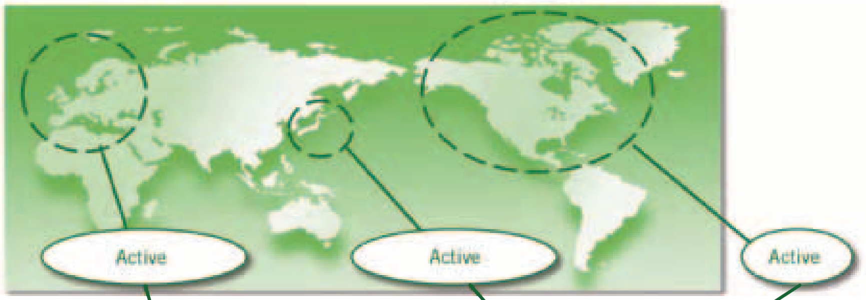
Approximately 84 % of the CO_2 produced here results from the rolling resistance when the tire is in use, so the reduction of the hysteresis losses of the tire rolling under load is the main challenge if one has to improve tires. The entire communication program of the company is accompanied worldwide by education and information measures which contribute towards consolidating the environmentally specific communication process.

As a result of the CO_2 reduction required and demanded worldwide, more and more countries have now introduced new directives for tires. The rules prescribed include reduced rolling resistance values which are to be staggered over the years to come. ❶ shows the directives using the example of the EU, Japan and the USA. Using this background, Bridgestone developed a new eco tire called Ecopia with low rolling resistance which is described in detail in the following.

ECOLOGICAL LINE OF TIRES FOR PASSENGER CARS

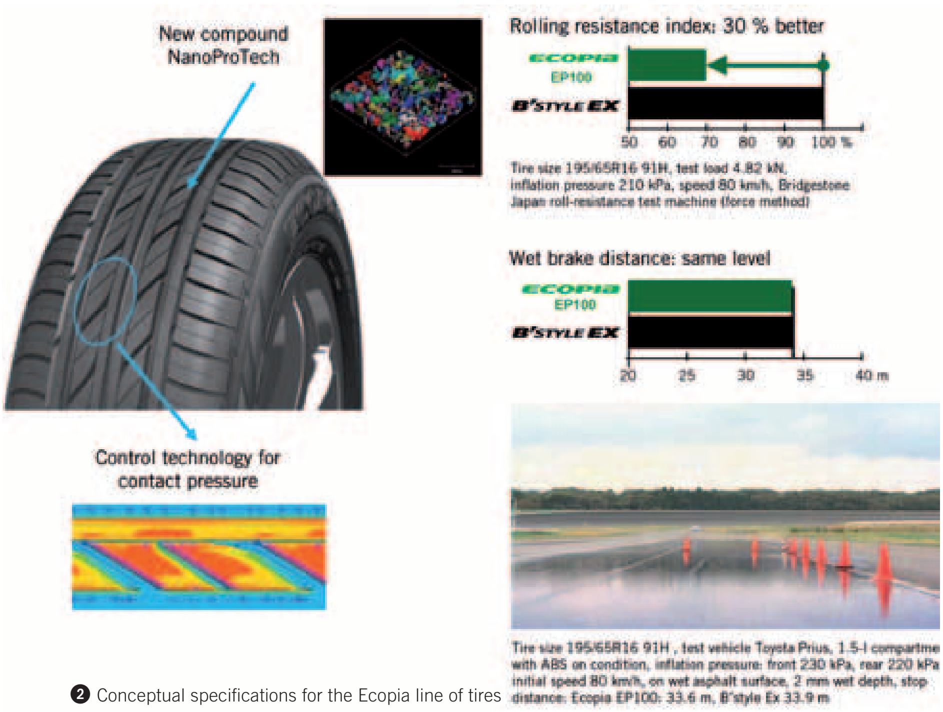
In keeping with the company philosophy of not developing and offering tires which favour certain properties to the detriment of others, for example by neglecting the safety properties, the innovative line of tires were designed. In doing so, challenging goals were set: a high degree of safety during handling, braking, wet performance; environmentally sound properties; low rolling resistance and reduced weight as well as optimal handling characteristics.

❶ Global trend towards the introduction of new tire directives in the EU, in Japan and the USA



REGION		EU	JAPAN	USA
SITUATION		New regulation starts November 2012	Tyre industry started the labeling system voluntarily from 1st January 2010	49 CFR Part 575 (Some items are not decided)
REGULATION	ROLLING RESISTANCE	Maximum limit		
		Grading	Voluntary grading	Grading
	NOISE	Maximum limit/ Grading		
	WET GRIP	Minimum limit		
		Grading	Voluntary grading	Grading
	WEAR			Grading

It has been shown that the wet grip and the rolling resistance are very well correlated with the hysteresis loss properties. For tire compound design the hysteresis loss properties are often of major concern in reducing tire rolling resistance with keeping the wet grip unchanged. It was vital to develop a new tread compound. ❷ shows all requirements at the Ecopia tire of the concept in a nutshell for the Japanese market. The tire combines eco-friendly features such as higher fuel efficiency via a reduced rolling resistance without undermining safety criteria (wet driving and braking).



❷ Conceptual specifications for the Ecopia line of tires

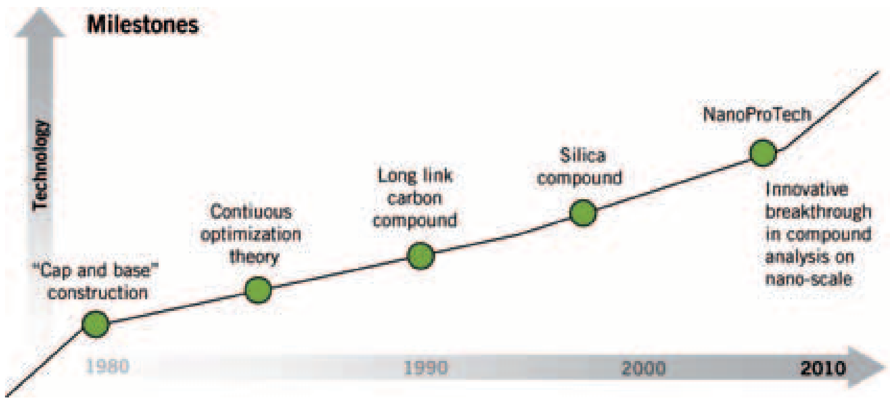
The quest to reduce the rolling resistance has basically been known for many years. The milestones in the history of the last 30 years from the viewpoint of Bridgestone [3] are shown in ❸. The last few years have seen a whole series of trailblazing innovations, from "cap and base" tread designs to silica tire compounds and the technology NanoProTech. Summing up, one uses state-of-the-art technology for the Ecopia tire – in terms of structural design, tread design and materials.

Throughout the development of this new line of tires, the focus at all times was on producing a tire which combines maximum wet grip with a low rolling resistance and dynamic handling characteristics:

: excellent tire compound based on the technology NanoProTech; it reduces hysteresis losses and saves energy
: special block and rib design on outside tread
: weight-reduced tire design – quiet, comfortable and economical in use.

TIRE TECHNOLOGY NANOPROTECH

❹ shows what NanoProTech innovative tire technology is all about: the controlled interaction between polymers, filler materials and other rubber chemicals used in tire development. This allows the nano-scale



❸ Milestones in innovative technologies by Bridgestone for reducing rolling resistance and improving wet grip

layout to be optimized; components which generate losses can be removed, and the polymers optimized in this way ultimately show lower hysteresis losses [4].

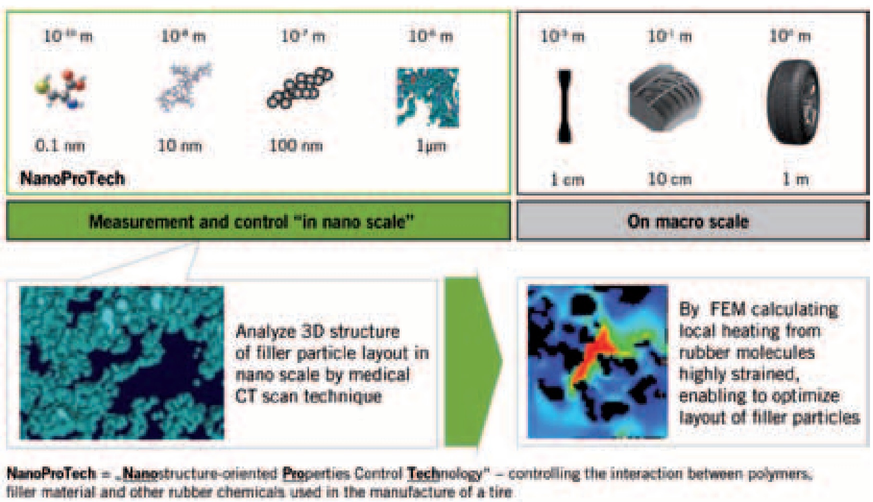
With the conventional polymer, the clustering of filler particles causes the material to heat up in places as well as generating energy losses. In contrast, the Ecopia polymer reduces losses by using modified polymer chain ends. This allows the distribution of filler particles and the friction to be improved considerably.

OUTSIDE TREAD DESIGN

The outside tread design can be characterized in five points:

1. The shoulder blocks are connected with the adjacent circumferential grooves via "3D cut design". This results into higher block stiffness for better handling performance, especially at cornering.
2. Connected block to thin rib for a more even contact pressure distribution, achieving higher braking force and grip in wet.
3. Drainage of excessive water from rib area into main grooves for increased resistance to hydroplaning. The central lugs are attached at high angles (around 60°) and assist the braking ability and ensure a good distribution of pressure. They reduce the tread noise.
4. A solid circumferential rib design (flat contact crown radius shape) helps to minimize block deformations and thus prevent irregular wear.
5. A convex block design for Adaptive Contact Surface (ACS) to reduce air volume velocity, resulting into lower pattern noise emission

❺ and ❻ sum up the main points for designing tread and pattern of the Ecopia tire.



❹ NanoProTech – molecules in nano and macro scale

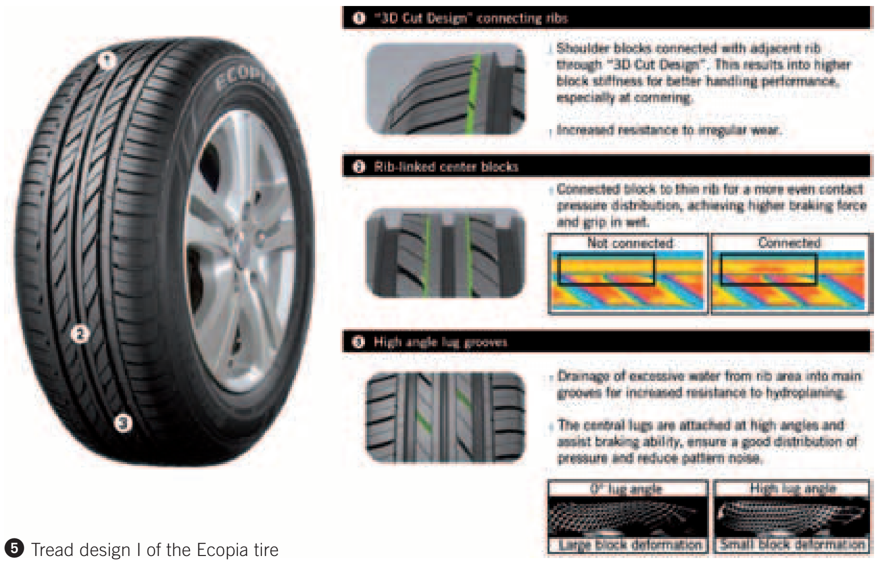
ACHIEVED PRODUCT IMPROVEMENTS

Finally, the following will describe and assess the product improvements achieved in the Ecopia tire line in comparison with a predecessor tire with regard to all relevant criteria like safety, driving dynamics and handling, weight, rolling resistance, comfort and lifetime. Safety is examined using the example of braking on a wet road. The Ecopia EP150 eco tire allows 5 % shorter braking distances from 80 km/h to a standstill than its predecessor B250, that means a reduction of 1.5 to 2 m.

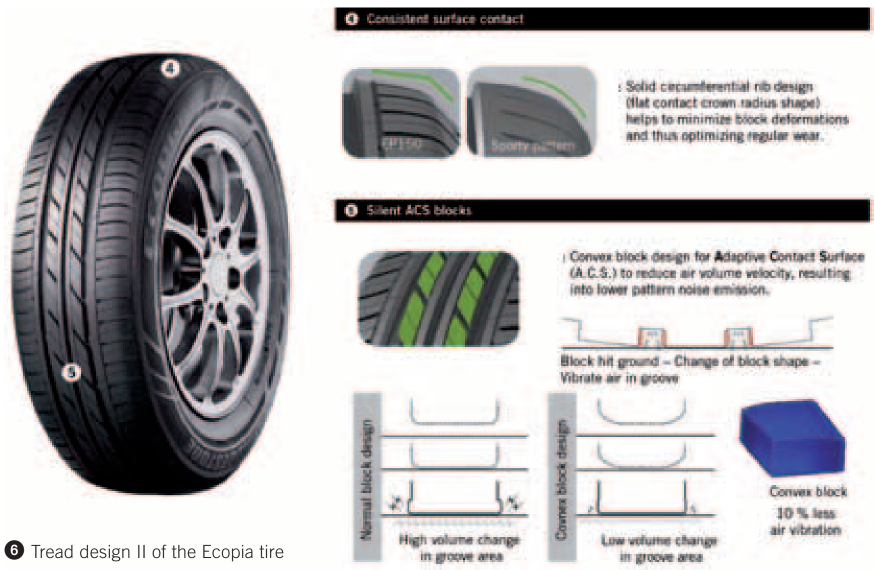
In terms of dry handling, Ecopia is in the same class as its predecessor, and in wet handling it is one point better. Driving precision and Driving pleasure are not adversely affected in any way. The newly selected materials and the innovative technologies allowed the tire weight to be reduced by up to 9 %. Depending on the tire model, it was possible to reduce the rolling resistance by up to 14 %. This corresponds with a decrease in consumption of approximately 1.7 %, thus making a contribution to CO_2 reduction. Like dry handling, it was possible to keep continuous-travel comfort on the same high level as that of the predecessor, and there was an improvement of up to 15 % with regard to tire mileage and lifetime. The on the whole convincing results are shown in diagram form in ❼.

ADVANTAGES AND OUTLOOK

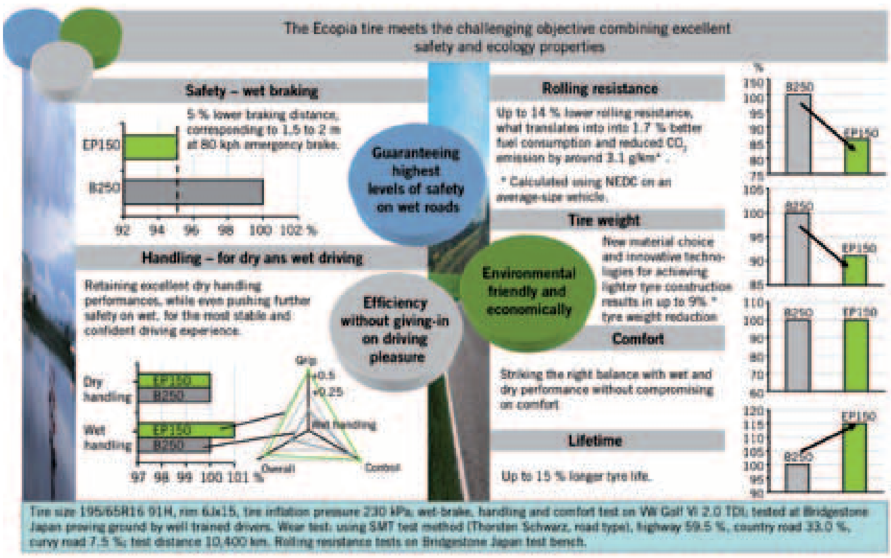
The Bridgestone Ecopia EP150 concept of an eco tire is a convincing proposition due to its well-balanced tire characteristics while showing clear advantages with regard to rolling resistance, noise and tire mileage. It is now in the worldwide field



❺ Tread design I of the Ecopia tire



❻ Tread design II of the Ecopia tire



❼ Development of tire characteristics – the eco tire Ecopia EP150 in comparison to its predecessor B250

in various characteristics and tread lines and is also in widespread use in tires for trucks and buses.

Pneumatic tires on motor vehicles are as old as motor vehicles themselves. Like the vehicles, however, they have changed and developed considerably over the last 125 years. Bias tires have been completely replaced worldwide by steel radial tires. There has been a dramatic improvement in tire safety, that means braking performance and handling, and their uniformity as well as their sensitivity to damage have also improved to a remarkable degree. In addition, run-flat tires were successfully introduced in the 1980s with the intensive cooperation of Bridgestone. They were constantly developed as a own segment because they allow the spare wheel to be dispensed with completely.

The beginning of the age of electric mobility with its completely innovative vehicle concepts, in particular for the emission-free megacity field of the future, also confronts tires with new challenges. Low to extremely low rolling resistance has already been mentioned here; the diameter/width ratio and the designing of the tire performance as a whole still have to be developed and consolidated in a constant dialog with the vehicle industry. From today's point of view, all that is certain is that, for the next ten to twenty years at least, pneumatic tires will be showing no signs of falling into disuse.

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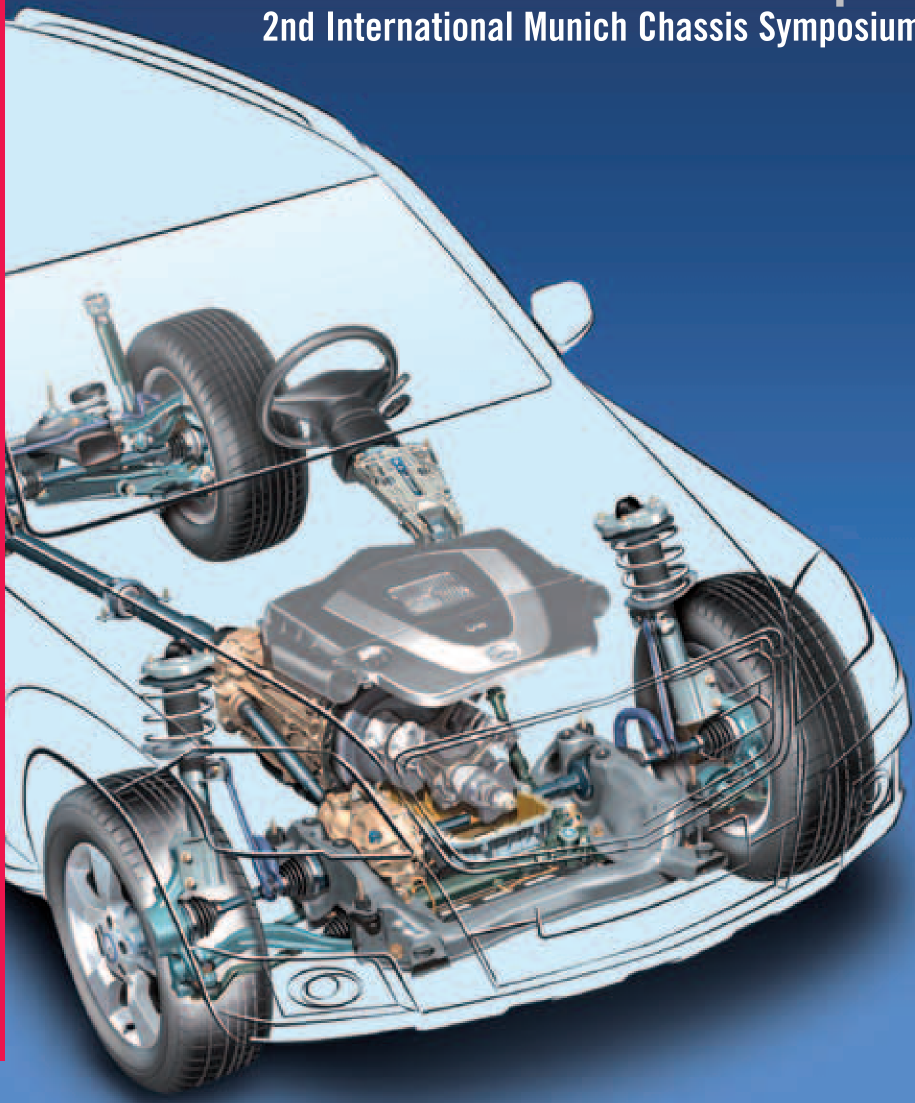

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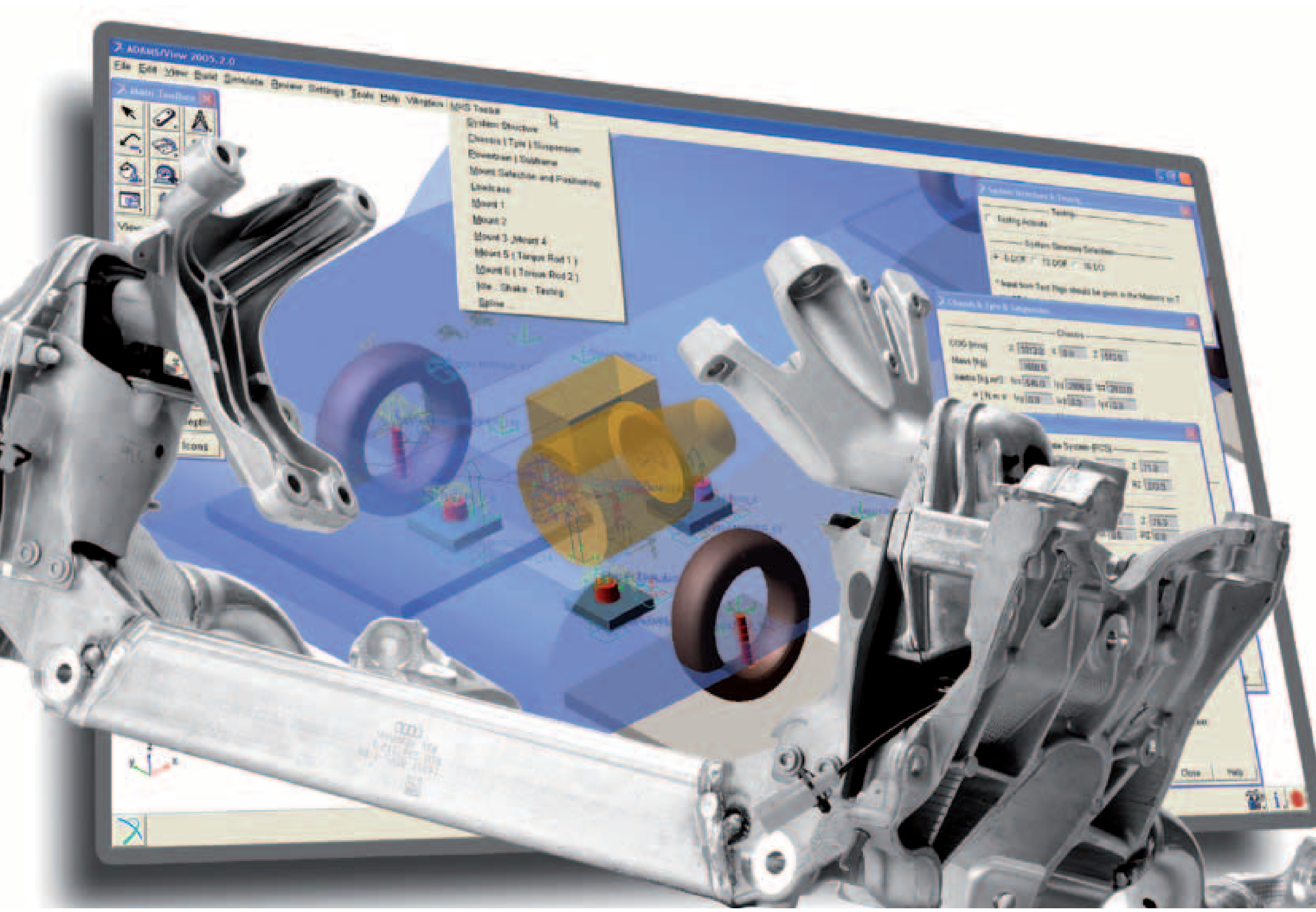
NVH SIMULATION OF ENGINE MOUNTS VERIFIED IN VEHICLE TESTING

In computer-aided simulation of vibration and noise generated by the powertrain, it is important to describe the properties of the individual rubber-metal mounts in a mathematically precise manner. In the case of switchable hydraulically damping engine mounts in particular, the properties vary significantly not only with the frequency and excitation amplitude but also with the respective switching state. In cooperation with the RheinMain University of Applied Sciences, ZF Lemförder was able to validate powertrain suspension systems calculated by the application of multi-body models in vehicle tests.



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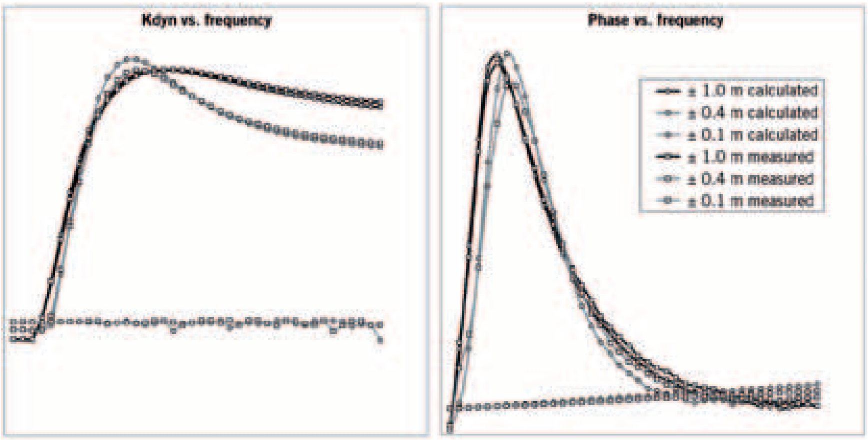
TASK

The abbreviation NVH (noise, vibration and harshness) stands for acoustic and mechanical vibrations and their subjective perception. Since noise and vibrations are generated by external vibration sources such as the road surface, chassis suspension and powertrain, the transmission and introduction of such vibrations to the passenger compartment can be prevented using appropriate components. In order to achieve an optimum NVH result for the suspension system, properties such as noise insulation and vibration control as well as natural frequencies are pre-calculated in a multi-body simulation. The validated multi-body models used as a basis for pre-calculation take into account the extensive expertise and knowledge of a manufacturer regarding the complex properties of conventional and hydraulically damping elastomeric mounts. Optimisation and improvement measures may thus be adopted swiftly at a very early stage due to an uninterrupted development chain ranging from simulation and testing of an individual mount and a single component right through to the complete system, thus reducing development time and cost. A solid foundation can be established in particular for new drive technologies concerning a reduction in CO_2 emissions that come along with engine downsizing and lightweight construction, which in turn leads to even greater vibration and noise-related problems.

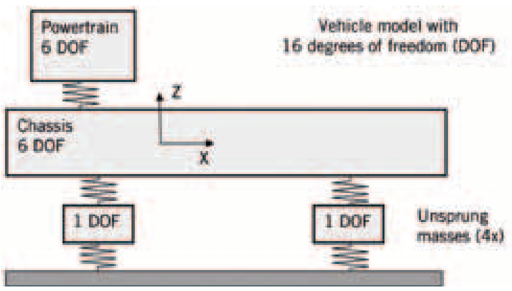
The requirements for engine mounts are usually contradictory. On the one hand, high stiffness is required in order to bear loads of the powertrain resulting from curb weight, engine torque and travel. On the other hand, low stiffness can help to insulate structure-borne noise. Then again, increased damping is required when it comes to decoupling natural vibrations such as engine shake to ensure a more comfortable ride. Such natural vibrations lead to annoying resonance effects, so that an overlap of the natural frequencies of the powertrain mount system with the excitation and natural frequencies of other vehicle components is to be avoided. In addition, engine mounts must ensure service strength, in other words they should withstand large deflections over a vehicle lifetime without damage.

MODELLING OF ENGINE MOUNTS

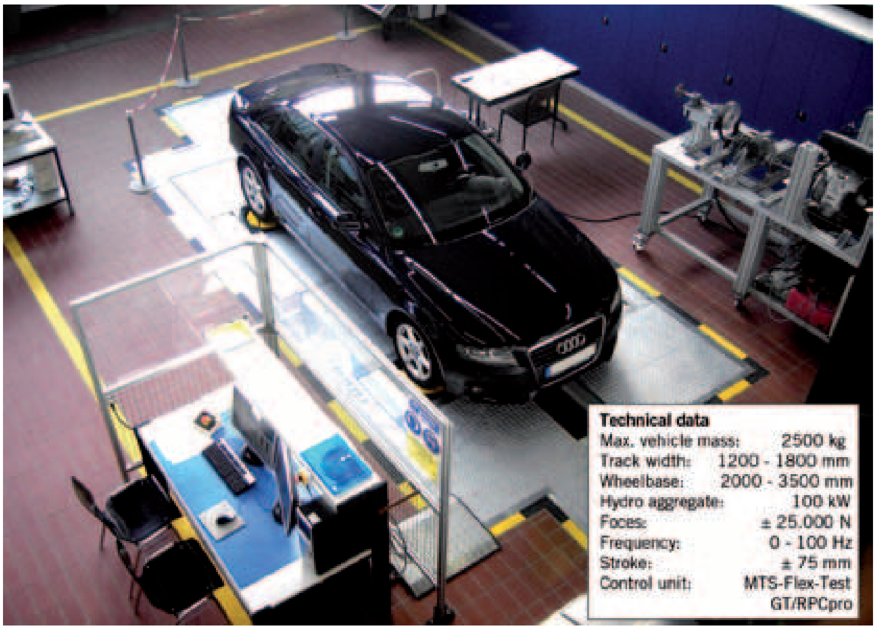
To cover all these different conditions and in order to find an optimum compromise, maximum optimisation effort is required. In this connection, simulation plays an increasingly important role. A mathematical description of all properties of the elastomeric mount that is as precise as possible is a fundamental prerequisite for computing the design and calculated dimensioning of engine mount systems. Across the entire range from conventional mounts through to active hydraulic damping mounts, their properties, such as transfer stiffness and damping, are non-linearly dependent on the load, excitation frequency and amplitude. With the support of the central research and development department of ZF in Friedrichshafen, tailor-made numerical mount models were developed. The calculation facilities at the RheinMain University, Rüsselsheim, also play an important role in this. When describing elastomeric mounts, first of all the non-linear, frequency and amplitude-dependent stiffness and the damping of the basic material rubber itself must be taken into account. In the case of hydraulically damping engine mounts, the resilience of the main spring component and decoupling system compared with the hydraulic internal pressure must also be considered. In addition, the fluid mechanical conditions of the liquid column in the orifice channel system must be calculated. Depending on the frequency range, this acts as a vibration absorber (mass damper), is in resonance or is even left undisturbed. The more complex the design of the mount, the more properties of the individual components must be modelled. The individual components also influence each other. Experience shows that only the rubber-metal part manufacturer fully understands these correlations, which can be changed sen-



❶ Dynamic stiffness and loss angle of an engine mount in relation to the frequency at different excitation amplitudes



❷ Multi-body model with 16 degrees of freedom for designing a powertrain suspension system



❸ RheinMain University's roadway simulator at the Rüsselsheim Campus

sibly in optimisation runs. In order to validate these numerical models of the mount, the parameters used as a basis must be adjusted in such a way that the mount characteristics calculated conform to the mount characteristics measured. ❶ shows a comparison of the measured and calculated dynamic stiffness and damping in relation to the frequency of a hydraulically damping engine mount.

ENGINE MOUNT SYSTEM

The sub-models of the mounts thus created, including the parameters determined, are then used in the multi-body model of the vehicle to pre-calculate properties such as isolation and damping, and also natural frequencies with the aim of ensuring an optimum NVH result of the complete engine mount system. In order to be able to simulate the effect of engine mounts in the low-frequency range for design purposes, simple rigid body models which represent the full vehicle are used. The powertrain, car body and wheels are represented as rigid bodies that are interconnected by elastic elements for the wheels, chassis and mount. The mounts are, however, not simple spring-damper elements, but are described by the above-mentioned numerical mount models. ❷ shows a simple multi-body model with a powertrain suspension mount.

In the subsequent optimisation process, the characteristics of the engine mounts are tuned iteratively, taking into account all interdependencies, until the development objective has been achieved. The design finally results in static and dynamic stiffness characteristics of the individual powertrain suspension mounts in all directions in space with a linear range and jounce and bounce progression that meet all powertrain suspension system requirements. In the following design stage, a mount design is derived from these characteristics. The intended mount characteristics are in turn confirmed in the subsequent FE analysis. In the event of a deviation, the design needs to be further optimised until synchronisation with the required characteristics is achieved. The hydraulic damping characteristics of the engine mounts are also determined by FE analysis. From this, the complete dynamic characteristics can be calculated.

VALIDATION IN VEHICLE TESTS

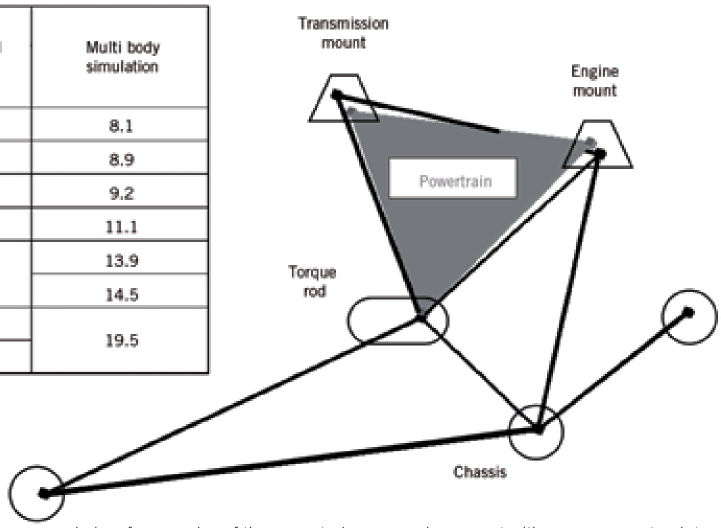
As described in the following, this model was used to carry out various calculations. At the RheinMain University, relevant tests with different vehicles were conducted on a newly developed four-poster hydropulse test stand, ❸, to verify the calculation models.

A first impression of the vibration characteristics of an engine mount system is provided by modal analysis. Whereas this analysis can be carried out easily while applying real boundary conditions in simulation, it is, however, much more difficult in the test to obtain information on modal natural vibrations. In this particular case, the vehicles were excited via the wheels by different signals on the hydropulse test stand. In the vehicle, the accelerations on the engine mounts were measured at the engine side and the body, as well as at different positions of the body, such as at the seat rail. With the aid of an operational modal analysis of the measured signals, eigenfrequencies and mode shapes of the powertrain were determined. ❹ shows a comparison of the resultant eigenfrequencies from calculation and measurement, which largely match in the case of the transversely installed engine shown. Such correlation can also be found in the case of a longitudinally installed engine. This clearly demonstrates that the calculation models used actually correspond to the facts in reality. ④ also shows the measurement positions in the vehicle, the motion of which can be animated for analysis of the natural frequency modes.

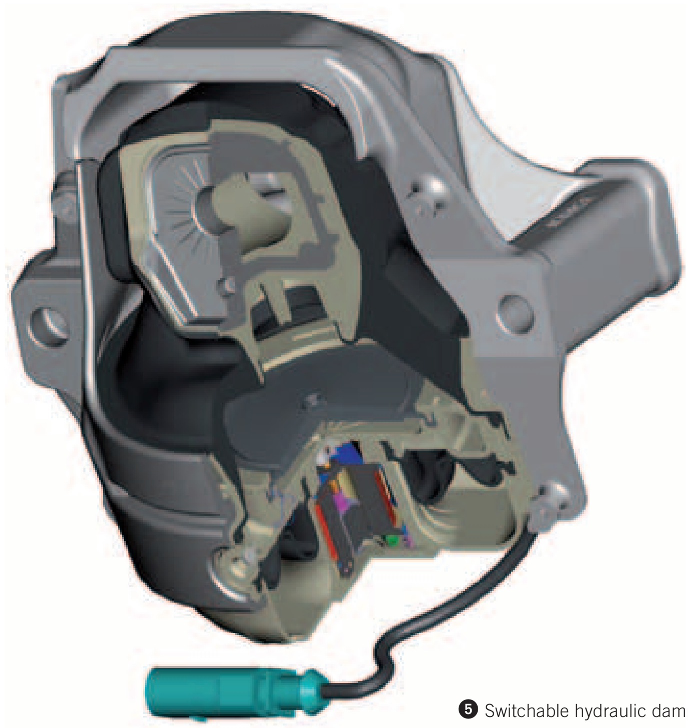
While the modal analysis provides information on the vibrations of the engine block (eigenfrequency) to be attenuated, the damping effect of the mounts is represented by the transmission of vibrations to the chassis. The resultant vibrations on the vehicle floor close to the occupants, which are normally measured at the seat rail, are the measure of this transfer. Vibrations of the unit are transmitted via the mount to the chassis, causing perceptible accelerations at particular frequencies. The aim of a good NVH design is to minimise such vibrations over a broad frequency range. An optimisation algorithm can be used in simulation to adjust the stiffness and damping of the mounts in such a way that this aim is achieved. The effect of hydraulic damping that is precisely tuned to a required frequency point was demonstrated by a switchable engine mount from ZF Lemförder Boge Rubber & Plastics, ❺. For this purpose, the mount was manually switched, contrary to normal operation. The stochastic excitation on the hydropulse test stand generated engine block vibrations, the accelerations of which were evaluated by a fast Fourier transformation in the frequency range of the mount engine and chassis side. ❻ shows the decrease in acceleration due to higher hydraulic damping.

Natural frequencies are, however, only one part of the NVH design of an engine mount system. In addition to damping, another important task of engine suspension mounts is to isolate vibration and noise generated by the powertrain. In the following example, this isolating effect is analyzed when the engine is idling. Also in this case, the switchable engine mount was manipulated to allow two different transmission stiffnesses to be examined, since in particular when the engine is idling, the mounts automatically switch to soft characteristics to increase the isolating effect, whereas when the vehicle is in operation, these

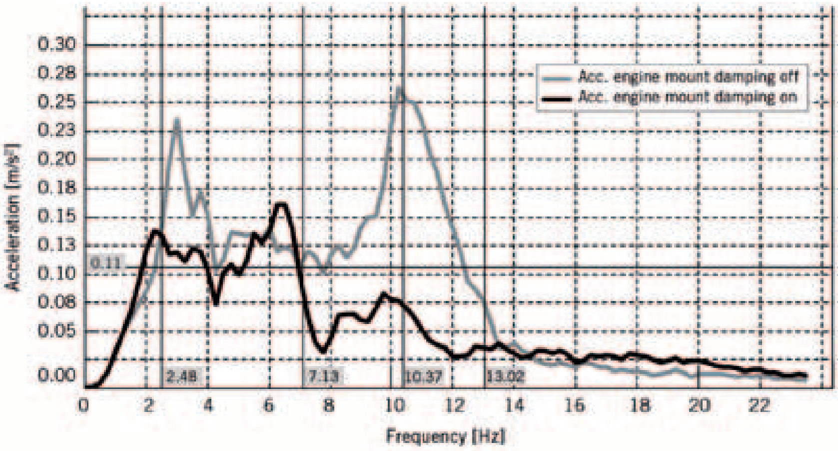
Operational modal analysis	Multi body simulation
7.2	8.1
7.8	8.9
9.7	9.2
11.4	11.1
14	13.9
	14.5
17.1	19.5
19.1	



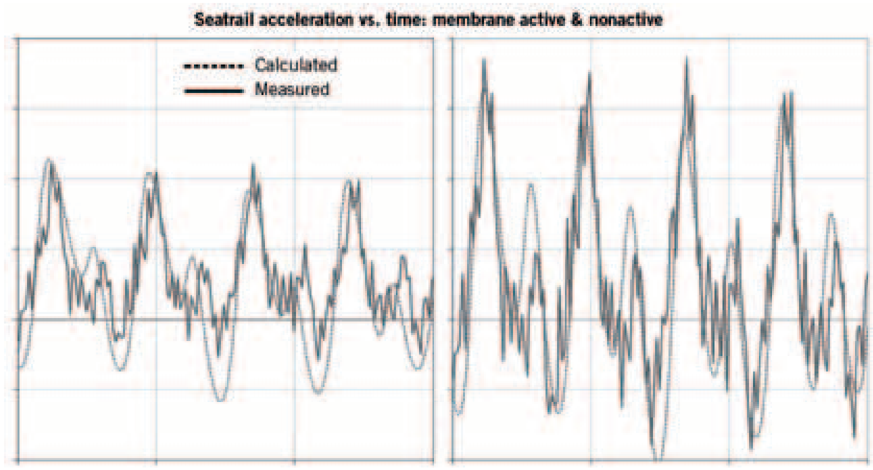
❹ Calculated and measured eigenfrequencies of the powertrain suspension mount with measurement points shown in grey at the unit and in black at the chassis



❺ Switchable hydraulic damping engine mount



❻ Comparison undamped (grey) and damped acceleration measured at the engine mount



❼ Comparison of measured and calculated vibration at the seat rail when the engine is idling with high and low stiffness

mounts automatically switch back to a higher dynamic stiffness to ensure damping and thus provide a more comfortable ride. ❼ shows the response of the vibrations generated by gas forces and inertia forces in the engine by the position of the seat rail, each measured and calculated respectively. As expected, the two mount states with the different transfer stiffnesses lead to different amplitudes. What is also particularly interesting is a good correlation of the amplitudes between calculation and measurements, which argues in favour of a well verified calculation model.

CONCLUSION

Experiments in vehicle testing have shown that an engine mount system can be designed and optimised early in the simulation phase by a fine and well-tuned calculation method. The calculation models were verified and will also be used in the future with a higher level of detail in more complex full-vehicle models. Thus, the simulation is integrated into a closed optimisation loop when it comes to the design of engine mount systems. It is a useful tool for gaining a better understanding of influences and developing new optimisation potentials. However, the road test remains an irreplaceable component of validation, since it additionally also provides a subjective impression. The main benefit for the car manufacturer is that the entire expertise and knowhow of elastomeric mounts is taken into account in the design of engine mount systems.

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SYSTEMATIC INVESTIGATION ON FUEL SLOSH NOISES

Fuel tanks belong to a group of components which are rather unobtrusive in an automobile. With the increasing application of start-stop systems and the on-going hybridisation of drive train fuel slosh noises may be perceived better in the passengers' compartment. Therefore the Institut für Kraftfahrzeuge (ika) of RWTH Aachen University investigates fuel slosh noises and develops methods for simulation based analysis and optimisation of fuel tanks as well as their vehicle integration.

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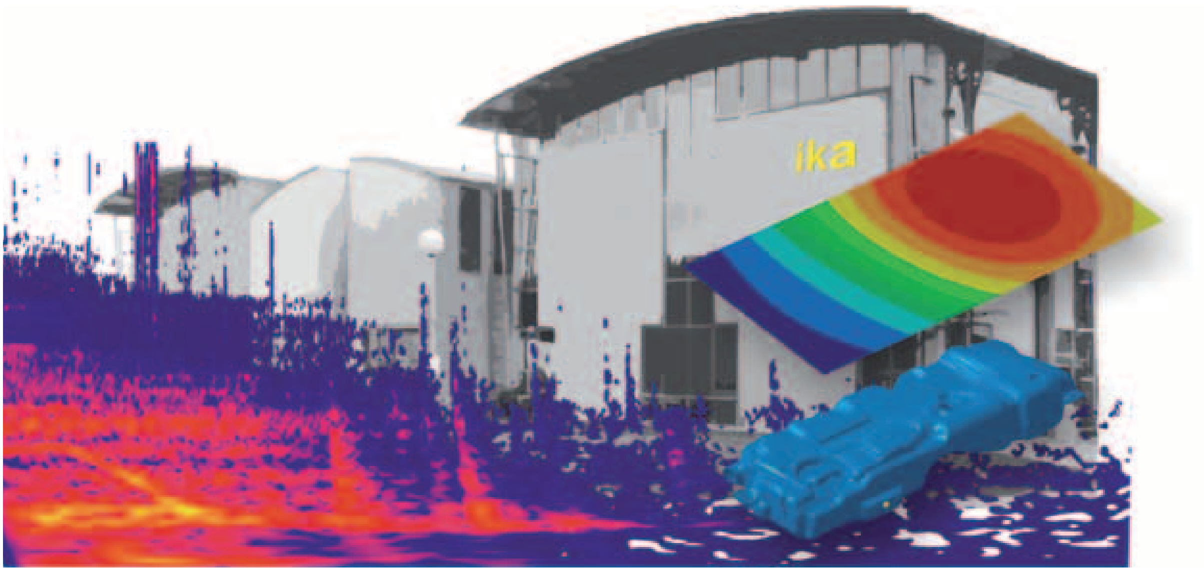


1 INTRODUCTION

Today an automobile has to fulfil several expectations regarding its attributes. Especially for automobiles of the premium segment the optimisation of NVH properties (noise, vibration and harshness) is in the focus of the development. Quickly changing economical and societal boundary conditions lead to shorter and shorter development periods and to a remarkable increase of platform derivates with a high number of shared parts at the same time. Thus, the automotive research aims for appropriate simulation tools and processes. With such tools it becomes possible to state reliable results regarding the NVH behaviour of an automobile in an early stage of development.

The proceeding of hybridisation and the equipment of vehicles with start-stop systems is a major and still ongoing trend in automotive industry. In the lower speed range and during stops the combustion engine is switched off in both concepts. Thus, its masking noise emissions are lacking. Additionally, in these operating states aerodynamic as well as tire-road noises are not relevant. In order to show the effect of hybridisation, a hybrid prototype vehicle built up at the Institut für Kraftfahrzeuge (ika) of RWTH Aachen University is compared regarding the interior sound pressure level in electric and conventional drive, ❶.

The difference between both curves is about 3 to 5 dB up to a speed of 20 km/h. Due to an absence of the combustion engine noises masked sound sources, i.e. the corresponding components and aggregates, come forward. One of these emerging sound sources is the fuel sloshing in the tank. This especially applies to sources of low frequency sound, for example the fuel sloshing in the tank.

2 APPROACH OF INVESTIGATION

In the developed approach of investigation the acoustic phenomenon “fuel slosh noise” shall be analysed holistically and systematically. Initially it starts with an in-depth investigation of the phenomenon in driving manoeuvres, in order to gain design parameters for an appropriate test bench and to derivate detailed component investigations. The results also will be used to validate simulation approaches and models. In a concluding step measures to reduce or avoid noise generation will be developed systematically basing on simulations. The investigation regards to the slosh events in the inner tank, to dynamic properties of the tank structure and to the sound transfer in the passengers compartment.

2.1 INVESTIGATION OF FLUID DYNAMIC EVENTS

Mainly, fuel slosh noises are perceived during or after braking manoeuvres during stops and slow driving. Before the braking at constant speed the fuel is in its equilibrium. When the vehicle is decelerating, the fuel piles up at the tank front due to its inertia. Potential energy is saved as in a compressed spring. When the deceleration becomes smaller, the potential energy is converted to kinetic energy and the fuel flows to the backside of the tank. After initial deflection this process reiterates until the energy is dissipated. This motion of liquid is called sloshing. In case of wave fronts hitting the tank wall or each other, slosh noises are generated [1].

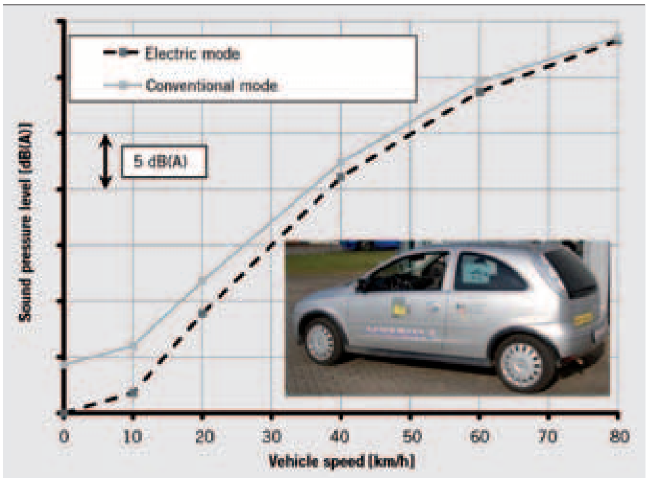
Baffles and special foams are well known counter measures. Baffles affect a resistance to the lengthwise liquid flow in the tank. Foams can be attached to the inner side of the tank wall in order to disperse the wave fronts and to attenuate their impact.

Especially automobiles with hybrid or start-stop systems provide acoustic boundary conditions, which forward the perception of slosh noise. Hence, a start-stop system is simulated by switching off the combustion engine during braking. ❷ shows a corresponding wavelet analysis of the interior noise measured on the driver seat. The single impulsive and low frequency slosh noises can remarkably be identified. In this particular example the slosh noises can be perceived until approximately 12.5 s after the vehicle stopped.

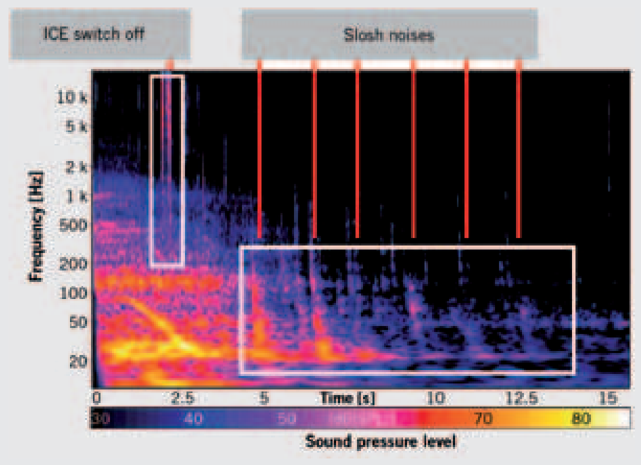
For reproduction of the relevant manoeuvres in a semi-anechoic chamber without disturbing background noise a special test bench is designed and built up [2]. The test bench concept has to fulfil the following criteria:

- reproduction of longitudinal motions analogue to the test drives
- one-dimensional excitation and motion of the tank
- capability to rebuild the contact points of tank and vehicle body
- enhanced accessibility of the tank
- engagement of an acoustic camera to locate sound sources
- programmable drive to enhance the repeatability of the excitation.

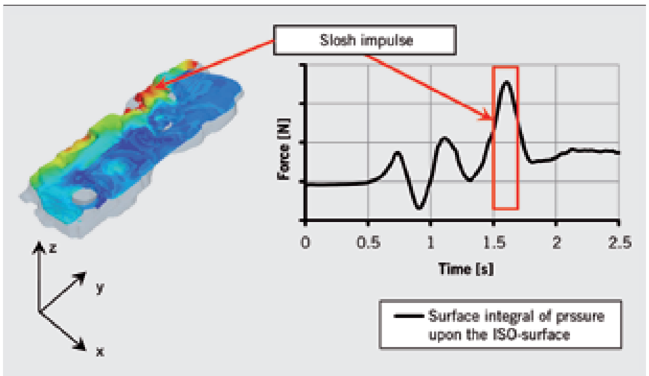
The test bench corresponding to these requirements possesses a sled, which the tank can be mounted on. Besides the manual excitation the sled can be accelerated by an electro-mechanical motor. Thus, a higher repeatability of the excitation can be achieved. With the test bench it is capable to determine relevant interface



❶ Interior sound pressure levels for electric and conventional drive



❷ Sloshing noises in the passengers' compartment



❸ Simulation and visualisation of sloshing

values between tank and vehicle body as sound pressure and accelerations of the tank wall.

Furthermore, the fluid dynamic events are numerically simulated by using CFD methods. Based on CAD data a geometrical model of the tank is generated. That means the inner of the tank is segmented by numerous single cells. In the following step the physical modelling is set up. Besides several other aspects, the modelling of the separation of gas and liquid phase has to be mentioned in particular. In the current approach the interface of multiphase flow is modelled by the volume of fluid method [2, 3]. With this approach the free surface or rather the border of the liquid phase can be calculated. This interface between the two phases is also called ISO surface. Altogether it can be stated, that the numeric results of the CFD simulations represent only approximate solutions. Besides the approximations of the geometrical modelling, the physical modelling bases on differential equations which implicate several simplifications. Additionally, the equation system is solved iteratively.

By now there is no direct way to determine acoustic values as sound pressure for slosh noise simulation. Furthermore, for a sound pressure level of 80 dB the sound pressure is just 0.2 Pa. This share can partially be overlaid with the error of the simulation. Thus, feasible indicators have to be developed and used, which provide at least a qualitative statement regarding the acoustics, in order to compare different tank versions. ❸ shows an exemplarily picture of a tank model with visualised pressure distribution upon the ISO surface at the moment of the slosh.

Besides the run of a force, which is calculated by a surface integration of the pressure upon the ISO surface, the corresponding pressure contribution is represented on the left side. The red areas represent areas of high pressure.

2.2 INVESTIGATION ON THE STRUCTURAL PROPERTIES OF FUEL TANKS

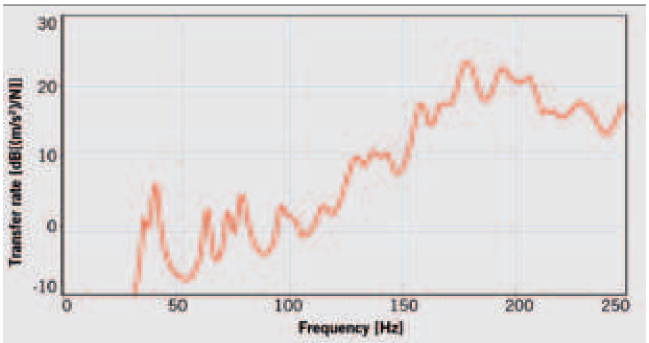
Because of the impulsive excitation of the tank wall airborne sound is radiated und structure borne sound is led in the vehicle body via the holding straps. If the tank ceiling lies against the vehicle body, each of such contact points is further transfer paths. Thus, the tank wall is a relevant element of the transfer path from noise generation to perception.

PE tanks are manufactured via the extrusion blow moulding process [4]. Hereby, a thermoplastic cast in a viscid state is inserted to a half shell mould. The mould is closed and pressure is built up by a blow pin, in order to press the cast against the mould wall. After a cooling period the parison is removed from the mould and burs are removed.

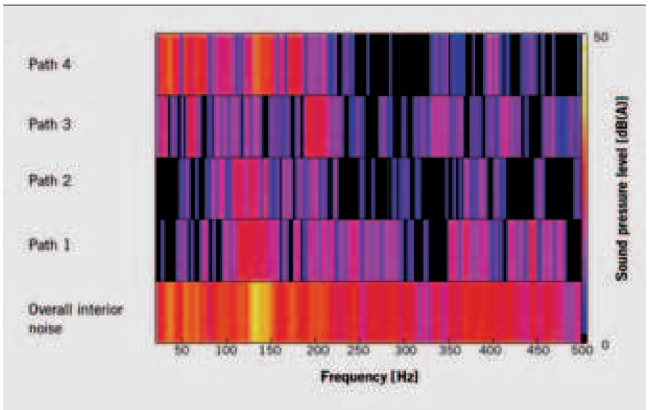
Starting from the outside of the tank a multilayer tank wall consists of soot-blackened PE-HD, an adhesive promoter layer, a thin layer of ethylene-vinyl alcohol copolymer resin, a further adhesive promoter and a milky-white inner layer of PE. The ethylene-vinyl alcohol copolymer resin layer reduces molecular emissions. PE-HD is a weakly branched semi-crystalline thermoplast. Its chain-like molecular structures are only connected by cross bridges at a few points. Instead of steel the dynamic behaviour of thermoplasts does not fulfil the requirements of linear material characteristics. Magnitude and duration of loads have an important influence [5].

In order to determine the structural behaviour an experimental modal analysis of a fuel tank is performed. ❹ demonstrates the result of this measurement. The single transfer functions are averaged to one curve.

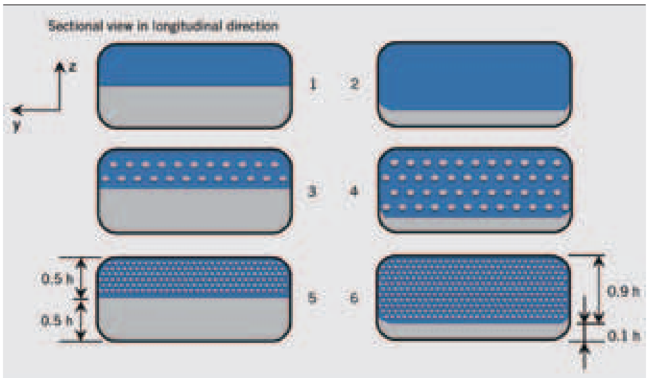
It can be remarkably noticed, that there are peaks in the run of the curve below 200 Hz. In this frequency range also characteristic peaks of slosh noise occur. Following investigations aim to clear, with which structural measures the peaks can be attenuated and which effect this will have on the slosh noise in the complete system "vehicle". Therefore, initially a FE model of the tank is validated with the transfer functions determined in the experimental modal analysis.



❹ Result of the structural investigation



❺ Transfer path analysis for a slosh noise



❻ Overview of baffle variations

2.3 INVESTIGATION ON THE TRANSFER PATHS FROM THE TANK TO THE PASSENGERS' COMPARTMENT

By utilising the transfer path analysis a common analysis tool is adopted, in order to determine, which shares structure borne and airborne sound transfer paths possess on the slosh noise in the passengers compartment [6]. For that, the transfer functions of a vehicle from relevant excitation points at the tank to an artificial head measurement system as well as the inertances of the excitation points are measured.

The information regarding the transfer behaviour of the vehicle is combined with measurement data from test drives. Using an analysis tool the interior sound pressure level generated by each excitation point can be calculated. ❺ shows the resulting interior sound pressure level for structure borne sound excitation. The share transferred via airborne sound can be neglected in this case and according results are not shown here.

In the bottom row the frequency spectrum of the interior noise is shown. Furthermore, the figure demonstrates which share of the interior noise each is transferred via the single transfer paths. It can easily be seen, that the slosh noise is mainly transferred into the passengers compartment via path 4. In the following it has to be investigated, which noise reduction can be achieved, when the tank is decoupled from the vehicle body by elastomer elements.

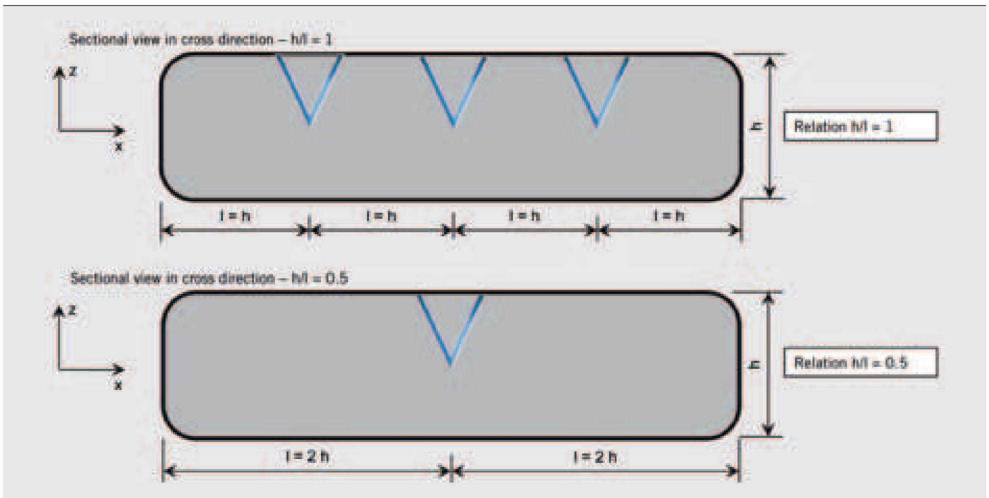
3 DEVELOPMENT OF NOISE REDUCING COUNTER MEASURES

After the analysis of noise generation and propagation on the base of test and simulation results measures to reduce slosh noise are developed. The different variations are compared by simulation results. Measures to influence the fluid dynamic events, structural measures to influence the vibrational behaviour of the tank wall and measures for an appropriate vehicle integration are discussed.

3.1 MEASURES IN THE INNER TANK

❻ exemplarily shows different baffle variations, which are integrated in a fuel tank geometry within a CFD model. The baffles on the left hand side screen the tank cross section by 50 %, baffles on the right hand side screen the tank cross section by 90 %. Furthermore, the variations differ by number and size of holes, which avoid hard and noisy impacts of wave fronts.

Besides the design of the baffles their position is varied, too. Especially the segmentation of the tank by the baffles and the resulting relation between chamber length and height has an influence on the sloshing [8]. In the current case the relations of h/l = 0.5 and h/l=1 are differed, ❼.



❼ Positioning of the baffles

		BAFFLE DESIGN					
	Relation h/l	Variant 1	Variant 2	Variant 3	Variant 4	Variant 5	Variant 6
Filling level 1	1	∘	+	–	∘	++	++
	0,5	––	–	–	––	++	++
Filling level 2 & 3	1					+	++
	0.5						+
Filling level 3 & 4	1						++

❽ Comparison of the tank variations

VARIANT	6.5 mm	8 mm	10 mm	15 mm
5 %	–8.3 dB(A)	–9.8 dB(A)	–6.8 dB(A)	–4.2 dB(A)
10 %	–5.9 dB(A)	–7 dB(A)	–9 dB(A)	–15.3 dB(A)
30 %	–8.1 dB(A)	–2.4 dB(A)	–6.5 dB(A)	–15.3 dB(A)
50 %	–8 dB(A)	–3.4 dB(A)	–8.4 dB(A)	4 dB(A)
	> –2.5 dB(A)	–2.5 ≥ –7 dB(A)	–7 ≥ x > –9 dB(A)	≤ –9 dB(A)

❾ Differences of sound pressure level at 132 Hz

The tank models with a relation h/l=1 consequently possess three baffles. As CFD simulations even today require a high computing performance, not all relevant filling levels for all variations are simulated here. The variations are rated by the following calculated values:

- surface integral of pressure upon the ISO surface
- derivation of the run mentioned above
- run of pressures at virtual measuring points
- derivation of the run mentioned above.

The maximum of each rating value is determined. The results of each tank variation are rated by these values. Five filling levels are investigated in total. ❽ demonstrates the results of this CAE-based investigation in a table.

All ratings are related to the original state tank without any measures. As expected, variation 6 provides the best result. It is remarkable that some variations show even worse results although baffles are implemented. It also means that not every measure leads to a better result and that the simulation helps to derivate and rate effective measures.

3.2 MEASURES ON THE TANK STRUCTURE

In the following FE models are set up which have optimised wall thickness distributions. Then the sound radiation of the models is calculated. The results are compared with the original state tank. Utilising a structural optimisation the following variations are analysed:

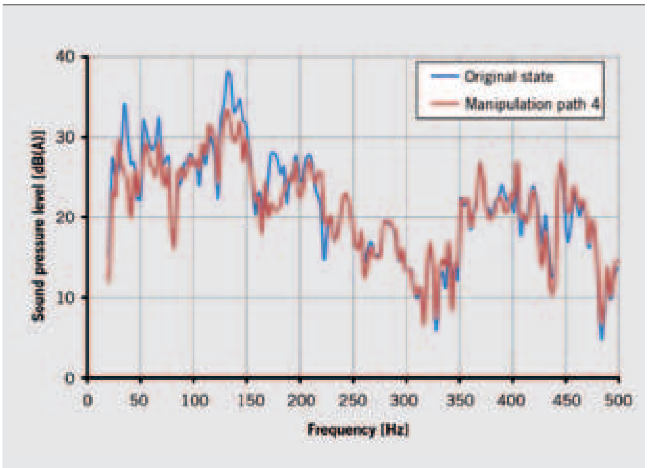
- For the optimisation a volume increase of the tank wall of 5, 10, 30 and 50 % is allowed.
- For the optimisation a maximal wall thickness of 6.5 mm, 8 mm 10 mm and 15 mm is allowed.

In the current investigation this leads to finally 16 variations, which are compared regarding their sound radiation around 132 Hz, ❾. By this it will be considered, that some resonance peaks are slightly moved in the frequency range due to the structural modifications.

It can be noticed, that the highest level reduction can be achieved with a maximal allowed volume increase of 10 % and a maximal allowed wall thickness of 15 mm.

3.3 ADAPTION OF THE VEHICLE INTEGRATION

The transfer path analysis provided the result that path 4 is mainly responsible for the transfer of the slosh noise into the passengers compartment. In the following it will be shown, which effect the manipulation of this single path has on the interior noise. Manipulation here means that the holding strap is decoupled from the vehicle body with elastomer elements. In ❿ the results for the original state and for the variation with a manipulated path 4 are demonstrated. The diagram contains two curves. The blue curve represents the spectrum of the interior noise for the original state. The red curve demonstrates the result after manipulation of path 4. In order to gain a remarkable noise reduction the implemented



❿ Influence of the transfer path manipulation on the interior noise manipulation is not sufficient, because the overall sound pressure levels up to 500 Hz differ only about 2 dB(A). For a noticeable impact the difference should be about 6 dB(A). Thus, a further adjustment of the tank attachment to the vehicle body is necessary for a higher reduction of the interior sound pressure level. This is not always feasible, so that measures like baffles or structural measures are required additional to a manipulation of the transfer paths.

4 CONCLUSION

With the increasing application of start-stop systems and the ongoing hybridisation of drive train fuel slosh noises may be perceived better in the passengers' compartment. In order to investigate the sensitivity of a fuel tank and its vehicle integration a holistic approach was developed at ika. The approach will be further refined and then can be utilised in the industrial product development.

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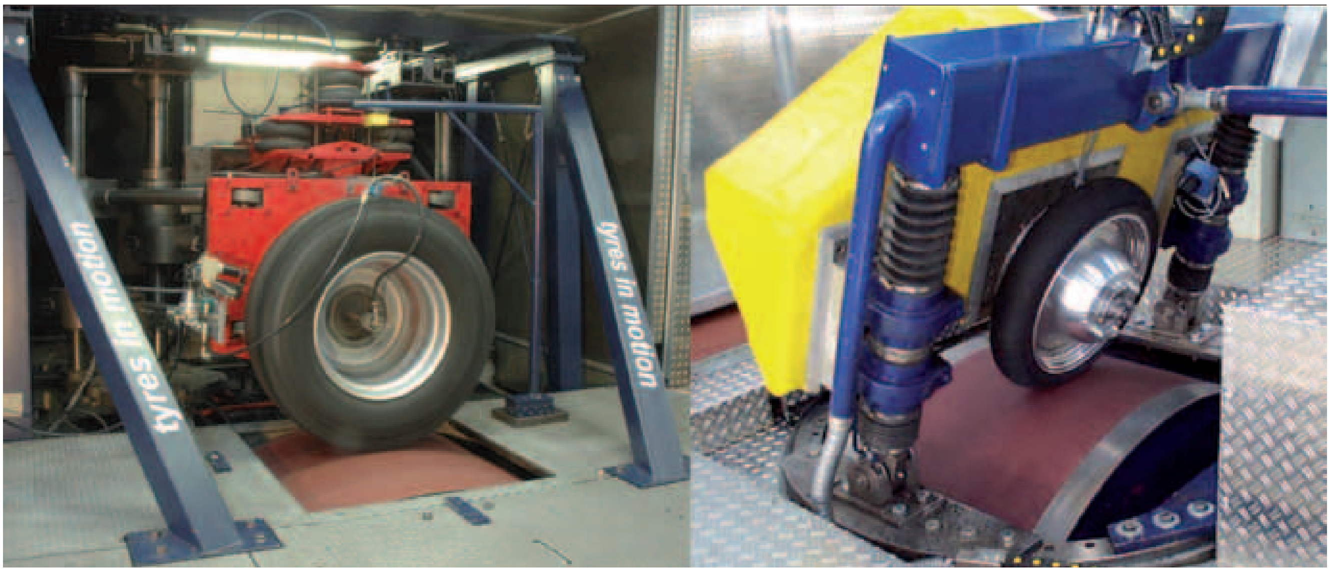
MOBILE TESTING EQUIPMENT FOR THE MEASUREMENT OF TIRE/ROAD FRICTION COEFFICIENTS

The chassis department of the Forschungsgesellschaft Kraftfahrwesen mbH Aachen (fka) and the Institut für Kraftfahrzeuge (ika) of the RWTH Aachen University have jointly developed two mobile test rigs that enable tire characteristics to be measured on real road surfaces. They supplement the external drum test rigs that have already been available for a long time for measuring tire properties under laboratory conditions.



❶ Indoor tire test rigs at ika and fka: heavy duty truck tire test rig NuReP (left) and motorcycle and passenger car tire test rig MoReP (right)

1 INTRODUCTION

The force transfer between vehicle and road takes place within the four contact patches of the tires. The complete driving dynamics of a motor vehicle are significantly determined by the characteristics of its tires. Due to the importance of the tires with respect to the overall vehicle, there is a special group in the chassis department of the Forschungsgesellschaft Kraftfahrwesen mbH Aachen (fka) and the Institut für Kraftfahrzeuge (ika) of RWTH Aachen University working on that subject. The team's fields of activity include the measurement of tire characteristics and their simulation with corresponding tire models. Another focus of their work lies in the conception, development and construction of entire tire test rigs. In addition to the two external drum tire test rigs that have already been available at ika and fka for the measurement of tire characteristics under laboratory conditions, two new self-made mobile test rigs have been launched recently. Those enable the measurement of tire characteristics on real road surfaces under different environmental conditions.

2 EXTERNAL DRUM TIRE TEST RIGS IN THE LABORATORY

For the measurement of the static, steady-state and dynamic force transfer characteristics of all common motorcycle, passenger car, light truck and heavy duty truck tires with a maximum wheel diameter of 1300 mm and within a wheel load range of up to 40 kN two external drum test rigs are in use at ika and fka. These are the heavy duty truck tire test rig NuReP [1] with a drum diameter of 2.54 m and the motorcycle and passenger car test rig MoReP [2, 3] with a drum of 1.59 m, ❶. The surfaces of these two external drum tracks are coated with the Corundum grit P80 abrasive sandpaper. These two test rigs have been developed and constructed by ika and fka. They have constantly been refined and are continuously kept at the state-of-the-art of measurement and testing technologies.

The tire measurements that are performed on these two test rigs are evaluated in a standardised process and converted into the measurement data format Tydex [4] in order to apply them within the software tools for the parameter identification of all common tire simulation models, such as FTire [5, 6] and Magic Formula [7].

3 MOBILE TEST RIGS FOR THE MEASUREMENT OF TIRE/ROAD FRICTION COEFFICIENTS

For different applications the measurement of tire characteristics also requires the investigation of the force transfer characteristics of the particular tire on different real road surfaces under different conditions. Especially the existence of intermediate layers, such as water, snow, ice or dirt, significantly reduces the maximum friction potential of a tire. But even road surfaces that are classified as "dry" can show diverse maximum tire/road friction potentials [8, 9, 10].

3.1 MOBILE TIRE TEST RIG

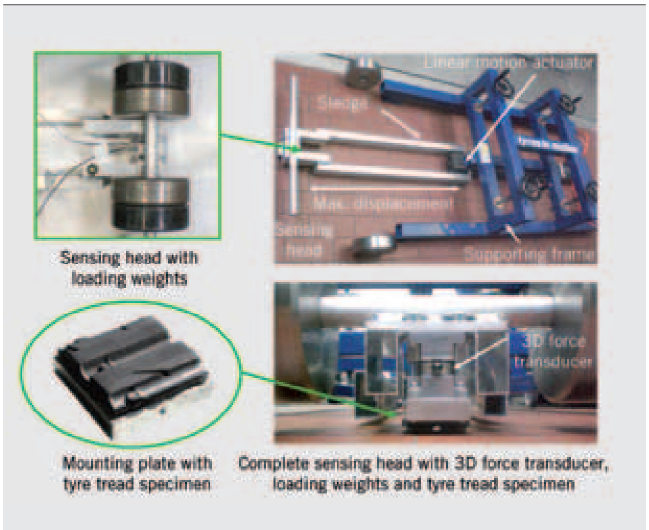
For the measurement of force transfer characteristics for motorcycle, passenger car, light truck and heavy duty truck tires on the surfaces of public roads and proving grounds, ika and fka have developed and constructed the mobile tire test rig FaReP [11, 12]. The concept of this mobile test rig allows the determination of tire characteristics on real roads outdoors as well as under laboratory conditions indoors on the corundum coated (2.54 m) external drum of the heavy duty truck tire test rig with identical measure-

❷ Mobile tire test rig FaReP on the external drum at the ika laboratory, as well as in measuring operation at the test track of Aldenhoven Testing Center (ATC) near Aachen

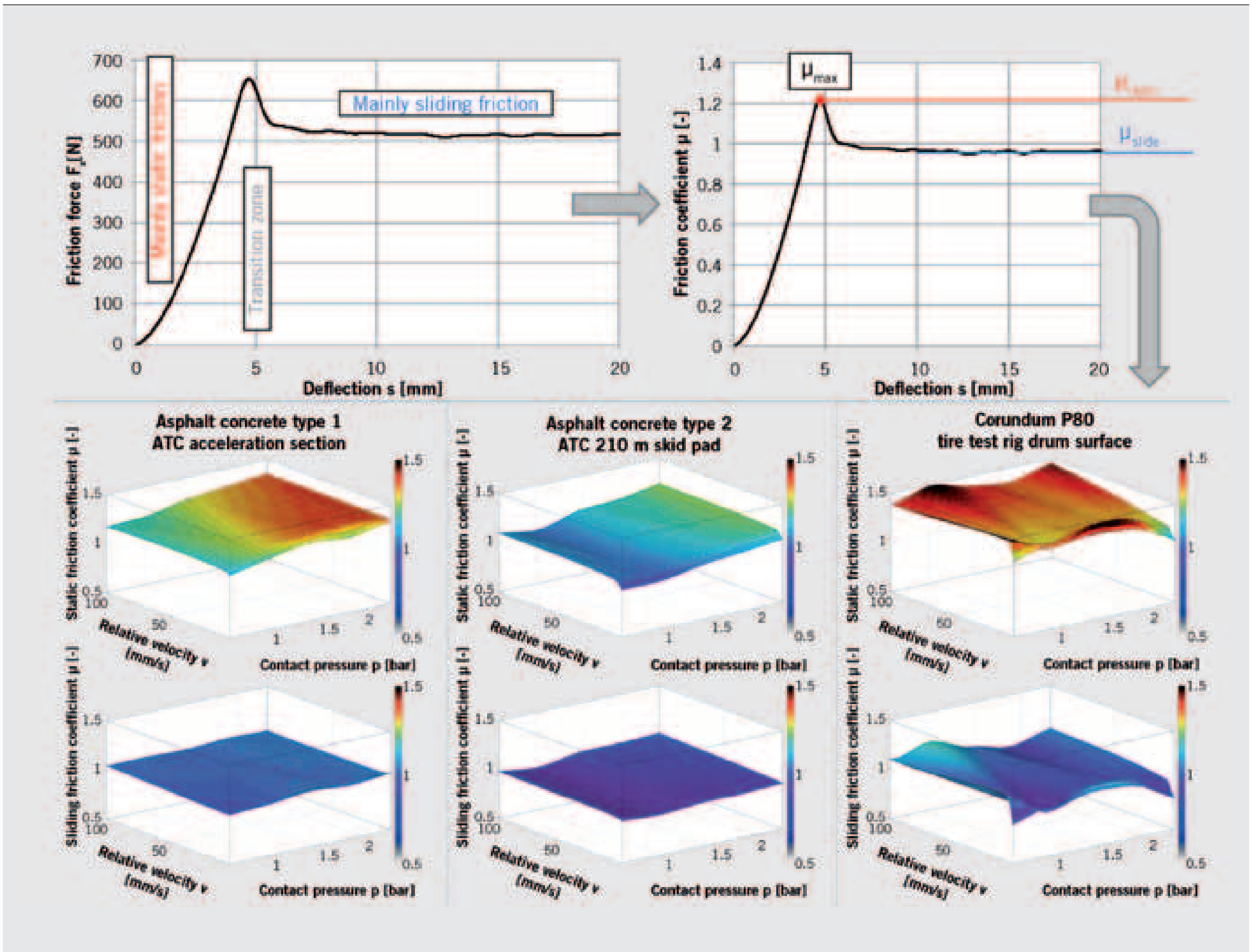
ment equipment, ❷. That way, a possible influence of the measurement equipment on the evaluation of tire characteristics under outdoor and indoor conditions can be eliminated. In principle, all vehicle tires with an outer diameter between 560 mm and 1240 mm can be investigated.

The mobile tire test rig consists of a trailer which is moved by a semi-trailer tractor. Driving speeds with a maximum of 100 km/h are possible. The trailer itself has two steering, or rather actively steerable rear axles to compensate possible slip angles of the trailer during the actual tire measurement. The tire test rig itself is located in the centre of the trailer between the rear axles and the driving axle of the semi-trailer tractor close to the overall centre of gravity. Due to this arrangement the wheel forces that are generated by the measurement tire have all in all the lowest effect on the driving dynamics of the entire vehicle with its gross weight of 22 t. During the measuring operation the motion unit of the tire test rig adjusts the required wheel load and the requested side-slip, camber and brake slip values to the tire. The set point commands for the tire measuring point of operating can be configured freely and are transmitted by a computer controller to the actuators of the test rig. The tire camber (up to $\alpha = \pm 10°$) and side-slip angle adjustments (up to $\alpha = \pm 15°$) are actuated hydraulically. The longitudinal slip is adjusted by a hydraulic disc brake with a maximum brake torque of approximately 24 kNm. The maximum wheel load of 50 kN for the measured tire is adjusted by an air suspension combined with a hydraulic cylinder. This active wheel load controller unit allows the compensation of low-frequency wheel load vibrations which can be induced by track excitations and vehicle body movements. The required hydraulic power of 17 kW and the electric power of 2.5 kW for the energy supply of the tire test rig and its measurement equipment are provided by a so-called power pack. It is made up of a diesel engine that drives a hydraulic pump and an electric generator and in addition supplies the compressed air network with 10 bar system pressure with the help of a suitable pump. The gauging and capturing of the tire forces and moments is done by a five-component measuring hub that has been developed by the Kistler company in cooperation with ika and fka. The measurement range goes up to 50 kN for the wheel load and up to ± 50 kN with respect to the longitudinal forces and lateral forces.

The main field of application for the mobile tire test rig is the measurement of force transfer characteristics of different tire types on real road surfaces, as well as the determination and evaluation of those characteristics with respect to the different road pavements that are used in today's road construction. Furthermore, the influence of different environmental conditions on the tire behaviour can be studied. The FaReP also allows tire force measurements on wet tracks that are needed for tire labelling required by European legislation. It is also possible to perform tire noise measurements on different road surface pavements. All standard tire simulation models benefit from the measurement data of the mobile tire test rig, too.



❸ Concept of the linear friction tester LiReP



❹ Friction coefficient mappings of the artificial track surface Corundum P80 (right) and of different asphalt concrete tracks (centre and left) on the proving ground of ATC (below) that have been determined by the linear friction tester

3.2 LINEAR FRICTION TESTER

With the linear friction tester LiReP [12], ❸, another testing instrument is in use at ika and fka in order to analyse the maximum force transfer potentials of different tire/track combinations. The LiReP allows the measurement of entire friction coefficient mappings on different track surfaces of single tread block elements separated from the complete tire. A typical friction coefficient mapping is described by the local static and sliding friction coefficients of the combination of tread rubber and track surface according to the operating parameters "local contact pressure" and "sliding and displacement speeds". In individual cases the operating condition of different "track and rubber temperatures" will also be added. The diagrams in ❹ show typical measurement results of friction coefficient mappings generated by the LiReP on different track surfaces.

The core piece of the linear friction tester is its linear motion actuator, which is realised as an electro-mechanical ball screw drive. It drags a weight loaded measuring sledge at a defined speed over the surface of the track. A rubber sample of 60mm x 60 mm, that was cut out from the tread of the analysed tire will be applied to a mounting plate and fixed to the sensing head of the sledge. With the help of loading weights between 3 kg and 60 kg local contact pressures in the range between 0.3 bar and 3.5 bar can be simulated, according to the size of the contact patch of the sample. During the actual test the fixed and vertically loaded tread lug specimen is dragged over the track surface from standstill at a constant moving speed. Displacement or rather sliding speeds in a range between.001 m/s and 1.2 m/s can be achieved. All forces that act on the rubber sample during the test procedure are recorded by a high definition 3D force transducer that is installed on the measuring slide. This sensor continuously measures the vertical load, the traction force (respectively friction force) and the lateral force that act on the tread block rubber sample. Additional displacement and speed sensors complete the sensing equipment. Furthermore, a climate chamber is available that enables the analysis of the effect of temperature on the friction coefficient mappings. Due to the fact that the LiReP is a mobile testing facility, it enables the measurement of friction coefficient mappings of single tire tread lugs on real road surfaces (such as public roads and test tracks) as well as on tire test rig tracks (such as external drums and flat belt tracks), ❺.

Besides the evaluation of the force transfer characteristics of different tread rubber compounds the friction coefficient mappings

❺ Mobile linear friction tester: general view with carrying wheels for moving from one measuring point location to another (above right), test rig in operation at the ATC test track (below left), as well as in measuring operation with a climate chamber (above left); comparison of the surface roughness of asphalt concrete and Corundum P80 (below right)

measured by the linear friction tester also allow a characterisation of different track surfaces under all possible environmental conditions. Furthermore, the measuring data can be used within the parameterisation process of complex 3D tire simulation models such as FTire [5, 6].

4 SUMMARY AND OUTLOOK

In addition to the two "classical" external drum test rigs available at ika and fka for the measurement of tire characteristics under laboratory conditions, two mobile testing facilities are in use that allow for the characterisation of force transfer characteristics of tires on the track surface of real road pavements. While the mobile tire test rig FaReP enables the measurement of tire force mappings for longitudinal and lateral slip characteristics on test tracks and public roads, the linear friction tester LiReP gauges the local friction coefficient mappings of single tread lug specimens separated from the complete tire on these tracks.

Therefore, ika and fka are equipped with the testing equipment necessary to perform detailed and holistic analyses of the force transfer characteristics within the tire/road contact patch under diverse environmental conditions.

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