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06 June 2010 | Volume 112

NEW PICKUP VW Amarok

AERODYNAMICS and Aeroacoustics of the New Audi A8

POWERSHIFT MODULE Combines Friction Brake and Planetary Gearset

LIFE CYCLE APPROACH to Lightweight Automotive Design

SOUND of Passenger Car Doors

INTEGRATED MUFFLER for Lines in Air-conditioning Loops

HOLISTIC Software Development

REQUIREMENT-DRIVEN Electric Drives

LATERAL DRIVE Torque Distribution

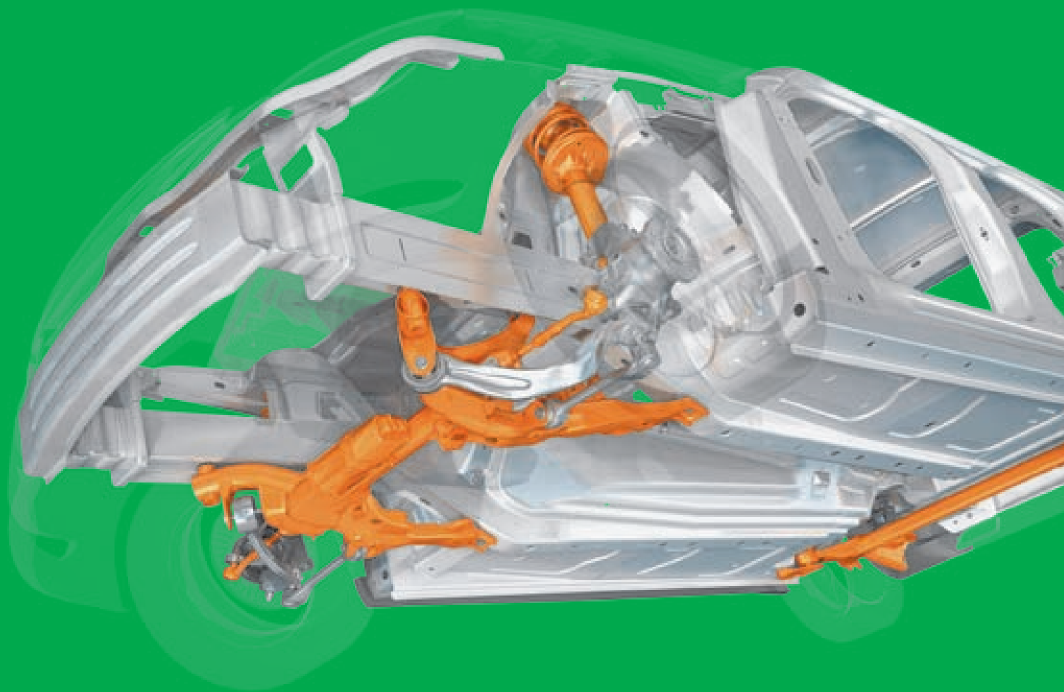
NEW CHANCES FOR THE CHASSIS

CONTENTS

COVER STORY

NEW CHANCES FOR THE CHASSIS

4, 8 I Like other vehicle components, the chassis must make its own contribution towards the development objective of cutting carbon emissions. Weight reduction and energy recuperation are just some of the key issues to be addressed. At the same time, costs need to be minimised. ZF is examining how intelligent lightweight design can help to meet these criteria. Bosch presents the synergies of a purely electric powertrain with the chassis, illustrated by the example of Torque Vectoring.



COVER FIGURE ZF
FIGURE ABOVE ThyssenKrupp

TOO MUCH OIL

Dear Reader,

That's all we need! After an Icelandic volcano spewed ash into the atmosphere and paralysed air travel in Europe for days, leaving everyone staring helplessly into a clear blue sky, we are now facing the challenge of a catastrophic oil spill. So much oil is being discharged from the deepwater oil well in the Gulf of Mexico that attempts to seal the leak have so far been unsuccessful.

On 21 April 2010, the Transocean drilling rig "Deepwater Horizon" caught fire after an explosion; it sank to the sea bed on the following day. 115 crew members were rescued, 17 of whom suffered injuries. 11 crew members tragically lost their lives.

The drilling rig was not a production platform but was carrying out initial exploration work on the promising Macondo Prospect. The word „promising" now takes on a different meaning as 5000 barrels of oil a day continue to flow out of three leaks. If this had been a large production platform, the disaster would have been many times worse.

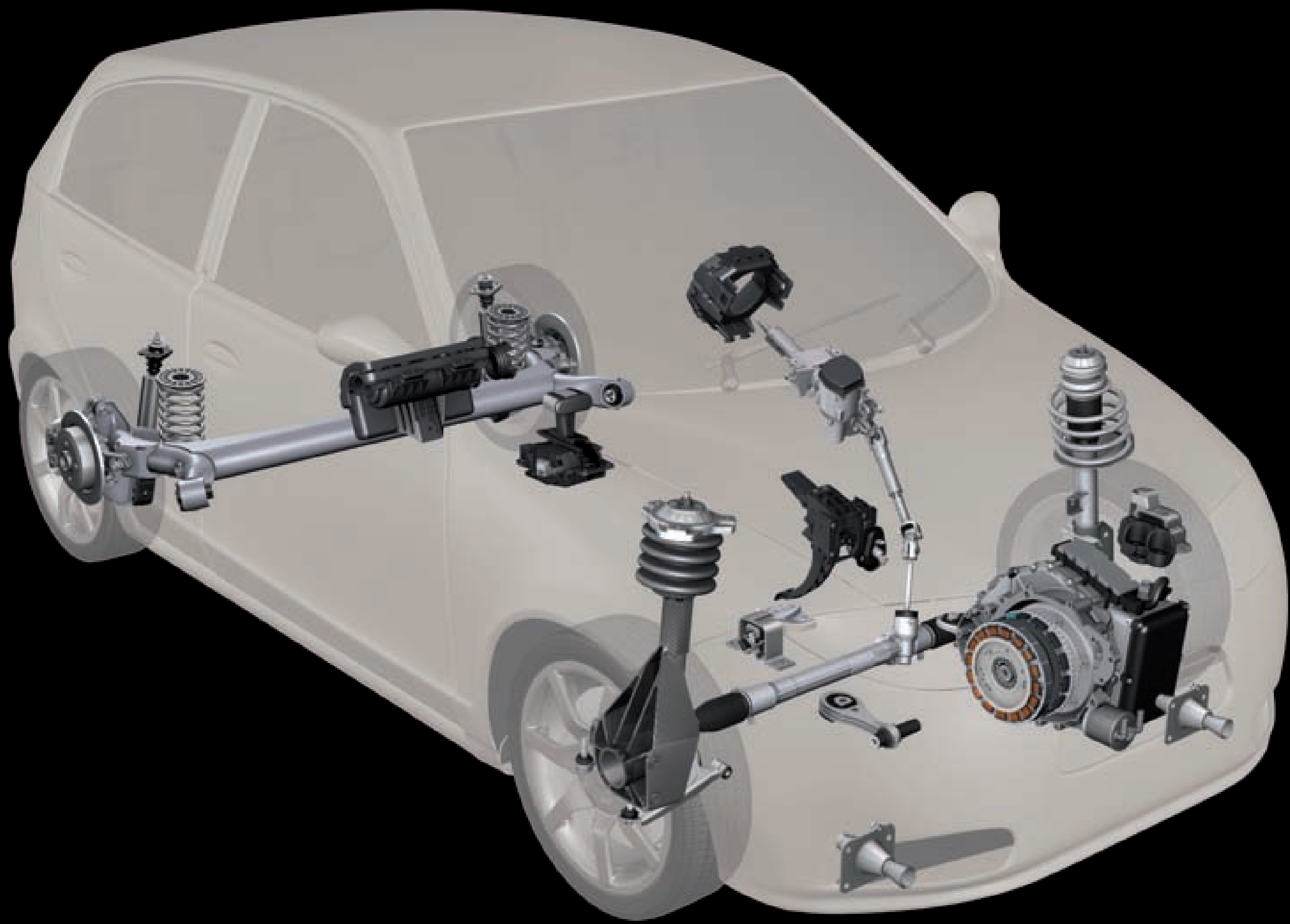
At the time of writing, the oil slick has already reached the Louisiana coastline, posing a serious threat to biotopes for marine life and plants. The livelihoods of the fishermen are also in danger, as their existence depends on intact oyster beds and fishing grounds for shellfish.

The WWF saw the accident as a reason to point to the growing number of offshore oil and gas production sites throughout the world, which they believe represent an incalculable risk. Environmental protection must take priority over the interests of oil production, they say. US President Barack Obama had only recently taken the decision to open up previously protected sections of the US coastline for new oil and gas drilling. Just how great the risk of accidents on oil platforms can be was demonstrated by an event in 2009: for ten weeks, all attempts to seal a leak on an Australian oil production platform failed.

Let us hope that the engineers in the Gulf of Mexico can get this problem under control more quickly, so that nature does not once again have to do pay the price of human error. This disaster should teach us to install even stricter safety measures, and to monitor them more closely. Apart from using oil more sensibly for plastics and pharmaceutical products, we still need oil and internal combustion engines. However, I wouldn't want my son to ask me one day, while sitting in his electric car, why we destroyed our environment so carelessly for such a long time.

Michael Reichenbach

DIPL.-ING. MICHAEL REICHENBACH,
Vice-Editor-in-Chief
Wiesbaden, 30 April 2010



LIGHTWEIGHT CHASSIS DESIGN

Weight reductions through alternative designs of passenger car chassis are given a particular timeliness: Initiatives of the automobile manufacturers on emission reductions and vehicle concepts – such as the electric car – require new, lighter chassis. Different approaches by the systems supplier ZF Friedrichshafen show that many measures are needed for lightweight chassis design.

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MOTIVATION

The use of alternative materials or production methods, the optimization of component structures, and the integration of functions are measures for smart lightweight design. They also offer opportunities for more economical chassis solutions, particularly if overall component design, including the manufacturing, is simplified in addition to weight reduction

It is true that the lion's share of a car's fuel consumption does not depend directly on the chassis. Indirectly, however, the chassis has an influence on many consumption-related factors and therefore offers potential to save fuel and thus to reduce CO_2 emissions. With lightweight design and the integration of functions, a rule of thumb in the vehicle is: 3.5 g CO_2/km or 8.5 g CO_2/km (including performance compensation/secondary effects) is saved per 100 kg of weight reduction. A lower chassis weight brings several benefits. It directly reduces fuel consumption and CO_2 emissions – and has indirect effects through the resulting possible use of smaller units (engine, transmission, brake, tank) in the vehicle. On the other side, reduction of unsprung mass enables improvements in driving dynamics, comfort and safety and as a result, the chassis becomes more agile with improvement in resonance behaviour. In past decades, the chassis design has become ever more complicated, many single components – such as in a multi-link rear suspension – place correspondingly high requirements on production and assembly. Therefore, not only lighter but also less complex chassis, which can be realized more economically, are desirable.

LIGHTWEIGHT DESIGN APPROACHES

There are different methods of lightweight design in the chassis, which mostly have to be bundled in order to achieve the desired results. Initially, the use of lighter materials offers obvious advantages – replacing steel with aluminum is already normal in many chassis components. Other materials, such as fiber composite materials, offer additional potential.

This includes – secondly – the optimization of structures by use of the finite element method. For example, the physical loads exerted on dampers or chassis components already have an impact on the design. The intelligent use of anisotropy of the new materials can save material and weight, respectively. Alternative materials therefore open up new possibilities for component designs.

There is a third important lightweight approach: the integration of functions, in other words, several components are merged into one. ZF already offers product concepts which weigh less than the previously used components including the joining material. Integration of functions often only becomes possible as a result of using new materials or production methods.

ZF applies all these lightweight design approaches and combines them to create new products, some of which are ready for the market and others are currently in concept study phase, ❶.

LIGHTWEIGHT DAMPERS

ZF produces lightweight aluminum dampers, ❷, which save 4 kg per vehicle and are already used in medium and luxury class production car models. This is possible, on the one hand, through a manufacturing method in which a variable wall thickness of the

MEASURE	STATUS	WEIGHT REDUCTION PER DAMPER (g)
Hollow piston rod for suspension struts	Volume production	260 - 400
Cold-drawn aluminum reservoir tubes with very different wall thicknesses for suspension struts/axle dampers	Volume production	360 - 480
CDC shock absorber with internal instead of external valve	Volume production	approx. 1500
Aluminum spring seat for suspension struts	Volume production	120 - 180
ET 30: monotube shock absorber with reduced diameter for rear axle	Volume production	300 - 400
Monotube shock with plastic eye and thin-walled steel and aluminum cylinder tubes	Close-to-production advanced development	560 - 650
Fiber-composite suspension strut & knuckle module	Concept study	3000 - 4000

❶ Summary of lightweight design potential for shock absorbers

❷ Lightweight aluminum damper

❸ Lightweight suspension strut

	Material		
Component	Steel	Aluminum	Fiber composite material
Suspension strut	3.49 kg	2.40 kg	integrated
Knuckle with joining material	2.92 kg	1.52 kg	
Total	6.41 kg	3.92 kg	1.95 kg

❹ Weight comparison of different assemblies and materials

aluminum reservoir tube is realised that fits to the mechanical load distribution on the shock absorber – utilizing the material almost ideally. Moreover, depending on their length, hollow piston rods allow for weight savings of 20 to 30 % in comparison to traditional, solid-steel piston rods; moreover, used in controlled dampers (CDC), they have the advantage that the required cable can run directly through the piston rod.

Another method is the concept of the ultra-light suspension strut & knuckle module made from fibre reinforced plastics (FRP), ❸. This study, initially targeted for supermini and subcompact cars, and presented for the first time at the IAA International Motor Show 2009, includes a complete strut module with integrated knuckle. Compared to the lightweight suspension strut and knuckle made from conventional metal materials, this FRP module reduces the weight by another 50 % when combined with a closed FRP spring. The great potential for lightweight design mainly results from the combination of the knuckle and suspension strut assemblies to form one component, which is only facilitated through the intelligent use of the lightweight fiber composite materials, ❹. For example, heavy steel components such as reservoir tubes and spring seats are superfluous in this damper concept as a result of the functional integration.

Despite its low weight, the FRP-damper has all the qualities of a modern, conventional, or electronically controlled damper. The damping inherent in the plastic material also has a positive impact on the NVH properties.

SUSPENSION DESIGN WITH WHEEL-GUIDING TRANSVERSE LEAF SPRING

One of the main targets when developing the suspension with the wheel-guiding transverse leaf spring was to improve the economic efficiency. It has been possible to reduce the number of components considerably by using new materials and functional integration – without any trade-off in terms of vehicle handling. One substantial feature of the new suspension design, ❺, is a transverse leaf spring made of unidirectional glass-fiber reinforced plastics. In addition to springing and anti-roll functions, this individual component also takes

❺ Suspension design with wheel-guiding transverse leaf spring made from glass-fiber-reinforced plastics (colored green)

❻ Multi-compliance-twist-beam-axle

over wheel guidance – and thus also essential tasks for the vehicle's directional stability and safety. Through the expanded functionality of the transverse leaf spring, the previously required anti-roll bar and the two anti-roll bar links with the corresponding mountings, as well as two control arms, and the conventional coil springs can be omitted. Apart from the reduction in weight, a lean manufacturing and assembly process is facilitated as a result. Reduced unsprung masses and the adjustable longitudinal flexibility over the longitudinal links permit good comfort. Under lateral force, the suspension goes toe-in and thereby supports the vehicle's under-steering. The lateral stiffness is adjusted to the requirements by the transverse leaf spring, the toe link, and the supporting length of the damper. The spring and roll behavior of the suspension can be determined by the mounting points of the transverse leaf spring and its cross section. The omission of the coil springs also improves the floor load width compared to the traditional McPherson suspension. In addition, the free space in front of the transverse leaf spring can be used, for example, for batteries or gas tanks in alternative vehicle concepts.

ALTERNATIVE TO THE MULTILINK REAR SUSPENSION

Another development approach – also with the aim of combining lightweight design and economy – is being followed by ZF with the „Multi-compliance-twist-beam-axle", ❻. This study showed an innovative and economical rear axle design on the basis of a twist-beam rear axle, that renders nearly the performance of a multilink rear suspension, but without considerable additional cost. In order to achieve the kinematic and elastokinematic properties of a multilink rear suspension, it is usually necessary to separate the functions and thus have a targeted design for the chassis-specific properties. The transverse force support, above all, was found to be dicey here – a considerable disadvantage of twist beam rear axles when it comes to operating dynamics. To minimize the tendency to oversteer, wheel-correcting bearings are typically used to connect to the vehicle body. When braking, standard twist-beam rear axles also generate a slight toe-out, which can lead to an edgy axle behavior when braking.

This behavior is counteracted by the new rear axle design. The new suspension of the wheel carrier generates a virtual pivot which moves the wheel in toe-in, both for transverse and longitudinal forces. This principle of a multi-compliance twist beam axle presents driving characteristics which were previously only possible with multilink rear suspensions: lateral force understeering, toe-in under brake force, and an optimized bump toe-in with reciprocal spring compression. The suspended wheel carriers also have a positive effect on vibration decoupling. The elastokinematic axle parameters offer automotive developers and manufacturers additional leeway for design. The Multi-compliance-twist-beam-axle can thus bridge the gap between the classical twist-beam rear axle and multilink rear suspension.

INTERFACES HAVE TO FIT

Lightweight design approaches in the chassis place high demands on the interface between product development and production know-how. Here, ZF relies on the tools profiled during chassis development in virtual product development. They help to shorten development times, save costs, avoid errors, and they support the documentation of the development stages. Implementing these measures in volume production requires corresponding production know-how. On the one hand, new process steps have to be introduced and mastered in the case of material substitution. On the other hand, functional integration of components can result in completely different assembly sequences with economic potential.

SUMMARY

Weight reduction in the chassis makes sense for two reasons: Firstly, the dynamic driving properties can be improved, and secondly, it contributes significantly to fuel economy and reduced emissions. ZF Friedrichshafen AG combines all typical approaches toward lightweight design in the chassis with a third objective – more profitability – to form smart lightweight designs. New concepts for axles and chassis components therefore not only result in less weight on the scales – it also results in virtually no loss in ride comfort, vehicle handling and safety. Thanks to the integration of functions, these products can also be manufactured with fewer components in fewer assembly steps, and therefore are more economical.

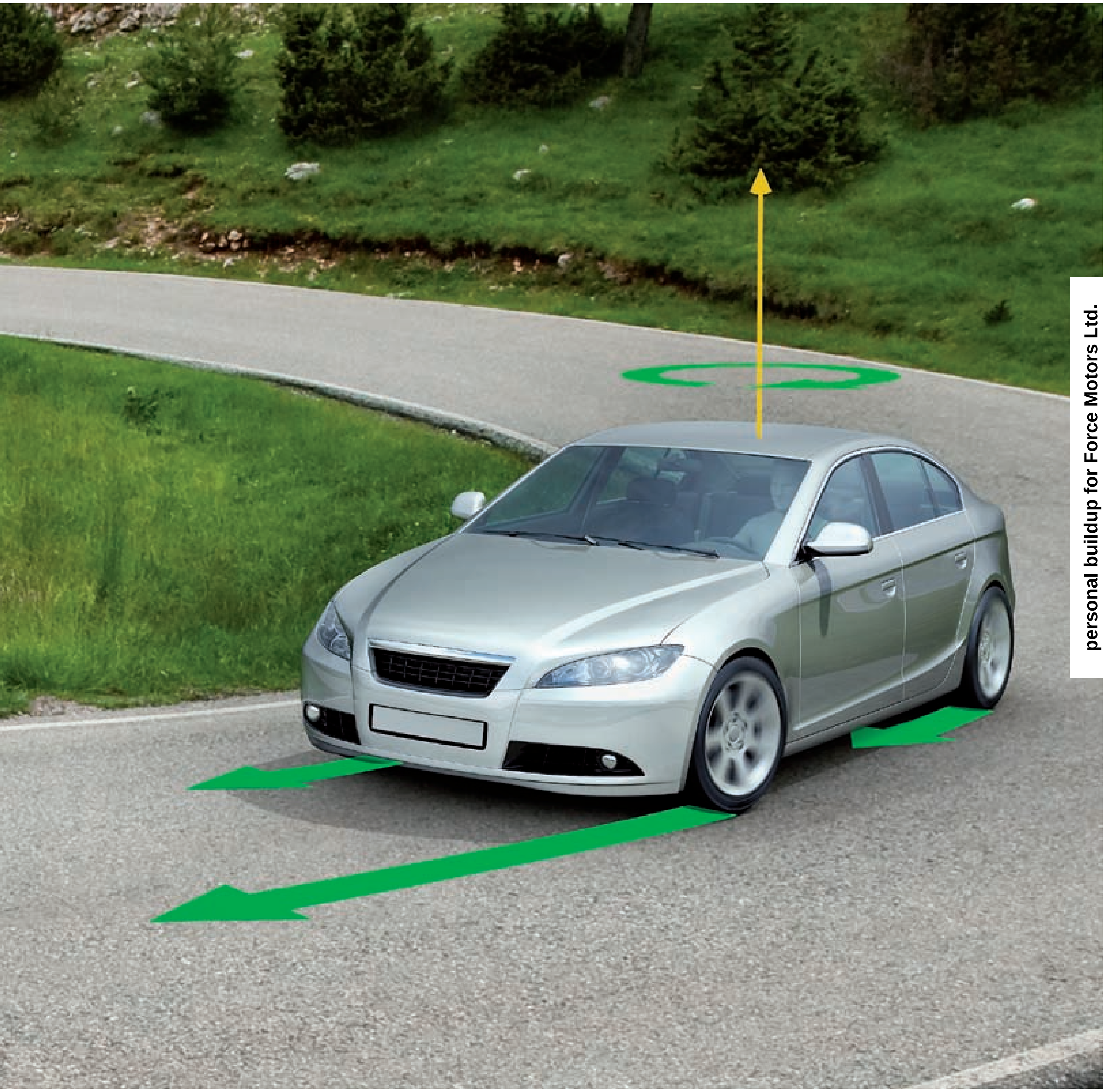
TORQUE VECTORING A NEW LEVEL OF FREEDOM FOR ELECTRIC VEHICLES

Much discussion on the launch of electric vehicles focuses on the reduction in emissions. Regarding vehicle dynamics there are also interesting new approaches to define vehicle characteristics, especially with multi-motor concepts. Bosch Engineering GmbH has demonstrated such concepts in a test vehicle.

MOTIVATION

Everyone is talking about electric vehicles, and they are a common sight in the automotive industry‘s research laboratories. First models already have appeared on the roads of various countries. Compared to the high profile and the share at motor shows, one could expect to see electric vehicles taking over our roads in the very near future. According to reputable studies [1], however, significant numbers of electric vehicles will only appear from around 2020 onwards. Despite this – or precisely because of it – this is the ideal time for engineers to act. The competition in terms of technology and expertise is already fierce.

This also provided the motivation for Bosch Engineering GmbH to showcase the entire scope of its engineering services in the form of an electric vehicle. The goal was to create a ready-to-drive sporty vehicle with impressive performance data. This was achieved by using four electric motors to convert a current series vehicle from all-wheel drive to individual-wheel drive. An interdisciplinary team completed all the necessary steps within just a few months – selecting and integrating components, redesigning the electrical system, developing an appropriate drivetrain control system (including new driver information system), adapting the ABS/ESP, and



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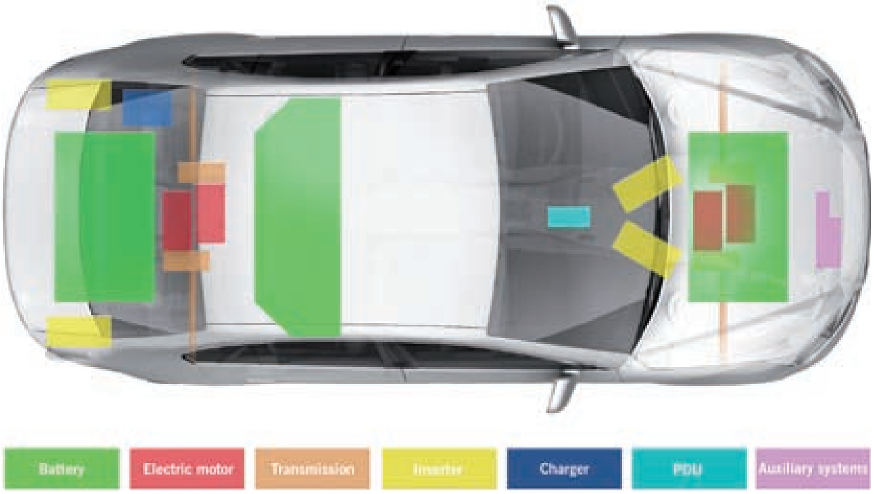


❶ Arrangement of components in the vehicle

VEHICLE	BATTERY
: Output: 240 kW : Drives close to wheels (4 x 60 kW) : Max. torque: 60 Nm : Max. speed: 130 km/h : Acceleration: 0 – 100 km/h < 7 s : Range > 200 km (FTP75) : Weight: 1.9 metric tons	: Lithium-ion-battery (390 V) : 216 individual cells (60 Ah) : 3 battery packs (~ 500 kg) : Capacity: 45 kWh : Air-cooled

❷ Performance data test vehicle

networking all the functions. The final step was developing new vehicle dynamics functions, including the torque vectoring explained more detailed in this article.

TEST VEHICLE

Firstly, the combustion engine, transmission, and exhaust system were replaced by the components required for electrification. ❶ shows the arrangement of these components in the vehicle.

The key element is the high-voltage system with three lithium-ion-batteries (390 V, 45 kWh). Each of the electric motors arranged close to the wheels has a maximum output of 60 kW. This power is transmitted by specially developed transmissions (ratio 13:1). A further electric motor operates the auxiliary systems such as the power steering pump and the air-conditioning compressor. The performance data is as shown in ❷.

TORQUE VECTORING

The focus of vehicle dynamics engineers on torque vectoring (TV) pre-dates the launch of electric vehicles. The term refers to a wheel based distribution of braking/drive torques on a wheel-by-wheel basis to improve vehicle safety and/or agility. The principle involves using different wheel torques to give the vehicle a yaw moment and thus modify its driving dynamics. Various actuators are used for this purpose.

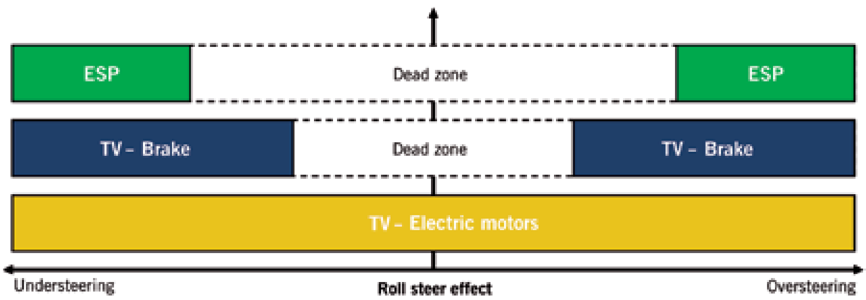
One option is active brake intervention, which is possible without additional components. Depending on the adhesion potential, braking torques prevent, for example, understeering and enlarge vehicle agility. This function – called "Dynamic Wheel Torque by Brake"– is used in various series vehicles as part of the Bosch ESP.

Yaw damping during cornering is possible using a locking differential, for example. Adjusting the wheel speeds creates drive torques on the inside of the curve and braking torques on the outside. Active differentials enable torques to be distributed as required between wheels on the same axle. This also makes it possible, for example, to improve the vehicle's agility.

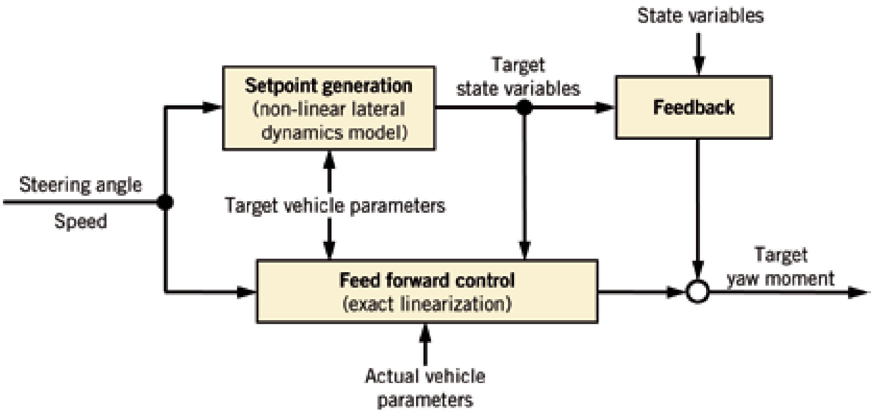
In hybrid and electric vehicles, torque vectoring can be designed directly from the drive, providing at least two electric motors for different wheels. Using four motors creates even greater freedom of control and optimizes the potential of torque vectoring.

❸ shows the range of action of torque vectoring functions compared to an ESP. Torque vectoring using electric motors enables continuous control without a dead zone. This means the software can improve the driving dynamics in a way that would otherwise only be possible by adapting the suspension. This optimization means the vehicle needs to be stabilized by the ESP less often. Generally speaking, torque vectoring using electric motors offers the following benefits:

: excellent control dynamics
: very large control range in both positive and negative directions
: completely separate control elements on individual wheels
: no limit on the maximum control variable due to different driving conditions
: continuous control possible
: no additional component wear
: energy-efficient distribution of braking and drive torques.



❸ Range of action of various TV functions



❹ Control algorithm

CONTROL ALGORITHM

In order to fully utilize the design benefits of electric motors assigned individually to the wheels, it is necessary to apply a controller that does not work on a situation-dependent basis. The objective was therefore to come up with a holistic algorithm that covers the entire scope of lateral dynamics, involves as few application parameters as possible, and works well with the existing Bosch ESP.

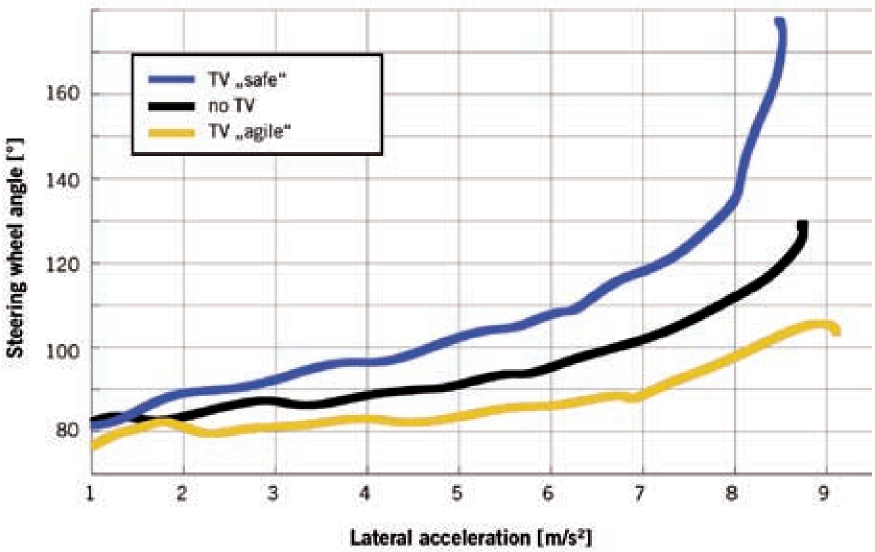
Due to the characteristic saturation behavior of rubber tires, the lateral dynamics of motor vehicles operating at their limits can only be accurately depicted using sufficiently precise non-linear models [2]. Taking this behavior into account, the TV system described here uses a model-based, non-linear control algorithm, ❹. In the “setpoint generation” function block, the setpoint values of all relevant state variables (e.g. yaw rate, slip angle, roll angle, etc.) are determined. A feed forward control based on the principle of exact linearization [3] ensures that the target vehicle behavior is precisely complied with as long as there are no model uncertainties or external disturbances. These unpredictable influences are compensated using feedback that determines the compensatory proportion of the control variable based on the difference between actual and target state variables. The control variable corresponds to a yaw moment about the vehicle‘s vertical axis. An arbitrator uses this to generate drive and acceleration torques for the individual wheels, taking into account the accelerator pedal travel and the current adhesion potential of all four wheels.

One benefit of the model-based design methodology used here is the controller‘s ease of application. Once the actual vehicle parameters (e.g. mass, moment of inertia, center of gravity, etc.) have been determined, the target vehicle behavior can be set by specifying a small number of physical parameters. If the target moment of inertia is defined to be smaller than the actual moment of inertia, for example, this reduces the phase displacement between steering angle and yaw rate. If the front axle‘s lateral force stiffness is raised in the target vehicle model, the TV-controlled vehicle achieves a higher yaw rate and lateral acceleration with the same steering angle than with TV switched off. A further parameter can be used to adjust the balance (understeering/neutral/oversteering) at the maximum lateral acceleration. It is also possible to balance a vehicle operating at its limits by adapting the suspension (stabilizers, instantaneous roll centers, springs, dampers, etc.). However, TV offers the benefit that these adjustments can be made by the software based on speed and dynamics. This enables neutral driving at very low speeds without the vehicle tending to oversteer at high speeds. Similarly, the balance can be corrected in the direction of understeering for highly dynamic maneuvers, for example to ensure stability when changing lanes at high speeds, even though the vehicle is neutrally balanced in steady-state cornering.

TEST RESULTS

Controlling vehicle dynamics is different to other problems involving control technology because the driver plays a key role in the control loop. Ultimately, the objective is to change the vehicle behavior in a way that the TV-controlled vehicle “feels better” than without individual drive torque distribution. For example, steering angle and lateral acceleration measurements can be used to evaluate the vehicle and the driver‘s performance at slowly increasing speeds during stationary cornering, ❺.

The form of representation selected in ④ enables the vehicle‘s self-steering gradient to be read off for small lateral accelerations



❺ Impact of TV on steady-state skidpad testing (see e.g. [4]). This corresponds to the change in the steering wheel angle with increasing lateral acceleration and can be manipulated by TV. Configuring TV for "safe driving" increases the self-steering gradient compared to the initial configuration and the vehicle feels less sensitive in terms of the steering response, which is generally comfortable regarding high speeds. An "agile" TV configuration enables the self-steering gradient to be reduced, so that the vehicle feels sportier and more responsive. In safe driving mode, TV was designed for understeering at the operating limits, so the steering angle increases significantly on approaching the maximum lateral acceleration. In agile mode, a neutral setting was selected. As can be seen from ⑤, the driver even has to reduce the steering angle slightly on reaching the maximum lateral acceleration. In addition to vehicle balance, TV also impacts on the maximum achievable lateral acceleration at the vehicle's operating limits. In safe driving mode, this is approximately 0.3 m/s^2 lower than in the initial configuration and in agile mode it is around 0.5 m/s^2 higher. This effect is explained by TV-based differences in the utilization of the rear axle's adhesion potential (see [5]).

CONCLUSIONS AND PROSPECTS

Until now, discussions relating to electric vehicles have often focused on ecological aspects. The demonstration of a torque vectoring function shows that switching to an electric drive can also result in a more enjoyable driving experience. Torque vectoring can be used for comfort-related functions, too, such as a lane-keeping assist system. One challenge at present lies in the large number of possible drive concepts at still relatively low volumes. In particular, networking functions and systems without negative feedback reaction should not be underestimated. The solution is to quickly establish standards to keep the outlay required for testing and validation at a reasonable level with an appropriate number of variants. Integration skills of automotive suppliers are very helpful in this respect. Their role could be to offer automakers the required levels of freedom to define driveability while at the same time limiting the development outlay by reusing generic solutions.

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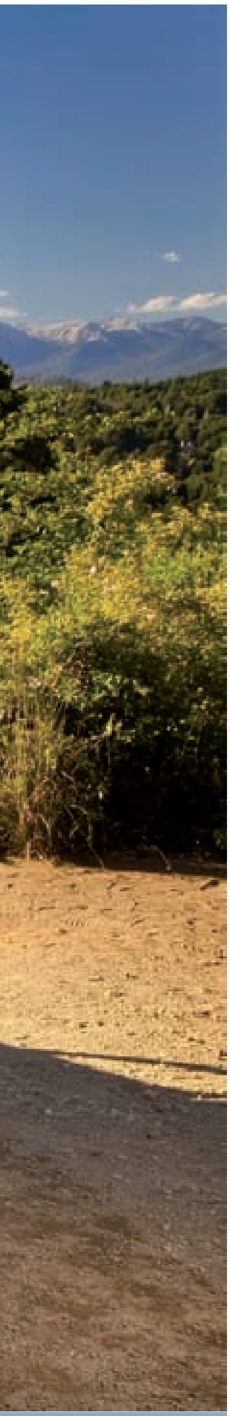
THE NEW PICKUP VW AMAROK

With the Amarok, Volkswagen Commercial Vehicles is launching one of the most modern pickup trucks in the world onto the market. No competitor in its class is as economical. Many of the technologies implemented in the Amarok are being used for the first time in this segment. These include bi-turbocharging for the top version of the TDI diesel engine with 120 kW (163 PS) power. The 5254 mm long vehicle in ladder frame concept is available with rear-wheel drive as well as non-permanently and permanently four-wheel drive.



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DEVELOPMENT TARGETS

The pickup VW Amarok is built in Pacheco (Argentina) and it completed its baptism of fire at the start of 2010 as a support vehicle in the Dakar Rally. The market launch in South America followed in February. The design and technology of the Amarok have the potential to establish pickups in Europe as well. First deliveries to the customers will take place in late summer 2010. The development specifications for the Amarok defined the following objectives right from the start:

- minimum fuel consumption and maximum performance
- highest safety standards for a vehicle in its class
- outstanding driving properties, even when fully loaded
- highest standards of robustness and safety combined with the best possible comfort in conjunction with a new, innovative ladder frame concept
- three drive concepts with standard rear as well as non-permanently and permanently engaged four-wheel drive
- ergonomic interior concept with a high-quality appeal
- optimum payload, load quality and storage compartment areas
- use of robust and easy-to-service electronic components
- body shape on the basis of the new Volkswagen design DNA.

TESTING

During the pre-development phase, endurance trials were undertaken with competitors‘ vehicles on various proving routes in order to establish the load profiles. This meant that duty cycles could be calculated for the component and overall vehicle design at a very early phase. These results were channelled into the operating strength calculations for the ladder frame, the cab, the load space (cargo box) and the chassis (running gear). Endurance trials with prototypes and series production vehicles were undertaken on the basis of the service life specifications that had been calculated in this way, making it possible to ensure structural strength and durability.

In particular, the high durability of the technically demanding, completely new top-level engine – a bi-turbo diesel with 120 kW (163 PS) power – was demonstrated in numerous test rig and in-vehicle endurance trials. The four-cylinder engine impressively demonstrated its performance capabilities during the Dakar Rally, above and beyond the standard tests. The new TDI, as expected, was able to negotiate mountain passes at altitudes of up to 5000 m without difficulties.

STYLING

In terms of concept and style, the Amarok, which is up to 5.25 m in length, is a copybook pickup, ❶: Uncomplicated, down-to-earth, straightforward and high-quality. Its design is entirely in line with the new Volkswagen design rule (DNA).

The most striking features of this DNA include the horizontal lines of the front end. This is made clear by taking the example of the Amarok‘s radiator grille: like blades mounted one on top of another, the two black slats extend across the front, and are uprated with chrome strips in some equipment levels. As a graphical element, they are continued as far as the headlights with their highly technical design, and significantly influence the

ENGINE	2.0 L 120 KW CR TDI BITURBO R4			2.0 L 90 KW CR TDI MONOTURBO R4		
DRIVELINE	4×2	4×4		4×2	4×4	
	REAR-WHEEL DRIVE	NON-PERMANENT DOG CLUTCH	PERMANENT TORSEN-DIFFERENTIAL	REAR-WHEEL DRIVE	NON-PERMANENT DOG CLUTCH	PERMANENT TORSEN-DIFFERENTIAL
Length [mm]	5254					
Width over door pull [mm]	1944					
Height body at ML1 [mm]	1834					
Wheel base at ML 1 [mm]	3095					
Overhang front / rear at ML 1 [mm]	898 / 1261					
Ground clearance ML1 [mm] with / without underride protection	249 / 26					
Track gauge front / rear ML 1	1648 / 1644					
Loading area length / width [mm]	1555 / 1620					
Loading width [mm]	1222					
Loading platform gate height [mm]	508					
Height boot sill rear ML1 [mm]	780					
Turning circle [m]	12.95					
Drive slope front / rear ML1	28° / 28°					
Ramp angle ML1 with underride protection / without underride protection	21.4° / 23°					
Payload [kg] **comfort**	938		862	948		872
Payload [kg] **heavy duty**	1137	1047		1147	1057	
Gross vehicle weight rating (GVWR) [kg] **comfort** / **heavy duty**	2820 / 3040					
Maximum front axle load [kg] **comfort**	1325	1385		1315	1375	
Maximum front axle load [kg] **heavy duty**	1335	1385		1325	1375	
Maximum rear axle load [kg] **comfort** / **heavy duty**	1600 / 1860					
Roof load [kg]	100					
Trailer load, braked, 12 % / un-braked [kg]	2800 / 750					
Vertical load [kg]	120					
Allowed trailer set mass [kg]	5500					

❶ Dimensions and vehicle data of VW Amarok for the variant ML 1

layout of the headlights. The concave shape of the lower edge of the front bumper also emphasises the considerable ground clearance of the Amarok. The stylistic unit comprising radiator grille, headlights, bumpers painted in body colour and robust lower termination gives the front of the vehicle a masterful character which is entirely in accordance with a pickup of this size.

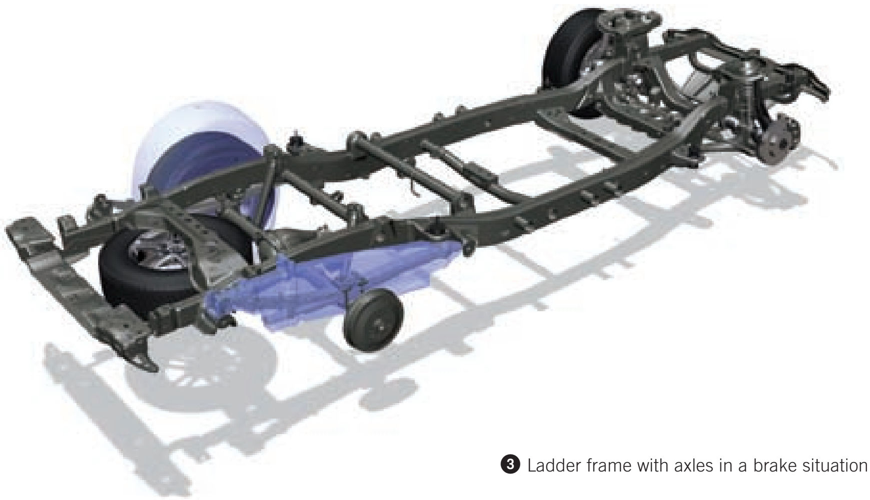
Powerful elegance determines the side view of the Amarok. A striking feature in this area: the flared wheel arches. Combined with the homogenous door and wing surfaces, they make for a very muscular look. The rear end with the split bumper is characterised by uncompromising functionality. This provides space in the middle of the vehicle for a step positioned at a lower height, in order to facilitate access to the load bed. At the same time, precisely fitting shapes also make it possible for the weight-bearing tailgate to be folded down through 90°, thus producing a surface that is flat and contiguous with the load bed. When closed, the precisely designed, smooth surface of the tailgate shows off the chrome-plated VW logo in its centre.

BODY CONCEPT

The Amarok will be launched as a four-door double cab version; a two-door single cab variant will follow in 2011. All of these types have their body made up of the cab and frame modules. The generously sized cross sections of the A and C-pillars help to deliver particular stability in the single and double cabs. At the same time, decoupling the cab from the ladder frame produces an optimum degree of torsional resilience. This has a noticeably positive effect on driving and acoustic properties when driving on poor quality roads. The specific use of high-strength steels also gives the chassis a particularly high level of crash safety and sets new standards in this class.

Volkswagen Commercial Vehicles has consistently tailored the concept of the Amarok towards optimum function. Take the cargo box for example: the leaf springs were arranged alongside the frame in order to produce a load sill height of 780 mm that is as low as possible. The cargo box is bolted onto the frame, and itself offers the

❷ Cargo box with Euro pallet loaded sideways, carrying tyres and with lighting of the load space



❸ Ladder frame with axles in a brake situation

❹ Cockpit – front seat row with long front/back adjustment range

largest dimensions in its class. The width of 1222 mm between the wheel houses is also „best-in-class" – giving sufficient space to load Euro pallets sideways in order to save space, a unique first for a medium-category pickup, ❷. Thanks to these impressive dimensions, the low load sill and a payload of up to 1.15 tons, it is possible to transport even large implements with ease.

LADDER FRAME

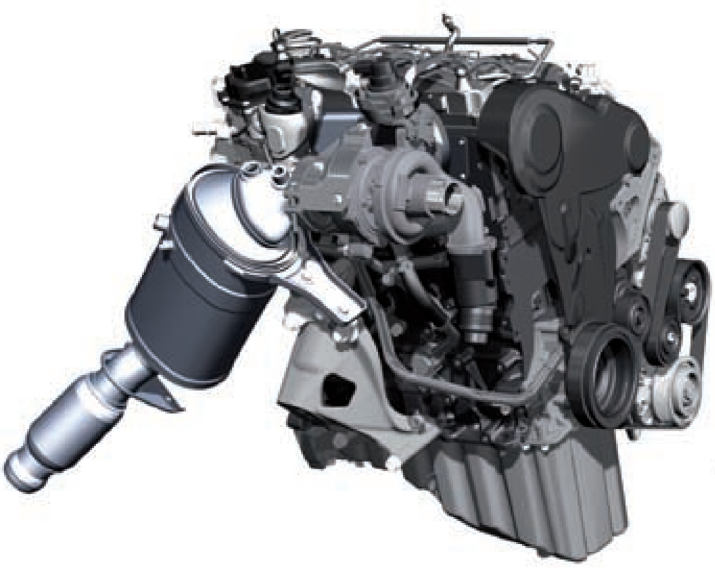
Achieving an optimum interplay between the ladder frame and the cab, the cargo box and the running gear, represented an overriding development objective. The configuration of the frame's static and dynamic rigidity was optimised using computer programmes and tuned during test drives. Overall, the ladder frame of the Amarok offers exemplary torsional rigidity which is attributable to its closed chassis rail sections and additional cross members with a closed cross section, ❸.

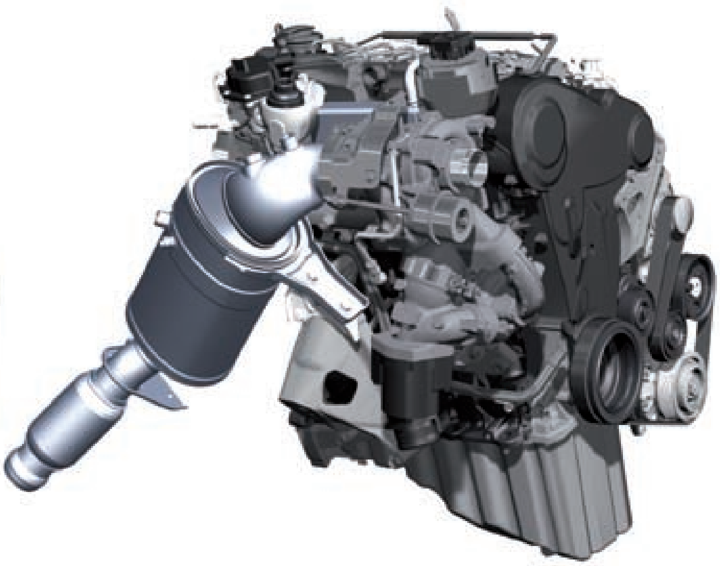
These latter components are inserted through the chassis rails as well as being welded, thereby offering an optimum basis for providing the best possible support for torsional forces. The single and double cabs have been designed to share the same ladder frame, thereby obviating the need to pay for a second development track. The only differences are in the area of the specific adapters for the cab and cargo box.

INTERIOR

High-quality, robust materials and ingenious operation are features of the interior – which, in the case of the double cab, is the largest in its class. Quality far above the average is provided by the height-adjustable front seats, ❹. In addition, the front/back adjustment range is the longest in the competitive environment. Three more adults can be seated comfortably on the rear seat bench. In order to optimise storage space within the cab, it is possible to fold up the seat bases with a 1/3 to 2/3 split. Furthermore, the backrest is folding.

As is typical for Volkswagen Commercial Vehicles, the cockpit meets especially exacting requirements in terms of ergonomics and operability. The additional functions for four-wheel drive and the powertrain have been positioned around the selector lever, for example, where they





❺ View of BiTDI (left) and TDI engine (right), both with 2 l displacement

can be used intuitively and are easy to reach. The audio system is positioned high up in the centre console, locating it directly in the driver's field of view. The heating, ventilation air-conditioning control has been arranged immediately below that. Furthermore, an automatic air-conditioning system is available on request, which sets a new top standard in this segment in terms of its performance capabilities. The air distribution settings are displayed in parallel on the screen of the infotainment devices RCD 310, RCD 510 and RNS 315.

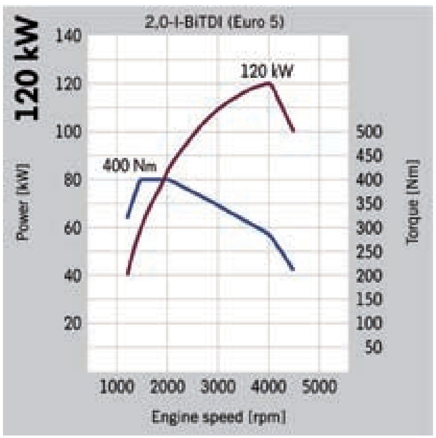
The Amarok also proves its high level of everyday practicality by offering numerous storage opportunities in the interior. For example, the large door compartments at the front can accommodate 1.5-l bottles (1-l bottles can be stored in the rear doors) whilst there are two cupholders in the centre console as well as, depending on the equipment level, in the rear. A new development concerns what is called the multi-fixing point in the front area of the centre console. This can be used for a range of items such as a hands-free unit, a ring holder for the cupholder, an ashtray or other accessories.

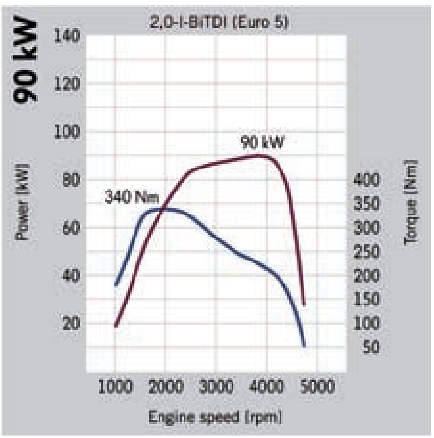
DIESEL ENGINES

With the Amarok, Volkswagen Commercial Vehicles is bringing the latest engine and powertrain technology into the most popular pickup segment in the world, that of the mid-size vehicle. Its progressive four-cylinder diesel engines are following the downsizing strategy. They feature direct injection and will appeal with frugality, powerful pulling power, high load capacity and constant power delivery. The pickup is being launched with two ultra-modern TDI common rail (CR) engines, ❺. The top engine is a 120 kW (163 PS), entirely new 2-l TDI engine with bi-turbocharger (BiTDI). The second TDI variant with 90 kW (122 PS) uses conventional supercharging.

The top-end TDI engine gives the Amarok an outstanding position amongst mid-size pickups. Thanks to its two-stage, controlled supercharging and highly modern CR injection, this four-valve engine complies with the Euro 5 exhaust standard and offers acoustic comfort and smooth running of the sort of a passenger car.

Maximum torque of 400 Nm is delivered with the engine speed as low as 1500 rpm; a good 350 Nm remains on tap even at 3000 rpm, ❻. Both of the highly responsive TDI engine is able to offer extremely low fuel consumption figures. Example of Amarok with 4 × 2 rear-wheel drive: average consumption only 7.6 l diesel fuel per 100 km. CO_2 emissions of the Amarok in the 120 kW TDI version are 199 g/km. In the non-permanently or permanently engaged 4Motion four-wheel drive combination, consumption is still very economical at 7.8 l per 100 km. In this case, the CO_2 emissions value is 206





❻ Performance curves for BiTDI (left) and TDI engine (right) g/km. These figures are matched by maximum speeds of up to 182 km/h (rear-wheel drive) as well as acceleration from 0 to 100 km/h which is within 10.8 to 11.1 s, depending on the powertrain.

The „small“ TDI engine with 90 kW (122 PS) and 340 Nm maximum torque (from 2000 rpm onwards) is even more frugal, ⑥. In this case, a single turbocharger with variable turbine geometry is used. Volkswagen Commercial Vehicles has also pushed ahead with development of this engine, particularly with regard to achieving Euro 5 exhaust limit values, low CO_2 emissions, ❼, and torque characteristics that are oriented towards traction. For example, newly designed compressor and turbine wheels optimise the efficiency of the four-valve engine which also has a 2 l displacement.

Used as a pickup powerplant, this TDI sets the bar all over again in terms of consumption. A value of only 7.5 l diesel per 100 km is achieved with rear-wheel drive, ⑦. The value for CO_2 emissions is 198 g/km. The 4Motion drive gives the 90 kW TDI a consumption figure of 7.6 l per 100 km. Here too, the CO_2 emissions are 199 g/km, putting them below the important 200 g/km threshold for this segment at agile driving performance.

POWERTRAIN AND CLIMBING ABILITY

Three different power transformation concepts are available for both diesel engines depending on the requirements profile: straightforward rear-wheel drive, rear-wheel drive with non-permanently engaged 4Motion four-wheel drive as well as permanent 4Motion four-wheel drive. The latter version features a Torsen differential, which is an innovation in this segment. Optionally, Volkswagen Commercial Vehicles is also offering a mechanical rear-axle differential lock for use under difficult offroad conditions. All three variants have an electronic differential lock (EDL) which functions by means of the brakes being applied automatically.

With rear-wheel drive, the newly developed pickup is intended for customers who are above all interested in the specific application spectrum of this body shape and who require flexible loading possibilities, but who predominantly use their vehicle on consolidated roads. This version is available with all the Amarok engines, giving the lowest possible consumption and CO_2 emissions combined with the highest payload values.

The four-wheel drive variants are drawn for pathless land. The Amarok with permanent all-wheel drive represents a special feature in the pickup segment. Its central Torsen differential ensures that propulsive force is optimally distributed between the front and rear axles under all application conditions, in order to guarantee excellent driving dynamics and high traction even when offroad. The usual ratio is 40:60.

The VW Amarok with non-permanently engaged four-wheel drive is a specialist recommended for the most difficult offroad requirements. This conventional solution is widely used in the pickup segment and features a transfer case and dog clutch, giving a rigid link between the axles at the push of a button. An additional offroad reduction stage (1:2.72) for the six-speed transmission permits driving at extremely low speeds. This is possible even on particularly steep gradients of up to 100 % (corresponds to 45°) with a full payload, ❽.

CHASSIS

The front axle of the Amarok has been designed as a double wishbone concept, ❾, with a tall support base. This design, combined with a suspension travel of

ENGINE	2.0 L 120 KW CR TDI BITURBO R4				2.0 L 90 KW CR TDI MONOTURBO R4				
DRIVELINE	4×2		4×4		4×2		4×4		
	REAR-WHEEL DRIVE		NON-PERMANENT DOG CLUTCH	PERMANENT TORSEN DIFFERENTIAL	REAR-WHEEL DRIVE		NON-PERMANENT DOG CLUTCH		PERMANENT TORSEN DIFFERENTIAL
Displacement [cm³]	1968				1968				
Power [kW, (PS)] / engine speed [rpm]	120 (164) / 4000				90 (122) / 3750				
Torque [Nm] / engine speed [rpm]	400 / 1500 - 2000				340 / 2000				
Driving performance v_{max} [km/h] / n [rpm], (depending on the model), 5th gear	182 / 4120		181 / 4100		163 / 3870		161 / 3820		
Acceleration [s] 0 → 80 km/h	7.5		7.7		8.4		8.7		
0 → 100 km/h	10.8		11.1		13.5		13.7		
Elasticity [s] 80→120 km/h in 5th gear	10.5		11.0		12.5		13.5		
Fuel consumption acc. to 1999/100 EG* for centrifugal class [lbs]	4250	4500	4500	4750	4250	4500	4250	4500	4750
City [l/100km] CO_2 [g/km]	9.4 246	9.5 249	9.6 251	9.7 254	9.5 251	9.6 253	9.6 253	9.7 256	9.8 259
Interurban [l/100km] CO_2 [g/km]	6.7 177	6.8 179	6.9 182	7.0 185	6.3 166	6.4 169	6.4 169	6.5 172	6.6 174
Total [l/100km] CO_2 [g/km]	7.6 199	7.7 203	7.8 206	7.9 209	7.5 198	7.6 199	7.6 199	7.7 203	7.8 206
Range [km]	1050	1030	1020	1010	1060	1050	1050	1030	1020

❼ Driving performance and fuel consumption data with BiTDI and TDI engine (* preliminary values for 2.0 l 90 kW CR TDI)

190 mm and permitted front axle load of up to 1385 kg, permits the necessary level of robustness to be achieved for a pickup, at the same time as offering very good comfort properties. Driving dynamics are promoted by an anti-roll bar with 28 mm diameter, as well as a steering gear with variable steering ratio which is used for the first time in this vehicle class.

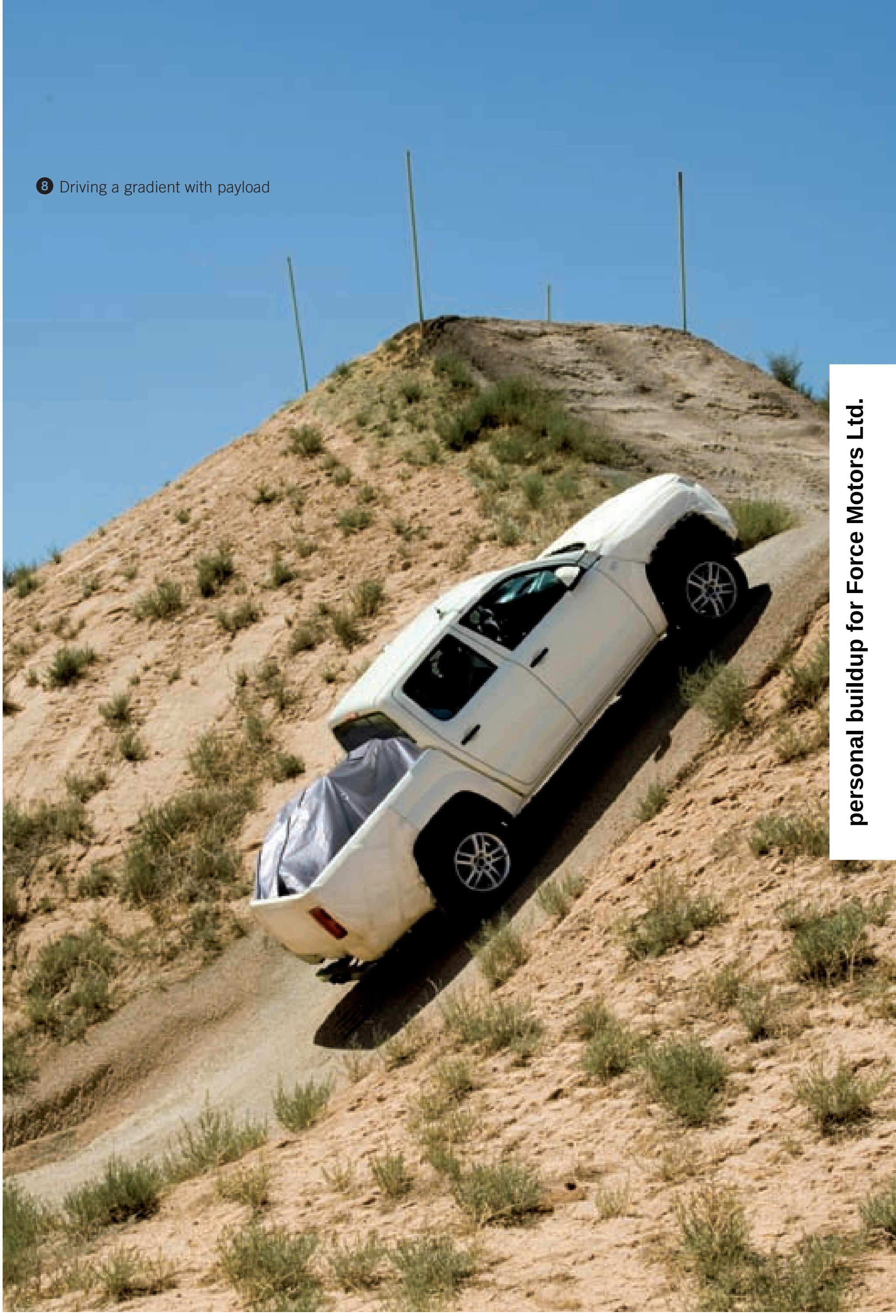
The rear axle is characterised by a maximum permitted axle load of 1860 kg in conjunction with a 3-plus-2 trapezoidal sliding spring. The leaf spring principle used with a specifically adjusted support layer results in a high progression and, in conjunction with the damper set-up, guarantees good overload reserves at the same time as a high level of suspension comfort.

When a particularly comfortable ride is required, it is possible to select an optional 2-plus-1 leaf spring with a permitted rear axle load of 1600 kg. This requires less progression during the rolling procedure of the support layer; in conjunction with optimised damper and buffer set-ups, the result is a level of comfort that has not previously been matched in this vehicle class. The necessary investigations of the S-shaped deformation behaviour were performed by simulations on a validated multi-body system leaf spring model.

The very good and agile driving characteristics of the Amarok are achieved by a marked toe-out setting of the wheels on the front axle. This delivers a significant level of driving stability during suspension compression and when exposed to lateral forces, as well as with a heavy load on the rear. Support is given here by the rolling understeer of the rigid rear axle which is generated by having the leaf springs angled forwards at 9.6°.

The front axle brakes take the form of ventilated 16-inch disc brakes with a dual piston sliding calliper principle. The brake system is rounded off by a drum brake on the rear axle. Even when subjected to heavy load, a 10-plus-10 inch tandem brake servo ensures there is adequate power in reserve even when the brake pedal force is low.

All Amarok versions in Europe are also equipped with ABS (including offroad mode), TCS, EDL and ESP. These electronic safety systems are also available to assist the driver even when not on consolidated roads. Take the example of offroad ABS: Pressing a button next to the selector lever reduces the braking distance significantly on unconsolidated ground. In conjunction with the ESP, a hill descent assist is also activated at speeds below 30 km/h. On extremely precipitous gradients, this applies the brakes specifically in order to prevent inadvertent acceleration of the vehicle – and it functions both in reverse and forwards. When climbing, a hill-hold assist ensures that the Amarok will not roll backwards on gradients when the brake is released.



8 Driving a gradient with payload

ELECTRICS/ELECTRONICS

The headlights are equipped with H7 bulbs for the dipped beam and H15 for the combination daytime running lights with full beams. They offer extremely effective illumination of the carriageway, whilst their low fuel consumption allows further reductions in CO_2 emissions. The complete integration of aerials (for AM/FM, GPS and mobile telephone) into the exterior mirrors is a special feature in this vehicle class.

❾ Double wishbone front axle with high support base and anti-roll bar

❿ Body and crash structure

The high-quality CD radios are the latest modules from the Volkswagen system, and guarantee outstanding sound quality with up to six loudspeakers. The optional RNS315 navigation system also allows cross-border navigation without difficulty, and is combined with a Bluetooth telephone hands-free system. Additional 12-V sockets in the dashboard, centre console and on the load bed guarantee a high degree of flexibility. Another new feature for this vehicle category is the high-level LED brake light with integrated load space lighting.

PASSIVE SAFETY

The structure and occupant protection concept of the Amarok has been designed for robustness and safety, with the help of results from component and crash tests conducted at a very early stage, as well as extensive simulations. The combination of high rigidity in the cab structure and precisely defined deformation zones forms the basis for good crash performance, ❿. The optimum behaviour of the front longitudinal members has been achieved by measures such as a graduated and controlled deformation behaviour in the area of the engine and running gear connections.

In the event of a crash, lateral support for the bodywork is provided on the frame structure by means of innovative deformation elements. This significantly reduces penetration into the passenger compartment in a side-impact collision.

The front protection system is chiefly composed of the driver and front passenger airbags in combination with three-point seat belts, belt force limiters, belt tensioners and an energy absorbing safety steering column. The specially designed head restraints on the front seat also counteract whiplash trauma in case of an accident. Head/thorax side airbags integrated into the backrests of the front seats also provide extensive protection in case of a side impact. The airbag triggering system comprises a control unit on the central tunnel with internal 3D acceleration sensors. Two sensors in the front area of the body combined with two satellite sensors ensure side-impact collision detection.

CONCLUSION

The Amarok by Volkswagen Commercial Vehicles sets new standards in its competitive environment. No mid-size pickup before has been as durable, comfortable and safe. Furthermore, the Amarok sets new standards in its segment with regard to specific practical advantages. The wide range of its positive properties gives it the potential to establish pickups in this category in Europe as well. The first feedback and test results clearly show that the Amarok is well on the way to becoming a trendsetter.

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The new Audi A8 in the aeroacoustic wind tunnel

THE AERODYNAMICS AND AEROACOUSTICS OF THE NEW AUDI A8

With a drag coefficient of $c_D = 0.26$ the new Audi A8 4.2 FSI is ahead of its rivals. Low lift coefficients at the front and rear axles provide a dynamic driving experience and ensure supreme dynamic stability at high motorway speeds. The standard of wind noise represents a new milestone in the luxury saloon car class and emphasises the harmonious overall impression made by the new Audi A8 in this segment. Aerodynamics and aeroacoustics therefore make a significant contribution to efficient, dynamic and comfortable travel in the new Audi A8.

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AERODYNAMICS – EFFICIENT, DYNAMIC TRAVEL

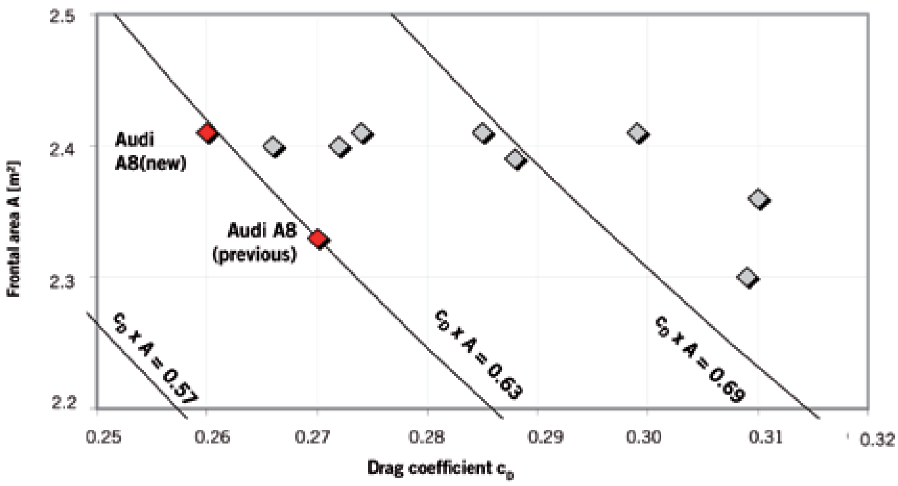
Aerodynamic development work for the new Audi A8 had to pursue an ambitious goal, since the previous model was already at the top of the competition. The biggest challenge was to keep the product of c_D and A, with its relevance to drag, down to the level reached by the previous model – in other words to compensate for an increase in frontal area A by reducing the drag coefficient c_D.

Although the frontal area had to be increased from 2.33 m² to 2.41 m² to satisfy the demand for a larger interior, a wider track and new statutory provisions regarding the size of exterior mirrors, targeted development work succeeded in lowering the drag coefficient c_D from 0.27 to 0.26 and thus compensating for the increase, ❶. As on the previous model, low lift coefficients at the front and rear axles permit tuning of the suspension for agility in the lower speed range, but provide the necessary dynamic stability at high speeds.

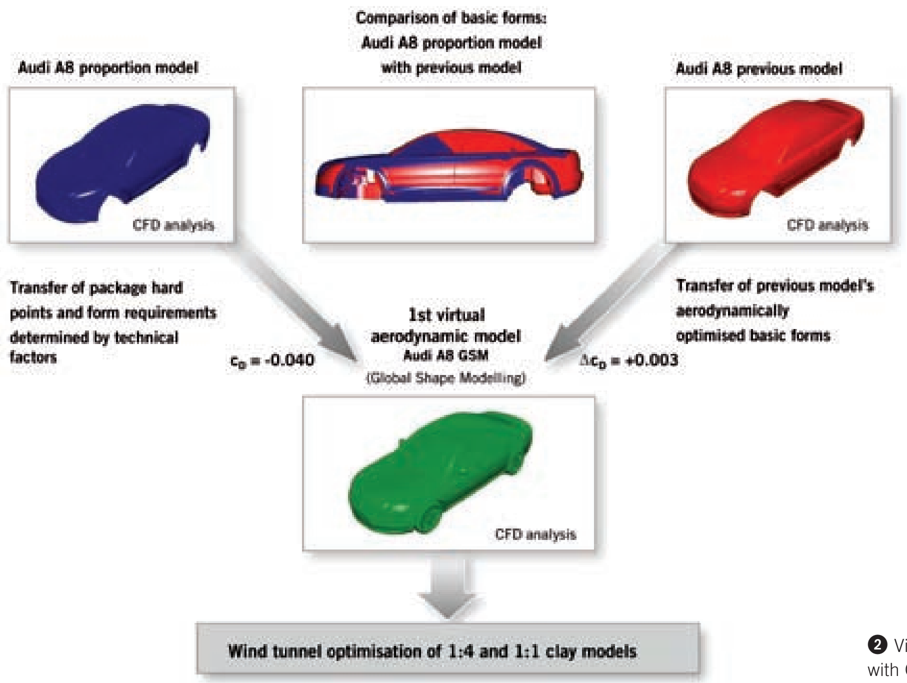
In order to achieve the ambitious aerodynamic objectives, development activity concentrated throughout the product creation process, that is to say from the concept and design phase to series-production development, on the following theme areas, which will be discussed below in detail:
- development of basic shape and detail optimisation of car's outer skin
- underfloor design
- efficient airflow through engine compartment and its interaction with airflow round the car
- wheels and tyres
- lowering the ride height with the air suspension.

The art of tuning the car's geometry in terms of aerodynamics is to shape the contours in such a way that drag and lift are reduced. The aerodynamicist, however, does not determine the shape of the car alone. Close cooperation was needed between the departments responsible for aerodynamics, design, dimensional concept and component design so that the aerodynamic potential in the various theme areas could be interpreted successfully on the series product.

CAR	c_D	A	c_D x A	c_{av}	c_{ah}
Audi A8 4.2 l FSI (previous model)	0.27	2.33	0.63	0.10	0.04
Audi A8 4.2 l FSI (new model)	0.26	2.41	0.63	0.10	0.04



❶ Chart of aerodynamic coefficients for the new Audi A8 compared with the previous model and competitors' c_D x A values



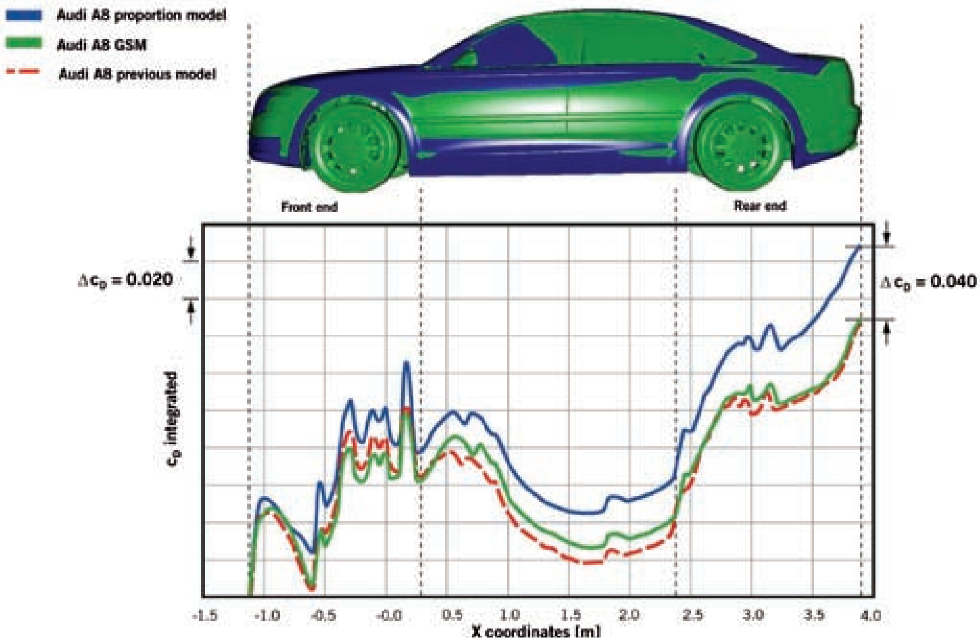
❷ Virtual basic shape optimisation process with GSM and CFD

DEVELOPMENT OF BASIC SHAPE AND DETAIL OPTIMISATION OF THE CAR'S OUTER SKIN

Development of the basic shape of the new Audi A8 was carried out to a large extent in the early concept phase with the aid of CFD (Computational Fluid Dynamics) numerical flow simulation. The main aerodynamic challenge when developing the basic shape were the specified dimensional concepts; these envisaged a longer front and a shorter rear body compared with the previous model, and also included above package changes. For the first time a surface modifying tool was employed for virtual shape optimisation: it enabled the outer skin of the previous model, with its excellent airflow, to be adapted to suit the new package (Global Shape Modelling or GSM). ❷ shows a virtual aerodynamic model (A8 GSM); this was the first time that a virtual aerodynamic model had been created using GSM in the concept phase of an Audi vehicle project.

In the CFD analysis of the various basic shape variants that followed, the complete 3D flow field round the car was computed; this enabled results such as the drag and lift coefficients to be extrapolated. ❸ illustrates assessment of the various basic shapes against the cumulative c_D values plotted against vehicle length (x coordinates). This diagram clearly indicates the points at which the basic vehicle bodies exhibit advantages or disadvantages.

The advantages of this procedure were the availability even in the early concept phase of a virtual aerodynamic model possessing high aerodynamic quality and also taking the "package hard points" into account and fulfilling shape requirements that were unavoidable for technical reasons. Compared with the first A8 proportion model, a c_D improvement of $\Delta c_D = 0.040$ was achieved, equivalent to a 15 % reduction in overall drag.



❸ c_D plotted against x coordinates (comparison between A8 proportion model, A8 GSM and previous A8)

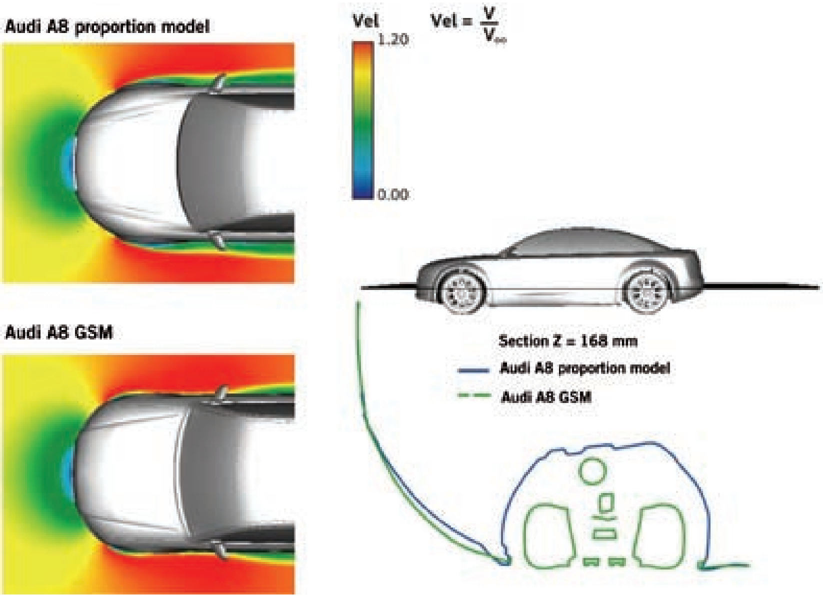
The aerodynamic potential identified on the A8 proportion model in the front end (for instance tapering the nose and embedding the wheel arches in the side panels for improved airflow, ❹) and rear-end areas (for instance the contour at the rear of the side panel where the flow breaks away, ❺) was incorporated into the design at an early stage.

The virtual aerodynamic model was therefore an ideal starting point for subsequent detail optimisation of the car's outer skin using aerodynamic and design models in the wind tunnel.

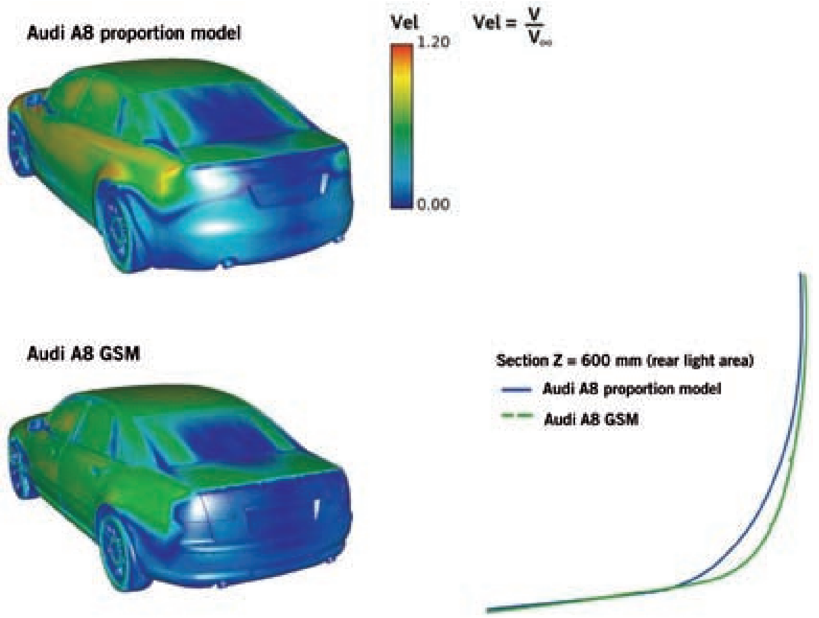
The main work area during detail optimisation on 1:4 and 1:1 clay models was the rear end of the car. Inward movement of the D-pillar, the height and length of the boot lid and the radius at its rear edge are important shape parameters when tuning the complete airflow from the rear end of the car and therefore for reductions in drag and rear-axle lift. The high pressure achieved by shape optimisation at the boot lid and basic rear-end surface makes a major contribution to the new Audi A8's low drag and low rear-axle lift. The dynamic rear-end taper and formally well integrated separation edge at the rear end of the side panel cause the defined amount of flow separation at the side that is necessary for dynamic stability.

During detail optimisation of flow at the rear end of the body, special attention was devoted to interaction between flow round the car, flow under the car and flow through the engine compartment. With its pioneering road simulation technique (boundary layer control, moving belt, rotating wheels) the Audi aeroacoustic wind tunnel is an ideal development tool. In addition to physical aerodynamic models with realistic representation of flow through the engine compartment, flow computing also plays a part in detail optimisation.

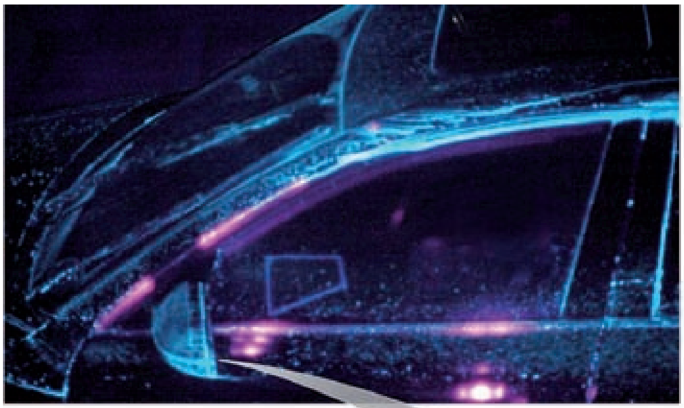
A further area to which special care was devoted during shape optimisation was the A-pillar area with rain gutter and exterior mirror. Since the exterior mirror projects from the body and is therefore a disturbance factor from an aerodynamic standpoint, and since at the same time the airflow is greatly speeded up in the A-pillar area, aerodynamic optimisation of the exterior mirror is especially important. The specific challenge when optimising the shape of the exterior mirror is to obtain the best possible combination of different characteristics: the size of glass called for by law, the aerodynamics, the aeroacoustics, the design and the need to keep the mirror glass and the side window free from dirt. In an intensive optimising process with several development loops, a mirror shape was obtained that satisfied all requirements in the best possible way. Thanks to the good flow properties of the basic shape, the proportion represented by the two exterior mirrors at $\Delta c_D = 0.009$ is only 3 % of total drag, and it was possible to reduce the noise of airflow round the mirrors to a level not perceptible inside the car. To ensure that the mirror glass remains clean when driving in the rain, the water droplets striking the



❹ Tapered nose, wheel arches embedded into side panels (CFD comparison of A8 proportion model and A8 GSM)



❺ Contour at rear of side panel without and with flow breakaway (CFD comparison of A8 proportion model and A8 GSM)



❻ Even when driving in the rain there is a clear view to the rear through the side window and on to the mirror glass

mirror housing are guided outwards by a groove on top of the housing and a deflector edge at the bottom. The water droplets thrown off there and carried along by the airflow no longer strike the side window in the view area. The rain gutter along the A-pillar also helps to keep the side window clear when driving in the rain. Rainwater moved by the windscreen wiper towards the A-pillar is trapped in the space provided under the rain gutter and diverted away along the A-pillar, either downwards into the water plenum chamber or upwards over the roof, ❻.

UNDERFLOOR DESIGN

After the basic shape had been developed, further major optimisation potential existed on the underside of the car, ❼. The development objective in this area was to keep the underside as smooth as possible and therefore promote good airflow over it.

The sequence of aerodynamically effective underfloor elements starts with a three-section front-end cover extending back from the lower edge of the front bumper cover. This allows the airflow to reach the centre section of the floor pan in a controlled manner and free from turbulence. Components of the car that could reach critical temperatures (such as the transmission) are systematically cooled by air guides with integral NACA ducts. These ensure effective cooling with only a slight increase in drag.

At the transition between the end section of the pedestrian protection panels and the engine compartment cover, a front spoiler lip reduces front-axle lift. Wheel spoilers are installed ahead of the front wheels and extend as far as the centre of the inside of the wheel arch lining. One of their tasks is to reduce flow losses between the wheel arch and the wheel; another is to keep splash water away from the front brakes.

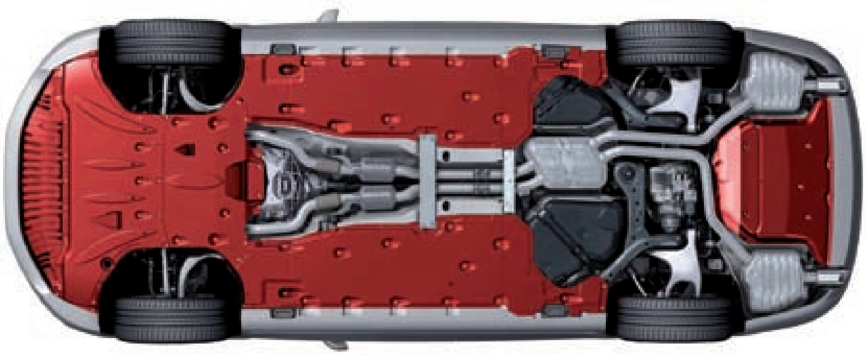
To the left and right of the transmission tunnel there are two smaller covers that provide a smooth transition from the front wheel arch lining to the two large-area central underfloor panels. The material used for the floor pan cover elements is highly absorbent and therefore has additional acoustic advantages. At the end of the centre section, two more covers with a drag-reducing function extend to a point just ahead of the rear wheels and cover the cavities in the vicinity of the fuel tank.

At the rear end of the car two covers at the spare wheel well, the underside of the secondary silencer and the underside of the rear bumper cover form the surface of a rear-end diffuser, which reduces drag and rear-axle lift.

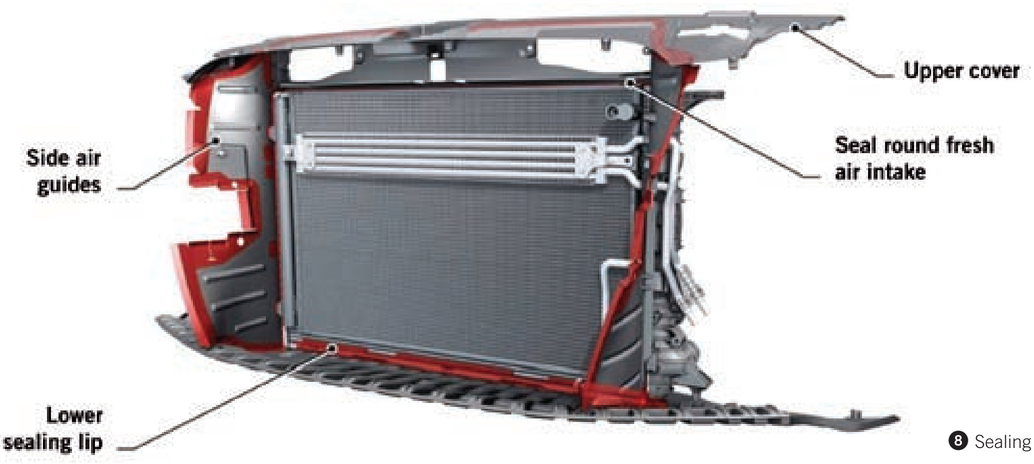
Altogether, this large number of individual underfloor measures achieves an improvement in the drag coefficient of $\Delta c_D = 0.033$, representing 13 % of total drag.

EFFICIENT UNDERHOOD AIRFLOW AND ITS INTERACTION WITH AIRFLOW ROUND THE CAR

In unfavourable circumstances the proportion of total drag accounted for by airflow through the engine compartment can be as high as 13 %. Important factors are internal friction and loss of momentum, but also leaks and the interaction between the air leaving the engine compartment and the airflow around the car. For the new Audi A8 the aim was therefore to make the airflow through the engine compartment as efficient as possible, with low drag.



❼ Underfloor of the new Audi A8 designed for good airflow



❽ Sealing measures round the radiator

Cooling air reaches the radiator almost without leakage, thanks to optimised sealing in the radiator area, ❽. In addition to the air guides at the sides, two sealing elements above the radiator and a sealing lip against the end of the pedestrian protection element below the radiator ensure that the airflow is free from leaks. Altogether, these measures resulted in an improvement of $\Delta c_D = 0.006$ to the drag coefficient, and reduced the proportion of total drag attributable to airflow through the engine compartment to only 8 %. At the same time, the sealing measures around the radiator increase the significant air mass flow for cooling the mechanical assemblies by 25 %.

Depending on the amount of cooling air needed by the engine and on the national-market version (hot or cold country), a fixed grille closure is inserted into the cooling air inlet and reduces cooling air drag still further.

The choice of apertures through which the air can leave the engine compartment also influences aerodynamic drag, since in this area undesirable interaction with the airflow over the car's body can be avoided.

The two-section front wheel arch lining is used without the need for complex additional parts to vary the pressure drop from engine compartment to wheel arch in such a way that the air leaving the radiator is diverted to a considerable extent toward the transmission tunnel. The outgoing airflow then flows in an effective, controlled manner to the rear via the tunnel and combines with the flow under the car.

WHEELS AND TYRES

The flow of air round rotating wheels also accounts for a high proportion of the vehicle's total drag; aerodynamic optimisation is therefore worthwhile in this area too. In addition to tyre geometry, wheel design has a major part to play. When wheels are designed, care has to be taken to provide an adequate flow of air to the brakes, and the wheels must not be too heavy. The new Audi A8 4.2 FSI has extremely light wheels as standard equipment that also ensure good airflow.

LOWERING THE RIDE HEIGHT WITH THE AIR SUSPENSION

Air suspension, which is standard on the new Audi A8, offers further aerodynamic improvement potential. Depending on the air suspension's operating mode (Comfort, Automatic or Dynamic) and the car's road speed, ground clearance and with it aerodynamic drag are reduced. Above a speed of 120 km/h in the Automatic or Dynamic operating mode, the car's ground clearance is lowered by 20 millimetres from the standard ride height to the "motorway" height. This reduces the drag coefficient by $\Delta c_D = 0.008$.

Compared with the initial A8 proportion model, a total improvement of $\Delta c_D = 0.050$ was achieved on the series-production car, which reduced its fuel consumption according to the NEDC cycle by 0.18 l/100 km (and its exhaust emissions by 4 g CO_2/km). Customers who regularly cover long distances on motorways or similar roads will, at an average journey speed of 130 km/h, save 1.3 l of fuel per 100 km and reduce the car's emissions by 30 g CO_2/km.

AEROACOUSTICS – TRAVEL REFINEMENT

Above a road speed of approximately 120 km/h, wind roar becomes the dominant source of noise inside the car; other noise sources, such as those emanating from the engine, driveline or tyres play a less important part. In a large luxury saloon with the emphasis on comfortable travel, which may be used frequently for high-speed motorway journeys, wind noise therefore has a strong influence on overall interior noise and therefore on the degree of refinement provided by the car. A low level of wind noise even at high speeds also makes it easier for the car's occupants to converse normally and understand what is being said.

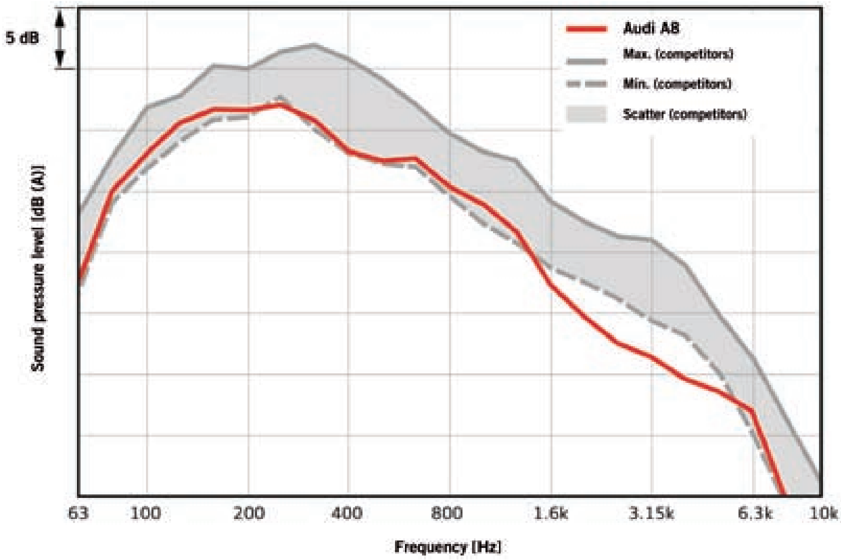
Wind noise is normally perceived as broad-band spectral noise, and can have several causes. The initial factor determining occupant comfort is for it not to be possible to locate the sources of individual aeroacoustic noise sources directly and that no cumulative level, as for instance with whistling noise, should be present. Furthermore, a low level at frequencies between 1 kHz and approximately 5 kHz is an important factor in subjective noise perception, since this is the frequency range in which the human ear possesses the greatest sensitivity, and broadband noise is therefore sensed as being particularly disturbing.

When the new Audi A8 project commenced, very high standards of interior refinement were laid down. The target was for the previous car's wind noise level to be reduced and a leading position occupied in relation to competitors' cars. The wind tunnel measurements in ❾ indicate that the wind noise levels of the new Audi A8 are at the lower end of the scatter recorded by competitors' models, and actually below them in the decisive frequency range between 1 kHz and 5 kHz.

For aeroacoustic development work on new vehicle projects, Audi operates one of the world's quietest aeroacoustic wind tunnels. Using the latest dummy head measuring technique, the wind noise reaching the ears of all the car's occupants is recorded and subjected to spectral analysis. However, the efficacy of individual wind noise reduction measures is not assessed by means of objective measurements alone. Subjective assessment of noise quality by an experienced acoustic engineer, both in the wind tunnel and during road tests, remains essential. His or her personal impressions of sound volume and character are helpful when the objective measured data are evaluated later, for instance when conducting comparative listening tests in the acoustic studio.

To achieve the ambitious target in this case, aeroacoustic development activities were concentrated on the following topic areas:

: reducing noise sources on the outer skin (airflow over body)
: provision of adequate noise insulation between unavoidable external noise sources and the car's interior (e.g. window glass)
: sealing the body to prevent wind noise sources from reaching the interior directly (e.g. door seal system).



❾ Wind noise levels for the new Audi A8 compared with competitors

REDUCING NOISE SOURCES ON THE OUTER SKIN – AIRFLOW OVER BODY

Since airflow over the body has a strong influence on the interior noise spectrum across the entire frequency range, much detail work was invested in reducing this source of wind noise. The basic shape of the new exterior mirrors was aeroacoustically optimised and, in their lowered position, they were integrated optimally into the airflow around the A-pillar, ❿. With its rounded-off front edge, the rain gutter on the A-pillar prevents disturbance caused by turbulence noise, and the windscreen wipers, which are deeply recessed into the scuttle, do not interfere with the airflow. At the front end of the car, stepped areas in the surfaces and joint gaps were kept as small as possible in order to avoid turbulent breakaways with negative aerodynamic and aeroacoustic effects. Joint gaps exposed to the airflow in the windscreen and rear window areas were sealed by spraying round the glass, to prevent disturbance arising as a result of the airflow reaching the cavities below them. On the outside, the window guide seals are chamfered where they contact the glass; the resulting airflow is almost entirely free from turbulence and therefore avoids unnecessary wind noise.

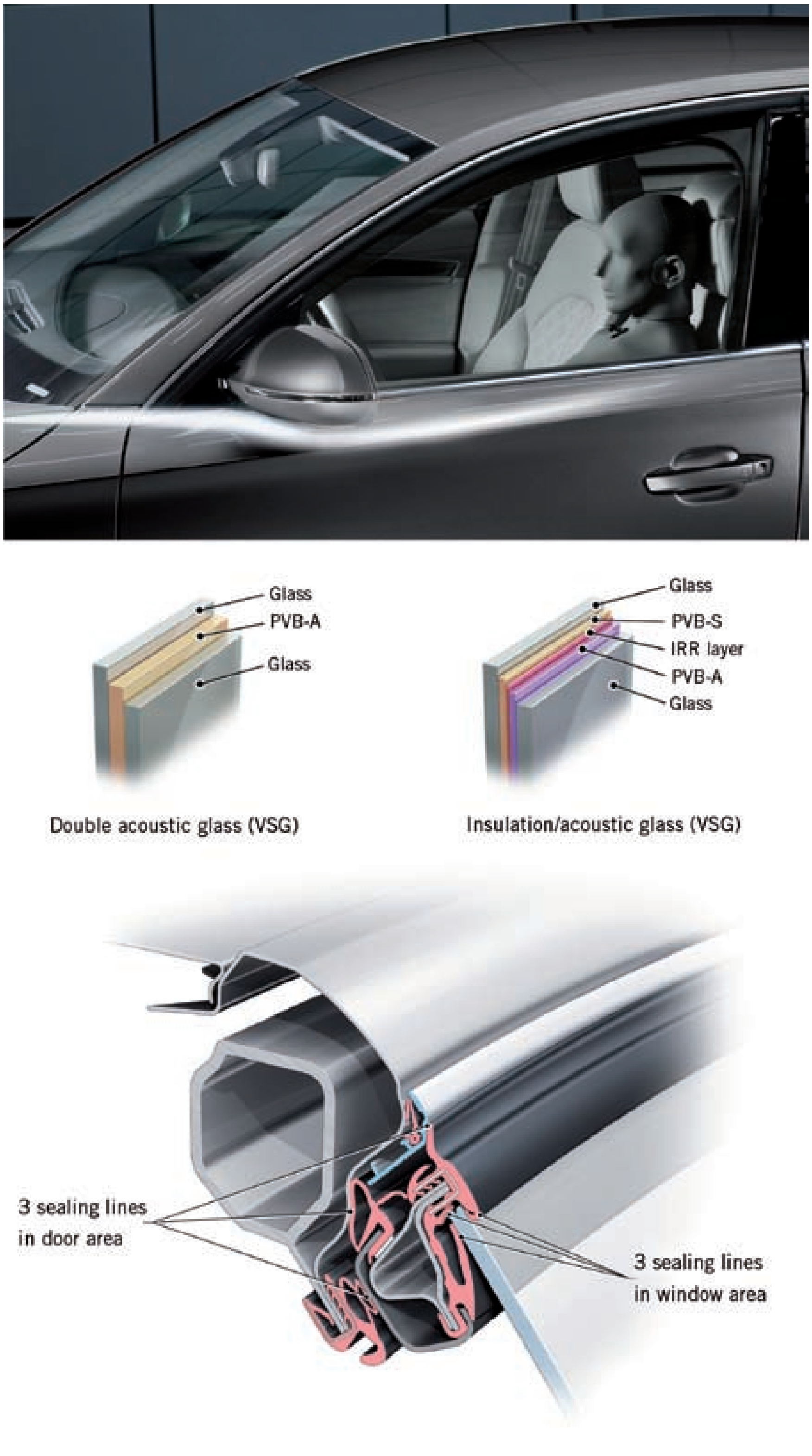
NOISE INSULATION BETWEEN UNAVOIDABLE EXTERIOR NOISE SOURCES AND THE CAR'S INTERIOR

As a means of satisfying the most extreme demands for refinement, the door windows are available in two versions, each in 5 mm thick laminated safety glass (either double acoustic glass or insulating acoustic glass, ⑩). Both versions feature a special acoustic foil intermediate layer that reduces the breakdown in airborne noise insulation that takes place for physical reasons at the coincidence frequency. The noise insulation provided by the door windows is specifically raised in the frequency range between 1 kHz and 5 kHz, which is relevant to wind noise refinement, and the interior noise in this frequency range thus reduced.

To reduce the noise input from the front, which is due to the airflow, the windscreen also has an acoustic interlayer. The laminated safety glass improves the wind roar situation and also suppresses the noise of passing vehicles more strongly and therefore adds still further to the sense of comfort inside the new Audi A8.

SEALING THE BODY TO PREVENT WIND NOISE SOURCES FROM REACHING THE INTERIOR DIRECTLY

The door concept of the new Audi A8 is a systematic development based on the Audi A4, Q5 and Q7, but designed to suit the lightweight material, namely aluminium, used for this car's body. In contrast to the previous aggregate door concept with its separate door case and door frame, the new Audi A8 has a fully pressed sheet-metal inside door panel, that is to say the door frame is integrated into the door structure. The main door-side seal therefore makes uninterrupted contact with the body. The inner panel shields the mirror base triangle against external noise sources. A sealing concept specially developed for the door-in-white makes the process sustainably reliable. For assembly



⑩ Aeroacoustic measures: optimised flow around A-pillar and mirror, door windows in double acoustic glass or insulating acoustic glass and new door concept (fully pressed) with 3 sealing lines in door and window area

reasons a few apertures remain between the inner door panel and the inside door trim: these are carefully sealed with plastic caps. This new construction principle distinctly increases door frame rigidity, thus reliably preventing the door from being sucked out by negative wind pressure or angled airflow at high speeds. A high standard of sealing and therefore of acoustic refinement is assured by the provision of three sealing lines in the door and door window areas, ⑩.

Special attention was devoted to transitions during detail optimisation of the individual profiles and moulded elements of the sealing system. The inner window aperture seal for instance is vulcanised to the window guide section to reliably prevent noise from reaching the car's interior from the door case. Other detail improvements concerned the loop-pattern door handles, the concept of which provides more effective sealing than the previous version.

CONVENIENT SUNROOF

The sunroof's wind deflector is controlled according to the car's road speed. It has two positions: the lower one is the high-speed setting, intended to keep broadband noise low and avoid draughts when the sunroof is fully open. The wind deflector's upper setting has been chosen to suppress judder at the Helmholtz resonance, which has been deliberately selected to occur at low road speeds. Side trim panels on the sunroof panel keep interior wind noise low when the sunroof panel is raised.

A further important factor in creating a harmonious overall impression is for the body to have good acoustic quality. Very high dynamic rigidity leads to reduced sensitivity to excitation from the engine or the road surface. Noise transmission through structural body cavities is counteracted by cavity bulkheads and foam material at defined points. Body joints are systematically blanked off by means of fine joint-seam seals. Since the body-in-white can behave as a thoroughly effective megaphone when noise is introduced, it is also necessary to prevent sheet-metal surfaces from vibrating, for instance when excited by the wind. Weight-optimised ratings for anti-drumming and reinforcing foils were closely coordinated with overall vehicle acoustics.

The combination of all the measures stated here enables the new Audi A8 to set a new standard in the luxury saloon car class for wind noise suppression quality, and emphasises the harmonious overall impression that it creates in this model category.

POWERSHIFT MODULE
COMBINATION OF FRICTION BRAKE AND PLANETARY GEARSET

Enabling a seamless powershift in layshaft-type transmissions requires (at least) two independently operable transmission systems between engine and driven wheels. Preferably, these two transmission systems each have their own electronically controlled clutch, which can be of dry or wet type. For one of these two clutches, Drivetrain Innovations (DTI) proposes an innovative mix of a friction brake and a planetary gearing. This constitutes a new powershift module, and by using more carry-over parts of the manual transmission, new and cost-effective powershift transmissions can be designed.

NEW POWERSHIFT MODULE CREATES A DCT OUT OF AN AMT

A vehicle's fuel economy and driving comfort relates closely to the adopted transmission technology. Around the year 2000, the offerings of transmission types where almost tripled with the advent of continuously variable transmissions (CVT), automated manual transmissions (AMT) and dual clutch transmissions (DCT), complementing the existing manual transmission (MT) and converter automatic transmission (AT) technologies. The main drivers for this fast transmission diversification lies in finding new compromises in low cost, good fuel economy, driveability, power level and brand identity by the automotive industry. In Europe, the transmission innovations AMT and DCT rely on the wide scale experience with the manual transmission, whereas in Japan an alternative for the AT was primarily found in CVT.

Drivetrain Innovations (DTI) focuses here on AMT and DCT improvements adopting the new powershift module conceived by DTI that can both enhance the AMT and DCT transmission types. In fact, the AMT can be improved approaching DCT functionality, while still relying on the original and simple MT transmission layout. Secondly, current DCT structures can be made more cost-effective and approach torque converter-like functionalities. DTI's solutions aim at reduced packaging while utilizing as many MT carry-over components as possible. This approach reduces both the transmission development cost and the production cost to a minimum.

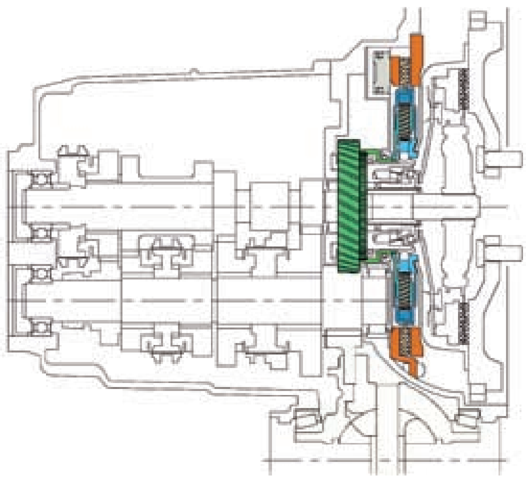
Most importantly, automation of manual transmission technology (both single clutch and dual clutch) is a promising step to take within the context of oncoming new CO_2 regulations. CO_2 reductions up to 7 % over the MT can be achieved by optimized gear shift points. Particularly when the automated shifting occurs without hesitations, interruptions or other uncomfortable attributes, such CO_2 improvements can be achieved with no compromises on driveability and performance. This causes powershifting transmissions to become more and more important. This clear and outstanding transmission trend could be extended over the entire application field of passenger cars and light commercial vehicles. For the lower priced cars, on the one hand appropriate powershift technology cost should be found. On the other hand, dry clutch DCT should also be applicable to higher torque/power vehicle applications, where up to now only wet dual clutches apply. DTI's powershift module complies with these application fields.

POWERSHIFT TECHNOLOGY

From planetary automatic transmissions it is known that a fixed gear ratio with a planetary set can be selected by connecting either one of its three rotational members (ring gear, sun gear or the carrier) to the transmission housing. This connection is usually established through a wet multi-plated brake unit such that synchronization with the housing is possible while retaining power transfer.

DTI's powershift transmission technology uses such a brake-controllable planetary gearset as a short term torque bypass while the main transmission shifts towards a next gear. One may compare the functionality to a limited slip differential, which controls the distribution of left and right drive shaft torque using a slip-controllable clutch. The functionality of the brake-controllable planetary gearset can be used to fill up torque gaps that occur when shifting an AMT from one gear to the next.

The setup of a brake-controllable planetary gearset is functionally equivalent to a fixed gear ratio and a controllable by-pass clutch. However, the installation of such a by-pass clutch and gear requires redesign of



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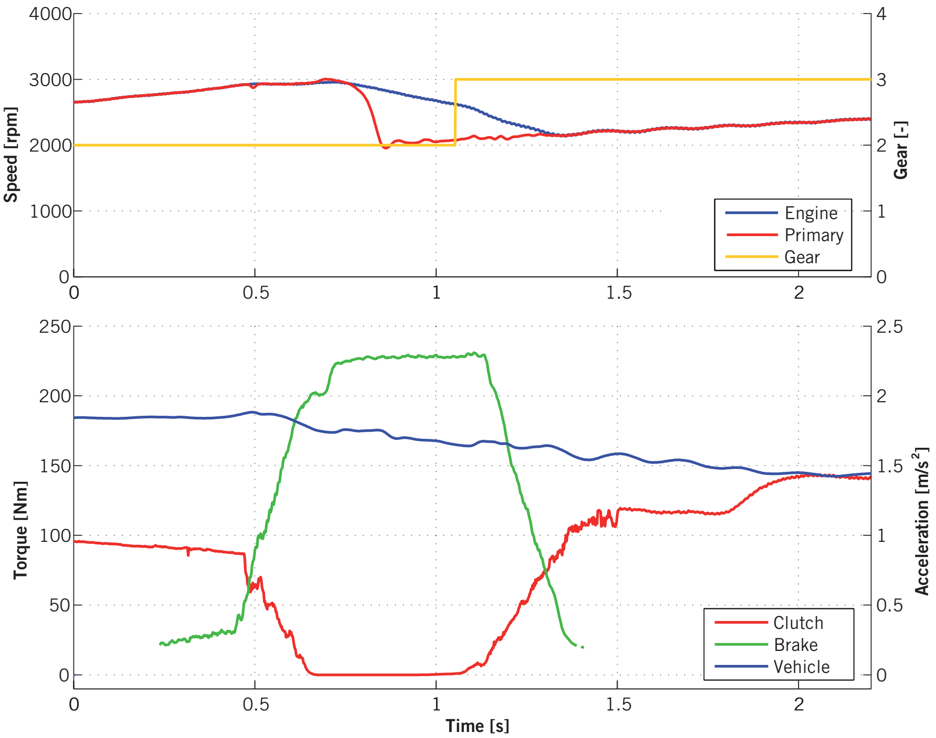
1 Cross-section example of a five-gear (automated) manual transmission including the new powershift module



❷ Double cross-shift of clutch and brake: planetary gearset transmits torque only during shifting of the main transmission (here: from second to third gear)

NO.	CLUTCH	PLANETARY GEARSET	BRAKE	NUMBER OF SHAFT SEALS
1	Dry	Dry/grease	Dry	2
2	Dry	Wet	Dry	3*
3	Dry	Wet	Wet	2*
4	Wet	Wet	Wet	1

❸ Useful combinations of clutch, planetary gearset and brake (*one of the shaft seals has zero rpm speed difference at a closed clutch)

the entire transmission if package neutrality is to be reached. On the contrary, the installation of a planetary gearset and brake in the bellhouse leads to package neutrality for central slave cylinder equipped AMTs, ❶. The combination of an AMT and the brake-controllable planetary gearset is termed "Powershift AMT" by DTI here.

The Powershift AMT still has a few interrupted AMT-typical shifts in the gears higher than fourth gear, the rationale of which will be explained further in the following. The main benefit of the Powershift AMT is that a standard MT can be "converted" gradually towards near-AT and -DCT functionality. In conclusion, the powershift module appears to be a very lucrative step when producing the MT, AMT and Powershift AMT systems on the same production line.

OPERATING MODE

The powershift module contains a planetary gearset where (usually) the ring gear can be torque controlled using the brake unit. The sun gear is connected to the main clutch cover (thus the engine crankshaft) and the planet carrier is connected to an output gear meshing with the transmission's final drive gear. Obviously, there are no connections with the transmission's input shaft, being the shaft that needs to be by-passed during the gearshift.

In ❷, a power upshift from second to third gear is shown. Initially, the main clutch is closed and the vehicle is driven in second gear. Then the first cross-shift occurs where torque is transferred from the clutch to the planetary gearset. During the pressurized phase of the slipping brake, the clutch is entirely opened, the main transmission is shifting to the third gear, the engine torque remains at steady or increasing level and the engine speed (shown in the upper diagram in ②) is decelerating towards the third gear speed. After synchronization of the third gear within the transmission, the second cross-shift is initiated. This implies that the clutch is re-engaged while the brake is depressurized. When the brake is entirely released, the clutch re-engages the engine to the primary shaft.

The powershift module including gear establishes a torque amplification factor similar to third gear. This realizes a torque shift functionalities as shown in ② for upshifts to second, third and fourth gear or downshifts towards third gear and lower, for example a kick-downshift from sixth gear to third, can be realized with retaining driveline power.

VARIANTS OF THE POWERSHIFT MODULE

The powershift module can be designed for different application purposes. The main characteristics of intended purposes are:
: to fill torque gaps of AMT shifts
: to use within a DCT structure
: to use the braked planetary gearset as a launch element.

DTI mainly focus here on the mechanical design where all functional elements are dry in combination with the first characteristic, thus used as an AMT torque fill module. An important design input of such a torque fill application is that the planetary gearset is only loaded during the very short shift periods.

The intermittent loading of the planetary gearset during shifting takes less than a second per shift and therefore this gearset does not necessarily require oil cooling for heat dissipation. For this reason, grease lubrication of the gearset also suffices for friction reduction. Such grease lubrication leads to the option of both a dry (grease filled) planetary and a dry brake. This dry-dry combination essentially constitutes a dry brake module next to the already existing dry main clutch. As such this configuration requires two shaft seals resembling the number that is also seen in dry DCT. Besides this "all-dry" combination, ❸ shows other useful combinations that can be designed for the clutch, planetary gearset and brake.

An advantage of the planetary gearset is its ability to trade teething width for planet gear count. In the current design, six planet wheels where chosen to accommodate the required strength for transferring the engine torque. In this way the planetary gearset can be designed as an extremely flat module and therefore enables installation around the central slave cylinder and behind the existing main clutch, ①.

The powershift module requires a modified bellhouse to accommodate the brake pressure plates and three actuation pistons (placed in 3 × 120° formation) for the brake, ❹. In totality, this design approach does not lead to any packaging penalties with respect to the existing (A)MT and can therefore be packaged easily in an (A)MT-based engine bay of the car. With respect to the donor AMT, the increase of weight due to the powershift module including its actuation and modified bellhouse, is about 7 kg.

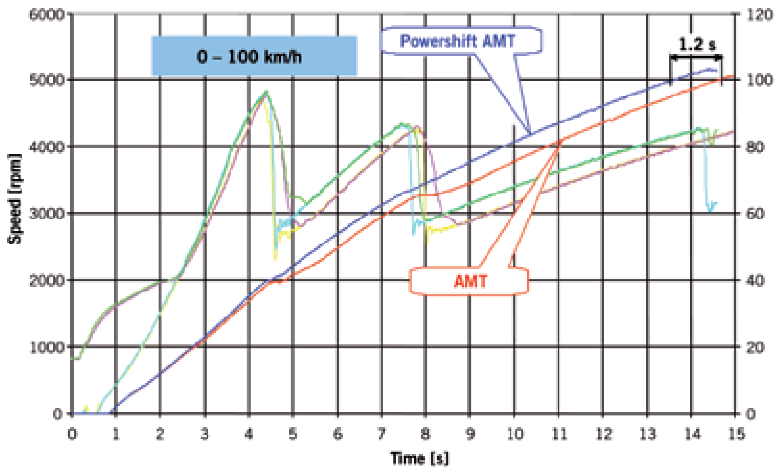
❹ Component breakdown of the brake-controllable planetary gearset (as powershift module)

TEST RESULTS

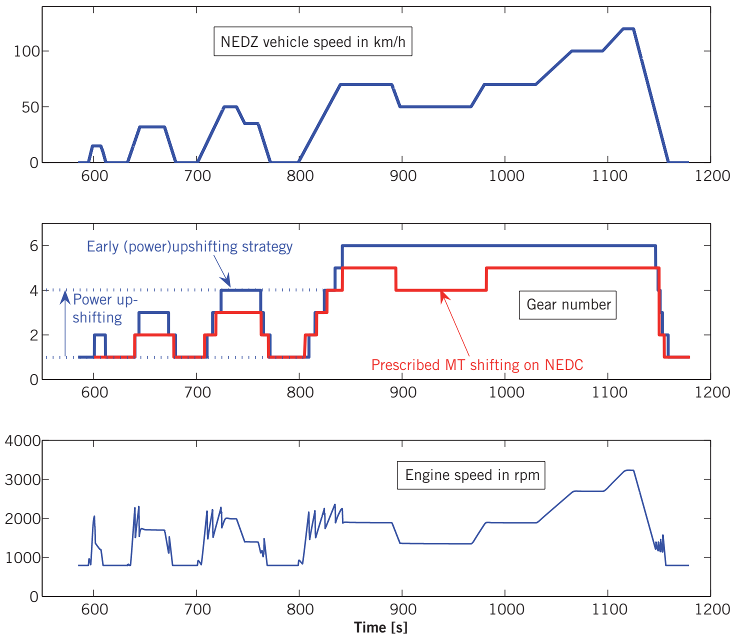
The powershift capabilities of the module are extensively tested with petrol and diesel engines. In ②, an upshift from second to third gear is shown for a B-class petrol car. From this ②, it can be seen that the double cross-shift procedure explained before is established within less than 1 s. In ❺, a 0 to 100 km/h acceleration with a B-class 1.4 turbo diesel engine car equipped with the powershift module is shown. Clearly in ⑤ to see, this sprint is improved by the torque fill capabilities by more than a second while the diesel engine and main transmission cluster are not adapted in any way.

In ❻, the fuel economy potential of the powershift module with "early upshifting strategy" is shown with respect to a prescribed manual shifting pattern on the New European Drive Cycle (NEDC). Adopting this early upshifting pattern, average engine speed is lower while the driveability is higher than in a normal AMT mode, which achieves a 6.3 % fuel consumption improvement.

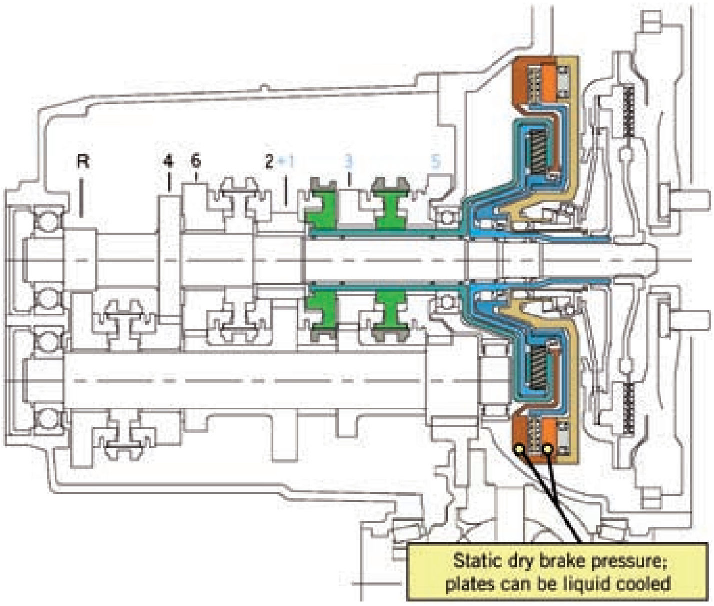
The upshifts towards fourth gear are powershiftable thus the higher frequency upshifting during city driving (ECE) does not deteriorate driveability and comfort level of the car. Next to this there is only one interrupted shift along the entire



❺ Sprint from 0 to 100 km/h for a diesel car with and without the powershift module activated in the AMT; engine and primary shaft speeds are also drawn for both cases



❻ Less fuel consumption by early power-upshifting strategy



❼ Usage of the brake-controlled planetary gearset as the “first clutch” within a six-speed DCT structure

NEDC drive cycle: the 4-5 shift during the first acceleration in the high speed part after 800 s. The 5-6 shift is performed at pedal release when the steady 70 km/h is reached. This shift is therefore not really felt as an interrupted shift.

POWERSHIFT MODULE IN DCT STRUCTURE

Item 2 and 3 mentioned in ③ require a more integrated design of the planetary gearset than the powershift module explained before. In fact, using the brake-controllable planetary gearset as the “second clutch” in a DCT structure requires a full lifetime specification of the planetary gearset and so must be oil lubricated. If it is used as the “first clutch” in a DCT structure, so including vehicle launching, creep, hill-hold etc., it also requires lifetime and cooling capabilities of the planetary brake system. ❼ shows a six-speed DCT where the engaged planetary gearset establishes gears 1, 3 and 5, whereas the engaged clutch establishes gears 2, 4 or 6.

Re-integration of a DCT structure where the “first clutch” is constituted by the brake-controllable planetary gearset renders a number of important advantages over existing DCT:

: Shorter transmission: The re-use of some gear pairs enables a shorter transmission and thus no multiple layshafts are required.
: Slender transmission: The second gear to the planetary gearset establishes a first gear equivalent. So, the first gear pair can be left out which enables a less asymmetric loading as well as a shorter center distance between input and output shaft and thus a more compact transmission.
: Active cooling: Regardless whether the brake system is dry or wet, it can be actively liquid cooled to guarantee the lifespan of the launch-brake when used for heavy hill/trailer launches, hill-creep and traffic congestion circumstances. Liquid cooling of the dry brake system can be established by integration of cooling channels in the static brake pressure plates and accompanied with a radiator system possibly integrated with the engine cooling system.

SUMMARY AND OUTLOOK

Drivetrain Innovations (DTI) shows that the brake-controllable planetary gearset is a new versatile mechanical clutch module that can be used for powershift functionalities in three ways: first as torque filling for automated manual transmission, second as one of both clutches in a dry or wet dual clutch transmission structure and third as an actively liquid cooled launch element.

The powershift functionalities addressed in this article can be used in a wide vehicle range. For A/B/C class passenger cars, where manual transmissions (and to a lesser extent automated manual transmission) are extensively applied, the new powershift module is a cost-effective upgrade. Higher torque automated manual transmission applications in sports-performance cars are also candidate for upgrade modules, essentially only to support city driving comfort.

Applications of the low-cost six-speed dual clutch transmission structure may fit well in the C/D class cars as well as light commercial vehicles. If for such an application the controllable brake is applied for launching the vehicle, thermal issues can be easily dealt with by active liquid cooling in the static brake pressure plates. As a result, the launching and vehicle creeping capabilities can be improved significantly which enables dry powershift technology in high torque applications found in D/E/Premium/M/J segment cars.

Finally, the module is also applicable to heavy duty trucks with automated manual transmission, where fuel consumption is a driver for powershifting capabilities. Together with the global automotive industry, DTI together with OEMs worldwide is developing powershift modules towards production within various applications ranging from B-class cars to heavy duty trucks.

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AN INTEGRATED LIFE CYCLE APPROACH TO LIGHTWEIGHT AUTOMOTIVE DESIGN

Modern lightweight design techniques are one of the key technologies in the development of efficient and sustainable mobility solutions. Volkswagen AG aims to ensure that as well as meeting all the necessary technical requirements, its products also deliver continuously improved environmental performance. Right from the start of the development process, these efforts are always focused on the full life cycle of the product. Life Cycle Assessments (LCA) based on ISO 14040/44 allow life cycle environmental impacts to be quantified, so that they can be taken into account in business decisions.

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LIGHTWEIGHT DESIGN LIFE CYCLE ASSESSMENTS

Present-day vehicles already rely heavily on lightweight design. Increasingly stringent legislation on vehicle CO_2 emissions due to be introduced in the coming years will require further significant reductions in emissions, and thus in fuel consumption, across all classes of vehicle [1], ❶. Given that approximately one third of a passenger car‘s total NEDC (New European Driving Cycle) fuel consumption depends wholly or partly on vehicle mass, ❷, lightweight design is one of the most obvious ways, along with more efficient powertrains and improved aerodynamics, of increasing fuel efficiency.

When assessing the environmental impact of a vehicle, emissions at all stages of the life cycle must be taken into account, that is to say not just emissions from vehicle use but also emissions from the manufacturing and recycling phases. Volkswagen AG aims to continuously develop and improve its products in such a way as to always achieve an improvement in total life cycle environmental performance from one model generation to the next [2]. Such total life cycle environmental impacts are quantified using Life Cycle Assessments based on ISO 14040 and 14044 [3, 4].

The Life Cycle Assessment records and evaluates all environmental inputs (resources, energy) and outputs (emissions, energy) associated with a product system in the course of its manufacture, use and recycling. Unlike other environmental management tools such as material flow analysis and risk assessment, the Life Cycle Assessment is product-oriented, quantitative, integrative and scientific [5]. It is therefore an appropriate way of comparing environmental impacts at the product level, preventing problems being simply shifted from one area to another, and providing top-quality data to support decision-making purposes [6].

The LCA approach is particularly relevant to the field of lightweight automotive design, since lightweight materials like aluminium or magnesium are well known for their energy-intensive production processes, ❸ [7]. From the climate change standpoint, it is important to ensure that such environmental impacts at the manufacturing stage, first and foremost emissions of Kyoto greenhouse gases [8], are outweighed as early as possible by reduced fuel consumption in the service-life phase. Only then does Volkswagen speak of „intelligent“ lightweight design, ❹.

METHODOLOGY

Volkswagen AG has many years‘ experience with automotive Life Cycle Assessments. The company was the first manufacturer in the world to publish such a study, in 1996 [9]. Since then, this instrument has been systematically refined for in-house applications and integrated into internal procedures to support decision-making. One core area in which LCAs are used is lightweight design. While relatively plentiful commercial or public information is available on the manufacture of lightweight materials [7, 10-13], the focus at Volkswagen lies particularly on the vehicle service-life phase.

A recently published paper [14], presents the current state of knowledge in this area, which will be described here from a manufacturer‘s standpoint. The key parameter in lightweight LCA studies is the Fuel Reduction Value (FRV). This value states the amount by which fuel consumption is reduced if the weight of a passenger car is reduced by 100 kg. For conventional internal combustion-engined vehicles, this fuel consumption effect can be calculated from the gradients of the Willans lines (differences in efficiency) [15]. The FRV is approx. 0.15 l/100 km · 100 kg for petrol vehicles, based on NEDC, and approx. 0.12 l/100 km · 100 kg for diesel vehicles. If the weight reduction also permits powertrain modification (smaller displacement and/or higher gearing, with no reduction in performance), the fuel consumption effect increases significantly, to around 0.35 and 0.28 l/100 km · 100 kg respectively [16], ❺.

When comparing a reference component with a lightweight version that exhibits the same functionality and quality, the weight-induced fuel saving can be calculated using Eq. 1.

EQ. 1

$$\Delta C_{comp,i} = (m_{comp,i} - m_{comp,ref}) \cdot V_{100\,kg,\,NEDC} \cdot 0.01$$

where

$\Delta C_{comp,i}$: weight-induced fuel saving for component version i [l/100 km]

$m_{comp,i}$: weight of component version i [kg]

$m_{comp,ref}$: weight of reference component [kg]

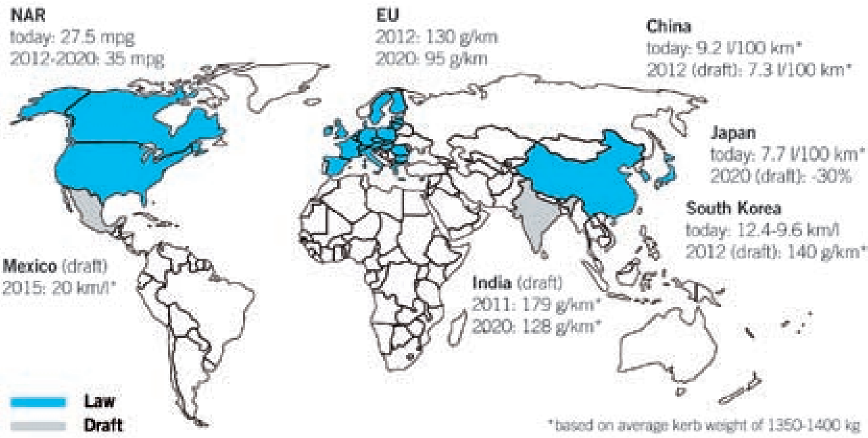
$V_{100\,kg,\,NEFZ}$: Fuel Reduction Value [l/100 km · 100 kg].

For example, if a lightweight design measure reduces the weight of the body by 35 kg, and if performance remains unchanged, then the reduction in NEDC fuel consumption for a petrol model can be calculated with Eq. 2.

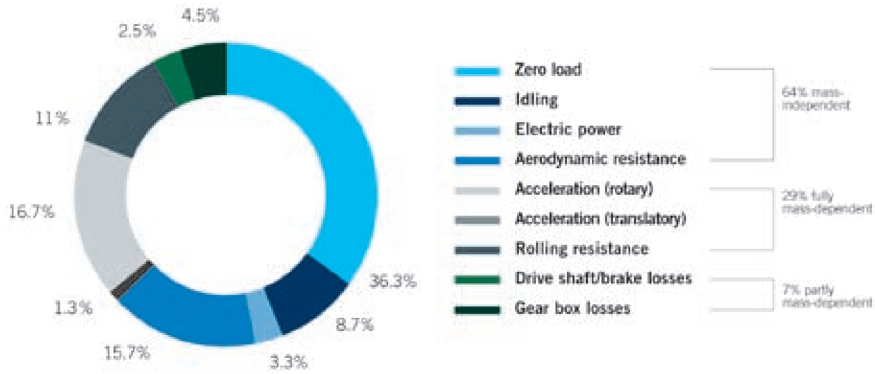
EQ. 2

$$\Delta C_{comp,1} = -35\,kg \cdot 0{,}35\,l/100\,km \cdot 100\,kg \cdot 0{,}01 = -0{,}12\,l/100\,km$$

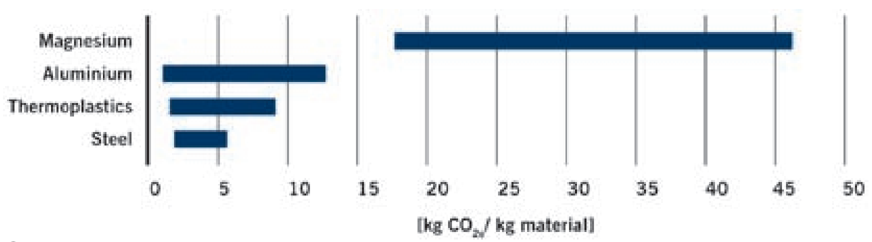
Without powertrain modifications, the saving would fall to -0.05 l/100 km. Whether to include powertrain modifications is therefore a fundamental issue in all such calculations. For the purposes of a Life Cycle Assessment, this assumption is essential, in order to ensure a comparison between two systems – before and after the lightweight design measure – that are functionally equivalent in terms of vehicle performance. However, the assumption is largely a theoretical one, since in practice the implementation of powertrain modifications depends on many different factors, including baseline vehicle weight, inertia weight class, the total sum of all weight reductions implemented in the vehicle, the available choice of engines and transmissions within the company and, last but not least, the timing of the finalised measure within the overall development process. The greater the number of individual lightweight design modifications finalised early on in the development of a new vehicle, the greater the achievable savings and the greater the chances that a powertrain modification will take place. Whether the sum total of all lightweight design measures will lead to a powertrain modification or not is therefore a question that is only answerable at vehicle level, taking into account all the various individual measures. Often, however, the only information available concerns the specific lightweight design measure being assessed. So as far as LCAs relating to lightweight design are concerned, associated powertrain modifications should only



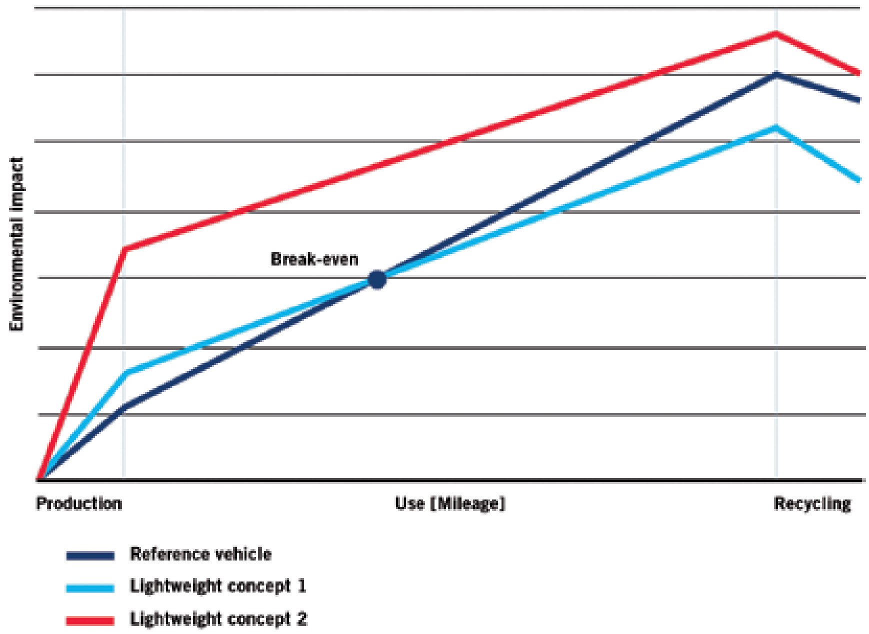
❶ Worldwide CO_2 legislation for passenger cars



❷ Fuel consumption breakdown, Golf A6 1.4 TSI 90 kW M6



❸ Greenhouse gas emissions of different materials [7]



❹ Lightweight design and environmental break-even

	PETROL	DIESEL
With secondary measures → unchanged vehicle dynamicy	0.35	0.28
Without secondary measures → enhanced vehicle dynamics	0.15	0.12

❺ Fuel Reduction Values (FRV) based on the New European Driving Cycle (NEDC) in l/100 km·100 kg [16]

be regarded as the most probable scenario, not the only possible scenario.

The savings in terms of absolute fuel consumption, as shown in ④, can be calculated using the sum total of all individual measures, Eq. 3.

EQ. 3
$$C_{veh,j} = C_{veh,ref} + \sum_{i=1}^{n} \Delta C_{comp,i}$$

where

$C_{veh,j}$: fuel consumption of vehicle concept j [l/100 km]

$C_{veh,ref}$: fuel consumption of reference vehicle [l/100 km]

$\Delta C_{comp,i}$: weight-induced fuel consumption of component version i [l/100 km]

n: number of all lightweight design measures in vehicle concept j.

Thus absolute fuel consumption information is only generated at overall vehicle level. For individual measures (components/assemblies), a negative fuel consumption delta is calculated, using equation (1). In terms of ④, this would appear as a negative slope (hence fuel reduction value, and not fuel consumption value). The reference component would be represented by the baseline of the graph (X-axis).

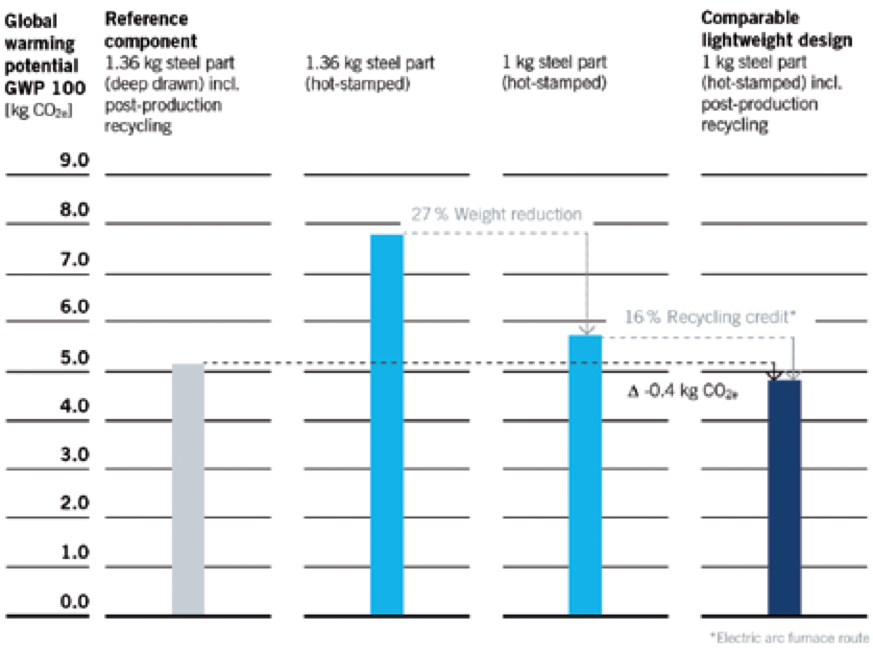
PRACTICAL IMPLEMENTATION

Volkswagen AG takes a variety of approaches to lightweight design, depending on the individual case. Amongst other things, such approaches include choice of materials and functional integration to reduce the number of different parts. For the bodyshell, which contributes around 35 % of the weight of a passenger car, the general preference in volume production today is for lightweight steel or lightweight aluminium construction, or some combination of the two (combined design).

Lightweight steel construction involves the use of special grades of steel and/or special forming technologies. The resulting improvement in mechanical component characteristics such as strength or rigidity allows the number of components and quantity of materials used to be reduced, while at the same time maintaining or improving functionality. An example of this approach is hot stamping of high-strength steels, a technique used in many Volkswagen Group vehicles. The principle could not be more simple. After being heated to a temperature of 900 degrees Celsius, the low-alloy, specially coated metal blanks are simultaneously control-cooled and formed. The structural transformation that occurs during the cooling process greatly improves the mechanical properties of the resulting sheet metal components compared with conventionally formed components. Compared for example with the Passat, the weight of the relevant sheet metal components has been reduced by approximately 27 %, and there has also been a sharp fall in the number of individual components.

Thanks to these weight savings and the resulting reduction in steel consumption, and also to post-production recycling of offcuts via the electric furnace route, the greenhouse gas impact of the hot-stamped sheet metal components in the Passat B6 is already lower than that of conventionally formed deep-drawn sheet metal parts, even when calculated on a „cradle-to-gate" basis, despite higher natural gas consumption during the production process, ❻. The lightweight effect then brings a further reduction in emissions during the service life of the vehicle. As a result, the total life cycle greenhouse gas impact of the Passat is reduced by approximately 6.1 kg CO_{2e} per kilogram of hot-stamped sheet metal used, ❼.

In the case of lightweight aluminium construction meanwhile, the best possible use is made of the density advantages of this material. Aluminium can offer weight savings of between 10 and 40 % over steel, depending on application and on the type of steel structure used. However, the general preference today is normally for a combined design featuring an optimal combination of aluminium and steel. Combined design presents certain challenges due to the different properties of aluminium and steel, particularly when it comes to joining the two materials. Combined design is featured in present-day models like the Audi TT and Audi A8, and has been successfully used in volume production since 1994. An aluminium/steel



❻ Greenhouse gas impact of hot-stamped parts in the Passat B6 (cradle-to-gate)

bodyshell can bring weight savings of around one third over a comparable all-steel construction. In our example concerning models in the mid-sized segment, the actual weight saving achieved through combined design was 36 %, ❽.

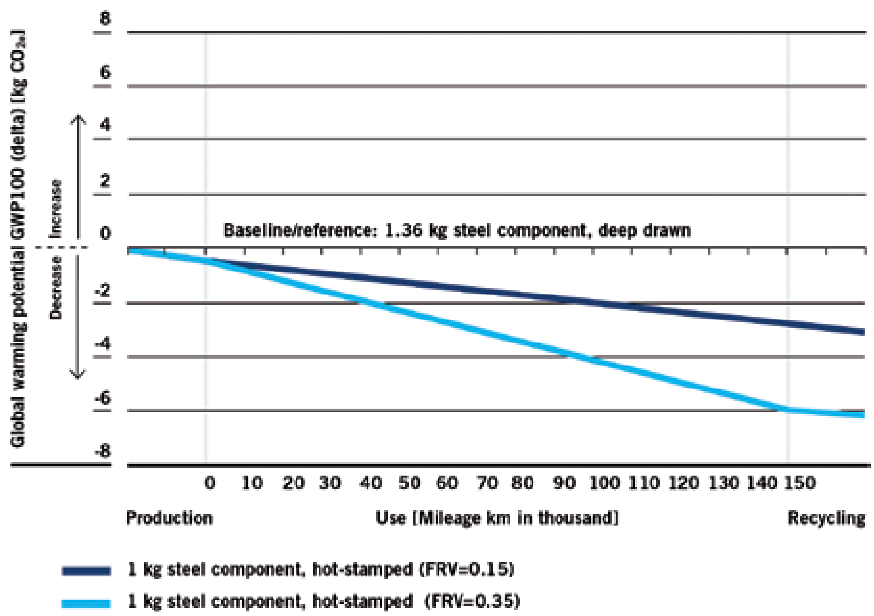
A further significant reduction in greenhouse gas emissions in the manufacturing phase can be achieved by efficient post-production recycling of sheet metal offcuts. For example, on the production lines for the current A8, Audi AG sorts sheet metal offcuts by type before returning them to the supplier for closed-loop sheet metal recycling with no loss of quality. Despite these measures however, the greenhouse gas impact of the cited aluminium combined design in the manufacturing phase is higher than that of a conventional steel construction, ⑧, – although it could be further reduced by the use of wrought aluminium alloys, produced using 100 % hydro-powered electrolysis (cf. Scandinavia).

Due to the lightweight effect on fuel consumption, an environmental break-even point versus a steel body can nevertheless be achieved during the vehicle use phase. In the present example, this point comes at 90,000 km, assuming powertrain modifications that maintain equivalent performance, ❾. The greenhouse gas impact can also be influenced by the recycling stage, where the size of the credit for secondary materials will depend on the degree of downcycling.

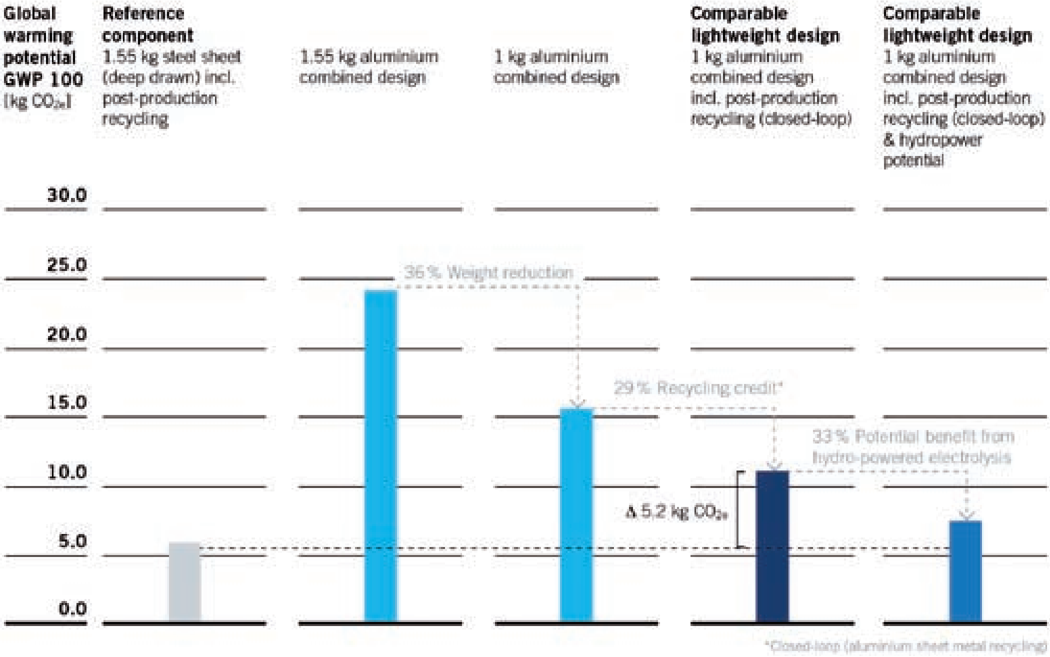
The above remarks all relate to a comparison between standard (reference) components or systems and corresponding lightweight versions. If the perspective is extended to the full vehicle however, further potential for fuel savings can be identified.

For example the significant weight savings from a modern, aluminium-intensive hybrid body provide a springboard for additional, weight-reducing secondary measures such as smaller suspension components and brake systems, as well as additional, non-mass-related measures such as cd improvements, reduced friction losses, etc.

Studies by Audi show that when all fuel-saving measures at vehicle level are taken into account (current model versus predecessor), the break-even point can even be reached at below 40,000 km.



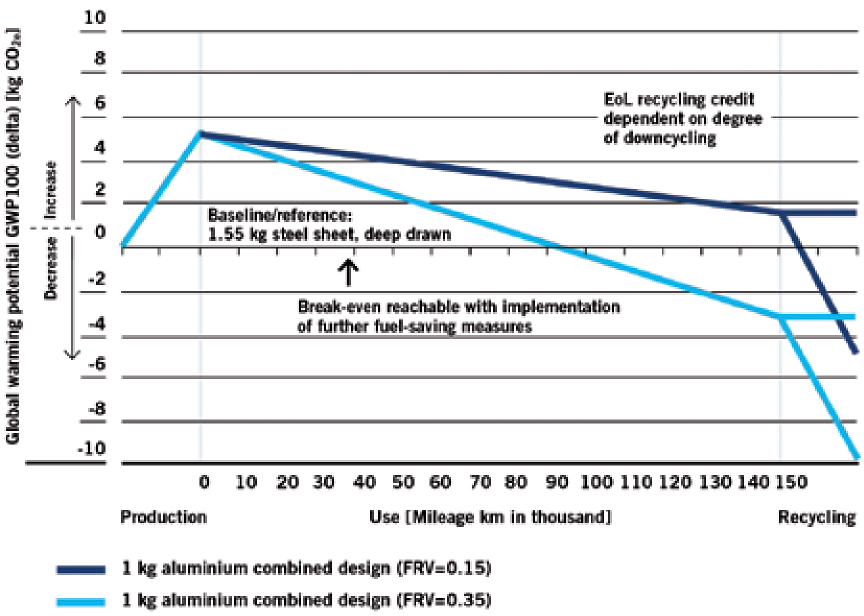
❼ Life cycle greenhouse gas impact of hot-stamped parts in the Passat B6 (delta)



❽ Greenhouse gas impact of a selected aluminium combined design (cradle-to-gate)

What is also clear from the above examples is that it is not possible to make general claims along the lines that „material A is always better than or always worse than material B". Whether a lightweight design measure reduces life cycle greenhouse gas impact or not will depend primarily on the following factors: the size of the weight saving, whether powertrain modifications can be implemented and the quality of secondary (recycled) materials derived from the end-of-life vehicle. In practice, one and the same lightweight design measure might allow powertrain modifications to be made on vehicle A but might not, by itself, be sufficient to warrant such modifications on vehicle B. That said, it is often the case in practice that components or assemblies are optimised and assessed in isolation from each other, so that the significance of each measure in the causality chain that leads ultimately to powertrain modifications cannot always be clearly determined. Nevertheless, the influence of powertrain modifications is such that this parameter should be mentioned whenever lightweight design findings are communicated by indicating both FRV values – one value describing solely the weight-induced fuel saving and the other, higher, value also incorporating resulting powertrain modifications.

As can be seen in ❿, the potential benefits of the various lightweight materials extend even further. The table shows the main actors – and potential actions – at the different life cycle stages. For example the material manufacturing stage offers the opportunity to significantly reduce specific CO_2 emissions per kg of material produced by reducing specific energy consumption and/or through the use of renewable, low-carbon energy sources. Use of secondary (recycled) materials can likewise help to reduce environmental impacts – for example some cast alloys already use up to 90 % recycled content. On the process side, too, measures such as use of climate-friendly shielding gases as a replacement for SF6, or the reduction of offcuts and scrap, all have a part to play. In the vehicle use phase meanwhile, fuel-efficient vehicle design measures by the OEM must be complemented and maximised through optimal use of this



❾ Life cycle greenhouse gas impact of a selected aluminium combined design concept (delta)

LIFE CYCLE STAGE AND MAIN ACTORS	ASPECT	MEASURE
PRODUCTION (SUPPLIERS, OEM)	: Energy demand : Energy mix : Secondary resources : Process-specific aspects	: Reduced energy consumption : CO_2-reduced energy mix : Increased use of secondary resources : Optimized consumables
USE STAGE (OEM, CUSTOMER)	: Lightweight effect (massinduced, secondary measures, e.g. downsizing) : Customer behaviour	: Strict design in favour of fuel efficiency : Avoidance of unnecessary ballast, optimal tyre pressure, etc.
END-OF-LIFE (RECYCLING-INDUSTRY)	: Recovery of materials : Availability : Scrap quality : Secondary material quality	: Installation of collection systems : Development of separation technologies : Enhancement of yield/quality

❿ Environmentally friendly lightweight design, aspects and measures

potential by customers. Finally, at the recycling stage, reprocessing of lightweight materials into high-quality secondary materials will influence the range of potential applications of such materials and thus the associated positive environmental impacts.

Seen from this overall perspective therefore, it is clear that environmentally friendly lightweight product development is a multi-faceted science and offers a wide range of potential opportunities. In addition to the engineers in the automotive industry, the other stakeholders in the value chain (materials manufacturers, suppliers, recycling companies) likewise have their part to play in further improving their products and enhancing the life cycle environmental impact of vehicle concepts of the future.

CONCLUSION

As has been demonstrated above with reference to greenhouse gas emissions, Life Cycle Assessments based on ISO 14040/44 are an environmental management tool that allows the environmental impact of lightweight design measures to be quantified and evaluated across the full life cycle of the product. In this way, Life Cycle Assessments prevent problems being shifted from one area to another and allow the net impact of any given measure to be assessed in a transparent and scientifically sound manner. When assessing the service-life phase of the life cycle, the calculation model presented here – based on Fuel Reduction Values – should be used.

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PSYCHOACOUSTIC PROCEDURE FOR THE SOUND CHARACTER OF PASSENGER CAR DOORS

A solid and harmonic door closing sound is one of the most important characteristics for high perceived quality performance of new vehicles in the show room. To fulfill this high customer quality requirement Opel uses optimization measures to be early implemented in the development process. Repeatable measurements with objective sound metrics are the prerequisites for a successful implementation.

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DOOR CLOSING – A FIRST IMPRESSION

The customer decides during a few tenth of a second whether the sound character of the door closing is solid and of high quality or if it gives a cheap or low-priced impression. Internal studies at GM/Opel showed that even this sound impression could influence the buying decision subconsciously. To analyze this short event psychoacoustic tools need to be used. The classical acoustic analyses like loudness level in dB and frequency spectrum are no longer sufficient.

In the past, the door closing sound was usually rated subjectively and the acoustical optimization was executed by individual tests. Since some years worldwide and general test procedures and single metrics were implemented successfully within GM. In the early phase of the development process these procedures enable to quantify the performance by objective values. Measures could be defined to meet the defined performance targets.

The long term experience was successfully applied to the latest Opel models Insignia and Astra, but of course also to the new Meriva (model year 2010). Already in the show room the customer get high quality impressions by a solid door closing sound character.

TEST PROCEDURE FOR LOUDNESS AND SOUND CHARACTER

To define a test procedure that gives a distinct description of the door closing sound test sequences at the test bench were executed followed by extensive rating sessions in the sound studio. This approach aimed to transfer long-standing and approved subjective rating criteria into objective metrics which can be recorded by a simple and repeatable procedure.

Studies in the sound studio resulted in two crucial subjective criteria for the door closing sound. These are the maximum loudness value and the sound character. Both should be described in detail by a suitable metric and a procedure to get a representative analysis tool for an objective assessment.

LOUDNESS

The major step for an objective description of door closing sound was the standardized recording and the repeatable playback in the sound studio. Unweighted analyses of sound pressure using Pascal units as well as the well-known dB-weightings did not correspond to the subjective feeling at all. Short sound events like the door closing sound impress the subjective rating in maximum 200 ms and therefore the conventional analysis methods are not suitable.

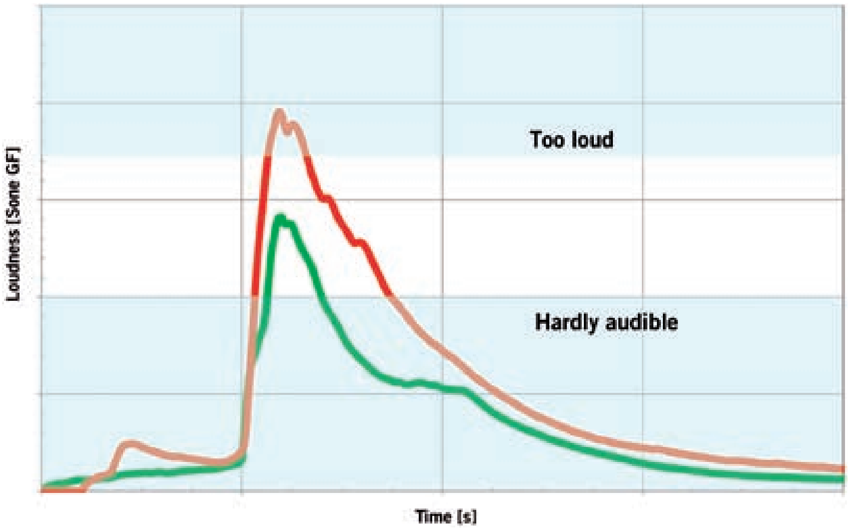
The most appropriate method to measure the psychoacoustic loudness is the so-called Zwicker loudness procedure [1]. This method is based on the calculation of the loudness value from overlapping third-octave frequency bands. Dependent on the sampling frequency every third millisecond a weighted Sone value is calculated, which gives a temporal characteristic and enables the detailed description of the door closing sound, ❶. Many hearing sessions in the sound studio confirmed that the maximum value of the loudness curve corresponds to the subjective feeling very well.

The door closing sound can be exactly classified according to the loudness level with this analysis method. A maximum loudness value should not be exceeded, to avoid an annoying perceived quality. On the other hand the loudness value should not fall below a certain level in order to give the customer an adequate response signal of the closing door. Within this bandwidth the current target values for the loudness of the door closing sound should be defined.

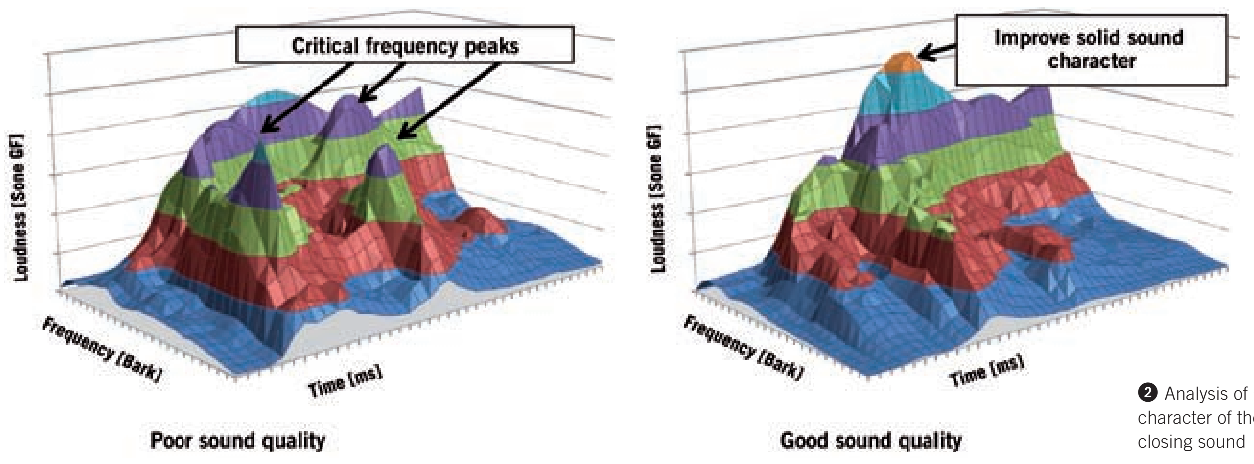
SOUND CHARACTER

As a further quality criterion the sound image is as important as the loudness. The loudness level could fulfill the required target area but the door closing sounds still tinny or not of high quality caused by a poor sound character.

To describe the sound character for an engineer a psychoacoustic metrics is used. The frequency axis is divided in different frequency bands similar to third octave analysis by using the unit Bark [1]



❶ Bandwidth of loudness for the door closing sound according to Zwicker loudness procedure



❷ Analysis of sound character of the door closing sound

to map the anatomic formation of the human ear.

The frequency spectra of door closing sound changes very fast in millisecond time steps. Therefore, a detailed representation of sound character will be displayed favorable in a three dimensional depiction of the frequency spectra as level versus time, ❷.

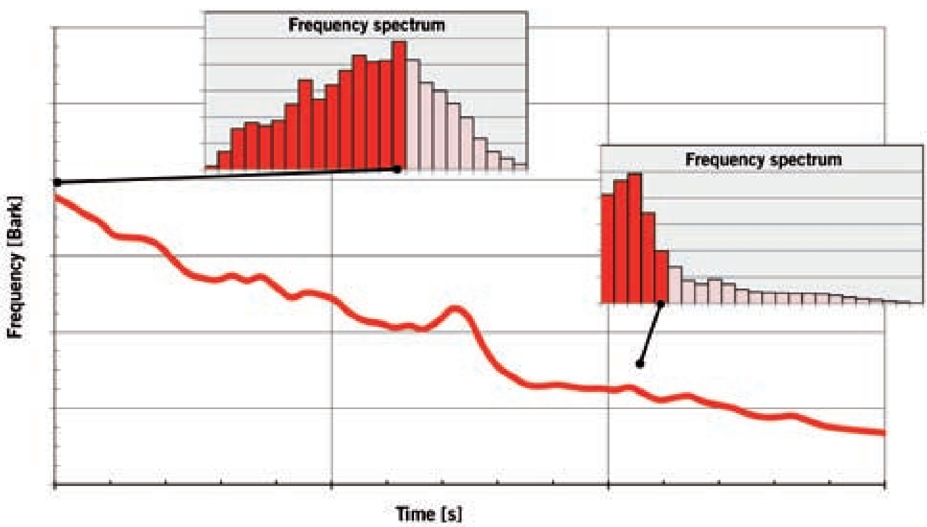
The time sequence analyses of all single events during the door closing show that the frequency peaks after the impact of the latch at the striker are the crucial criterion for sound quality, marked with arrows in ② left. These frequency peaks, which generally will be generated by mechanical operations within the latch, will not be recognized as single events but define the sound character decisive. The reduction of these high frequency peaks become a substantial criterion for a good sound character of the door closing sound. On the other hand higher values in the low frequency area improve the sound character, ② right, because a low-pitch sound will be perceived as pleasant and of high quality.

The presented three-dimensional figures are essential tools to analyze and optimize the sound character. These graphs are not really suitable to classify and to set performance targets during the acoustic development. Charts which show comparable curves or tables, where numbers describe objectively subjective feelings, are necessary for objective assessment of every single development step.

The relevant information of the 3D-diagram will be reduced to a two-dimensional curve. That means a certain fraction of high frequencies will be displayed versus time. The so-called frequency limit curve [2] describes the share of the high frequency content after the impact of the latch to the striker versus time, ❸. The level declines clearly visible versus time. This gradient and the characteristic of the curve is the baseline for a new sound character metric.

The deviation out of a measured test result compared to a reference curve which was defined out of many studies as a reference "frequency limit curve" describes the metric of sound character with a single value as sound character number. A low value stands for a high-quality and a solid sound character.

The results of the presented studies identify the loudness level as well as the sound character as the decisive quality criterions for a high-quality door closing sound. Appropriate psychoacoustic metric and test procedures were developed to gain a reasonable correct description of the event. Now it is possible to describe the subjective feeling with an objective metric.

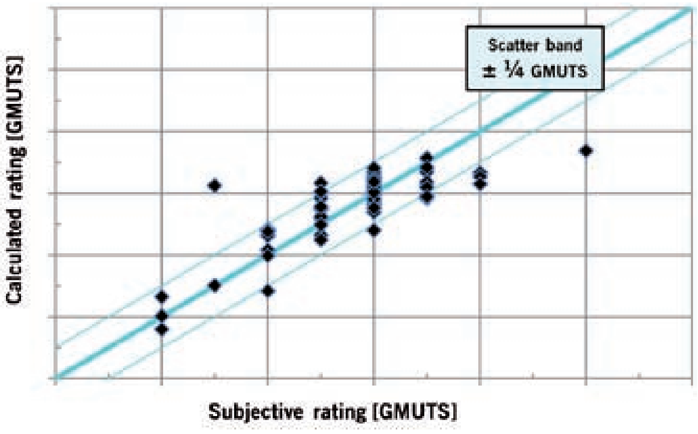


❸ "Frequency limit curve" represents the sound character of the door closing sound

VALIDATION

Before new test procedures can be applied in global development at GM/Opel successfully a validation is absolutely essential. The objective metric should describe the subjective feeling of the customer in all cases.

Prerequisite for a reasonable comparison is a standardized rating system, which represents the customer feelings as a subjective rating. GM uses a scale from 0 to 10. A



❹ Validation result of the objective metrics

rating of 10 corresponds to a really high-quality door closing sound. The customer perceives a noise rated beneath 6 as tinny and low-cost and even annoying.

The acoustic quality of the door closing sound is defined by the loudness value (in Sone) and the sound character number. An increased loudness level as well as a higher sound character number reduces the quality performance of the sound. For the quantification of the correspondence between subjective ratings and objective metrics the following linear formula in Eq. was presumed:

EQ.	Subjective rating = base rating (A) – loudness value *B – sound character number *C

With this linear approach the correlation value was calculated by more than 80 %. It will be estimated as a reasonable correct correspondence between calculated and subjective rating, ❹. Only solitary deviation can be indentified outside a scatter band of ± ¼ rating unit.

An adequate validation of both single metrics was achieved. The objective assessment of door closing sound by the maximum loudness value and the sound character number could be done from now on. Therefore, the method can be applied purposive during the vehicle development in the future.

APPLICATION EXAMPLE

The presented test method was applied during the development of the Opel Insignia first time and was continuously improved. The correct target setting as well as the repeatable test recordings and analysis create the suitable prerequisite to develop a high-value and solid sound character for door closing sound.

The door closing sound performance of the Opel Astra and of the new Opel Meriva with its new FlexDoor concept (doors are opening in the opposite direction) benefits from all the experiences during Insignia development. The consequent retaining on the approved Insignia strategy (benchmarking – target setting – monitoring of development steps) turns out to be the base for a high-value and solid quality performance for the door closing sound.

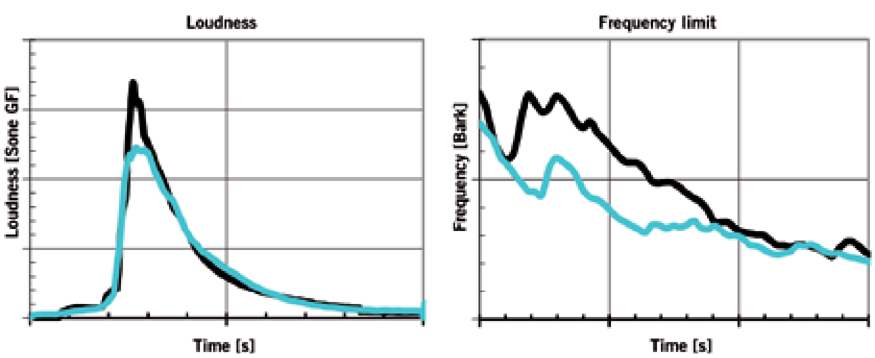
❺ shows the objective analysis results of the door closing sound for the new Opel Meriva compared to its predecessor. Significant improvements were achieved for loudness as well as for the sound character with the new procedure. All four doors close with a solid and high-value sound quality.

SUMMARY

Many different development areas inside GM/Opel took actively part in to analyze and optimize the door closing sound with objective metrics. A further step was successfully done to describe the subjective human aural impression with an objective metric.

The door closing sound can be clearly described by two quality criteria: maximum loudness level and sound character. These metrics are calculated by psychoacoustic methods. The maximum loudness value is described by the Zwicker loudness method in Sone and the sound character by the sound character number based on the frequency limit curve for the high frequency noise share during the door closing. Both single metrics enable to define clear development targets and to assess single development steps by an objective value.

The procedure was successfully validated and implemented in the development process of Insignia, Astra and the new generation Meriva. All vehicle doors close with a solid and high-value sound character and convince the customer of the high quality of Opel vehicles.



❺ Analysis of door closing sound of the new Opel Meriva compared to its predecessor representing the new defined metrics loudness and "frequency limit curve"

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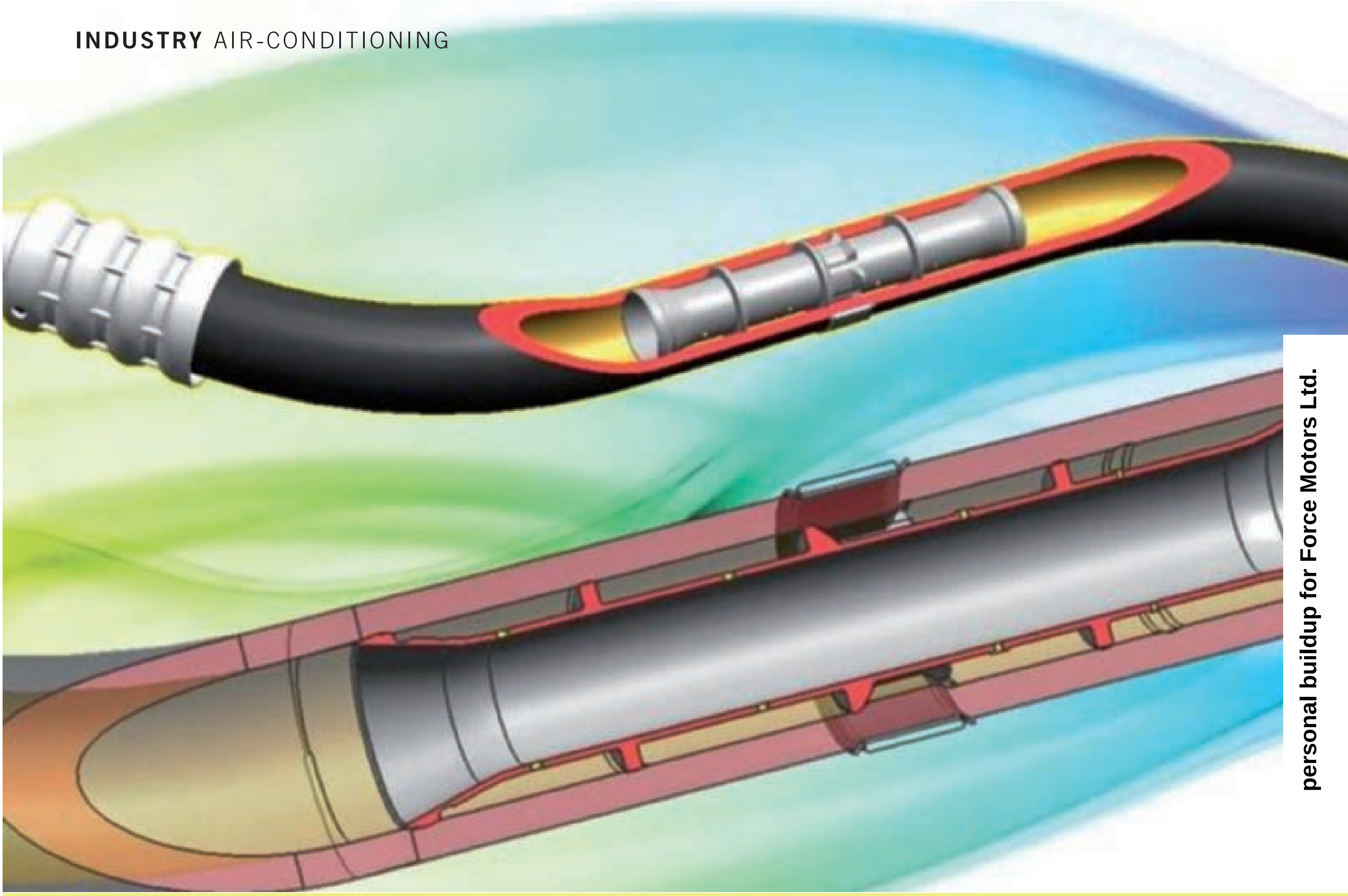
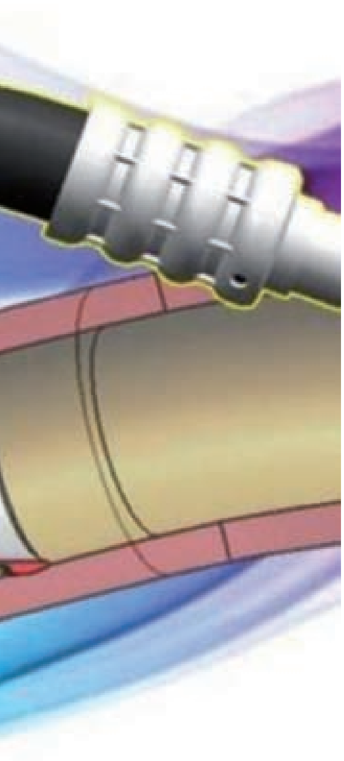


INTEGRATED MUFFLER FOR LINES IN AIR-CONDITIONING LOOPS

Over the course of specific validation tests to measure the acoustic performance of air-conditioning loop lines at TI Automotive, the opportunity came up to develop and introduce a brand new type of muffler. TI Automotive was able to develop a new muffler that even solves several acoustical problems. Additionally further targets were accomplished to reduce the complexity of the production process as well as the enabling of cost saving potentials and at the same time reduce of pressure losses in the system.



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NOISE REDUCTION

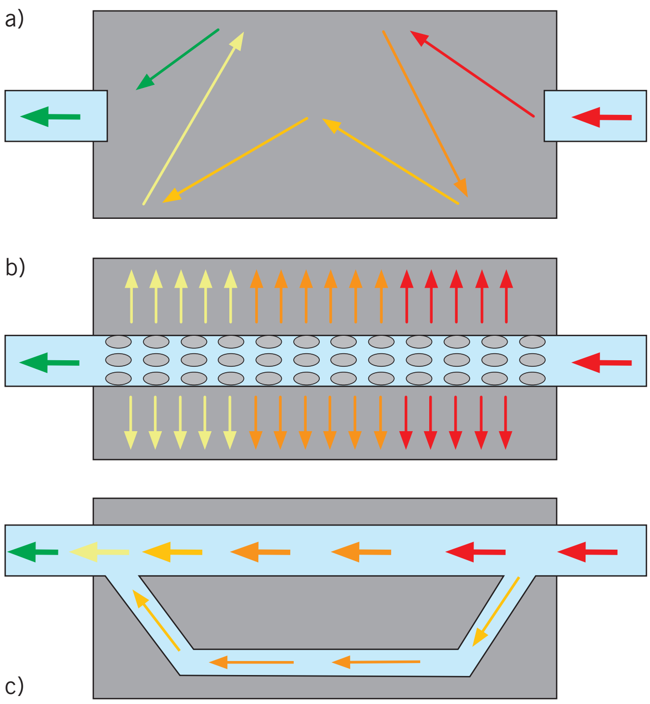
Today one of the key topics of vehicle development is the reduction of noise in and at the vehicle. Increasing comfort and quality demands have made it difficult, since the vehicles are equipped with great amounts of new additional systems. Such aggregates now emit via there drives (compressors, pumps) and connections (hose, tube, lines) noise into the passenger compartment.

Accordingly air-conditioning (AC) system loops have such undesirable noise levels – not only through turbulences and pulsations, from valves and connection blocks, but also from the compressor itself, which is emitting its noise directly into the line system. That is why current systems use a so called 'bottle-muffler'. This component now has actually a relatively good acoustical dampening function, yet, it does bring a substantial pressure drop and also it requires quite extensive packaging space, which, is hardly available in modern vehicles. This can lead to a removal of the bottle-muffler and therefore the higher noise level is accepted. The acoustical quality of such a system is then drastically reduced.

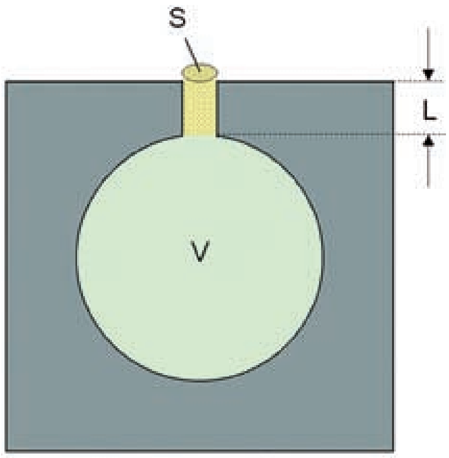
The solution to this dilemma is the newly developed integrated muffler from TI Automotive. This means that the muffler can be placed in the tube and as well in the hose. The routing of the lines is therefore not influenced! Through the choice of the material it is possible, to position the muffler in the bend of a hose, which increases the flexibility within the routing enormously.

BASIC PRINCIPLES TO DAMP NOISE

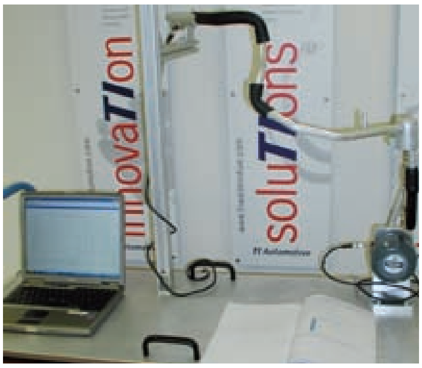
There exist three basic principles to dampen noise within tube and line systems, ❶. The conventional solution (a) works with the principal of reflection. The upcoming sound is reflected several times inside the increased volume and loses with that its energy. A second option (b) is to reduce the unwanted noise by absorption.



❶ Three principles to reduce noises: a) reflection, b) absorption, c) interference



❷ Helmholtz principle with *S* as area of the opening, *L* as length of the resonator-throat and *V* as volume of the enclosed gas



❸ Test stand with test-line and loudspeaker as noise-generator (below right)

In this case, noise-waves guided through a specific absorption material and the energy is reduced by friction. A third solution (c) uses sound interference. Here, a counter noise is generated, which eliminates the annoying noise by phase displacement.

Since it is a key target to reduce the package volume the solution a) is not further considered. Also, the use of form-free absorption material per definition b) is not further tracked, as the potential loss of such material within the AC loop could bring much larger issues as it would negatively influence the system's components.

INTERFERENCE SOLUTION GUIDES TO NEW MUFFLER

Only solution c), generating of a noise interference, was successful. Early in the testing, with relatively simple inserts into the hose, some good potential was recognized. The Helmholtz principle, ❷, clearly showed a high potential of dampening for such an application. The principle is generally based on a gas volume that is linked by defined opening and to a corresponding defined length in the main system. Through the elasticity of the gas volume in agreement with the inert mass of the gas in the main stream, it creates a spring-mass system with an according own resonance. After several optimizing loops a first prototype of the new, integrated Muffler "TI Automotive Silencer" was developed. It can be produced as quite precise injection molding component.

The actual noise encourages this system to swing or move in answer to a contradictory wave. Accordingly, this muffler can be seen as a noise generator, which produces anti-noise. Both noise waves overlap each other and it ends in a reduction of the noise. The resonance frequency f_0 can be calculated via the following Eq.:

EQ.	$f_0 = \frac{c}{2\pi}\sqrt{\frac{S}{V(L+2\Delta L)}}$

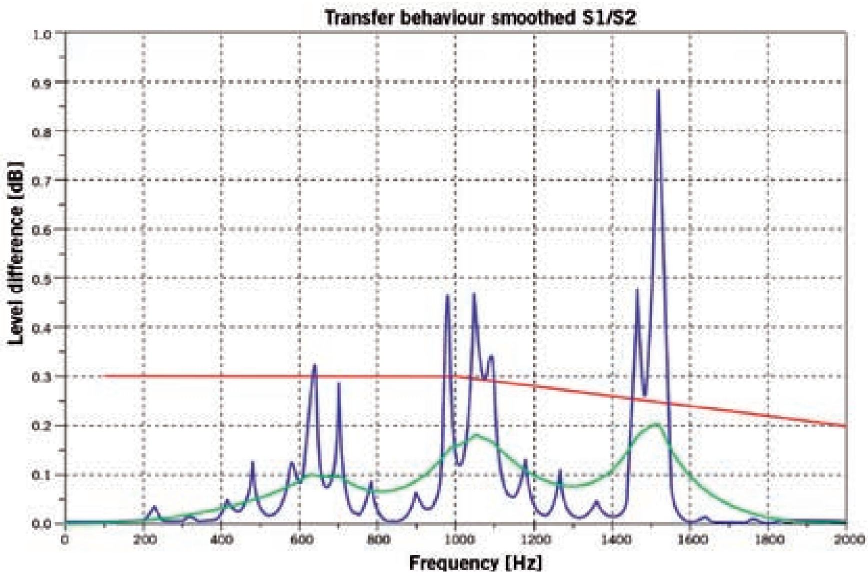
Here c is the speed of the noise, S is the area of the opening, V the volume of the enclosed gas, L is the length of the resonator throat and $2\Delta L$ the orifice correction factor.

ACOUSTIC MEASURING OF THE NEW MUFFLER

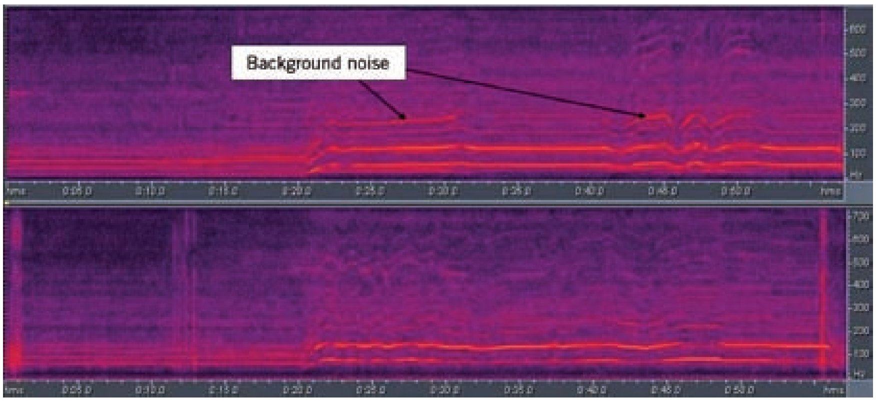
The prototypes of the new, integrated muffler were first tested according to the specification of a European OEM within an AC-line system. The function-testing process was based on the typical validation test of such an assembly. The test-stand consists of the following components: Loudspeaker as noise generator, two microphones to measure signal of input and output side of the line, a rack to bring the line in position per vehicle definition and software to generate, collect and calculate the noise parameters, ❸.

To measure the function a so-called white noise is generated in the loudspeaker and brought into the line. On either side microphones are collecting the signal of the noise level. Following this, the software calculates the ratio between both signals and forms accordingly the system-answer (transfer-function). The result of a typical measuring is shown in ❹.

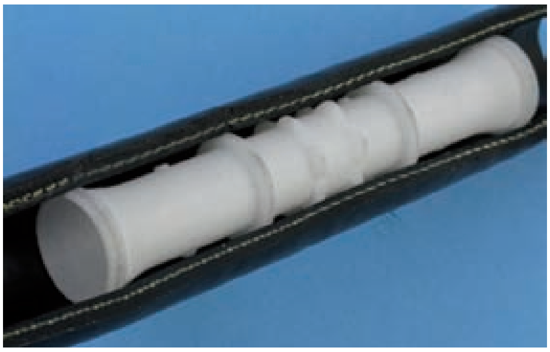
Further in the development, more measuring was done in straight tubes. At the same time it was tried to reduce the influence of the typical AC-line routing and on the other hand to establish precise numbers to predict the function in the vehicle. The advantage of the Helmholtz principle is that the resonator can be well focused on a specific disturbing frequency. At the beginning of the tests, this required a precise positioning of the muffler insert within the line. Background for that is that the Helmholtz resonator only dampens at the point of the wave-belly (maximum peak). In order to solve this task, several Helmholtz chambers are combined back to back, were each is fine-tuned on one spe-



❹ Average transfer function (green) and the borderline (red); the borderline equals the minimal requirement of the European OEM



❺ Frequency spectrum for a serial solution (above) and with the new muffler (below); the AC system was switched off at 30 s and switched on at 40 s test duration



❻ Muffler in a design version as multi chamber silencer

cific resonance frequency. The muffler's wide range of functionality is that it can be positioned in the line to allow a wide range of tolerance.

The fact that the relevant resonances (within vehicles AC systems typical between 20 and 2000 Hz) and the geometrical size of the muffler can be quite small makes this principle quite interesting for an integrated solution. The new muffler obtains its full function when assembled to the hose or tube. The Helmholtz volumes are than so small, that the free area of the 'normal' line is hardly reduced. Also the pressure drop increases only slightly. In comparison to a typical "can muffler", it is even lower. During the theoretical test phase, it was already possible with first simple prototypes to see that the requirements within the targeted specification could be met.

PRACTICAL TESTS

Following the tests on the test-stand, specific versions and prototypes were built and tested practically in a test vehicle. The acoustical performance was measured with a specific microphone within the passenger compartment. In preparation for these tests, it was tried to subjectively identify the relevant noise from to the AC system and objectively measure it.

With this vehicle test it was possible, to determine a specific noise at 1500 to 2000 rpm of the engine. This was clearly identified as being generated by the AC unit. With graphical displacement of the noise it was accordingly possible to show and also to give it a quantitative value, ❺. The AC system was switched off at around 30 s and switched on at approximately 40 s test duration.

Simultaneously it was possible to assign a frequency ("disturbing noise" in ⑤), which allowed an optimized fine-tune of the muffler. After review of all results, it was recognized, that the new muffler brought a clear improvement of the system's noise. Comparing to the serial solution an improved dampening of 12 dB was achieved, which relates to a quarter of the original level. In the sequel of the project further NVH tests with a well-known OEM were done. The performance of the new muffler was confirmed, too. The new muffler can be focused on one frequency (Single Chamber Silencer) and as well on several resonances' (Multi Chamber Silencer, ❻).

SUMMARY AND OUTLOOK

TI Automotive offers an insight into the development of a specific noise dampener for vehicle air-conditioning systems. This new muffler, called "TI Automotive Silencer", is space-saving and integrated into the line and works per the Helmholtz principle (Interference). The dampener can be designed on a single frequency (Single Chamber Silencer) respectively on several resonances' (Multi Chamber Silencer).

The advantages of the insert-muffler lie in the precise tuning, the high dampening potential, and the low pressure drop in the system and as well the specific form. This type of design allows integration of the muffler into straight tubes and as well into formed or flexible hoses. There is no need to give any special attention to the "routing" of the lines in the engine compartment.

Additionally, the three basic functional principles of noise dampening in tubes are explained: the reflection, the adsorption and the finally successful interference. TI Automotive provides an overview in carrying out the validation testing to quantify the acoustical behaviour of AC lines. The test-stand and the typical results are presented and the practical tests addressed.

In conclusion, it can be said that the new muffler brings a substantial improvement to the current conventional solution. In respect for the production this component is a major improvement since a precise injection-mold part can be manufactured.

The future goal is to equip serial AC lines of new vehicle models with the new dampener "TI Automotive Silencer", respectively replace currently used variants like the bottle muffler. The patent is pending. Further development to adjust for new alternative refrigerant fluids like CO_2 is being planned for the near future.

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HOLISTIC SOFTWARE DEVELOPMENT
DEGREES OF FREEDOM FOR PROCESS AND PRODUCT QUALITY

Are the following statements actual correct and still appropriate? Product quality is generated or confirmed by a HiL system. Sufficient process conformance and quality is confirmed with a Spice assessment report. Adequateness and efficiency is gained with process tailoring at project level. The SQS Software Quality Systems AG and the Marquardt GmbH attempt a critical view of the above statements to show how software can be validated better; although beyond the scope of the Herstellerinitiative Software (HiS).

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CHALLENGES AND DEGREES OF FREEDOM

Increasingly complex system development forces holistic approaches for software quality assurance. This requires a scope beyond project limits to guarantee the delivery of products with a constant quality for an organisational unit. This objective can only reached by a holistic and strategic approach of the quality assurance and the development units.

The Automotive Spice standard [1] has been established to rate the software development process. The purpose of this standard is to evaluate the performance of the development processes for ECU suppliers in the automotive industry. The standard will be used for supplier rating by the OEMs, which are requiring the supplier to conform to the HiS scope [2]. OEMs like Audi, BMW, Daimler, Porsche and Volkswagen are focusing activities on this initiative which also includes the process scope for capability determination. This selection reduces the Spice process which is currently in excess of 30 processes to 15 processes. In general the OEM restricts the supplier rating on the process scope which is defined by the HiS. However there is a risk of "tunnel vision" because everything beyond the HiS scope will be hidden initially. The quality of software is originated only in the development phase; this is because software is constant and copied without production variations. Therefore typical measures like ppm and guarantee & goodwill costs are not adequate for rating a software product. A better approach for rating the quality of software for example should be a metric for software problem reports. Another metric is the rate of software changes after implementation of 100 % functionality; this can be used as a rework indicator. For these metrics, there are not general applicable guidelines available from the OEMs. This requires adequate objectives to be identified and metrics for the software quality which are tracked and measured until their realisation.

The main challenge is to gather all the required aspects of process and product quality. The presented challenges show clearly, that quality cannot be tested into a product at the end of development phases using methods of hardware in the loop (HiL) test systems and a "post-documentation" for Spice conformance.

A beneficial way to increase quality is an optimized workflow of the organization unit for product development. Therefore the "degrees of freedom", which are allowed by the OEM guidelines, must be exploited. The following degrees of freedom can be used if the product quality is not restricted and the approach is adequate in terms of Spice:

: The strategy for achieving Spice conformance is not defined within closed borders.
: The metrics for process and product control are not specified in detail.
: The quality scope is not limited to the HiS-Scope.

The Holistic approach to software development, as defined by Marquardt GmbH, a manufacturer of electro-mechanical and electronic switches and switch systems, is a balance between as much quality as possible and only using what is required to ensure quality.

A HOLISTIC SOLUTION APPROACH

A suitable starting point for defining the degrees of freedom are as follows:

: function orientation – allowing a flexible and specific validation
: project portfolio management – to create a general and not product dependent comparability
: product lines – to start a project on a high initial quality level.

This approach combined with a risk based and preventive quality strategy is described below in more detail. The specific setup of the approach is oriented on the circumstances of the organisational context of the Marquardt GmbH and can be easily adapted to other organizations.

FUNCTION	Q-RELEVANT-ASPECT 1	Q-RELEVANT-ASPECT 2	...	Q-RELEVANT-ASPECT N	Q-FOCUS-POINT-VALUE	VALIDATION-MEASURE 1	VALIDATION-MEASURE 2	VALIDATION-MEASURE 3	...	VALIDATION-MEASURE X	PRODUCT-Q-INDICATOR 1	...	PRODUCT-Q-INDICATOR Y
Function 1	low impact	middle impact	...	low impact	5		X		...		N/A	...	N/A
Function 2	high impact	high impact	...	middle impact	13	X	X	X	...	X	low	...	middle
Function 3	middle impact	middle impact	...	middle impact	9	X	X	X	...		N/A	...	N/A
Function 4	high impact	low impact	...	low impact	7	X			...		N/A	...	low
Function 5	low impact	middle impact	...	high impact	9		X	X	...	X	middle	...	low
Function 6	middle impact	low impact	...	middle impact	7	X		X	...		middle	...	N/A
...	...	...	...	...	...	...	...	...	...	...	...	...	...
Function M	high impact	low impact	...	low impact	7	X			...	X	N/A	...	low

❶ Presentation of a functional refinement of a product, the evaluation of the relevant quality aspects for specific validation measure definition including the effectiveness monitoring by product quality indicators

FUNCTION ORIENTATION

The term function orientation is defined as a refinement of products into its functional entities. A functional entity is defined by its stimulus and processing which leads to a reaction. Based on the systematic refinement of products, the functionality can be identified. The specific validation of the function is defined after the awareness of the importance and its contribution or value to a product with the explicit presentation of functionality [3], ❶. The functional importance is called "focus point value" and is derived from the quality relevant aspects. The validation considers not only the function, but also is based on existing quality risks of the organisational unit. A (software) product, which is generally developed in a project, is refined in smaller entities by this approach. The functional entities are validated with a specific package of quality measures to reach the validation targets. The packages of quality measures are primarily based on analytic measures. With static measures, early function specific error detection is realized, which is then rounded down by dynamic measures such as tests on a HiL system. The effect of the quality measures is determined by the "product quality indicators". Through effective feedback and cyclic checks of the relevance of the quality measures, an iterative measure optimization is possible. If this approach is realized as a part of the quality strategy of a project, it is possible to assign resources more precisely.

A further step is to take constructive measures to promote the initial quality level. These are generally methods and processes. Below two constructive methods are shown: the project portfolio management and the product lines.

PROJECT PORTFOLIO MANAGEMENT

An organisational unit generally processes many projects in parallel time slots; this makes it necessary to manage the validation transparently based on the quality risk and at a sufficient granularity level. This allows the entire organizational unit to achieve a constant quality level.

The logical result from the facts mentioned above about product specific and function oriented quality planning is applied to the project portfolio. Therefore all quality relevant aspects have to be comparable and constant over the entire organisational unit. To ensure more comparability, the derived functions can be grouped into function trees to ensure the reuse of conceptualization.

The result is a representation in which the management realize the validation not on the explicit project level but rather implicit project level by function validation. Therefore the general project management approach, for example the quality lump sum, is extended with a specific control and resource allocation on the function level.

PRODUCT LINES

Product lines provide a means of strategic reuse. With strategic reuse, the high maturity level of elements can be used selectively. Although a process for reuse is defined in the automotive Spice, it is not in the HiS-scope.

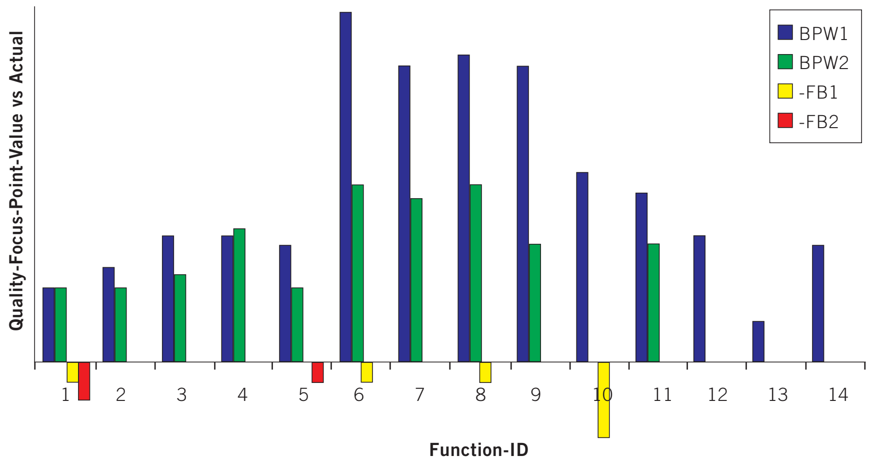
The reuse approach is based on the variation point concept [4], but is extended by specific elements for product configuration as well as aspects for release planning, ❷.

The variation point concept for product line management uses the customer specification as base element from which all possible variants are derived. For example, all identical features are gathered in this base specification, non identical features are defined in the variants.

VARIANT MANAGEMENT	VARIATION POINT	DEPENDENCY	VARIANTS
	VariationPoint 1	XOR	Variant 1.1
			Variant 1.2
	VariationPoint 2	optional	Variant 2.1
	...	...	...

PRODUCT CONFIGURATIONS	PRODUCT	CONFIGURATION	RELEASE
	Product A	Variant 1.1	1. Sample
		Variant 2.1	2. Sample
	Product B	Variant 1.2	2. Sample
	...	...	...

❷ Generic structure of the data for the management of the reuse



❸ Comparison of two products of a product line regarding their initial quality focus value (BPW) and the result about the effectiveness of the specific validation measures based on the product quality indicator (FB)

The variation points of a product line are identified and documented with properties about the appearance of the variant; for example, "optional" or "exclusive-or". In a next step, a selection of variants (which includes variable features) is assigned to a specific configuration of a product line. This configuration is implemented by a product. Products are saleable variants of the product line.

The benefit of this approach is that an initial not for "taking over" intented specification is reusable, because the product line management is an additional element of the specification that helps identifying features which can be carried over in a new product. For example CRS for product A is 90 % identical to CRS of product B. Instead of analysing the whole CRS of product B only the delta and its impact of 10 % has to be analysed.

By the connection of the reuse management to the specification, it is also possible to reuse the test cases which are derived from the specification. This allows a high amount of quality assurance investment for the test case definition and automation on the HiL system is reusable for the functional validation of the integrated products, not just the reuse of individual validated elements.

A LOOK TO THE APPROACH BASED ON AN EXAMPLE

Finally a comparison of two products of the Marquardt GmbH manufactures based on the product line attempt has been made, ❸.

The data is anonymized and relativized, but the relevant patterns or relations which are discussed here are preserved. Easy to recognize are the functions (Function-ID) which are similar – the same for the product line. The quality focus value (BPW1 and BPW2) is on the "over taken" project (is marked with the index 2), which is based on the product line, is generally much lower. The quality focus is based on a specific set of validation measures that were defined and realized.

For control of effectiveness, the product quality indicator, "problem reports" (FB) is recorded based on functions. With this indicator, it can be seen that the over taking project (FB2) has a high initial maturity level in comparison to the first product (FB1) of the product line. The unexpected high number of errors on FB1, at Function-ID 10, originated in organizational circumstances, which are not covered by the functional validation. This example shows the limits of the approach, which can generally be seen as one of more indicators for quality control.

SUMMARY

The example from Marquardt GmbH is showing that process and product quality is improved by a holistic approach for software development, which is typically not established in classic organized development units. The principle suitability of the approach was confirmed by an OEM sanctioned Spice assessment in one of the projects mentioned above. Finally the established product, manufacturing and customer based quality view of a supplier was extended with a structured value based [5] quality view.

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DESIGN AND IMPLEMENTATION
OF REQUIREMENT-DRIVEN ELECTRIC DRIVES

The deployment of electric or electrified drivetrains is rapidly becoming a significant trend. Getrag demonstrates the appropriate steps required in electric drive design and the necessity of taking a holistic view. A comparative analysis based on simulation results reveals the advantages of single and two-speed transmissions in combination with high-speed e-machines.

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COMPLEX RELATIONSHIPS IN THE OVERALL SYSTEM

The Getrag Corporate Group is active both in designing gears, shafts and bearings in transmissions for electric drives and in the optimal design of complete drivetrains. Given the nature of electric machines, power electronics and energy storage, and the diverse operating expectations for a vehicle such as its ability to climb grades, accelerate in a given time period, and reach a defined final speed, a transmission is needed that put the properties of the electric components to their best possible use. In this manner the complete drive system fulfils the vehicle requirements.

Only by taking into account the individual component modules can an optimally designed overall drivetrain be achieved with concomitant reductions in total costs, energy savings and improvements in vehicle performance. Energy savings thus gained can be used to extend a vehicle's range and/or reduce energy storage space. The complex relationships in the overall system must be determined and taken into consideration. In particular, efficiency characteristic maps for electric machines, power electronics and energy storage systems play a significant role in this respect.

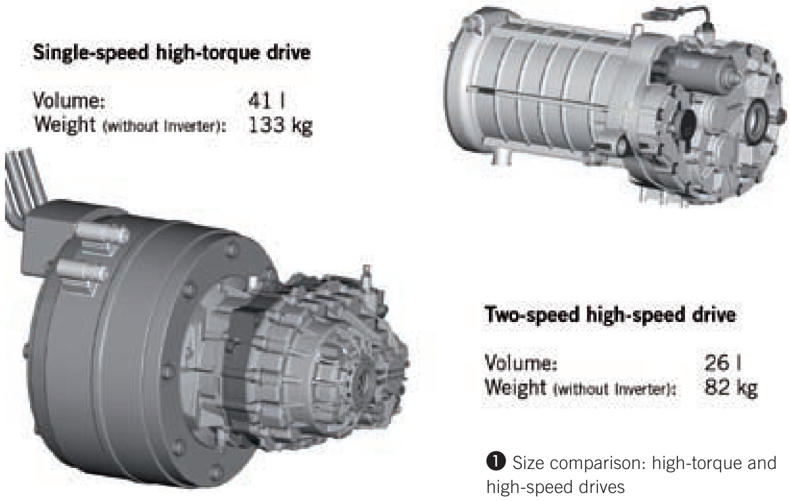
Therefore, Getrag has developed methods for taking a holistic view of electric drive systems and to design them for specific customer needs.The co-operation between Getrag and Robert Bosch GmbH in the field of hybrid drives has contributed to the continuous improvement of these complete drivetrain solutions. The portfolio of products available shows that many concepts have already been implemented and are in use as single-speed and two-speed transmission variants over a broad torque range. There is also an optional disconnecting device to remove the e-machine and its inertia from the drivetrain when not required.

HIGH-TORQUE VERSUS HIGH-SPEED

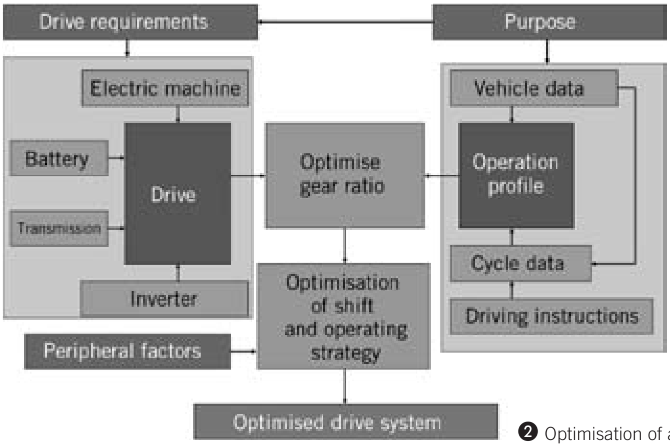
The design of electric drive systems starts with identifying the requirements for output torque and the maximum desired wheel or axle rotation speed. The torque range on a passenger car usually varies between 1000 and 4000 Nm at the axle depending on the size and type of vehicle.

Comparatively large electric machines, known as "high-torque electric machines", are needed to provide these torques. The torque, which is generated proportionally to the effective area of the gap between rotor and stator, varies quadratically with the diameter and linearly with the overall length of the active parts. It follows from this that torque firstly has a substantial influence on installation space, weight and costs. The second variable that contributes to performance, rotation speed, simply does not need installation space.

Downsizing becomes possible with the utilisation of "high-speed electric machines". The combination of a high-speed electric machine with an efficient helical gear transmission makes for a compact, lighter unit, but which delivers comparable output torque. For this reason, combinations of transmissions with electric machines



❶ Size comparison: high-torque and high-speed drives



❷ Optimisation of an electric drivetrain

(so-called gear motors) are widespread, not only in the automotive sector, but in all fields of drive technology such as electric hand tools or electric locomotives.

❶ shows electric drivetrain systems with the same power. The electric drive on the left is a single-speed high-torque drive, while that on the right is a high-speed electric drive with two gears. The size comparison clearly demonstrates the advantages of the high-speed solution. The two gears of the high-speed drive increase maximum torque and lead to considerably increased efficiency if the drive is designed optimally.

Two-speed drives can be designed as torque break shift or power shift units, depending on customers' requirements. Getrag has developed the technologies necessary for this and has already introduced them into series production [1].

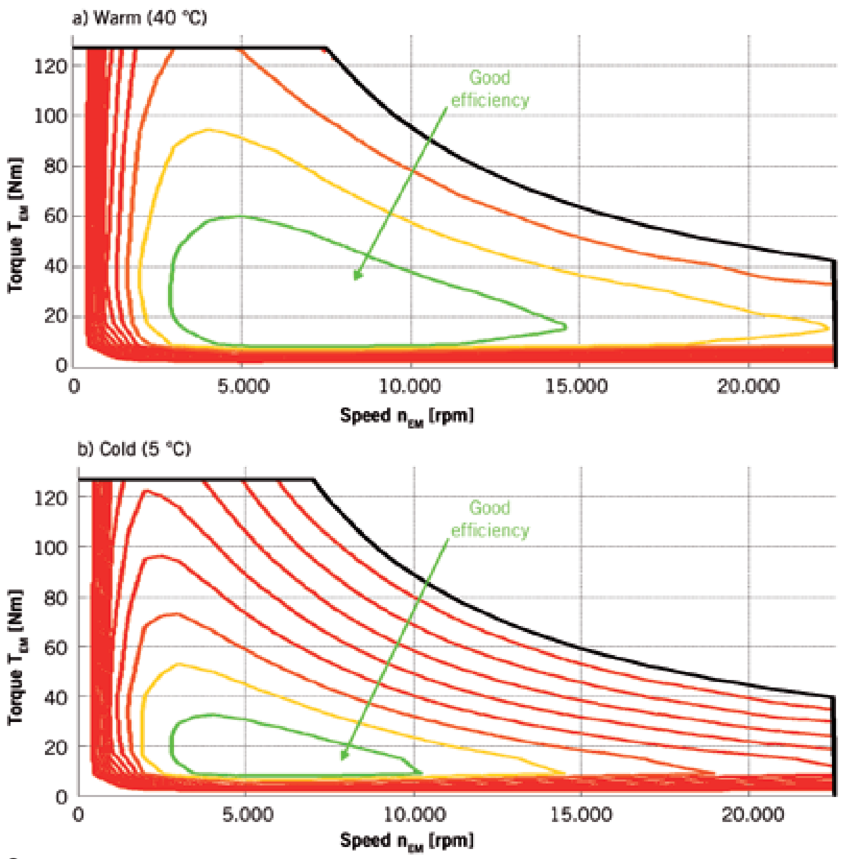
TOOLS FOR THE DESIGN OF ELECTRIC DRIVETRAINS

Using a comprehensive, thorough series of simulation tools, it is possible to deliver meaningful answers at a very early stage in projects to a variety of questions regarding proposed systems. Hardware prototypes are consequently conceived from the outset with the requirements "designed in".

Getrag uses its in-house developed programmes for the necessary simulations, and maintains a database of model parameters for a wide range of hybrid variants and topologies alongside pure electric drives [2]. These include a simulation programme that estimates consumption using conventional, hybrid and electric drivetrains, which has been validated with independent metrics [3]. When developing the necessary models, a physical description of the drivetrain components is generally used.

❷ is illustrating the design sequence of an electric drive system. First, the transmission ratios are set based upon the drive's requirements and purpose. Then an inverse quasi-static calculation of vehicle consumption with an optimised shift and operating strategy is made on the basis of the gear stepping that has been determined. The consumption, range and comfort of handling can be analysed as outcomes [4].

Physical models of the components take into account drag torques, moments of inertia, electric resistances, specific sources of dissipation and other factors, which can also be mapped as a function of temperature when necessary. ❸ shows the efficiency graphs of the electric machine, power electronics and battery from the same drive with the battery both warm and cold. The conclusion drawn from this is that the efficiency characteristic of an electric drive changes as a function of different parameters. It is possible to respond to these changes with the shift and operating strategy by selecting the gear with the higher overall efficiency rating for the electric drive system at any given operating point. By taking the influences of various ambient conditions into consideration, other potential gains can be identified and can be used to improve the overall efficiency in multi-speed drive systems.



❸ Efficiency maps for electric machine, inverter and battery at different battery temperatures

Optimization of operating point strategies is also available to create a warm-up strategy to optimize the overall energy consumption during the target driving cycle (New European Driving Cycle, NEDC). For this investigation interfaces to other systems as well as the auxiliary energy consumption are taken into account.

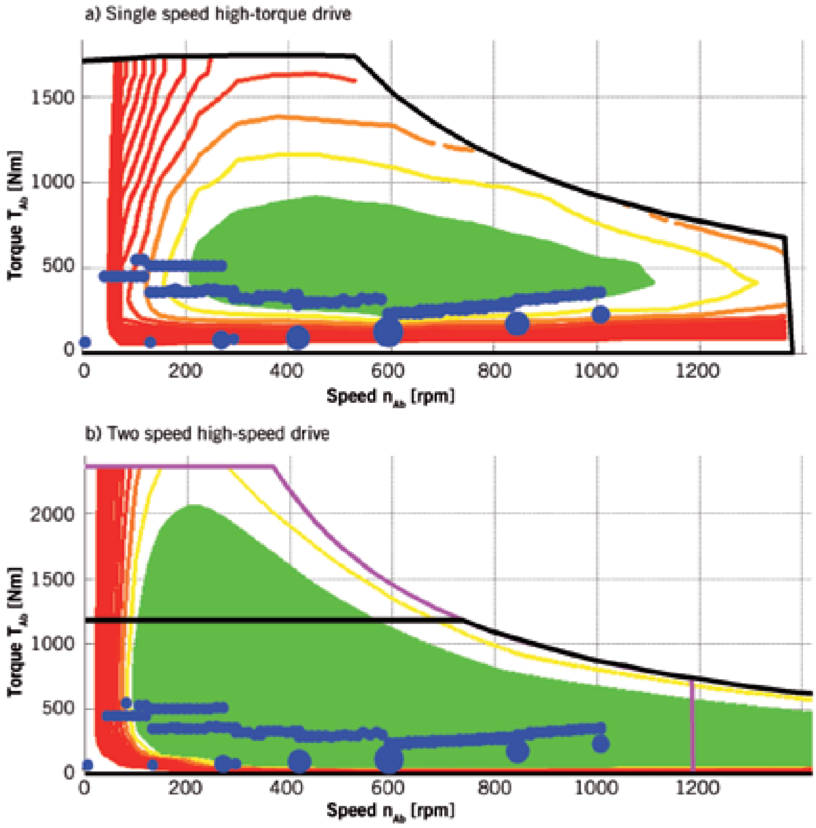
COMPARISON OF TWO DIFFERENT DESIGNS

❹ illustrates two different electric drivetrains. The two drives have a maximum output of 100 kW and use the same power storage device. On the top, a high-torque electric machine with a maximum torque of 550 Nm and a maximum speed of 4500 rpm is shown. The single-speed transmission has a gear ratio of 3.3. In the design variant on the bottom, a high-speed electric machine with a maximum torque of 127 Nm and a maximum speed of 22,500 rpm was selected. The transmission in this case is a two-speed unit with gear ratios of 19 and 9.5.

The electric machines are comparable from the point of view of their field weakening ratios and maximum efficiencies. The high-torque machine, however, has optimised efficiency in its maximum output range whereas the high-speed machine was optimised in a part-load range. A comparison of the drive systems' weights including the power electronics shows that the two-speed high-speed drive is 53 kg or 37 % lighter than the 143 kg single-speed high-torque drive.

As illustrated in ④ further on, the two-speed drive offers higher maximum output torque T_{Ab}, giving the vehicle better climbing performance and acceleration. The advantage of a two-speed drive is quite clear in acceleration comparisons in medium-class vehicles: the vehicle equipped with the two-speed drive accelerates from 0 to 100 km/h in 7.4 s, while the vehicle with the single-speed drive requires 8.4 s.

Looking at the isolines on the efficiency maps in ④, it is clear that the area where efficiency is highest is considerably larger in the case of the two-speed design. The area of highest efficiency for both gears can be enlarged by optimising gear ratio selection. The overall efficiency characteristic map can be achieved if the gear selected is always that with the higher overall efficiency.



❹ Efficiency maps for single-speed high-torque drive and two-speed high-speed drive with operating points of the NEDC

The operating points of the reference vehicle in the NEDC are plotted as blue spots in the respective efficiency characteristic maps in ④. The size of the spots indicates the relative time spent at each operating point. The areas corresponding to constant speeds in the NEDC can be clearly identified as large spots. Here, only a small output torque is required on level surfaces. This means that the second gear can be used for large portions of the time. In the design illustrated below in ④, second gear clearly achieves distinct advantages because of its better overall efficiency from speeds of approximately 35 km/h (corresponding to an output speed n_{Ab} of 300 rpm) [5].

An energy saving of approximately 18 % over the single-speed high-torque drive was achieved in the comparison selected here. This was accomplished by applying an efficiency-boosting shift strategy with a shift lock at high accelerations to the two-speed high-speed drive. Comparing a single-speed drive with a two-speed drive, both equipped with the same motor, the latter is capable of approximately 5 to 10 % lower consumption, depending on the vehicle configuration; in extreme cases this difference can be higher.

CONCLUSION AND OUTLOOK

Using simulation tools developed in-house, Getrag is able to configure an optimised overall system for vehicle hybrid and electric drives at a very early stage, while taking all the relevant parameters into account. Tried-and-tested development processes are used to implement designs right through to series production. Development of these new systems has been made possible by the many years of expertise in the fields of gear and transmission design, NVH, durability and industrialisation. The enhanced requirements with regard to transmission noise that result from the relative silence of electric vehicles demands detailed understanding of how that noise is affected by the transmission noise behavior.

To design electric drive systems in such a way as to exploit their full potential, it is essential to take the overall systems and their relevant peripheral factors into consideration. The weight of the drive and the space it occupies, as well as the use of costly magnetic materials, can also be reduced by using higher speed motors generating less torque. By optimally designing a unit with a two-speed transmission and a high-speed motor, the tractive force graph can be extended and the total energy requirement reduced. For the same vehicle range, this can be used to reduce the size of the electric power storage device. Based on current and forecast costs of electric power storage devices, this will lead to considerable cost reductions for complete systems.

As mentioned at the outset, today's vehicles with electric drive systems are often equipped with single-speed drive systems. These systems can be optimised by modifying the transmission ratio to take into account the peripheral factors described.

Apart from the pure optimisation of the electric drive, profitability plays an important role in this new technology with low production quantities. For small production runs, an optimised gear range is the most favourable option for improvement. A holistically optimised electric drive system with a specially developed electric machine and a matching two-speed gear transmission becomes advantageous when production numbers are high. Getrag offers modular solutions for all variants depending on the scenario.

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THE SEAL OF APPROVAL FOR SCIENTIFIC ARTICLES IN ATZ.
ATZ PEER REVIEW
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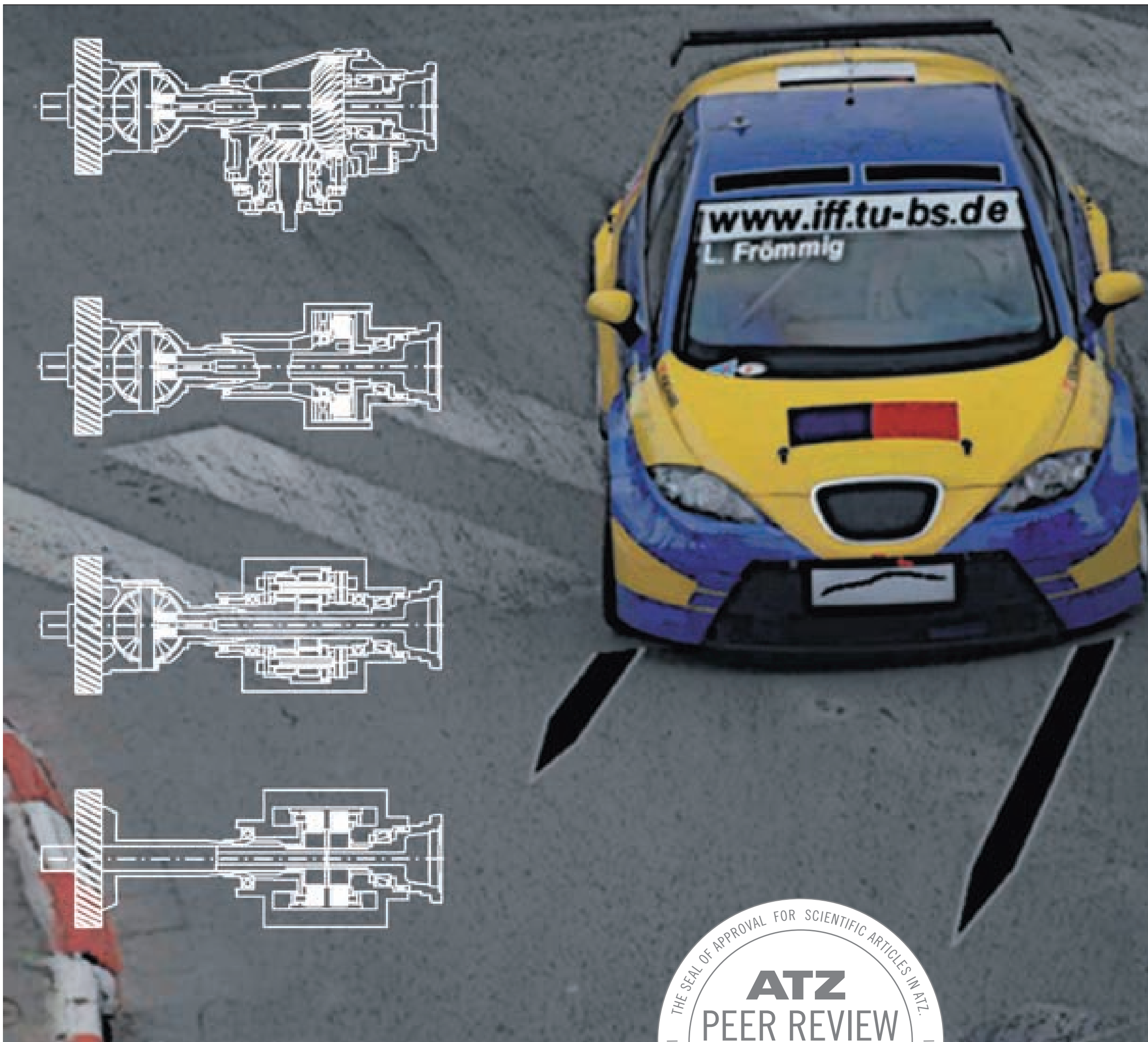
COMPARISON OF ACTIVE SYSTEMS FOR LATERAL DRIVE TORQUE DISTRIBUTION

Lateral torque distributing drive train systems allow the use of the adhesion potential adapted to the wheel load, which leads to a significant improvement of agility and vehicle performance especially in front-wheel drive applications. The Institute of Automotive Engineering of the Technische Universität (TU) Braunschweig conducted a simulation analysis and systematic concept comparison of these systems.



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1 INTRODUCTION

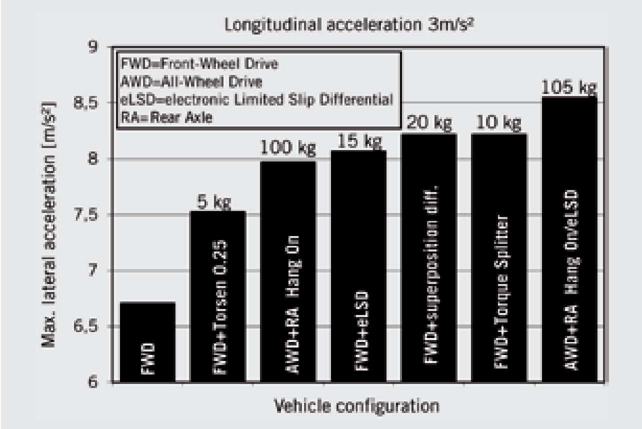
The ability of front-wheel drive vehicles with conventional differential for a coeval transmission of longitudinal and lateral forces is limited by the dynamic wheel load transfer.

During cornering, the traction slip of the unloaded inside wheel is higher than that of the outside wheel. In an extreme case, it tends to spin when reaching its adhesion limit, which is avoided by intervention of the acceleration slip control (ASC). The necessity of the ASC intervention therefore limits the maximum longitudinal and lateral acceleration even before the adhesion limit of the outside wheel is reached.

As an alternative to all-wheel drives the vehicle performance can also be improved by an asymmetric lateral drive torque distribution, which complies with the dynamic wheel load distribution.

At the Institute of Automotive Engineering of the TU Braunschweig an analysis and comparison of longitudinal and lateral distributing drive train systems as well as their combination was conducted. For this purpose internally developed simulation models were used. Passive systems were also taken into account due to their low overall costs. The present article focuses on the examination of lateral torque distributing drive train systems and their working principles.

❶ shows that lateral distributing drive torque systems reach maximum lateral accelerations similar to a hang-on all-wheel drive system. An advantage of the lateral distribution is the reduced additional weight of up to 20 kg as well as the low efficiency losses, which results in an additional fuel consumption of up to 0.1 l.

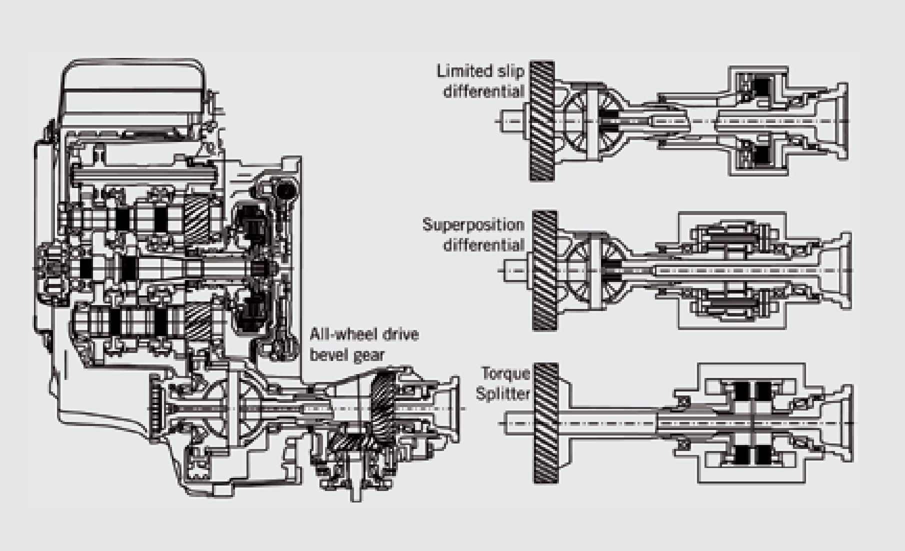


❶ Maximum lateral acceleration of different front and all-wheel drive configurations

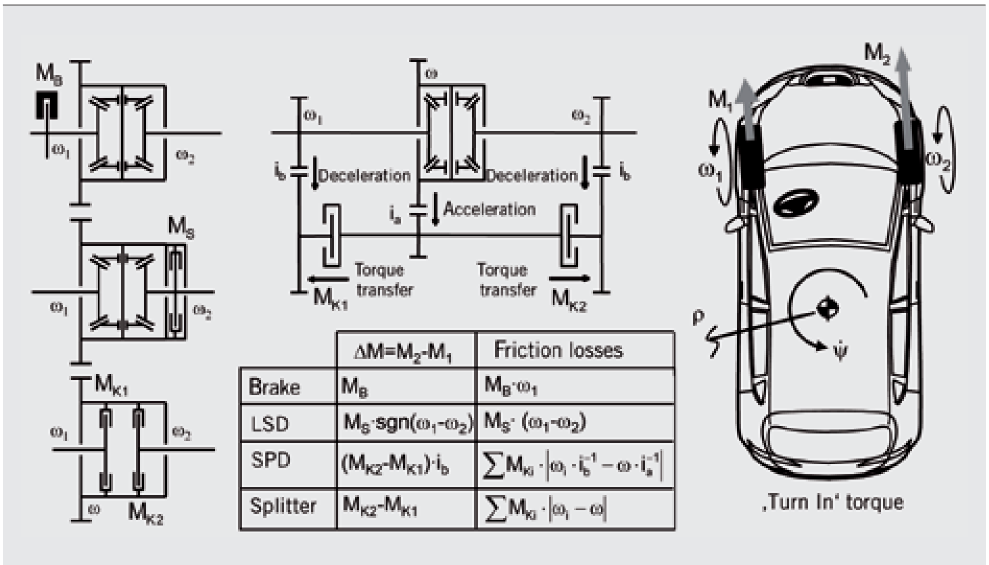
A challenge resulting from the application of lateral distributing systems in front-wheel drives down to the A-segment, in addition to the cost pressure, is especially the integration of transmission and final drive as well as the interaction with the steering system.

2 MARGINAL CONDITIONS WITH TRANSVERSE ENGINE

In front-wheel drive vehicles with transverse engine, the transmission and final drive are installed in one gearbox, as shown in ❷ for a VW dual clutch transmission. For the economic integration of torque distributing systems, no constructional modifications are required. The space for the all-wheel drive bevel gear may alternatively be used for a lateral distributing system if the requirement above is met. But only asymmetric constructions are possible which results in disadvantages in terms of system modularisation compared to active rear final drives. ② also shows construction concepts for a limited slip differential, a superposition differential, and a torque splitter (see also [1]). The lateral distributing system is linked in the same way as the all-wheel drive by splines



❷ Package situation with transversal engine



❸ Final drives for lateral torque distribution

of the differential cage. The output end is connected to the right drive shaft. A separate housing is used for the integration of the actuator technology.

The problem of the steering system interaction is well known for high powered front-wheel drives. This effect is amplified by the disturbing force lever arm in connection with a defined asymmetric drive torque distribution. McPherson axles, which are often used at the front axle, may have a disturbing force lever arm of up to 50 mm. Asymmetric driving forces therefore lead to a significant influence on the steering wheel torque. To explore the full potential of lateral distributing drive train systems, the steering system also has to be regarded. Two strategies are possible. The first one is a constructional minimisation of the disturbing force lever arm [2] as used in the Ford Focus RS with a torque sensing limited slip differential. A hardware-neutral and therefore inexpensive compensation of the disturbing torques by an electromechanical steering system is the second strategy. A similar strategy is already used in mass production cars for the compensation of the torque steer effect caused by drive shafts of different lengths.

3 LATERAL DISTRIBUTING DRIVE TRAIN SYSTEMS

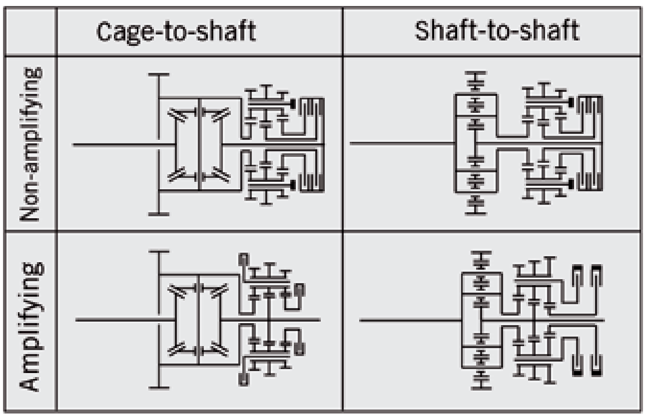
For the lateral drive torque distribution in mass production cars the systems illustrated in ❸ may be used. A simple solution is the extension of the electronic stability program (ESP), which produces asymmetric driving forces by a wheel-individual brake intervention. A major advantage of this system is that no additional weight is needed. But high friction losses occur when the system is activated, so the driver only profits from the system when driving at low speeds [4]. Passive and active limited slip differentials, superposition differentials, and torque splitters are much more powerful systems for the performance improvement of front-wheel drive vehicles.

In an electronically controlled limited slip differential (eLSD), the multiple disc clutch is fitted between the differential cage and a drive shaft. The locking torque is controlled from the outside, but its transfer direction is determined by the sign of the wheel speed difference.

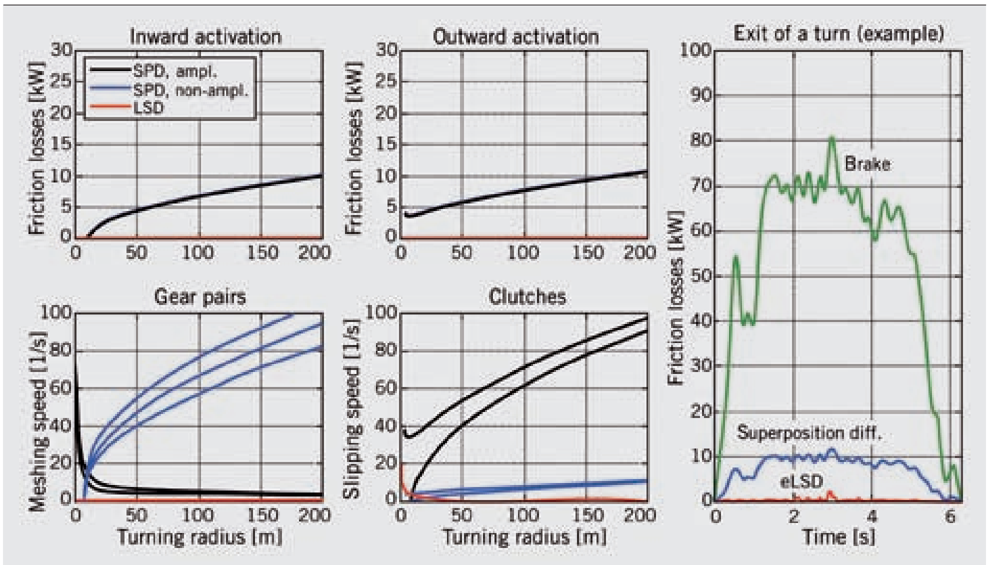
With a limited slip differential, torque transfer is only possible from the faster to the slower rotating wheel. This is usually the outside wheel due to kinematic conditions. Torque transfer to the outside wheel is only possible with increasing traction slip at the inside wheel, which occurs with appropriate longitudinal and lateral acceleration. Furthermore, the “turn in” torque difference cannot be kept up without the traction slip, without engine torque it is therefore only possible to produce “turn out” torque differences.

The dependence on wheel speed difference and engine torque is avoided by using superposition differentials with two active clutches. The working principle of a simple structure is illustrated in ③. By using a gear pair, the differential cage accelerates an auxiliary shaft, with a multiple disc clutch fitted on its end. The second part of the multiple disc clutch is fitted to another gear pair which decelerates the speed of the right drive shaft. This leads to a defined speed difference in the multiple disc clutches, so the activation of the right clutch results in drive torque transfer to the right wheel.

The torque difference may be kept up independent of the engine torque because the direction and the amount of the speed



❹ Structures of superposition differentials



❺ Efficiency behaviour

difference are almost independent of the traction slip. This characteristic behaviour will be discussed in connection with the vehicle dynamics in the following chapter.

All superposition differentials comply with this working principle, but even more compact gear structures are used in order to minimise the required space. They are divided into amplifying and non-amplifying as well as shaft-to-shaft and cage-to-shaft constructions. ❹ shows corresponding gear structures for application in a transversal engine.

The major design goal for a superposition differential is to guarantee a sufficient speed difference even for the turning circle. There is a minimal curvature radius where the speed difference of the "turn in" clutch becomes zero. Further criteria for these systems include the amplification and the required clutch capacity and the occurring power losses, divided into efficiency losses due to gear meshing and clutch slipping as well as activating friction losses.

It has to be mentioned that systems designed for the same minimal curvature radius – independent from the gear structure – also have the same friction losses, but may differ in meshing and slipping losses. The characteristic parameters are represented in ❺. Amplifying structures have the advantage of a low clutch capacity, but it is more important that the planetary gear set rotates in a block during straight-line driving. This means that meshing losses are minimised. The slipping speed of the clutches, however, is on the same level as the wheel speed. Therefore specific measures have to be taken to reduce drag torques.

The control precision is also influenced negatively by the efficiency of the planetary gear set. This and the low slipping speeds are the main advantages of non-amplifying structures.

In terms of drive train efficiency, the limited slip differential has a conceptual advantage: its integration does not require additional gear pairs and the speed difference during straight line driving is zero, therefore the lowest influence on the overall efficiency can be expected even without additional measures.

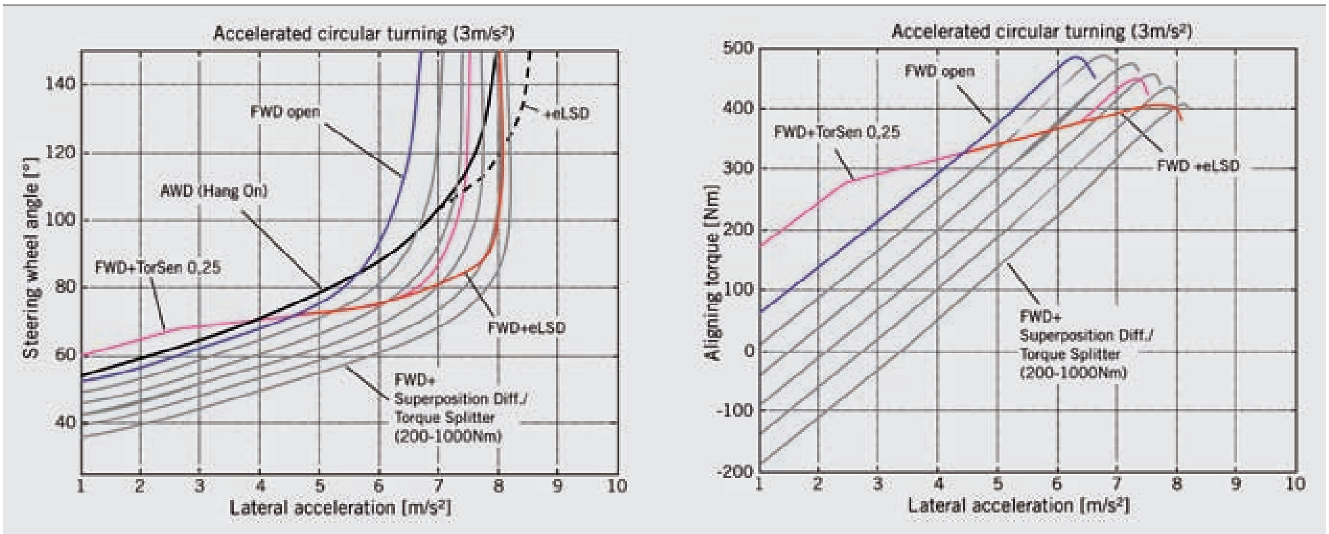
With its ability to stop speed compensation if necessary, it is the only system which is able to generate high torque differences at minimal friction losses. This minimises the power and heat input to the multiple disc clutch which reduces the life of the components.

The torque splitter does not use the 50:50 basic distribution of the differential, but direct control of the clutches. This concept eliminates the starting clutch, which may result in a low additional weight of only about 10 kg. Unlike the superposition differential there are no kinematic constraints with sufficient engine torque. Without engine torque the torque splitter can only be operated as a limited slip differential. Speed compensation can only be achieved by slipping clutches, which should be avoided as much as possible by intelligent control strategies.

The behaviour in case of failure as well as the high effort needed to coordinate the clutches argue against an application in a single-axle driven vehicle from a present point of view.

4 VEHICLE DYNAMICS

A different potential for the improvement of the vehicle dynamics results from the kinematic characteristics. ❻ left represents the characteristic behaviour of the described drive train systems. It illustrates the steering wheel angle in relation to the lateral acceleration in accelerated circular turning. The conventional differential (blue) shows the typical understeering of a front-wheel drive. The controlled limited slip differential (red) is activated when the inside wheel passes the outside wheel. If the speed compensation of the differential is stopped, the vehicle behaviour keeps linear and reaches a significant higher lateral acceleration. The maximum locking torque amounts to approximately 900 Nm and is limited through the state of the drive shafts rotating with the same speed. The added value of a controlled limited slip differential becomes obvious, when the TorSen-differential (magenta) is regarded. Under engine torque, it even locks in case of low lateral accelerations and therefore generates a "turn out" torque, which is also visible by the aligning torque, ⑥ right. The maximum locking torque or the locking ratio is limited by the required compromise between normal and sporty driving conditions as well as the μ-low behaviour. The performance of a controlled limited slip differential is not reached.



❻ left: Accelerated circular turning; right: Aligning torque of a conventional McPherson axle

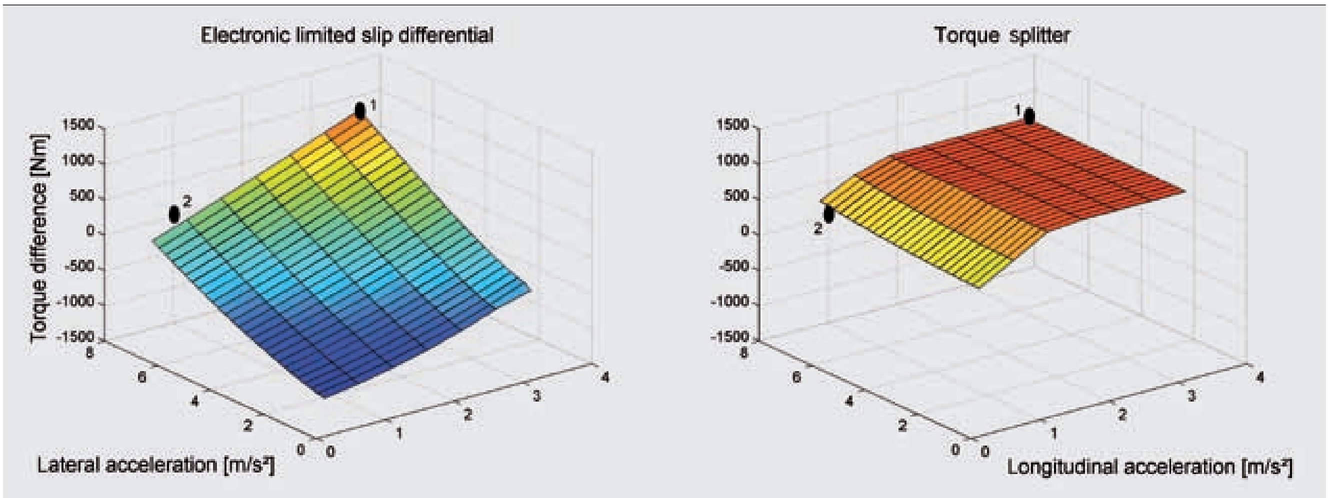
Superposition differentials and torque splitter (grey) allow a faster rotating outside wheel, which allows the generation of higher torque differences between the front wheels. But with higher drive torque, the outside wheel reaches the adhesion limit, so there is only a slight increase in vehicle performance at the stability limit. It has to be mentioned that compared to an all-wheel drive (black) the stability limit is reached more abruptly. The all-wheel drive shows a more consistent understeering behaviour.

The advantages of the superposition differential result from its capability to generate drive torque differences independent of the vehicle state. ❼ therefore shows the torque potential of a locked limited slip differential and a torque splitter.

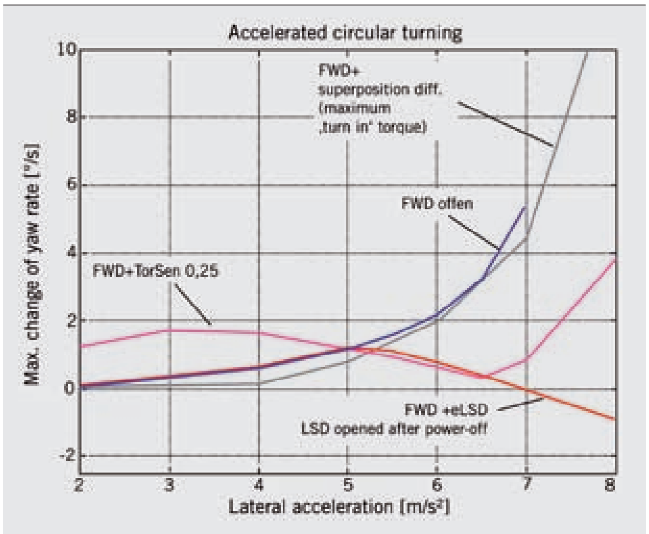
The limited slip differential shows its typical driving behaviour depending on the driving state in a way that “turn in” torques occur first with corresponding longitudinal and lateral accelerations. A sporty driving style is thus required for a benefit from this system.

But in this way the limited slip differential becomes only active when a sufficient aligning torque is generated by lateral tire forces (⑥ right). Active compensation is not inevitably necessary. The torque splitter is permanently able to generate “turn in” torques, but the full potential will only be reached with increasing longitudinal accelerations.

In case of a quickly released throttle (power off), characterised by the change of the working point from 1 to 2, the “turn in” torque cannot be kept up. Two effects therefore interact. Due to the increased front axle side force potential, the car tends to oversteer, but the elimination of the “turn in” torque leads to an understeer tendency. This causes a damping of the oversteering reaction even without keeping the differential locked. In situations with high lateral accelerations, this effect may become the dominant one. Then the vehicle understeers, which is illustrated in ❽ by the maximum yaw rate change after the throttle release.



❼ Differential torque potential of engine torque dependent systems



❽ Power-off behaviour

	Limited slip differential	Superposition differential	Torque splitter
Starting on: μ_{low}	Additional value only achievable with all-wheel drive		
μ_{split}	Only small additional value, if vehicle is already equipped with brake lock differential		
Vehicle response from constant straight-line driv.	o	+++	o
Stability intervention during understeering	o	++	o
Stability intervention during oversteering	++	+++	++
Reduction of power understeering	++	+++	+++
Increase of maximum lateral acceleration	+++	+++	+++

❾ Comparison of functionality

This has to be regarded as critical, because the situation will not be self-correcting anymore as wished. These statements are also valid for the torque splitter, which without engine torque behaves as a limited slip differential.

The superposition differential can keep up the torque difference at any time, so a neutral or slightly oversteering response can be realised. Another advantage of the superposition differential is that "turn in" torques can already be generated from low lateral accelerations without engine torque. So the normal driver can also experience this system, particularly through improved vehicle response. Measures with regard to the steering system, however, are required for this purpose. ❾ represents a table summarising the functional potential of the systems.

5 SUMMARY

As opposed to all-wheel drive vehicles, a limited slip differential in front-wheel drive vehicles is able to improve the vehicle's agility already beginning from medium lateral accelerations and is also able to increase the maximum lateral accelerations.

Due to the dependence on the wheel speed difference, only the sporty driver will profit from this system, which is also true for the torque splitter. Added value under normal driving conditions can only be achieved with a superposition differential, but to explore its full potential the steering system has to be regarded.

With increasing engine power or poorer road conditions, the lateral torque distribution reaches its limits, so the all-wheel drive is still superior to a single-axle drive under these conditions.

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