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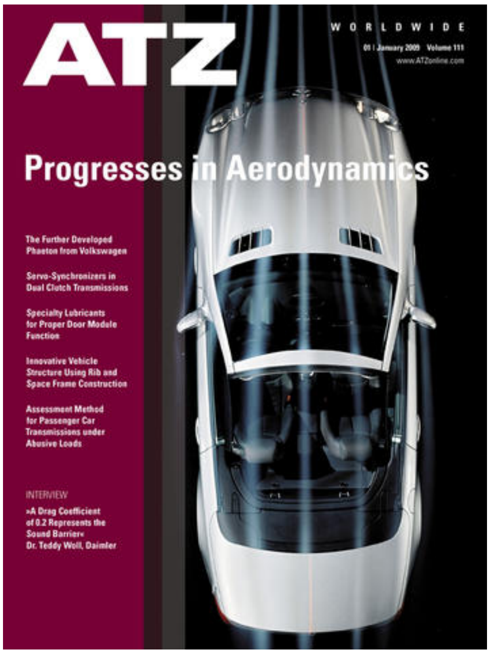

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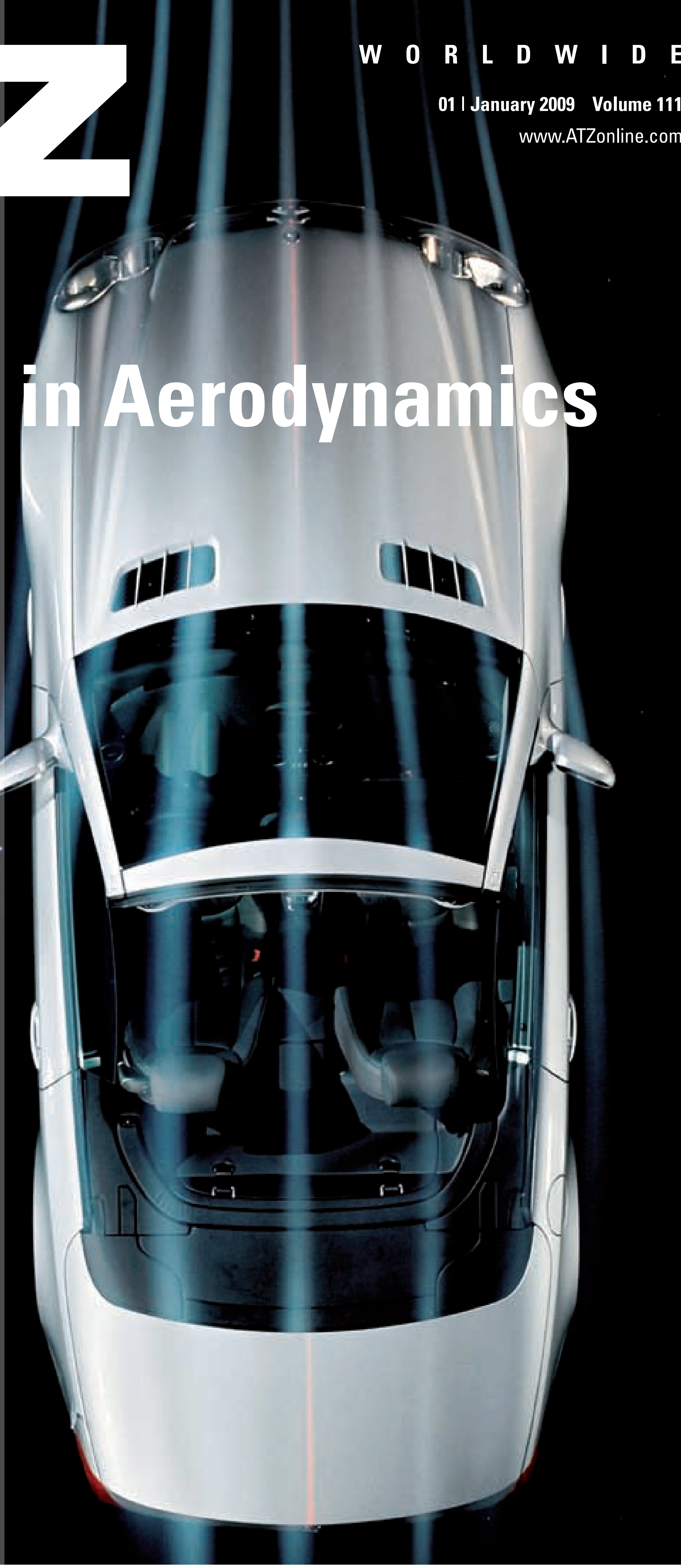

Progresses in Aerodynamics

The Further Developed Phaeton from Volkswagen

Servo-Synchronizers in Dual Clutch Transmissions

Specialty Lubricants for Proper Door Module Function

Innovative Vehicle Structure Using Rib and Space Frame Construction

Assessment Method for Passenger Car Transmissions under Abusive Loads

INTERVIEW

»A Drag Coefficient of 0.2 Represents the Sound Barrier« Dr. Teddy Woll, Daimler

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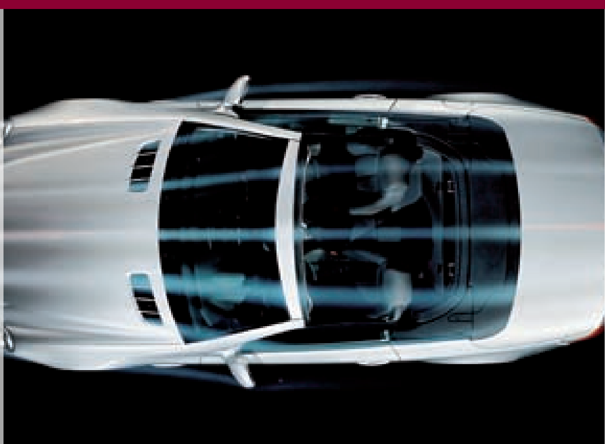
COVER STORY

Progresses in Aerodynamics



4
The days when **Aerodynamics** raise emotions are gone. Expert Wolf-Heinrich Hucho explains how to overcome the obstacle c_D=0.25. Audi and Werum show a control system together with a measuring technique for a trouble-free operation of climatic wind tunnels.

At the Top of the Agenda

Dear Reader,

Whenever I talk to heads of development and chief executives, there is really only one topic at the moment: what will 2009 be like? How badly will my company be affected? Will there be further cuts in production in the short term? These are questions that managers are forced to confront. And even excellent crisis management doesn't seem to provide all the answers.

There is no doubt that our industry is facing a time of structural change and paradigm shifts. Meeting the upcoming CO_2 limit values not only in Europe but also in the USA and China will require huge investment by manufacturers and suppliers. Research and development and, at a later date, the establishment of production facilities for the new technology will cost billions. In order to survive in the future, companies need to set their future course during the present crisis – simply tightening their belts is not the answer.

Providing strategic orientation for top management in the automotive industry is the aim of our new magazine "Automotive Agenda", which we are launching in the new year. Instead of offering hectic news journalism, our monothematic quarterly journal will present background articles written by some of the top decision-makers in our industry. For example, the authors writing for the first issue of this premium magazine, which will be dedicated to the issue of electromobility, will include Dr. Thomas Weber, Dr. Bernd Bohr, Dr. Thomas Neumann and Erich Sixt.

Each issue will focus on a topical issue that is of vital importance for automotive managers and examine it from various perspectives. The editorial team – led by Peter Gaide – sees itself as the organiser of a "think tank" that addresses the burning issues facing the largest German industrial sector and its future. The design of the magazine will set new standards in technical information. The quality of the paper, graphics and photography put it in the top league of high-quality business and cultural magazines. Want to know more? Then order a sample copy now at www.automotive-agenda.de.

I wish you all the best for what is likely to be a difficult 2009.

Johannes Winterhagen
Milan, 27 November 2008

Johannes Winterhagen
Editor-in-Chief

ATZ WORLDWIDE
01|2009

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

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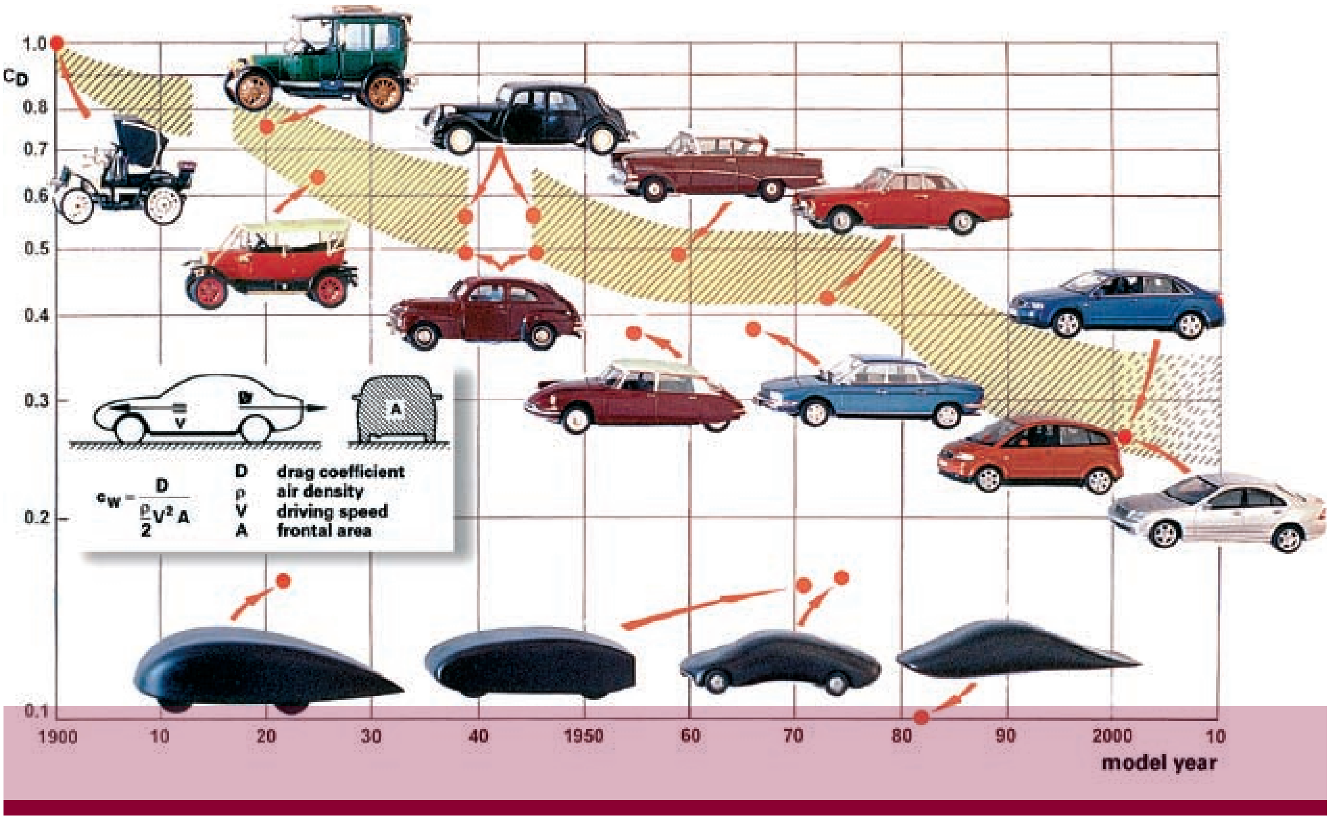


Threshold Strategy

Halving the Drag Coefficient Seems Possible

The drag coefficient of cars is gradually falling to the level of $c_d = 0.25$, but its progress is so hesitant that you could be forgiven for thinking that this figure represented a limit set by the laws of physics. It will be demonstrated how the apparent obstacle can be overcome. A further cut in fuel consumption can be achieved if the reduction in drag is accompanied by the corresponding use of lightweight structures and a specially designed powertrain.

1 Introduction

Things have been rather quiet recently in the field of vehicle aerodynamics. The times when aerodynamics aroused an emotional response and caused major controversy among designers have long since gone. Was aerodynamics not responsible for increases in performance? Did it not make a major contribution to reducing fuel consumption? So what is happening today?

It is true that aerodynamics engineers have not been completely idle. Every possible improvement has been made to the detailed features of cars in order to reduce wind noise and dirt, for example, but the reduction in drag, the key factor in aerodynamics, has come to a complete standstill. Despite all the work put into reducing the drag coefficient every time a new vehicle is developed, no progress has been made over the last ten years or so. However, the Audi A2 does represent a milestone in development, **Figure 1**, as it was the first production car to achieve a drag coefficient of 0.25 (the efforts involved have been documented by Dietz [1]). In the meantime, some other manufacturers have moved closer to this threshold, but no one has yet succeeded in passing it. It is almost as if the figure of 0.25 were an asymptote, but a number of different prototypes have demonstrated that this is not the case.

Similar stagnation has occurred in the past. From 1960 to 1975, when the so-called ponton-style body design was popular, a drag coefficient of 0.4 seemed to represent a limit. Some individual models were able to improve on the figure, such as the Citroën DS 19 and the NSU Ro 80, but at the cost of highly unconventional designs, which were not suitable for modification.

Progress only started again in this area when two strategies involving the optimisation of details and shapes were adopted. The working hypothesis on which these strategies were based was to take the flow around the individual adjacent areas of the vehicle body as far as possible back to elementary mechanisms [2]. The results of this approach are summarised in **Table 1**.

2 The Threshold Strategy

2.1 The Philosophy of the Threshold Strategy

The fundamental concept behind this strategy consists of two stages. The first involves identifying the drag coefficient that can be achieved for each of these areas without taking technical feasibility into account. In the second stage, technically feasible measures are developed that allow the actual drag coefficient to be brought as close as possible to this theoretical figure. The result was a further reduction of $\Delta c_d \approx 0.150$ when compared with the "optimisations". The

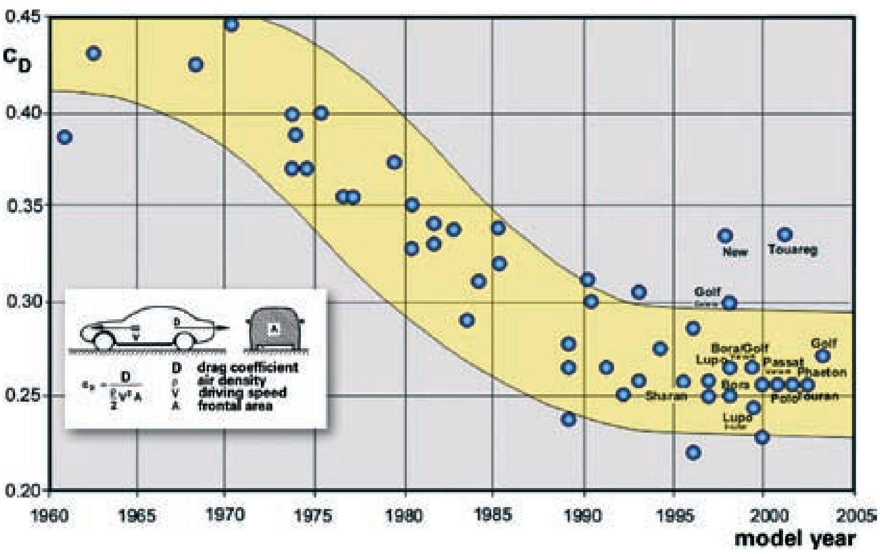
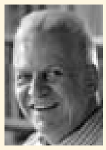


Figure 1: It almost seems that c_d = 0.25 is the lower limit of the drag coefficient for production cars (diagram VW)

The Author



Dr.-Ing. Wolf-Heinrich Hucho is a consultant engineer in Schondorf (Germany) and publisher of the Vieweg-Teubner book „Aerodynamik des Automobils" („Automotive Aerodynamics").

Table 1: Strategies for developing car shapes

Development strategy	Δc_d achieved/possible	Example
Optimising details (1970)	0.100	VW Golf I: c_d of 0.51 reduced to 0.41
Optimising forms (1980)	0.100	Audi 100 III: c_d = 0.30
Threshold (200?)	0.150	?

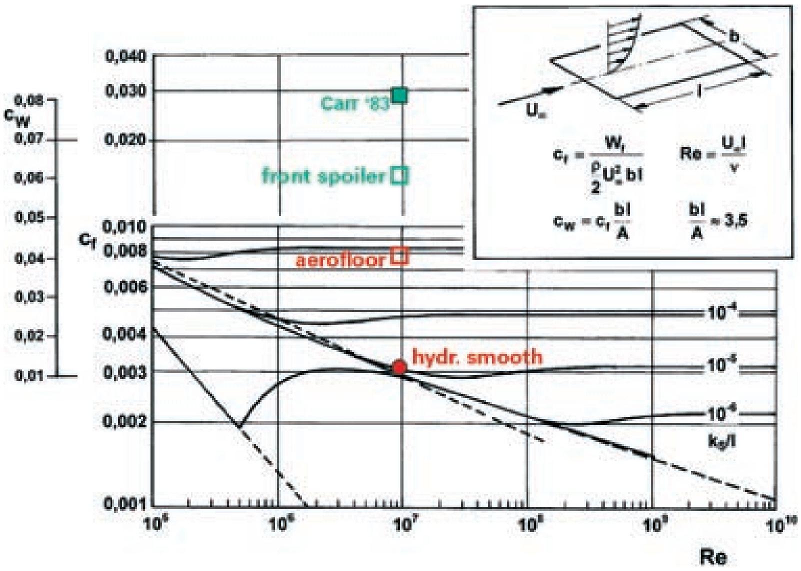


Figure 2: The „classic" panel diagram with an extension for vehicle underbodies

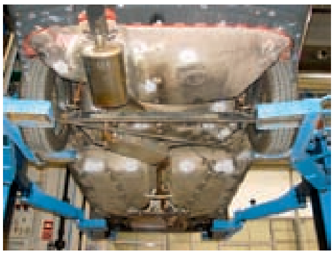
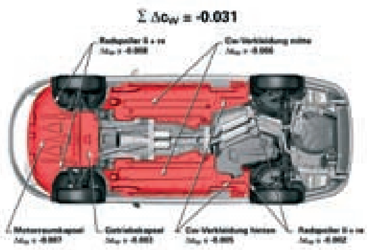
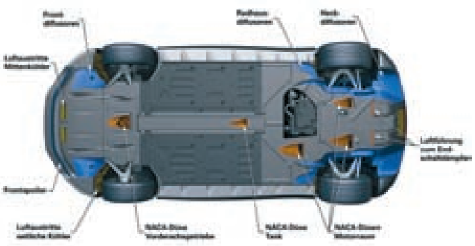
	Audi 80 (1985): „Normal" underbody
	Audi A8: Largely smooth underbody [4]
	Audi R8: Extremely smooth underbody with 3D elements [5]

Figure 3: From the indented ("raw") underbody to the smooth "aero-floor"

practical examples below show how this can be achieved.

2.2 The Underbody

Ideally, the underside of a vehicle should be completely smooth with no open cavities. The drag on this smooth panel would be purely frictional drag, which has been researched in great detail.

The results are summarised in a traditional panel diagram shown in **Figure 2** [3]. The box in the top right-hand corner of Figure 2 indicates how the coefficient of frictional drag produced by the wetted surface is converted into a drag coefficient.

Carr (1983) indicates that the drag coefficient of the underbody, which was originally highly indented, is $c_d \approx 0.08$, see the top picture in **Figure 3**. In order to represent this figure in the panel diagram, its ordinates had to be increased by one order of magnitude.

The front spoiler was the first means used to counteract this high level of drag. Later, some areas of the underbody were smoothed out, making the front spoiler largely redundant, see Figure 3 centre and bottom. Even the "aero-floor" was still a long way from being a hydraulically smooth panel. This is probably the result of the cavities needed to allow sufficient clearance for the wheels and to dissipate the heat from the exhaust system. In technical terms, it is possible to close these gaps, but determining whether it is worthwhile on a production scale would require a cost-benefit analysis and a comparison with other options for reducing drag.

2.3 The Wheels

Reducing the wheels' contribution to drag is particularly difficult. Because of the need to cool the brakes, a flow of air through the wheels is essential. The proposal to make the volume of the wheel arch as small as possible (investigated by Cogotti in 1983 [6]) could only be implemented to a limited extent because of different wheel and tyre sizes and snow chains.

The incoming flow of air to the wheels is at an angle, as a result of the displacement effect of the vehicle body. The slip angle at the front wheels can be up to 15° [7] with the result that the drag coefficient of the wheels more than doubles

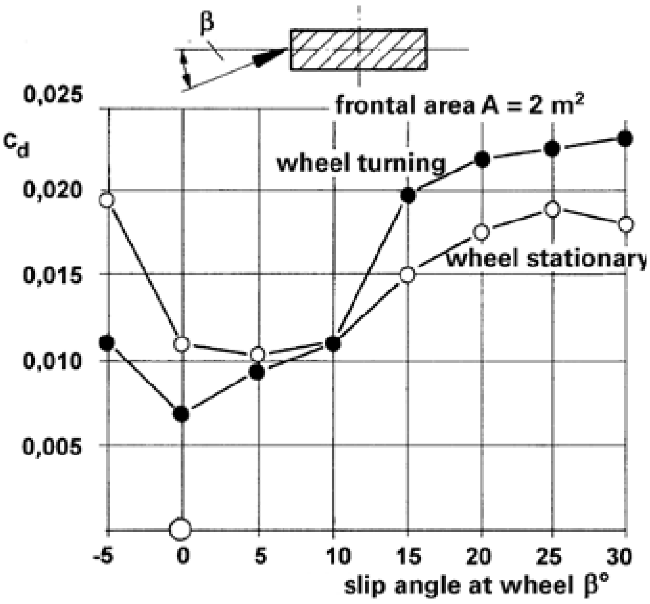


Figure 4: Increase in the drag of a wheel when the incoming air flow is at an angle [8]

[9], as shown in **Figure 4**. The air flow to the rear wheels is also at an angle of up to 10° outwards. However, if the vehicle has a diffuser, the incoming air flow to the rear wheels is at an angle from the outside. A wake then forms on the inside of the wheels, which disturbs the flow in the diffuser and negates its effect. This results in a significant increase in the drag of the rear wheels.

It is not yet clear whether the angle of the air flow to the wheels can be reduced by placing elements in front of them. One option is a cooling air duct, as proposed by Wolf and Preiss [10]. Designers have never really come to grips with side covers for the rear wheels, which were a common feature of cars in the past. Perhaps they will try them again, because the use of snow chains does not present a problem in the case of front-wheel-drive cars. The figure of $\Delta c_{dR} \approx 0.020$ is a conservative estimate of the wheels' potential for reducing drag.

2.4 The Rear

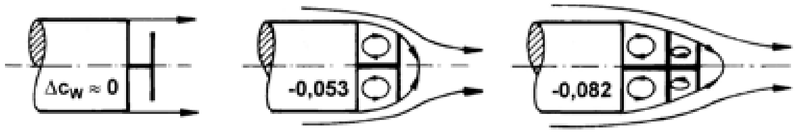
The rear end of the car is another area that comes under consideration in this respect. Once again, results obtained some years ago, as shown in **Figure 5**, indicate how the drag of a blunt object can be reduced by manipulating its wake [11, 12].

Figure 6 shows some technical approaches to achieving the ideal figure of $\Delta c_d = -0.105$. Several investigations have been made into the use of a „parapet" on commercial vehicles, but the reduction of between 2 % and 4 % in fuel consumption did not compensate for the additional expense involved in the design.

Inward curving panels, shown in Figure 6 variant d, seem to allow the flow to be concentrated more effectively than is the case with the step, variant b. The „fluid tail" proposed by Morelli and Di Giusto [13], shown in variant e, appears attractive in particular in combination with an active measure, which involves filling the wake of the rear wheels with air that is supplied by the rear wheels themselves acting as radial fans. This prevents the ring-shaped turbulence that forms at the rear of the car from becoming „broken open" underneath to create two longitudinal, horseshoe-shaped areas of turbulence. Tests in a wind tunnel using a Fiat Punto equipped in this way, **Figure 7**, resulted in a reduction in drag of $\Delta c_d/c_d = 18$ %.

2.5 Windscreen, A-Pillars and Mirrors

Even well-designed cars often have one area that, in terms of flow dynamics, is far from perfect. This consists of the A-pillars and door mirrors, which must be considered in combination with the windscreen. The flow in this area, shown in the diagram on the left of **Figure 8**, depends on two parameters: the angle of inclination δ and the radius r. The case where the angle of inclination is



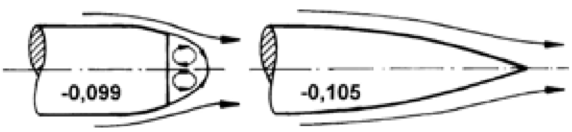
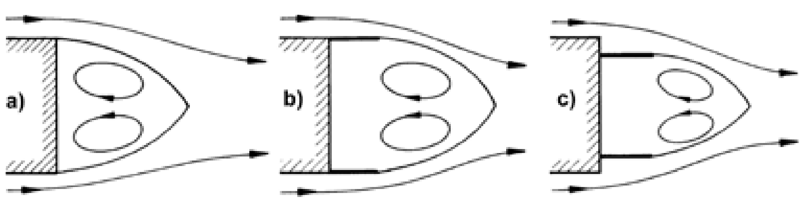


Figure 5: Reduction in the drag of a blunt body as a result of modifications to its wake [10]



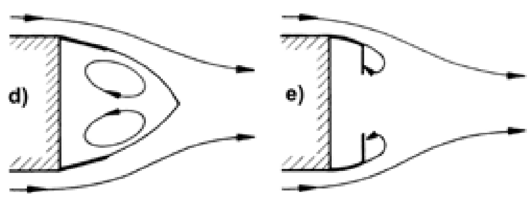


Figure 6: Options for influencing the wake behind a hatchback: a) no measures taken; b) "parapet"; c) "parapet" with step; d) inward "parapet"; e) fluid tail [12]

Figure 7: Fiat Punto with fluid tail [12]

90°, in other words where the front of the car is vertical, has been thoroughly researched. The result is shown in the diagram on the right of Figure 8. There is an "optimum" radius $(r/b)_{opt}$, beyond which the drag no longer continues to decrease. The figures for an angled windscreen (δ < 90°, dashed curve) are currently unknown.

In the case of relatively small radii, which are common today, the well-known A-pillar turbulence forms. This is part of a horseshoe-shaped area of turbulence as shown in **Figure 9**, which induces downforce A_{dS}, which in its turn is quadratically related to an induced drag $W_{i\,wS}$ as shown in Eq. (1):

$$W_{i\,WS} \cong A^2_{WS} \qquad \text{Eq. (1)}$$

An appropriately curved windscreen (large radius) prevents the A-pillar turbulence from forming and therefore avoids its contribution to drag (unknown figure) $W_{i\,WS}$. (It is worth noting here that the variation in the geometrical and flow parameters of radius, angle and Reynolds number shown in Figure 8 is ideally suited as a benchmark for CFD codes.) This gives designers the opportunity to try out alternatives to the increasingly flat windscreens currently in use. However, windscreen manufacturers will be less keen on this prospect.

The last factor is the mirrors, which should no longer present an aerodynamic problem. In the age of videos and display screens they are an anachronism. The mirror blind spot, which leaves many drivers in a state of uncertainty, should also become a thing of the past.

3 Summary

The strategy described here was used to analyse the essential contributory factors to a car's drag. The results are summarised in **Table 2** (further details in [14]).

$$\frac{\Delta c_d}{c_{d0}} = \frac{0.150}{0.320} = 0.47 \qquad \text{Eq. (2)}$$

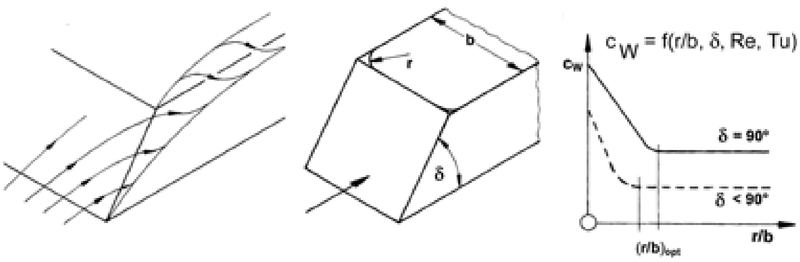


Figure 8: Flow around the A-pillar: diagram showing the "optimum" figures for the radius r and the angle of inclination δ

A reduction in the drag coefficient of a hatchback in the Golf class of Δc_d = 0.150 seems feasible. On the basis of a drag coefficient of 0.32, which is a very good figure for a compact hatchback, according to Eq. (2) this results in an improvement of 47 %. It is important to note here that Δc_d = 0.150 is the threshold for a hatchback with a specific body size (length : height : width). It is possible to move gradually towards this threshold as a new generation of cars is introduced, depending on the skills of the designers and the cost-benefit ratio. The amount of fuel saved (Δb) can be determined using Eq. (3) and Eq. (4), where the values with an index of zero are those before the reduction in drag:

$$\frac{\Delta b}{b_0} = \varepsilon \frac{\Delta c_d}{c_{d0}} \qquad \text{Eq. (3)}$$

$$\varepsilon = f[V_F(t)] \qquad \text{Eq. (4)}$$

The ε factor depends on the cycle in which the evaluation is carried out. In the case of the New European Driving Cycle (NEDC), ε equals 0.25 (according to [15]). The unrealistically low average speed of 32,5 km/h of the NEDC, which has been decided on by bureaucrats, has marginalised the influence of aerodynamics on fuel consumption. Some car manufacturers have carried out studies into how the cars they manufacture are driven by their customers. According to [16], the ε factor is two to three times higher than that of the NEDC. A comparison of different practices to identify consumption is to be seen in **Table 3**. The threshold for reducing the drag of a hatchback given here therefore represents a reduction in consumption of an average of 30 %. (This result is subject to the reservation that it comes from only one OEM. Two others have made off-the-record comments indicating that they have achieved similar results, but these have not yet been published.)

4 Conclusions and Proposal

In this assessment, only the rear axle was modified to ensure that the elasticity (the time needed to accelerate from 60 km/h to 120 km/h) in fourth gear remains unchanged. No other modifications were made to the car. However, if

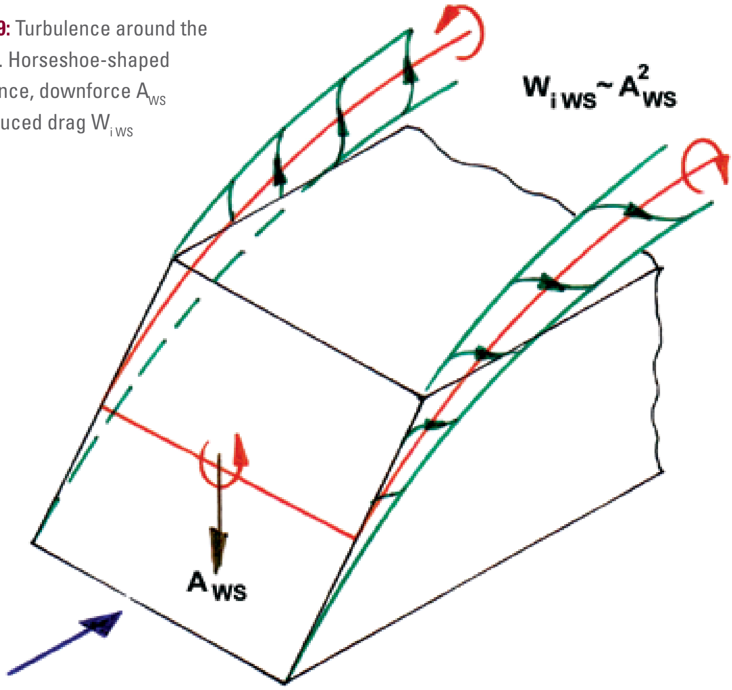


Figure 9: Turbulence around the A-pillar. Horseshoe-shaped turbulence, downforce A_{ws} and induced drag $W_{i\,ws}$

the potential of a reduction in drag is to be exploited to the full in order to cut fuel consumption, two further steps are needed (which have long since been identified). The engine power must be reduced to such an extent that a moderately low maximum speed cannot be exceeded, but this maximum speed must be the same as the previous model. The reduction in power will only be accepted by customers if it does not affect the acceleration. The essential second step is therefore to reduce the weight of the vehicle accordingly. These supporting measures that accompany the reduction in the drag coefficient have not been taken into account in the estimate of the fall in consumption given above. The proposal below has been developed on the basis of the arguments in this article.

To develop and construct a car with optimum fuel consumption and the following features:

- a drag coefficient close to the threshold
- lightweight structures
- a specially designed powertrain.

The Unicar project is one good example of a development of this kind.

Table 2: Possible measures for reducing the drag coefficient of a hatchback with a starting figure of $c_{d\,0}$ = 0.320

Action		$\Delta c_{d\,aero}$	$\Delta c_{d\,real}$
Chassis	Smooth underside	0.030	0.015
	Rear diffuser	0.025	0.025
	Wheels	0.025	0.020
Rear end		0.100	0.060
Cooling system		0.020	0.020
A-pillars and mirror		0.010	0.010
Total $\sum\Delta c_d$		**0.210**	**0.150**

Table 3: Possible consumption reduction, $\Delta b/b_0$

Cycle	Average velocity (km/h)	ε factor	$\Delta b/b_0$
NEDC	34	0.25	-12 %
1/3 Mix	76	0.40	-19 %
Customer-relevant (acc. to Reiser et al.)	≈ 60	0.5 bis 0.75	-0.24 bis -0.35

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"A Drag Coefficient of 0.2 Represents the Sound Barrier"

Cars can be designed to be as aerodynamic as possible, but still be safe and good to look at. This is obvious from the drag coefficient rankings, where Mercedes-Benz cars are among the leaders. ATZ discussed the market opportunities of highly aerodynamic cars, the existing potential for reducing drag in cars and the lowest drag coefficient possible in production cars with the head of aerodynamics at Daimler, Dr. Teddy Woll.

Dr. Teddy Woll
Senior Development Manager in the Mercedes-Benz Cars Aeroacoustics, Aerodynamics and Wind and Climate Tunnels Department

ATZ It is not long ago that people were discussing the contribution that aerodynamics could make to reducing fuel consumption. Has its potential been fully exploited?
Woll No, but when you look at the design of current mid-size saloons, such as the C-Class, their aerodynamic potential is already close to its limits. We have been able to reduce an already good drag coefficient of 0.27 to 0.26 using the aerodynamic measures from the Blue Efficiency package.

ATZ Experts say that the aerodynamics of a production car could be improved by almost 50 %. Do you think that this is realistic?
Woll It is important to explain what we mean by aerodynamics and what our starting point is. I understand aerodynamics to mean the effective drag, which is the product of the frontal area A and the drag coefficient c_d. Typical American SUVs have drag coefficients over 0.5 and frontal areas bigger than 3.8 m^2. This gives a c_dA of 2 m^2. The best European SUVs, including the M-Class, have a c_dA of just under 1 m^2 and are therefore 50 % better. A mid-size saloon is almost twice as good again with a figure of 0.54 m^2. That is the range covered by current production vehicles. If you want to reduce that by a further 50 %, you will find it very, very difficult. The most extreme low-consumption concept car currently on the market has a c_dA of 0.28 m^2, which is equivalent to 52 % of 0.54. At 1.40 m wide and 1.14 m high, it is narrower than the Smart and lower than the SLK. The central questions is: How far are you prepared to compromise on safety and comfort in order to reduce the drag coefficient and therefore the CO_2 emissions?

ATZ In your opinion what would a perfectly aerodynamic car look like?
Woll The perfect aerodynamic shape is a teardrop. It has a drag coefficient of around 0.05, which is the optimum figure. Cars cannot be rotationally symmetrical, simply because they have wheels that also increase the drag coefficient by around 0.1. The design of a perfectly aerodynamic car would need to head in this direction, which means that it would have to be as streamlined and smooth as possible above the midline.

ATZ Would it sell on the market?
Woll Such an extreme design probably would not. But the market opportunities for concepts that combine aerodynamics and low fuel consumption are constantly growing.

ATZ Which components of cars do you believe have the greatest potential for aerodynamic improvement?
Woll The greatest potential for aerodynamic improvement is at the rear. The longer, narrower and more angular it is – for saloons slightly higher, for estates slightly lower – the better the aerodynamic figures. We have equipped the current C-Class with air ducts, so that the outgoing air flows create a virtual angular rear end and therefore a clear aerodynamic trailing edge. The length of the rear end can reduce drag by between 5 % and a maximum of 15 %. However, in the case of a 15 % reduction, the rear end would be so long and narrow that it would probably be difficult to get a suitcase into it widthways. From the point of view of accidents, a long, narrow rear end is good, but it can make parking difficult. The ideal solution is to make the

"How far are you prepared to compromise on safety and comfort in order to reduce the drag coefficient and therefore the CO_2 emissions?"

car narrower behind the B-pillar, but this means that you can no longer seat three adults in the back. However, if you want to reduce drag by 10 %, these are the sorts of things that you have to take into consideration. Another relevant factor is the roof height at the back, in particular the trailing edge of the roof or the point where it joins the rear windscreen. The 'greenhouse', in other words all the areas surrounded with glass, offers a further potential reduction of 5 %. You would need to move to much rounder and smoother forms, which could result in conflicts with the heating and visibility requirements. The cooling air system can give potential reductions of between 5 % and 10 % and the wheels another 5 %. However, redesigning the wheels can conflict with road safety requirements. Finally, there are cavities in the underbody, such as the wheel arches and the exhaust tunnel. It would be possible to insulate the exhaust system completely and mount it above a completely smooth underbody, with the tail pipe (also well insulated) emerging from the bumper. However, when the exhaust needed replacing, this would be three or four times more expensive than it is currently. In principle there are solutions for everything, but some of them are very expensive. It is important to think carefully about how to identify the overall optimum solution.

ATZ What do you think is the lowest drag coefficient possible for a mass production car?
Woll In the 1980s, a few prototypes were created that were used to sound out the limits. For example, the GM Aero had a drag coefficient of 0.14. It had completely smooth wheels and a completely smooth underbody. The angle of the windscreens meant that it heated up to very high temperatures in the sun and it had no brake cooling, no cooling air system and no seams on the bodywork. If you were to use this shape, which would not fit into a standard garage, to create a production car, you would have to add at least 30 or 40 points, which would bring the drag coefficient up to 0.18. If you then shortened it to a length that is suitable for everyday use, the figure would increase to 0.2. I think this represents the sound barrier for production cars.

ATZ Over what period of time do you think this figure could be reached?
Woll I can imagine that it would be possible over the next decade.

ATZ How close do you think digital flow simulations come to real flow processes?
Woll Shortly after I took over responsibility for the department, I set up a CFD team that within a few years became one of the world leaders. We are currently among the best in the world at flow simulation. In the case of drag, our error margin is less than 1% compared with the results from the wind tunnel. The benefit is that we can look in detail at highly complex flow phenomena, for example the flow in the wheel arches, which previously were a mystery to us when we only had the wind tunnel available. However, in contrast in the wind tunnel we can take a large number of measurements in a very short time. On a 1:1 scale, we currently make between 20 to 25 measurements and on a model scale between 40 and 50 measurements a day. For a flow simulation we need several days. This is improving as computers become more powerful and it is likely that in the foreseeable future we will be able to do this in the same amount of time as it takes in the wind tunnel.

ATZ Dr. Woll, it has been a pleasure talking to you.

The interview was conducted by Roland Schedel and Bettina Merkelbach.

Dr. Teddy Woll

studied industrial and electrical engineering at the Technical University of Darmstadt (Germany). From 1987 onwards he worked as a research associate at the Institute of Electromechanical Design. His areas of research included measuring the internal pressure in the eye with the eyelid closed and later solar and electric-powered lightweight vehicles. Until 1995, he worked at Smart in the overall design department with responsibility for improving aerodynamics and weight and for developing alternative drive systems. In 1996, he joined Daimler in Sindelfingen to work in development. Since April 1999, he has been head of the Aeroacoustics, Aerodynamics and Wind and Climate Tunnels Department.

(Source: Audi)

Comprehensive Control System for Wind Tunnels

Modular Design for Adaptability and Extensibility

In the context of automotive development, modern wind tunnel technology constitutes an essential element for Audi AG to further optimize their vehicles. In this respect, the wind-tunnel centre in Ingolstadt with its three wind tunnels provides the engineers with unique test facilities. The same as with the already installed aeroacoustic wind tunnel, trouble-free operation is ensured by the comprehensive wind-tunnel control system from Werum Software & Systems and S.E.A. Datentechnik at the heart of the new climatic wind tunnel. The users are guided through all preparatory and testing phases by the control computer software. It not only permits rapid access to all resources but also the simple configuration of new tests. Adaptations and extensions are an easy matter thanks to the platform's modular design.

1 Introduction

Early in 2008, the German automobile manufacturer Audi took into operation its new climatic wind tunnel in Ingolstadt, Germany. This wind tunnel can be regarded as a performance benchmark. The installation permits the realistic simulation of a vehicle being driven on the road even under the most extreme environmental conditions. The wind tunnel is designed for passenger cars and SUVs, as well as for sports and racing cars. This means that Audi now has a wind-tunnel centre with three sectors:

– an aeroacoustic wind tunnel to carry out aeroacoustic and aerodynamic tests on models and real vehicles
– a thermo wind tunnel to test the cooling of combustion engines, gearboxes and brakes
– a new climatic wind tunnel for such tests as windshield de-icing, coolant-performance measurement at high temperatures, and heater operation at low temperatures.

Apart from Audi, other Volkswagen vehicle brands (Volkswagen Seat, Skoda) regularly use the facilities in Ingolstadt as guests. But also motor-racing teams, unaffiliated with the Volkswagen Group, utilize the wind tunnels, as do winter-sport teams, swimmers, cyclists and other industrial enterprises.

When the new facility was built, Audi also took the opportunity to completely modernise the wind-tunnel control system. As a result, the new wind tunnel is equipped with a control system based upon the Wind Tunnel Control System (WTCS) from Werum and S.E.A. As early as 2003, these two specialists for large-scale test-stand software planned and installed a WTCS control system for Audi's aerodynamic and aeroacoustic wind tunnel. In the meantime, both control systems – which basically have the same layout – have more than proved themselves in practice.

2 Demands on Wind Tunnels

Today, modern wind tunnels must comply with a wide range of technical and functional requirements. For instance, in the case of the new climatic wind tunnel, the enormous increases in engine power and cooling capacity of today's vehicles, together with the required flow velocities and vehicle dimensions, meant that, compared to the previous wind tunnel, many subsystems (installations and devices integrated in the tunnel such as fans, air-conditioning, rolling road, or wheel drive) had to be adapted to comply with the new requirements. Consequently, the climatic wind tunnel is well equipped to cope with the increasing demands resulting from new developments and a broader product spectrum. Apart from this, it is also necessary to be able to precisely set the ambient temperature, the sun's radiation level, and the humidity, and to keep these constant during long test runs.

In order to realistically test a vehicle's interior ventilation and air-conditioning, the wind tunnel is equipped with a 6 m^2 (approximately 65 ft^2) nozzle and a fan for generating wind velocities of up to 300 km/h (approximately 190 mph). Furthermore, the equipment of the climatic wind tunnel wind must be particularly corrosion-resistant considering the extreme ambient temperatures like for example from -25 to +55 °C, under which the wind tunnel is operated, the saline substances used and the high levels of humidity, including rain. Many of the tunnel's components are therefore made of stainless steel. The climatic wind tunnel also provides extra conditioning chambers for preparing the vehicle before the test run starts.

Thanks to computer technology advances in the field of automotive developments, relatively precise simulations are available even before the first test runs start. This means that in order to be able to apply real measurement results in verifying the increasing amount of virtual data, new, and above all, precise measuring systems are needed in the wind tunnel. For instance, Audi is preparing to increase the measuring frequency so that it can even better register dynamic phenomena. The control system of the climatic wind tunnel though is already equipped to handle these higher data rates.

On the functional side, it must be possible to easily and purposefully integrate the wind tunnel's large number of different subsystems and the measuring systems themselves. Since subsystems are not only independent units with specific

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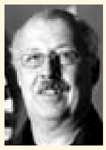

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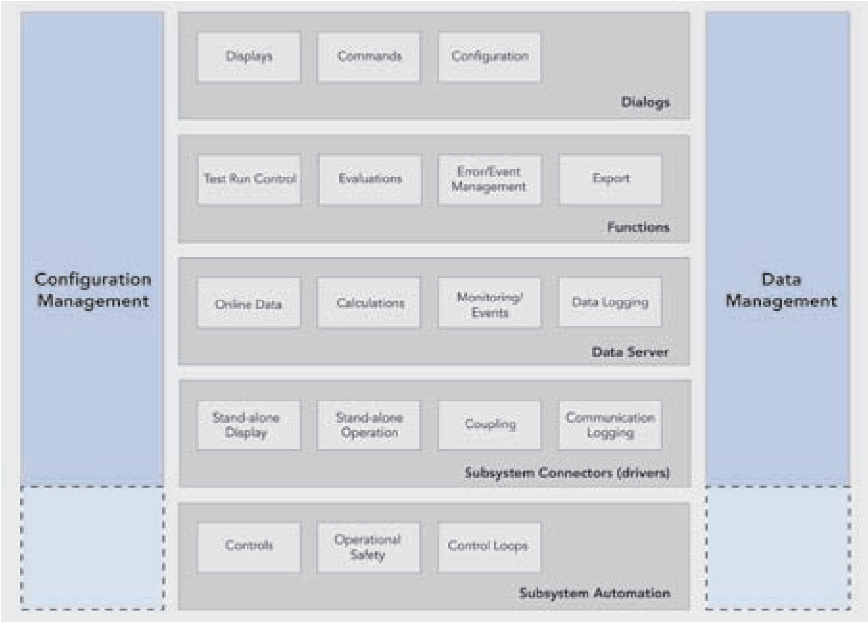

Figure 1: Component structure of the wind-tunnel control system WTCS with five levels (Source: Werum)

tasks but are also subject to short innovation cycles, they must be replaced relatively often by more modern systems – it is here that the control system's excellent integration capabilities are of great advantage. On the other hand, the measurement data must be rapidly evaluated, presented and processed, as well as being securely archived. In addition, the wind-tunnel control system in its role as a central component in wind-tunnel operation must be failsafe.

In the case of Audi, the wind-tunnel control system for the aeroacoustic wind tunnel had to be integrated without interrupting normal operations. Among other things, this meant that all available subsystems with their specific interfaces had to be integrated in the new control system – in cooperation with Werum, the Audi project team accomplished this assignment without any wind-tunnel downtime.

3 Design of the Control System

Werum's WTCS control system is based upon precisely defined, self-contained functional components. This platform is comprised of configurable core components and application-specific modules. The modular system is supported throughout by a client-server architecture which, for instance, enables any number of operating and display workstations to be set up. Each and every subsystem or measuring system is connected and triggered individually and independently.

Such essential functional components as the configuration and data-management, or the control of the test sequences, are standard Werum modules. The interfaces to the various subsystems and functions are designed so that the connection of standard tools is a simple matter. Further-ranging project-specific control-system functions and procedures were implemented by Werum in line with Audi specifications and in close cooperation with operators and users.

Depending upon requirements and architecture, the overall system can be distributed among several computers. Each functional component is allocated by configuration to a specific computer. The hardware is based on standard PC systems using Windows and Linux operating systems. The wind-tunnel control systems are can be operated autonomously. The data server for the central processes and databases is at the same time the interface to the Audi data network. The WTCS system components are organized in five levels, **Figure 1**:

– The dialog level comprises all displays, configuration, and operating functions. This level is used directly by the operator and other users. Individual dialogs and displays can be embedded in the interactive level. The layout of monitor windows, screens, and contents is also freely configurable.
– The function level comprises components which provide automated processes – for instance the sequence control, the evaluations and the export functions.
– The data server level is the central data hub of the system and is responsible for the distribution and availability of all data and information within the wind-tunnel control system. It is this level which permits the far-reaching decoupling of the other components. A computational module is used for online interlinking of arbitrary values (for instance, the measured quantities from subsystems), as well as for performing conversions or for deriving new quantities. Data logging is applied to continually store all information delivered to the data server in a database. A monitoring module is used in this level as the central supervisory function.
– For the most part, the subsystem connection level is comprised of the drivers. They connect these installations with the "data" via a generic interface.
– These subsystems are via automation processes on the subsystem level (for example, through programmable logical controllers). Basic function and operational reliability of the subsystems at this level. These functions are often already available in the various elements.

The configuration management and the data management are available for all component levels. The configuration management covers all data for the parameterization of the control-station functions and the coordinated operation (including, among other things, set-up definitions and sequence charts). The wind tunnel's quality-assurance measures are based on these data. The data management administers the measurement data, links these with the corresponding test configurations, and trans-

Figure 2: The operator console (left) and in the background of it the user workstation (Source: Audi)

fers them to the archive so that they are also at the disposal of the Audi engineers during tests at a later date.

4 Control System in Practice

At present, the WTCS users have more than 50 different dialogs and displays available for operation of the wind tunnel. The control system differentiates between the users according to their roles: Operator, Administrator, Test Engineer, Measurement Engineer and User. Depending upon his/her particular role, the person in question is allocated the corresponding authorizations.

This means that the wind-tunnel Operator has access to all control options on all installations, and is presented with all wind-tunnel data and measurement values, **Figure 2**. A User on the other hand, is allocated a separate workstation where the measurement data are “only” displayed visually. The user can in no way intervene in system operation. In order that he/she can work with the measurement data without delay, these are automatically imported into a table, along with the evaluations, and placed at the user’s disposal together with the test records.

Compared with the previous control system, the wind-tunnel operator is now provided with considerably more information from the wind-tunnel tests. For this reason, Audi and Werum concentrated on an uncomplicated operating interface which the operator can intuitively understand and control. The cockpit is the central display and presents the most important wind-tunnel parameters in a clearly arranged way, **Figure 3**. A status display indicates in colour the operating status of each subsystem, and in case of deviations from setpoint values immediately turns to red. This means that since the operator is warned visually in the event of deviations, he/she is no longer forced to directly monitor the process parameters.

Furthermore, upon demand, the cockpit displays online a selection of the most important process parameters such as wind velocity, humidity, and air temperature. After all, in addition to already predefined displays and presentations, the user can also work in the WTCS with individually designed display elements. The layout of the display interface was developed by the Audi wind-tunnel specialists in cooperation with Werum. Technically speaking, the control system’s display and operating interface (visualisation) is independent of the system’s actual operating procedures (for example data management, limit monitoring, event management). This means that the dialogs can already be tested before they are actually taken into operation in the wind tunnel.

Thanks to the sequence control, the person operating the test bench is able to automatically run through test sequences which have been completely configured beforehand. This contributes towards minimising faulty entries and sources of error. Formerly, the operator was forced to adjust many of the parameters by hand whilst the actual test run was taking place. Today, he or she simply calls up the already prepared test configurations from a database, or adjusts other tests to the current re-

Figure 3: Graphical presentation of the most important wind-tunnel parameters (Source: Audi)

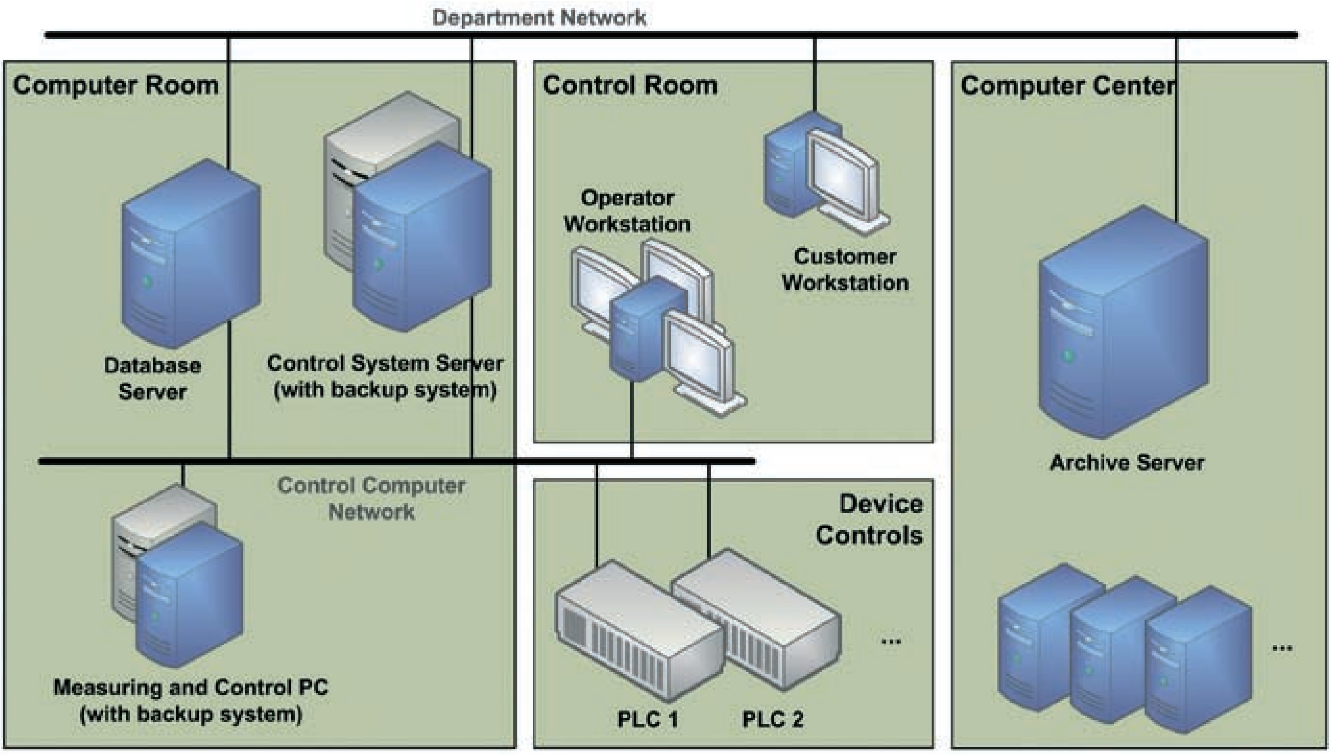

Figure 4: Reliability is trumps: High-level data integrity thanks to hardware redundancy (Source: Werum)

quirements. This rich stock of configurations also guarantees a high level of reusability for already existing test configurations, and permits the direct comparison between identical wind-tunnel tests performed on different dates. A further advantage of the uniform control system lies in the fact that the measured values for a given test are consistently time-synchronous.

For Audi, the short setting-up times for new test scenarios result in increased efficiency. Not least thanks to the test database and the clearly defined data management, this control system ensures a no-delay test start and a trouble-free test run. Compared to the former thermo wind tunnel, almost four times as many tests can be performed in a given period.

5 Reliability

The unusually high level of operational reliability is not least thanks to the concept drawn up jointly by Audi and Werum. With regard to the important technical components (for instance the control computer), the concept specifies redundant systems in order that in the case of malfunctions the switchover to backup systems takes place with a minimum delay.

With regard to the measurement technology the three wind tunnels operate as autarkic systems. Their data administration and data filing though are networked with each other. A central archive server is available for this purpose. Finally, all Audi test runs, including the respective configurations and measurement data, are transferred to the Audi tape library where they are retained for audit purposes, **Figure 4**.

6 Conclusion

In addition to economic considerations, above all operational and technical reasons were behind the Audi decision to select the WTCS control system by Werum and S.E.A.:

- software modularity: each and every subsystem has its own driver program, which sends all data to the central data pool and transfers the control-system commands to the subsystem
- data integrity and easy access to the data
- consistent Ethernet communication instead of proprietary communications systems
- client-server architecture
- the ability to arbitrarily distribute all components and the TCP-IP-based communication
- open interfaces for third-party applications
- standard hardware and software as the technical basis
- failure concept featuring many redundant technical components and an uncomplicated switching box for rapid manual change to the backup system.

Since extensions are an easy matter due to the open driver structure at the subsystem level, the new control system is a very future-oriented installation. Even a gradual modernisation of wind-tunnel hardware poses no problems. On the other hand, thanks to its flexibility and configurability, the WTCS is easy to integrate in existing wind tunnels. ■

The Further Developed Phaeton from Volkswagen

In autumn 2008, the Phaeton from Volkswagen underwent a substantial advancement in development. The key elements include a completely revised infotainment system, various improvements to the engine range and new design accents in the interior.

1 Introduction

The Phaeton has successfully established itself in the luxury car segment. It is valued for its long-distance comfort and its traction properties even in adverse weather conditions, thanks to its „4MOTION" permanent four-wheel drive system.

Some major innovations were already introduced in the spring of 2007, when the Phaeton became the first luxury car to comply with the future EU5 emissions standards with its V6 TDI diesel engine. The ACC (Adaptive Cruise Control) system was extended by the addition of the Follow-to-Stop function, and the new driver assistance systems were introduced: „Front Assist" with the "stopping distance reduction" function (pre-conditioning of the braking system and preventive brake intervention if the car gets too close to the vehicle in front) and „Side Assist" (lane change assistant for monitoring blind spots). In addition, an even more user-friendly telephone system with mobile phone connection via Bluetooth „rSAP" (remote SIM Access Profile) was integrated.

In autumn 2008 a further substantial advancement in development took place. The main components are a completely revised infotainment system, various improvements to the engine range and new design accents in the interior, **Cover Figure**.

2 New Generation of Info- and Entertainment

With the further development of the Phaeton — above all in the interior in the area of infotainment and comfort functions — customers from Autumn 2008 will have the benefit of innovations that allow an even higher level of functionality and user-friendliness. Highlights include the introduction of the new „RNS810" radio/navigation system, a twelve-channel high-end Dynaudio sound system and the changeover to white LED lights for the switch lighting. Furthermore, the car now addresses the growing demand among customers for solutions to integrate their own communication systems into the vehicle with the possibility of a standardised operating and display philosophy with the new infotainment system.

To implement the new system, the complete CAN bus-based infotainment architecture had to be functionally revised, **Figure 1**. Two newly developed components form the basis of the system: the

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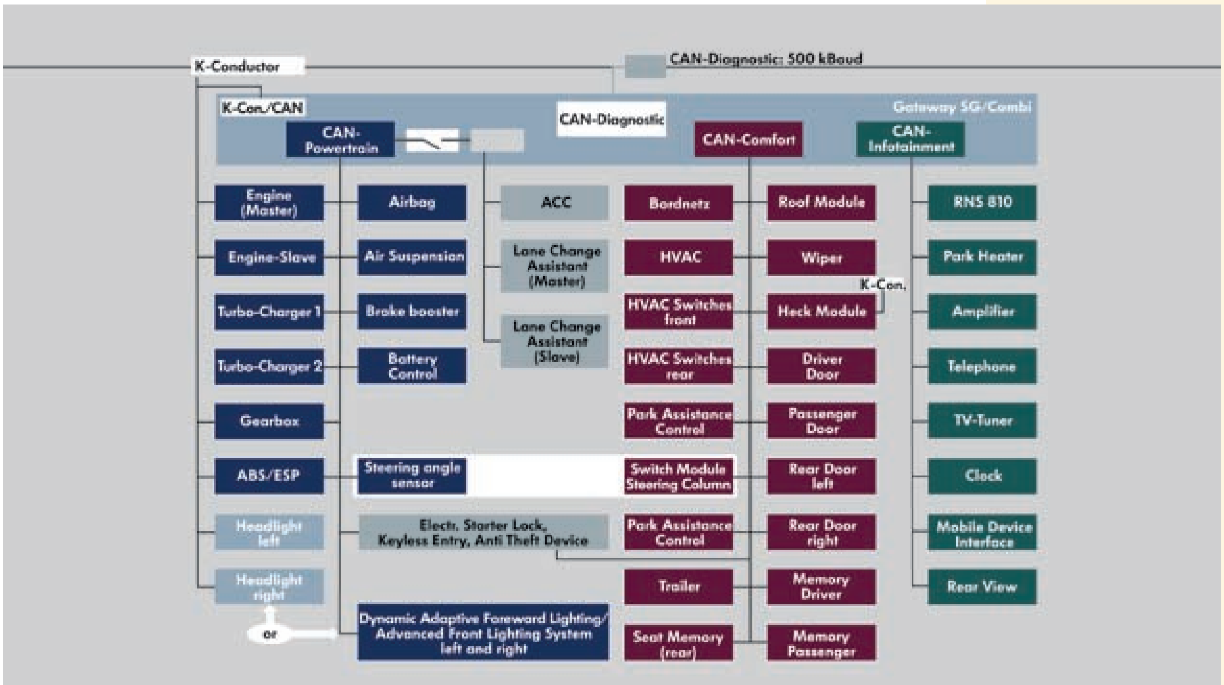


Figure 1: Topology of the revised infotainment and comfort network

Figure 2: The RNS810 radio/navigation system

instrument cluster with a 5-inch colour display that is specially adapted to the driver and a newly developed radio/navigation system installed in the centre console, Figure 2. The RNS810 makes use of the proven VW modular infotainment system and, for the first time in the Volkswagen brand, is being equipped with a touch-sensitive 8-inch colour TFT display. This means that the Phaeton is now also benefiting from the established VW operating philosophy via a touchscreen. The newly designed graphical user interface – which has been specially designed for touchscreen operation – is displayed with a high resolution (800x480 WVGA) to allow exceptionally clear representation of menus and maps. The high contrast and LED backlighting ensure clear images even under the most unfavourable lighting conditions. The combination of high-definition graphics and the 8-inch touchscreen display conveys an exceptionally high-quality appearance. The display is framed by a panel in piano finish and by chrome trim on switches and control elements, thus blending into the luxurious interior ambience of wood and leather.

The large screen presents the user with the many display functions in a clear manner, allowing simple operation of the integrated radio, audio, video and navigation functions as well as control of additional equipment such as Bluetooth telephone preparation, a TV tuner, a rearview camera and a CD changer. A particular highlight is the possibility to connect mobile equipment such as an “iPod” or “iPhone”, an mp3 player, an external hard drive and a USB stick via the Media-In interface, Figure 3. This allows even large music collections to be operated safely and conveniently while driving.

Further highlights include:

- integrated 30 GB hard drive for storing navigation and media data, partitioned into 10 GB for the navigation data and 20 GB reserved for media dataintegrated DVD drive and SD card reader
- player functions for MP3 and WMA audio data and audio/video DVD
- FM twin tuner with integrated Phase Diversity for optimum radio reception.

The navigation system calculates up to three alternative routes with route guidance via symbols and maps, optionally in 2D, 3D or topographical representation. In addition, point-to-point navigation is provided for off-road driving.

A telephone control unit with Bluetooth rSAP (remote SIM Access Profile) allows drivers to leave their mobile phone in their pocket. The telephone and telephone administration functions, such as the phone book, number entry, caller lists, etc., are conveniently performed via the interactive display of the RNS810.

A further high-level innovation is the twelve-channel high-end sound system, Figure 4. The system of twelve Dynaudio loudspeakers and a Blaupunkt amplifier – which has been specially tuned for the Phaeton, guarantees a high-quality experience.

The high-quality loudspeakers and their careful placement in the car ensure a balanced sound distribution throughout the vehicle interior. The high-end system has a continuous power output of 1,000 watts. This power is divided between the individual outputs, with the woofers accounting for the highest power consumption. The brand Dynaudio stands for the unmost in sound quality. Only the most lightweight materials are used, thus allowing the sound-generating surface area to be increased by three to four times without any increase in the weight of the loudspeakers, resulting in a corresponding increase in efficiency.

Figure 3: The „Media-in“ interface

In order to maintain the high-quality appearance of the Phaeton both during the day and at night, the lights on all the switches and buttons have been changed from red to white. The challenge for the VW engineers was to achieve a harmonious appearance in both colour and light intensity across all the groups of switches. The aim was to cancel out any lack of homogeneity caused by illuminating through different button and switch materials. Their expertise lay in harmonising the colour and light intensity classes of the white LEDs accordingly The results were verified by both light measurements and subjective assessments.

3 Modifications to the Powertrain and Chassis

All engines of the Phaeton already comply with the EU 5 standards from autumn 2008 onwards, even though Euro 5 does not become mandatory until 1 January 2011. The limit values for NO_x and particulate matter in particular have been significantly reduced.

A new addition to the engine range of the Phaeton is the V6 FSI gasoline engine with a displacement of 3.6 l and an output of 206 kW. This significant rise in output compared to the previous V6 engine (177 kW) is the result of the conversion to direct fuel injection. Torque has increased by 45 Nm to 370 Nm. Furthermore, a reduction in fuel consumption by 0.4 l/100 km (NEDC, combined) compared to the predecessor engine has been achieved, corresponding to a decrease in CO_2 emission by 15 g/km.

In the V6 TDI diesel engine with common rail fuel injection, which is greatly in demand particularly in Europe, torque has been raised by 50 Nm to 500 Nm, while output has been slightly increased to 176 kW, which noticeably enhances the car's agility. Modifications inside the engine have resulted in a further improvement in fuel economy, with the result that the engine has a fuel consumption of only 9.1 l/100 km (NEDC, combined; equivalent to 240 g/km CO_2).

For the model with the W12 high-performance engine, ceramic brakes will be available as an option from spring 2009. These offer an approximately 25 % higher friction coefficient and a more constant friction coefficient under load and high temperature. What is more, they contribute towards reducing the overall weight of the rotating masses.

4 Interior and Exterior

The interior has been revised and now has the option of the new colour "Corn Silk" for the leather trim. In addition, the decorative and console trim elements, which were previously in the colour "Anthracite", are now finished in the lighter colour of "Warm grey", which, together with the white switch lighting, further supports the pleasant driving experience. The customer can also choose from four new wood trims. The long-wheelbase version of the Phaeton is equipped as standard with two illuminated and swivelling vanity mirrors for the rear-seat passengers. The retractable mirrors are integrated into the roof lining, **Figure 5**.

The external appearance of the Phaeton has been modified, for example by three new wheel designs in the dimensions 17, 18 and 19 inches or the new radiator grille in chrome and gloss finish.

5 Further Developed Assistance Systems

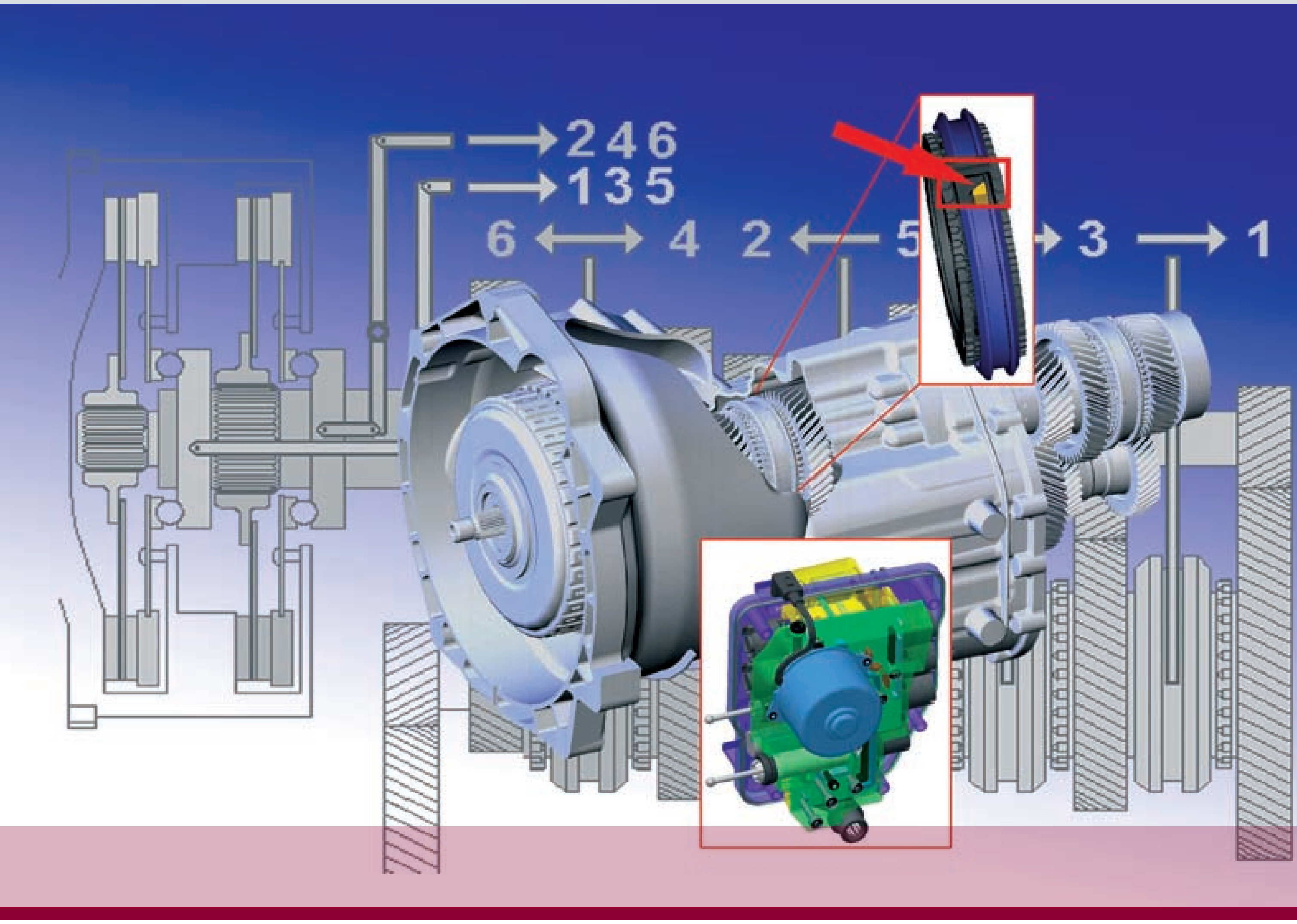
The Phaeton has also undergone further improvements in comfort and safety. These include the introduction of „Rear Assist", which has a rear-view camera that transmits its image directly to the infotainment system, and „Side Assist", a lane change assistant with warning lights in the outside mirrors, which, due to the reduction in the activation speed from 60 to 30 km/h and an increased range, is now even more practical.

6 Conclusion

The Phaeton is being continuously further developed as a technology carrier. It makes a major contribution towards positioning Volkswagen as an innovative high-volume manufacturer. ■

Figure 4: Dynaudio twelve-channel high-end sound system

Figure 5: Retractable vanity mirrors for the rear-set passengers



Servo-synchronizers in Dual Clutch Transmissions

Dual clutch transmissions provide a valuable contribution to the reduction of vehicles CO_2 emissions. Using servo-synchronizers can help to increase efficiency and to reduce weight without changes of interfaces and space. hofer powertrain has developed a new approach for reducing the energy for shift actuation. By using system simulation innovative tools were created to tap the full potential of the dual clutch transmissions.

1 Introduction

Since the launch of Volkswagen's DSG in 2003, dual clutch transmissions (DCTs) are in high volume mass production [3]. In the meantime, the second generation with dry clutches, seven and more gears and power pack contributes significantly to higher efficiency and thus to less CO_2 emission.

A further cost-effective and comparatively simple opportunity to reduce losses in the hydraulic supply system of DCTs provides the use of servo-synchronizers. The key component for that approach is a self-energising synchronizer, which can reduce the outside actuation energy for the shifting significantly. These systems use basically the same effect as self-energising brakes (e-brake) and self-energising clutches as they are used in all-wheel drive systems.

2 Functionality and Types of Servo-synchronizers

There are many patents and designs of servo-synchronizers. In general, all of them are using the friction torque via an active servo slope to get an additional force in the actuation direction adding to the actuation force, which is applied externally. Mostly, this is done by special thrust pieces.

Generally there is a difference between semi servo-synchronizers [4], which are using only a part of the synchronizing torque, and full servo-synchronizers, which are using the full torque and get more self-energising effect, **Figure 1**.

Several producers and suppliers have tried to develop servo synchronizers for serial production. The goal is to improve shift quality of manual transmissions (MTs) and to reduce the shifting energy in automated manual transmissions (AMTs) and DCTs. After hofer powertrain has verified the function and durability of semi servo-synchronizers in the past [4], now the even higher performance of full servo-synchronizers has been proven, Figure 1, with prototypes that have also passed durability tests, Figure 5. These systems have way better performance than systems available on the market. This break through has been achieved by using state-of-the-art simulation methodologies, **Figure 2**, in parallel to bench tests. With this systematic, efficiency potential and shift quality can be projected in early concept stage, which is accelerating the development process significantly.

3 Potentials of Full Servo-synchronizers

The hydraulic losses of DCTs with constant supply pump can be in the Kilowatt range and every reduction of the pressure level leads directly to an efficiency improvement. How big this influence is, is depending on the cycle, the

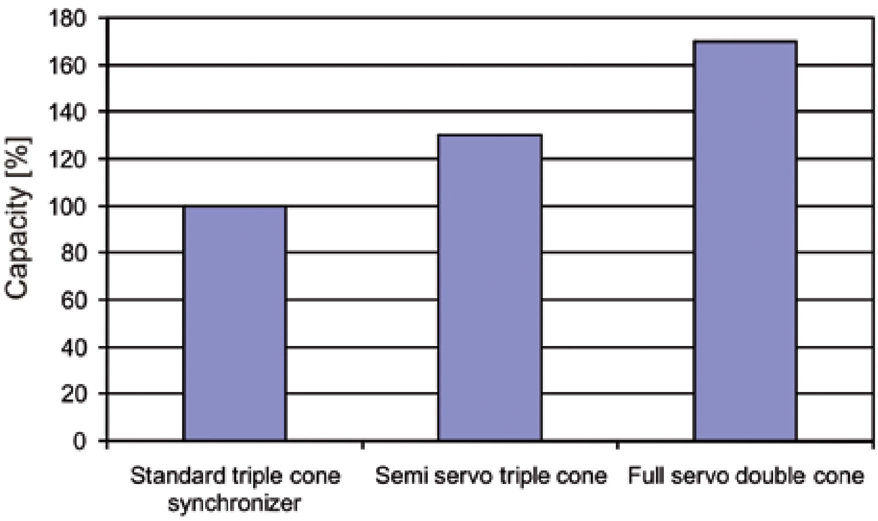


Figure 1: Comparison of synchronizer performance in Nm/N

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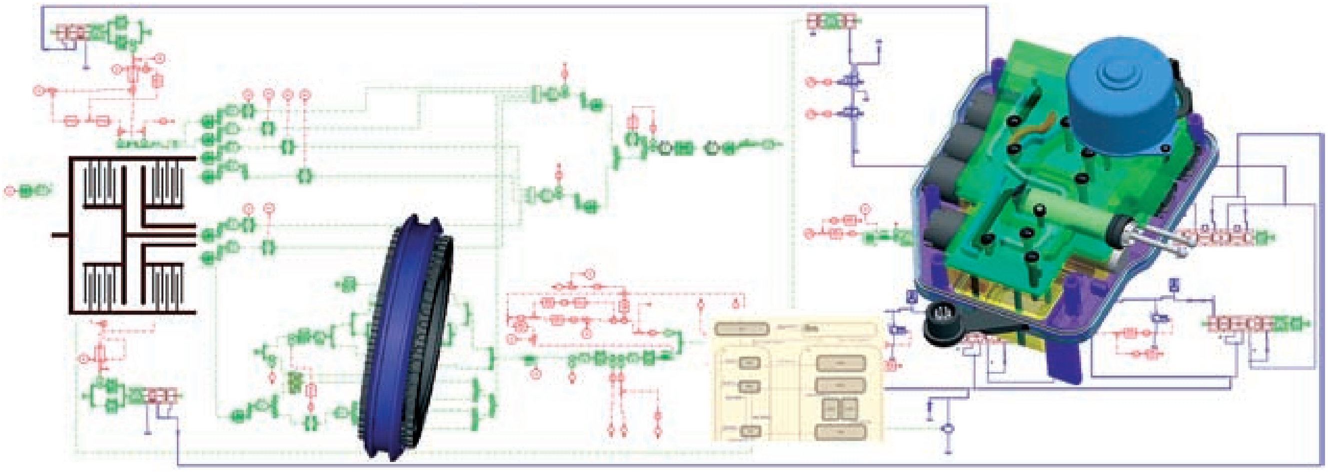

Figure 2: Simulation model of a dual clutch transmission

number of shifts and the leakage in the system. The maximum pressure level is defined by the necessary pressure for a shift and can be a multitude of the maximum pressure at the clutches. Of course, this reduction of pressure is only possible if the same or even better synchronizing times can be achieved (criterion double back shifts). This can only be achieved by a higher synchronizer capacity. The limit of conventional standard synchronizers is a triple cone type. Four-cone synchronizers have turned out to be no reasonable technical and economical solution.

That means that only a completely new synchronizer technology can disclose further potential. hofer powertrain has now found a full servo-synchronizer design, that can increase the synchronizer capacity by 70 % while reducing the number of cones from three to two. This means a big reduction of actuation force (minimum 40 % depending on detent forces, drag torque and shift speed). It also has a positive effect on the shift time Figure 3.

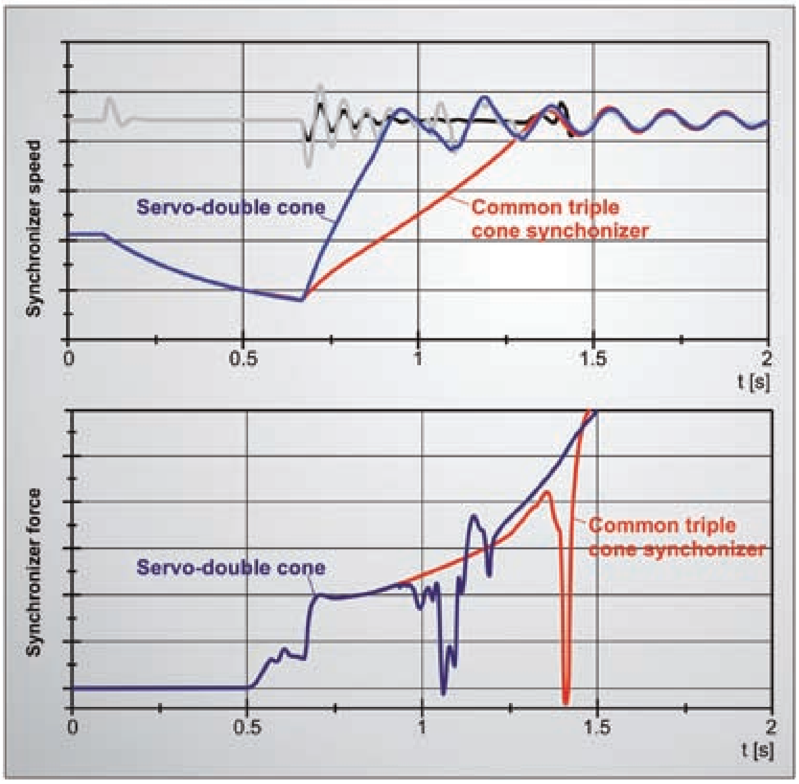

Figure 3: Relative comparison of simulation results between standard triple cone and full servo double cone synchronizer

Due to the lower forces the internal shifting system can be reduced in size and weight, Figure 4, and according to the reduced pressure level the leakage of the hydraulic system is reduced. The system can also be used to realise extremely quick shifts with the pressure kept constant. This might be interesting in high performance applications. Certainly, when using the system for shortening of shift time, the durability of the friction materials has to be considered.

A look at the market shows that DCTs currently in series production are using three or more triple-cone synchronizers. That means that the potential for servo-synchronizers has to be set at the same number minimum per transmission. A market study of actual and coming DCTs has shown that eight of twelve transmissions can reach high efficiency improvements by using full-servo-synchronizers. Regarding shift quality one can see the same for approximately 24 of 36 MTs, which are in the market or under construction.

As advantages of full-servo-synchronizers the following items can be mentioned:

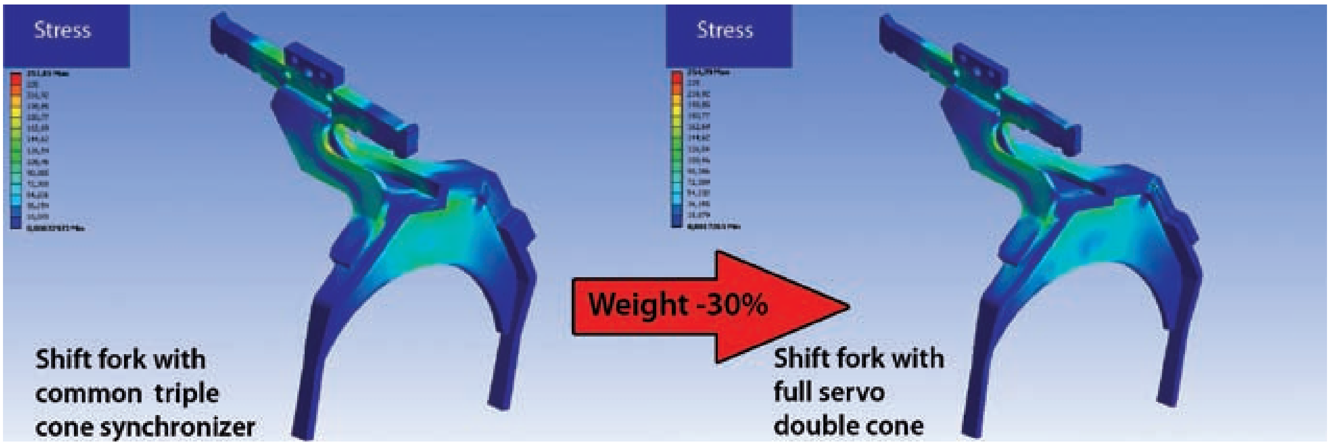

Figure 4: Weight reduction in a dual clutch transmission with full servo synchronizers, for example of a shift fork

- better shift times with same shift force
- reduced shift forces with same shift time, which lead to further advantages: lower hydraulic pressure and less leakage, weight reduction of internal shift system and hydraulic components due to lower forces and pressure, efficiency advantage due to less actuation energy, less shift noise
- reduction of cones (for example double cone replaces triple cone) and thus drag torque reduction and cost advantage
- in systems with hydraulic power packs, the piston area can be reduced at same pressure level to reduce pump and accumulator size and/or actuation time
- less current demand and smaller electric motors in DCTs with electromechanical actuation.

Depending on the actuation system, there is a combination of these advantages for every transmission. This can be projected in advance with the help of simulation. Since the interfaces of a full-servo-synchronizer are not any different to a common synchronizer, the application in serial production is possible. In DCTs and AMTs software adaption is only necessary.

4 Design Criteria

For the integration of servo synchronizers in DCTs, there are two necessary steps. To get the optimum out of the system, the first step is to identify the synchronizer positions ideal for servo synchronizers. The second step is the design of the servo synchronizer itself.

To identify the best positions for servo synchronizers, the main criteria to consider are performance and costs. For the design of the servo synchronizer itself, there are many criteria to consider. Friction material durability safety factors [1], synchronizer capacity, self lock safety [4] and unlock safety at low temperatures.

The choice of friction materials is an important point. For every tribologic system (friction material and counter material, oil, geometry and environmental parameters) there are design limits regarding specific pressure, relative speed, specific heat, specific performance and (for glued materials) shear stress to be considered.

If only the shift forces are reduced to achieve the same shift performance, the same friction materials can be used without danger also for servo synchronizers. If the shift performance is increased and the loads on the synchronizer materials increased, the design limits have to be checked. Durability tests at Miba Sinter Austria GmbH with shorter shift time using full servo synchronizers have proven full functionality during 100,000 shifts, **Figure 5**. Regarding lubrication there is

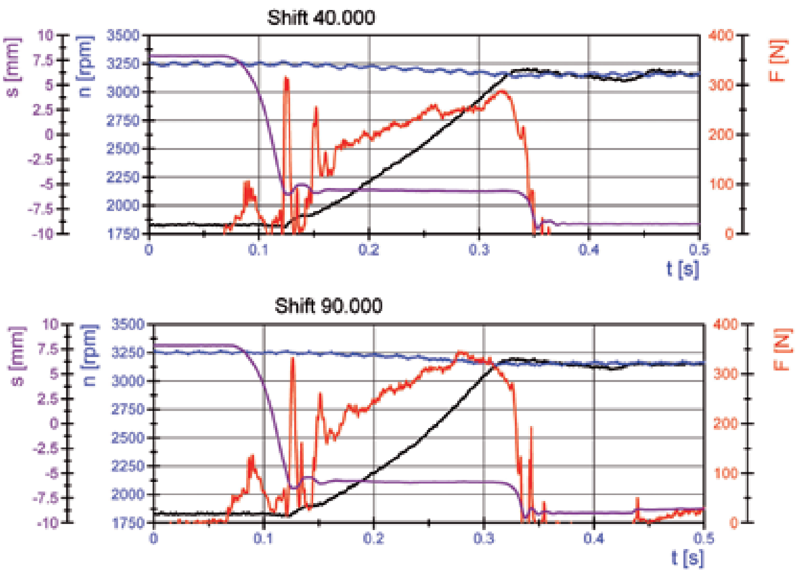

Figure 5: Measurements during a durability test of friction materials of full servo synchronizers with shortened shifting time (source: Miba Sinter Austria GmbH)

no additional topic to consider with servo synchronizers.

5 Development Systematics for New Transmissions

Since DCTs are very complex mechatronic systems, hofer powertrain uses entire system simulation tools starting from the specification. These simulation tools are fine tuned during the development process. The entire system simulation tools comprise the mechanical components of the transmission like clutches, synchronizers and gears, the driveline and an engine model, if needed hybrid components, side shafts, tyre-street contact, vehicle longitudinal dynamics model, the hydraulic control, the actuation system and the logic algorithms including, for example shift strategies, Figure 3.

These simulation models deliver quick answers in advance to verify functional features in an very early stage. hofer powertrain has fine tuned the modelling by constant calibration with measurements. So by using special macros, transient dynamic phenomena can be simulated during the shift process in detail. This enables us to optimize the geometry of a servo synchronizer without hardware.

The models are used in reduced complexity for virtual model-based software development. The hardware application in the vehicle is reduced significantly due to this virtual testing. This reduces precious development time with hardware prototypes.

6 Impact on Design Space and Transmission Weight

The design space and the interfaces do not change using a full-servo synchronizer compared to a standard synchronizer. But the weight of the internal shifting system of the transmission can be reduced significantly because of lower shift forces. For example, a reduction of shift forces by 40 % leads to about 30 % weight reduction at the shift fork without durability drawbacks and using the same material and manufacturing processes, Figure 4.

7 Conclusion and Outlook

New full-servo synchronizers from hofer powertrain have proven by simulation and testing, that the shift forces can be reduced by 40 % compared to standard synchronizers, while the life cycle requirements can be satisfied furthermore. Thus, the big advantage is proofed compared to available systems on the market.

Nevertheless there is still further potential in the area of synchronizers especially when considering the increasing performance of electronics and sensorics in the transmission. The mechanical blocking function of servo synchronizers could be replaced by an electronically controlled synchronizer completely. Self energising cone clutches could lead to the same advantages in automatic transmissions. This means an ongoing race between the different transmission concepts.

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Specialty Lubricants for Proper Door Module Function

Car doors contain a great variety of electro-mechanical and hydraulic components for opening and closing and a number of comfort and safety functions. All these components make new, taxing demands on special lubricants. Taking the example of locks, window regulators and bowden cables, this article by Klüber Lubrication illustrates how the design element lubricant contributes to improving the function, reducing energy consumption and increasing the driving comfort.

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1 Introduction

According to the materials used for the opposing bodies, the operating temperature, the ambient media and other OEM-specific requirements on the lubricant, a distinction can be made. Some of these parameters are:

– the lubricant has to be compatible with different plastics like PA, POM, ABS, PBT and PP, as well as with elastomers like NBR, EPDM and various lacquers
– it has to display excellent friction characteristics at low temperatures and offer lubricity over a wide service temperature range from -40 °C to +90 °C
– it should be resistant to water (rain and condensation), cleaning and de-icing agents and offer good corrosion protection (SKF EMCOR 0/1)
– good wetting and flow characteristics at temperatures as low as -40 °C are of major importance
– additional requirements are good noise dampening, the possibility of thin-film lubrication, no odor emission, no fogging as defined in DIN 75201 A and, of course, lifetime lubrication.

2 Side Door Locks: Higher Actuating Torques at Low Temperatures

A leading German OEM experienced a considerable increase of the release torques of the door catch bolts and locking pawls at temperatures between -20 °C and -40 °C. Starting and running torques of different lubricants can be compared on the IP 186 low-temperature test rig. In these tests, fully synthetic products of low viscosity show much lower release torques than semi-synthetic or mineral oil based lubricants with a higher viscosity. Due to the good test results achieved on the IP 186, a fully synthetic and low-viscous lubricating grease was validated for series application on side door locks. Field tests performed with the component show consistently lower actuating torques than a semi-synthetic or mineral oil based product in a wide operating temperature range from -40 °C to +90 °C.

Relative movements in the mechanism of side door locks generated considerable creaking noise in the full-vehicle test. Therefore, 36 locks from series applications were examined on a special lock test rig, **Figure 1**. A linear drive generated wobbling in order to simulate the

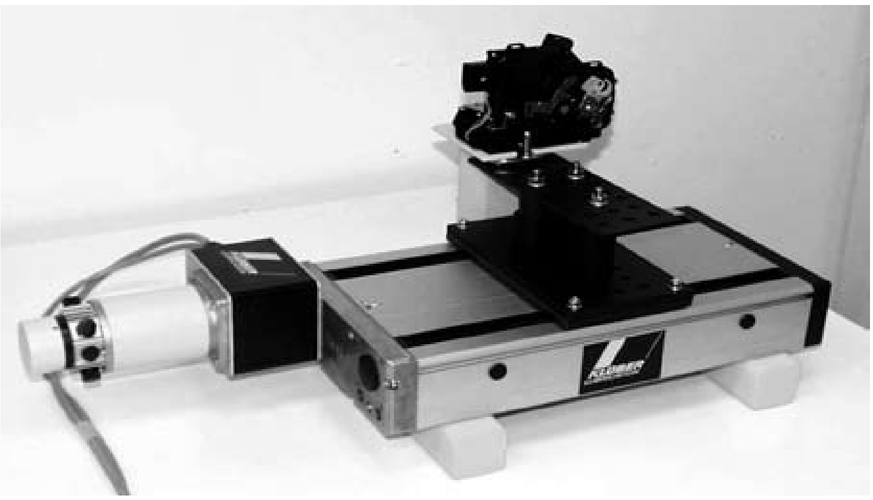

Figure 1: Test rig for creaking noise examinations

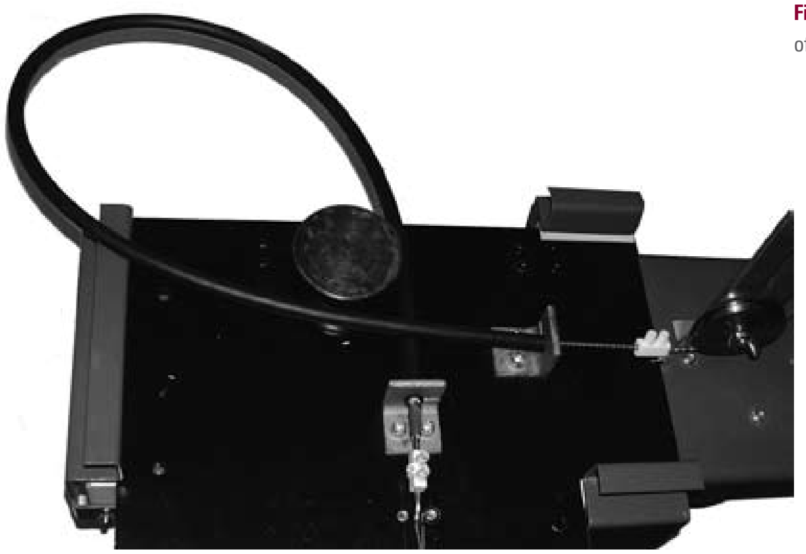

Figure 2: A test rig for bowden cables to simulate the influence of the lubricant on the operating forces at an angle of 270°

vertical impact of the vehicle in motion. 40,000 load alternations were performed – one opening and one closing is one load alternation.

Side door locks made by various manufacturers were tested. These side door locks were taken from the series application on a single platform and tested with different consistent lubricants as well as with bonded coatings. The results show that squeaking noises can only be generated when applying SAE J726 specified Arizona dust and occur at the contact points of the striker. A special lubricant, able to prevent this type of noise, was specified by a car manufacturer for these friction points and prescribed by his component supplier for series application.

3 Central Locking Systems and Small Gear Motors

The worm gears of a series window lifter suffered repeated backlash because friction was too low. The same phenomenon was observed in central locking systems. The obvious thing to do was to increase the self-locking of the gear. The locking of a worm gear can, among others, be influenced by the lubricant used. A suitable test rig for examining the locking effect of lubricants is the Tannert sliding indicator. On the Tannert sliding indicator two sliding surfaces are subjected to an oscillating movement and the load is increased continuously. The Tannert sliding indicator allows determination of sliding and stick-slip behavior of various lubricants at low sliding speeds.

For a mineral oil based lubricant, for example, the Tannert test rig shows an increase in the friction value with increasing load. A fully synthetic lubricant, however, gives a constantly low friction value. Therefore, in many gears of window regulator systems and car door modules requiring additional self-locking, special lubricants based on a mineral oil mixture are used.

4 Inside Door Handle and Bowden Cable

Besides the inside door handle bearing, the type of connection between the handle and the door lock, for example a bowden cable, determine the operating forces required. The ambient temperature and the laying of the bowden cable have a particularly decisive influence. With bowden cable laid in the side door at an angle of 90°, the operating force is much lower than with bowden cables laid at an angle of 270°. At temperatures below -10 °C the operating forces at the inside door handle increase considerably. A test rig was built for bowden cables in order to simulate the influence of the lubricant on the operating forces at an angle of 270°, **Figure 2**. Several test runs were performed with different lubricants based on synthetic hydrocarbons and silicone. The difference between the fluid (oils) and consistent lubricants of medium and high consistency and the fluid greases was considerable. Owing to the good results achieved at low temperatures a special silicone lubricant was chosen for bowden cables laid at an angle of 270°. For other applications with a laying angle of just 90° and at -20 °C to + 23 °C, the use of a lubricant based on synthetic hydrocarbons showed the best tests results.

5 Summary

The lubricant makes a decisive contribution to the proper function of door module components – if given due consideration at an early stage of design. Examples from field tests show which mechano-dynamical model test rigs can help in determining suitable lubricants for side door locks, window lifter motors and bowden cables. ■

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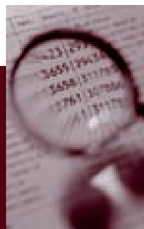

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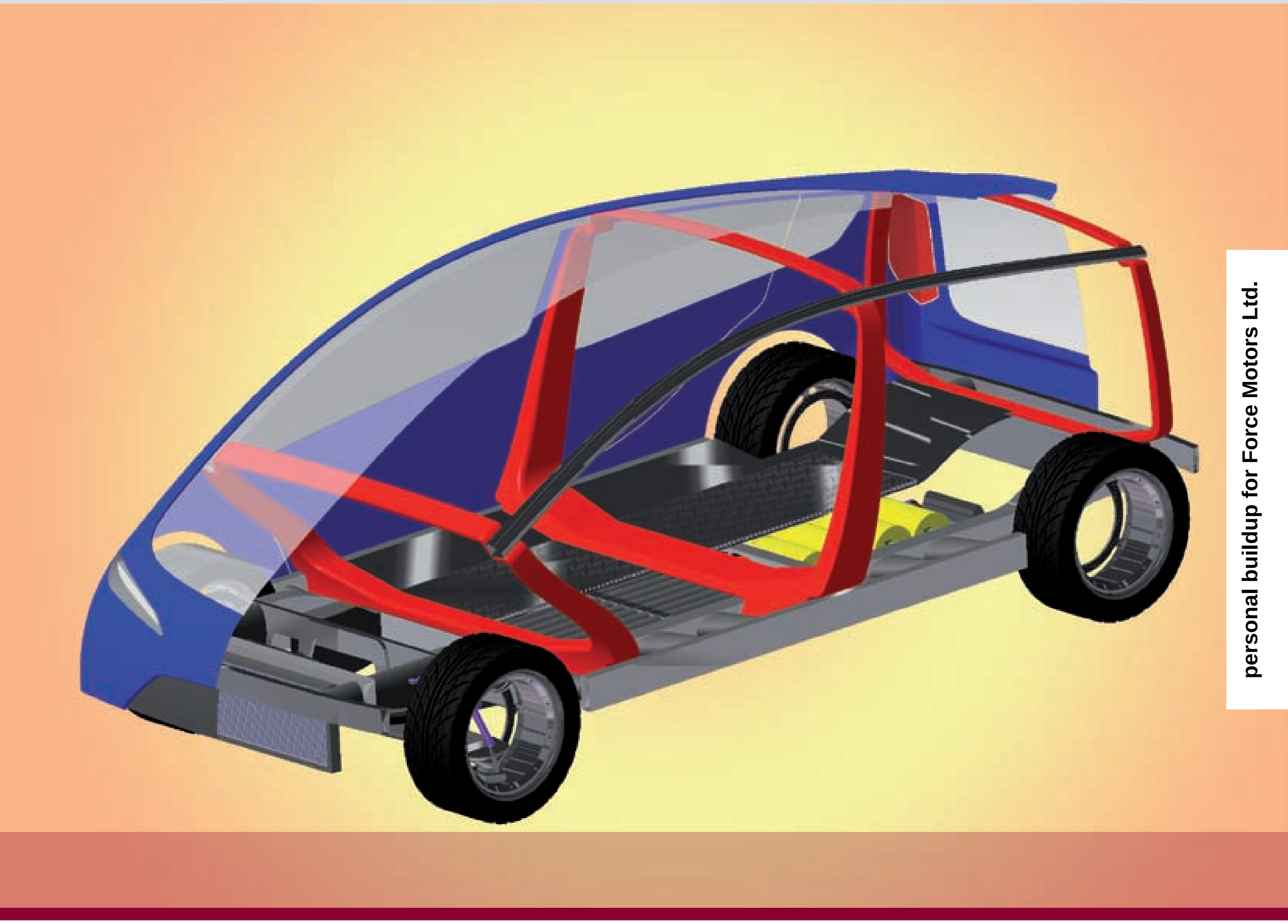

Innovative Vehicle Structure Using Rib and Space Frame Construction

The DLR in Stuttgart (Germany) is currently pursuing research under the title of "Novel Vehicle Structures" with the aim of developing concepts and structures for the vehicles of the future. The project described in this paper is concerned with the development of an innovative "rib and space frame construction method". The focus of this work is to achieve groundbreaking improvements in areas such as weight reduction, safety and modularisation strategies.

1 Introduction

The progressive growth of the world's population, increasing globalisation and the need to conserve resources in response to climate change are just some of the factors that will have a significant effect on future mobility. In this respect it is imperative that individual mobility in the years to come is environmentally sustainable, does not require sacrifices and can find the necessary finance. When developing forward-looking vehicle concepts, it is therefore necessary to consider requirements that go far beyond those that have been placed on vehicles to date [1, 2, 3].

Now, in addition to safety, comfort and driving performance, other aspects such as ecology and economy are coming to the fore. The aim is to contribute to the reduction of fuel consumption and environmentally detrimental CO_2 emissions by decreasing driving resistance [2]. Similar to various research projects (for example SuperLightCar) the potential for lightweight vehicle construction offered by the research project described here will initially be concentrated in the body of the vehicle, because this area typically makes up 25 % of the total vehicle weight.

Future drive technologies will be either partially or fully electric, with some also making use of other energy sources. The vehicle development programme is therefore also concerned with drive concepts such as fuel cells and hydrogen drives. This must be taken into account in the structural design. Especially important is that the necessary tank and storage modules must be integrated into the vehicle in such a way that they meet with packaging requirements and cannot intrude into the passenger cell during a crash.

A further challenge for future vehicle concepts is the trend towards individual vehicles, which means a reduction in the volumes produced for each model variant [4, 5]. An additional requirement was therefore defined for the project to ensure that solutions can be modularised and that there is the possibility of creating derivatives. Previously, these strategies have been implemented primarily in the areas of interior fittings and engines, and in the creation of modules for the assembly process. By contrast, for body and structural components the focus is on developing concepts for new, integrated solutions.

2 Vehicle Concept

Taking an upper mid-range vehicle as a starting point, the rib and space frame concept developed at the Institute of Vehicle Concepts has been designed in such a way that alternative drive concepts can be integrated safely and easily. The location of the drive units in the floor area is designed-in right from the concept phase, **Figure 1**. This is because new drive concepts, such as the free piston linear generator (FPLG) [10] developed at the DLR, allow packaging considerations to

Figure 1: DLR vehicle concept with intrusion resistant containment

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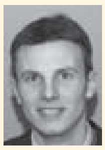

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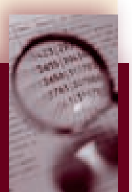

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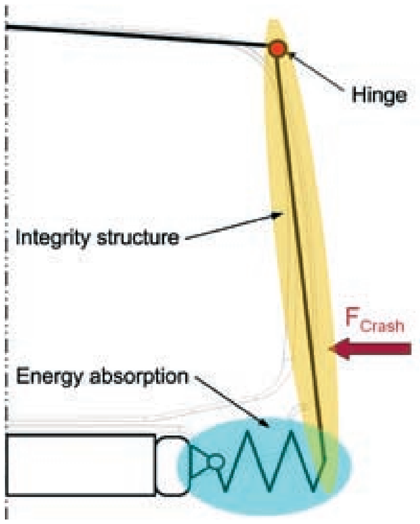

Figure 2: Mechanical principle used for the rib (schematically)

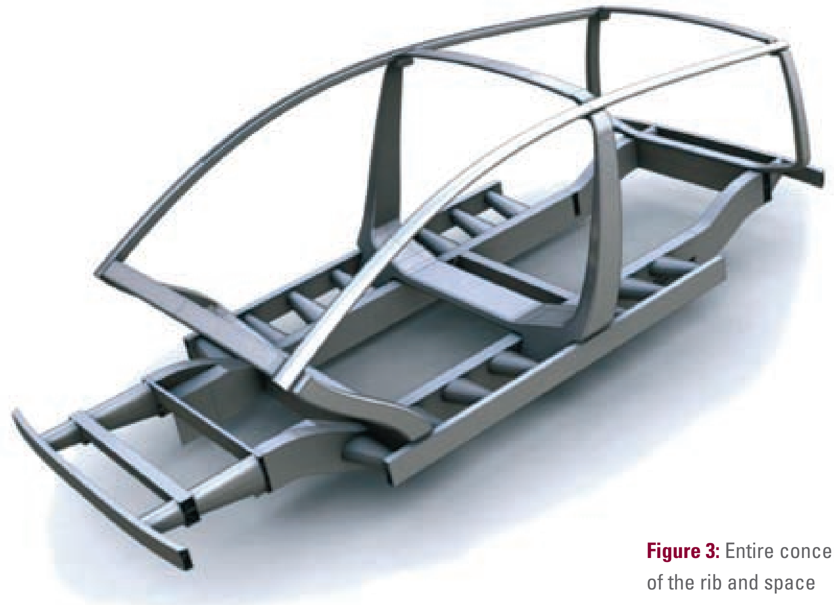

Figure 3: Entire concept of the rib and space frame construction

be optimised for weight and encourage the use of modular containment solutions. This has the advantage that alternative energy storage options such as compressed natural gas (CNG) or hydrogen can be located in an intrusion-resistant area, allowing the centre of gravity to be lowered and the axle load distribution to be optimised.

These requirements do, nonetheless, result in specific characteristics that have to be implemented as part of the body concept. The most significant load cases for the design of the floor assembly are those that emerge from both the side and pole impact test procedures. The components of the drive unit must not be forced against an incompressible object, as this would cause the acceleration values for the occupants to increase too sharply.

A construction was therefore chosen for the bottom of the car body that forms an outer zone to act as a "crash compartment". This zone extends from the sidewall to the continuous side members, which are used for their suitability for lightweight construction. By exploiting a mechanical principle and utilising energy absorbing CFRP crash cones (see also [6]), the zone absorbs the required energy for the IIHS side impact test, while also ensuring a defined level of intrusion, **Figure 2**. The front end of the "crash compartment" forms a cross-brace in the area around the A-pillar. This concept allows the complete front end to be designed to meet the requirements of a frontal impact. In the rear of the vehicle, the bracing is adapted to meet the respective packaging requirements.

The "base frame" for the space frame is formed from three ring structures (ribs) at the A-, B- and C-pillar positions, which function also as node elements that support the variable profiles used for the rocker panels, side members and roof beam, as well as distributing the loads that occur in the various load planes. This arrangement offers the advantage that, by scaling the lengths of the profiles, it is possible to derive different variants from a single base structure.

The basic construction principle used for the vehicle can be retained. The structure of the vehicle behind the B-rib can be modified, for example shown in **Figure 3**, to meet individual requirements for the design of the rear end.

Taking this fundamental concept as a basis, the DLR hopes to be able to make a significant contribution towards improvements in quality of lightweight construction and passive safety. As well, it likes to present an innovative means to implement modularity in structure-bearing components. Participants in the project are: the DLR Institute of Structures and Design in Stuttgart (Germany) and the DLR Institute of Composite Structures and Adaptive Systems in Brunswick (Germany), besides the DLR Institute of Vehicle Concepts (DLR FK) in Stuttgart (Germany). The DLR FK is also working in cooperation with external companies, such as ACE GmbH and ACTS GmbH.

Using the rib and space frame construction leads to a reduction in complexity and offers a considerable cost advantage. However, the lightweight construction necessary to achieve a reduction in CO_2 presents a particular challenge, because any increase in weight must be compensated for by the possibilities offered by modularisation and improved passive safety.

3 Key Component Hoop Rib

The hoop rib component lies at the heart of the path developing the complete rib and space frame construction method. As the centrepiece of this novel construction method, the rib demonstrates increased complexity in terms of design and dimensioning. The B-rib, which replaces the B-pillar of a conventional car, has been chosen as a representative example. The multi-material design combines materials and construction methods in such a way that the resulting lightweight construction is optimised in terms of economy for a particular production volume.

The ribs used in the DLR concept described here employ a fibre composite construction method. At 25–50 Euros/kg, the CFRP (carbon reinforced composite material) is certainly expensive; however it possesses the necessary mechanical properties in terms of stiffness, energy absorption and weight.

Using a relatively small amount of CFRP allows components and structures to be optimised according to local requirements and to achieve considerably better results in side impact crash tests than would be possible with conventional materials. This high-performance material is capable of absorbing energy inputs of up to 60 kJ/kg during a side impact, with the minimum amount of travel in the shortest time, **Figure 4**.

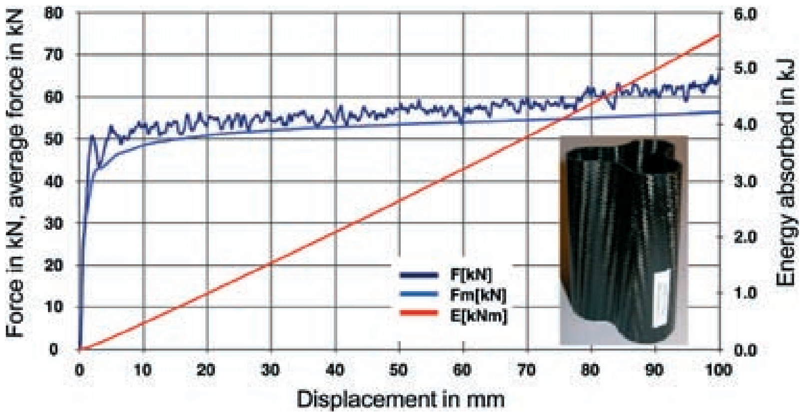

Figure 4: Example of a crash cone made of CFRP (bottom right) with resultant force-displacement curve

The material notwithstanding, the closed hoop structure of the ribs means they are predestined to accommodate the radial loads caused by lateral and roof impacts. The rib construction therefore represents an innovative approach that shows a great deal of promise in providing effective side impact protection.

The connection between the individual CFRP ribs can be implemented using metal profiles with simple geometric shapes. It is precisely this combination of expensive high-performance components in relevant structural areas and cost-effective materials and manufacturing techniques elsewhere that can make a contribution to a safe, lightweight solution at a reasonable cost.

4 Development Methodology for the Rib Construction

The development methodology used covers every stage of the process, from topology optimisation, design, and statistical and dynamic analysis, through to the construction of a prototype, which was tested in trials and serves to validate the results of the simulation process. During the early concept phase, analytical evaluations were conducted in respect of the side impact crash requirements. The result of these deliberations is the use of the mechanical principle illustrated in Figure 2.

When a side impact occurs, the rib in the vicinity of the roof beam acts as a deformable joint. By contrast, the area between the roof beam and the rocker panel is designed to be extremely stiff, so as to provide the best possible protection to the occupants in the head and shoulder region.

The resulting rotation about the deformable joint means that the crash energy is transferred to the crash elements in the floor region. These are located in the area between the rocker panel and the continuous side member and absorb energy through a crushing action.

This mechanical principle is particularly useful in that it allows the engineer to effectively evaluate the energy absorption in the floor, the maximum permissible intrusion and the acceleration of the occupants. The specific aim is to guarantee that there is little or no intrusion during a Euro NCAP side impact test, as well as affording passengers the best possible protection in the new IIHS tests [7, 11].

5 Optimising the Topology of the Rib Cross-section

Finite element analyses (FEA) were employed at an early point in the development phase in order to facilitate the design of the rib geometry and cross-section, **Figure 5**. The calculated optimal cross-section was determined on an abstracted section of a B-pillar with the aid of the Tosca optimisation tool. This result takes the form of a finely-fanned diagonal rib structure, which for stabili-

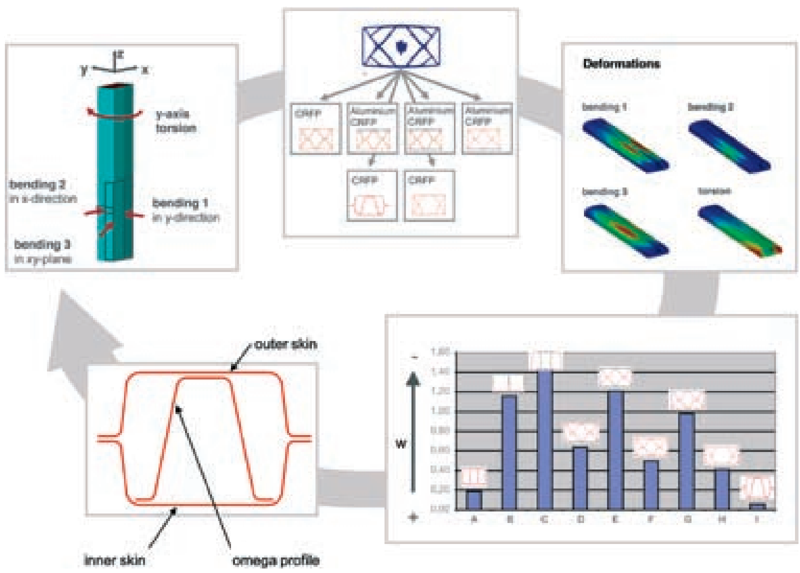

Figure 5: Topology optimisation and flexural strength calculation of the B-pillar structure

ty reasons extends from the corner regions to the centre of the opposite belts.

However, the cost and effort involved in manufacturing and joining this cross-sectional shape with fibre composite construction methods would be prohibitive, so several other types of simplified cross-section were designed and constructed with a view to better manufacturability. The cross-sectional designs that were developed were investigated for flexural strength by means of static analysis and stability calculations. The evaluation took account of a characteristic values w, determined in relation to a reference cross section, which was used to adjust for the relationship between the material utilisation factor and the mass. Two distinct cases were considered: for isotropic materials, the utilisation factor of the material is the ratio of the maximum stress that occurs to the maximum strength of the material; by contrast, for fibre composites the utilisation factor corresponds to the first ply failure index f [9]. The Tsai-Hill criterion was used to calculate the first-ply-failure index. The First-ply-failure index indicates the ratio of the occurring stress in a ply to the materials strength for a given load case. It is therefore the equivalent to the utilisation factor, used for isotropic materials. For isotropic material the Eq. (1) is used:

$$w = \frac{\frac{\sigma}{\sigma_{max}}}{\frac{\sigma_{Ref}}{\sigma_{max,\,Ref}}} \bullet \frac{m}{m_{Ref}} \qquad \text{Eq. (1)}$$

For fibre composites the Eq. (2) is used:

$$w = \frac{f}{\frac{\sigma_{Ref}}{\sigma_{max,\,Ref}}} \bullet \frac{m}{m_{Ref}} \qquad \text{Eq. (2)}$$

The formula signs are described in the **Table**. The cross-section "I", shown in Figure 5 (right, bottom), which has a monolithic construction and is made from CFRP, emerged as the best solution as a result of the investigation. This was therefore examined with regard to its stability, whereby the critical load case was the buckling of the top surfaces. This can be, however, be prevented by using a structural foam. This means that the rib consists of three components: inner skin; outer skin; and an omega profile that serves to stiffen the cross-section, Figure 5 (left, bottom). The three components of the rib are bonded together with adhesive at the contact surfaces of the half-shells.

Table: Meaning of formula signs

Meaning	Formula
First ply failure index	f
Material utilisation factor	$\frac{\sigma}{\sigma_{max}}$
Material utilisation factor for a reference cross-section	$\frac{\sigma_{Ref}}{\sigma_{max,\,Ref}}$
Mass, reference mass	m, $m_{Re\ f}$

6 Selected Results from the Dynamic Simulation and Validation

In order to minimise the effort and expense required to implement a prototype, while still creating a realistic structure for the investigation, a numerical vehicle replacement model was produced and compared to the side impact behaviour of the reference structure. The plane load of the side crash barrier was realized by using simplified doors in the simulation as well as in the prototype structure. The output model for the dimensioning of the rib is represented by a rigid connection between the centre of the rib and the outside of the doors, with the additional definition of discrete mass points in the centre of the vehicle to achieve the crash weight of 1250 kg, **Figure 6**.

In order to identify weak points, each CFRP component was initially assigned a constant, but low, wall thickness, so that failure zones occurred in the dynamic simulation with LS Dyna. Based on these results, the structure was improved over several iterations, in order to exploit the advantage offered by fibre composite construction of easily implementing variable wall thickness.

Trials were carried out on physical parts in order to validate these simulation results. To this end, the simulated structure was built at a 1:1 scale and impacted with a barrier of the type used for Euro NCAP test at a speed of 50 km/h. The rib structure was fitted with acceleration sensors in order to obtain better values for comparison than those provided by

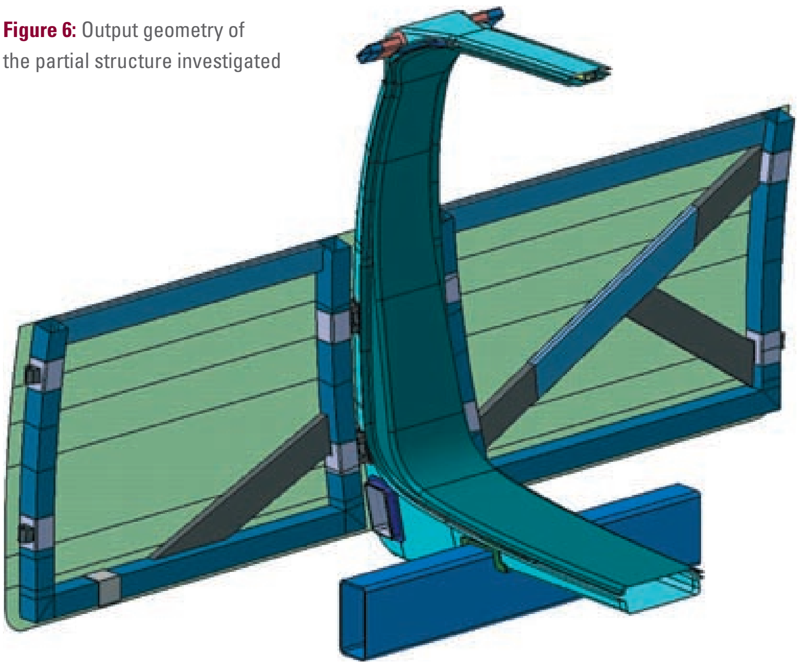

Figure 6: Output geometry of the partial structure investigated

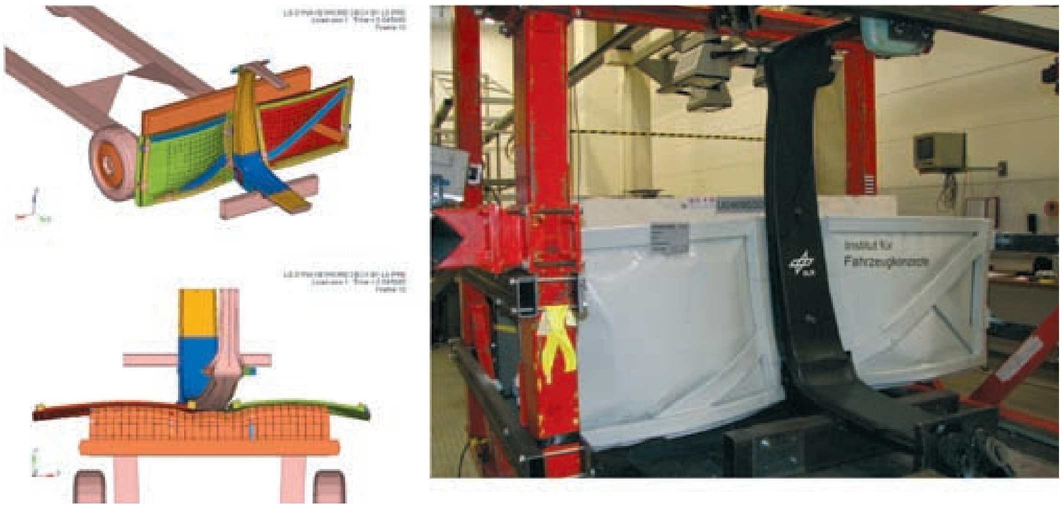

Figure 7: Comparison between the CFRP simulation (left) and a physical crash trial (right)

just the deformation of the barrier and the intrusion of the doors. **Figure 7** shows a comparison between the simulation and the physical trial.

Comparison of the acceleration sensors demonstrated that the FE model provides a good prediction of the experimental test curves, even without validation measures. Using the simulation results, however, it was established that the FE model had a greater stiffness than in reality. To date, the following major changes have been made to the FE model for the validation process, which is not yet fully complete.

- decoupling of the rigid connection between the lock panel and the door frame
- definition of the failure points for the hinge joints on the frame side
- improved modelling of the beading and reinforcement plate along the outside of the door frames
- flexible connection of the upper rib through modelling the frame carrier.

The acceleration curve demonstrated an oscillatory behaviour and was predominant in the upper region due to the flexibility modelled for the frame carrier. Furthermore, reducing the stiffness in the doors also improved the values obtained in the centre of the vehicle, **Figure 8**.

7 Results and Outlook

An innovative vehicle structure concept was developed using a geometric design, a rib structure, crash cones, an innovative door attachment and specially selected materials. This concept promises higher levels of safety for vehicle occupants, lower weight and more cost-effective options for modularisation than can be achieved with conventional construction methods. The first key component to be developed was the rib which forms the centrepiece of the rib and space frame construction. This was completed within 9 months and boasts a weight reduction of up to 50 % compared to a steel reference structure [8]. Physical testing, which followed the concept phase, the successful implementation of the structural design and the manufacturing of a prototype, demonstrated that during the trial implementation in line with Euro NCAP no component failure occurred and would therefore also be able to address the IIHS requirements with higher speed and barrier weight [7].

Areas that have not yet been investigated, but show great potential for further optimisation, include, for example, the use of self-adhesive foam for improved transmission of shear stresses through the rib structure. The material parameters for the FEM simulation also need to be optimised. In the case of the

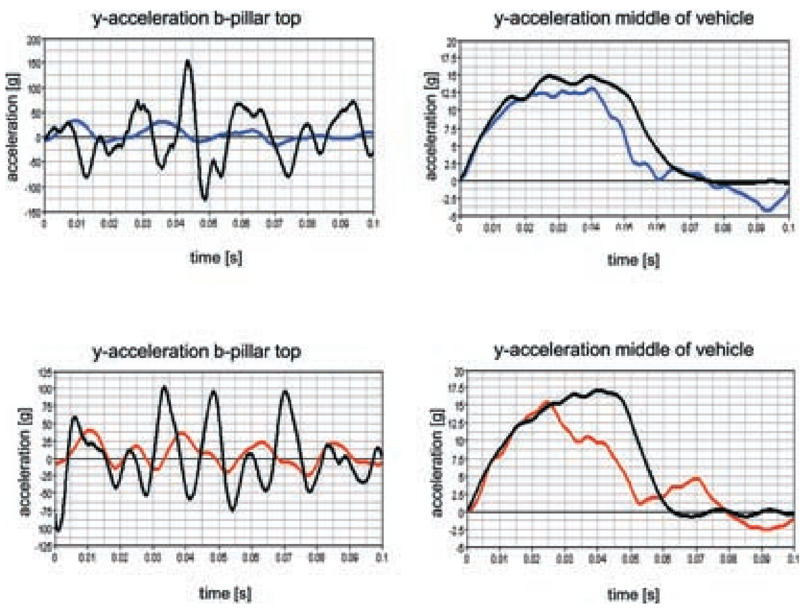

Figure 8: Acceleration values: blue line = test 1, red line = test 2, black line = FEM (with validation measures for two test procedures)

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joints, this is being done by means of additional trials on generic samples.

A significant phase in the future development will be the integration of the rib into the space frame environment. It is also planned to implement and test this complete theoretical model as a prototype, with the goal of achieving a weight reduction of between 20 and 25 %, whilst also improving safety and various other factors. It is hoped that the possibilities in terms of creating derivatives and functional containment solutions for alternative drives offered by the concept described will set a trend for low-emission mobility in the future.

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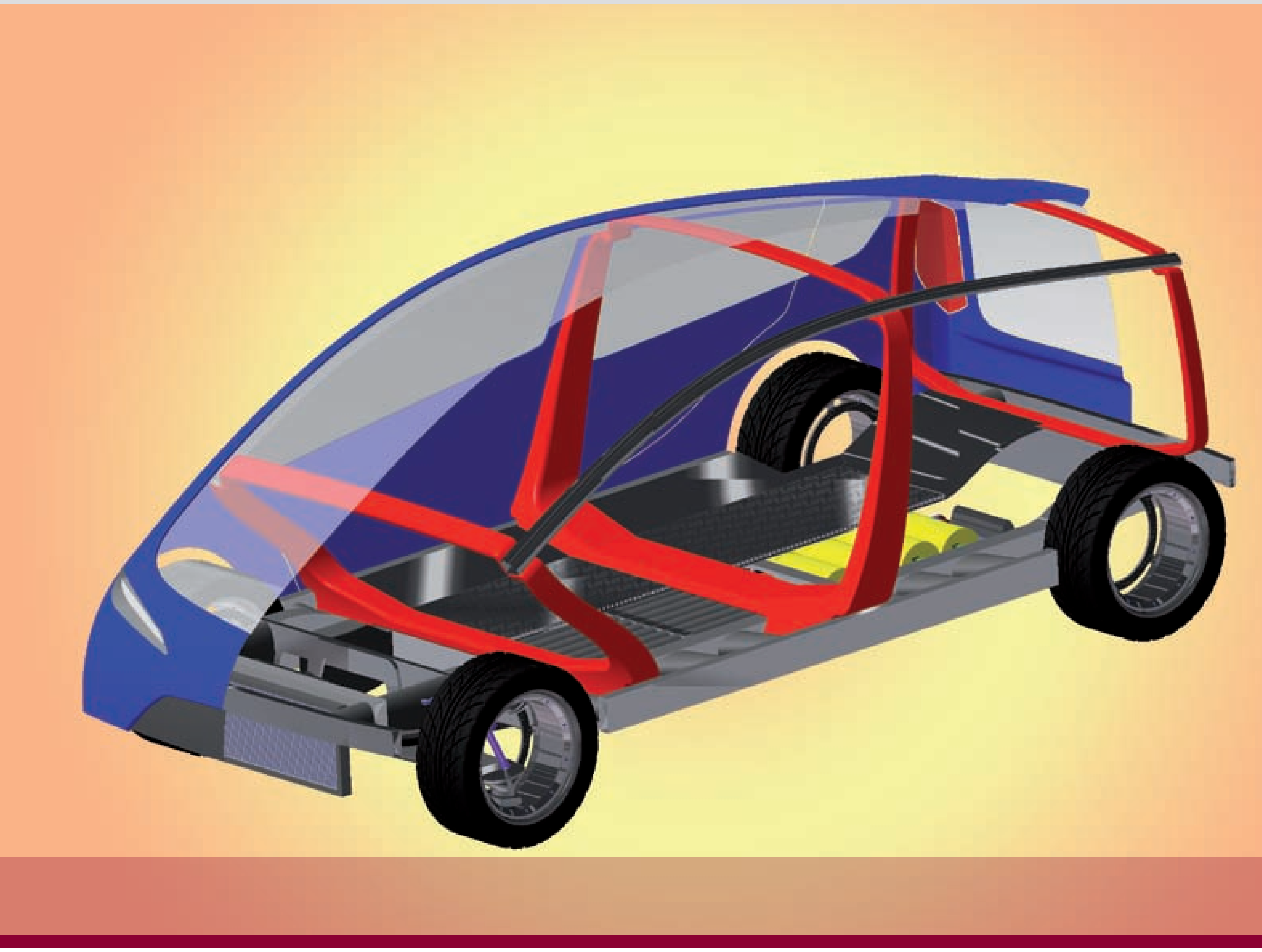

Innovative Vehicle Structure Using Rib and Space Frame Construction

The DLR in Stuttgart (Germany) is currently pursuing research under the title of "Novel Vehicle Structures" with the aim of developing concepts and structures for the vehicles of the future. The project described in this paper is concerned with the development of an innovative "rib and space frame construction method". The focus of this work is to achieve groundbreaking improvements in areas such as weight reduction, safety and modularisation strategies.

1 Introduction

The progressive growth of the world's population, increasing globalisation and the need to conserve resources in response to climate change are just some of the factors that will have a significant effect on future mobility. In this respect it is imperative that individual mobility in the years to come is environmentally sustainable, does not require sacrifices and can find the necessary finance. When developing forward-looking vehicle concepts, it is therefore necessary to consider requirements that go far beyond those that have been placed on vehicles to date [1, 2, 3].

Now, in addition to safety, comfort and driving performance, other aspects such as ecology and economy are coming to the fore. The aim is to contribute to the reduction of fuel consumption and environmentally detrimental CO_2 emissions by decreasing driving resistance [2]. Similar to various research projects (for example SuperLightCar) the potential for lightweight vehicle construction offered by the research project described here will initially be concentrated in the body of the vehicle, because this area typically makes up 25 % of the total vehicle weight.

Future drive technologies will be either partially or fully electric, with some also making use of other energy sources. The vehicle development programme is therefore also concerned with drive concepts such as fuel cells and hydrogen drives. This must be taken into account in the structural design. Especially important is that the necessary tank and storage modules must be integrated into the vehicle in such a way that they meet with packaging requirements and cannot intrude into the passenger cell during a crash.

A further challenge for future vehicle concepts is the trend towards individual vehicles, which means a reduction in the volumes produced for each model variant [4, 5]. An additional requirement was therefore defined for the project to ensure that solutions can be modularised and that there is the possibility of creating derivatives. Previously, these strategies have been implemented primarily in the areas of interior fittings and engines, and in the creation of modules for the assembly process. By contrast, for body and structural components the focus is on developing concepts for new, integrated solutions.

2 Vehicle Concept

Taking an upper mid-range vehicle as a starting point, the rib and space frame concept developed at the Institute of Vehicle Concepts has been designed in such a way that alternative drive concepts can be integrated safely and easily. The location of the drive units in the floor area is designed-in right from the concept phase, **Figure 1**. This is because new drive concepts, such as the free piston linear generator (FPLG) [10] developed at the DLR, allow packaging considerations to

Figure 1: DLR vehicle concept with intrusion resistant containment

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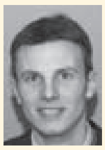

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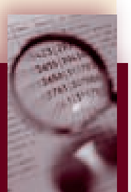

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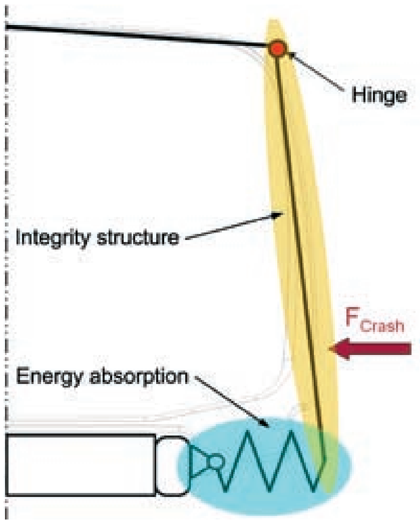

Figure 2: Mechanical principle used for the rib (schematically)

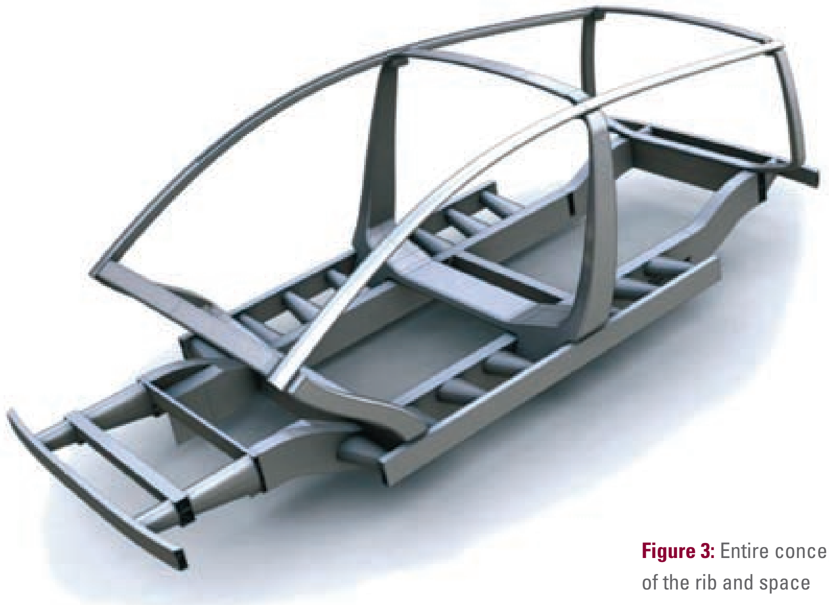

Figure 3: Entire concept of the rib and space frame construction

be optimised for weight and encourage the use of modular containment solutions. This has the advantage that alternative energy storage options such as compressed natural gas (CNG) or hydrogen can be located in an intrusion-resistant area, allowing the centre of gravity to be lowered and the axle load distribution to be optimised.

These requirements do, nonetheless, result in specific characteristics that have to be implemented as part of the body concept. The most significant load cases for the design of the floor assembly are those that emerge from both the side and pole impact test procedures. The components of the drive unit must not be forced against an incompressible object, as this would cause the acceleration values for the occupants to increase too sharply.

A construction was therefore chosen for the bottom of the car body that forms an outer zone to act as a "crash compartment". This zone extends from the sidewall to the continuous side members, which are used for their suitability for lightweight construction. By exploiting a mechanical principle and utilising energy absorbing CFRP crash cones (see also [6]), the zone absorbs the required energy for the IIHS side impact test, while also ensuring a defined level of intrusion, **Figure 2**. The front end of the "crash compartment" forms a cross-brace in the area around the A-pillar. This concept allows the complete front end to be designed to meet the requirements of a frontal impact. In the rear of the vehicle, the bracing is adapted to meet the respective packaging requirements.

The "base frame" for the space frame is formed from three ring structures (ribs) at the A-, B- and C-pillar positions, which function also as node elements that support the variable profiles used for the rocker panels, side members and roof beam, as well as distributing the loads that occur in the various load planes. This arrangement offers the advantage that, by scaling the lengths of the profiles, it is possible to derive different variants from a single base structure.

The basic construction principle used for the vehicle can be retained. The structure of the vehicle behind the B-rib can be modified, for example shown in **Figure 3**, to meet individual requirements for the design of the rear end.

Taking this fundamental concept as a basis, the DLR hopes to be able to make a significant contribution towards improvements in quality of lightweight construction and passive safety. As well, it likes to present an innovative means to implement modularity in structure-bearing components. Participants in the project are: the DLR Institute of Structures and Design in Stuttgart (Germany) and the DLR Institute of Composite Structures and Adaptive Systems in Brunswick (Germany), besides the DLR Institute of Vehicle Concepts (DLR FK) in Stuttgart (Germany). The DLR FK is also working in cooperation with external companies, such as ACE GmbH and ACTS GmbH.

Using the rib and space frame construction leads to a reduction in complexity and offers a considerable cost advantage. However, the lightweight construction necessary to achieve a reduction in CO_2 presents a particular challenge, because any increase in weight must be compensated for by the possibilities offered by modularisation and improved passive safety.

3 Key Component Hoop Rib

The hoop rib component lies at the heart of the path developing the complete rib and space frame construction method. As the centrepiece of this novel construction method, the rib demonstrates increased complexity in terms of design and dimensioning. The B-rib, which replaces the B-pillar of a conventional car, has been chosen as a representative example. The multi-material design combines materials and construction methods in such a way that the resulting lightweight construction is optimised in terms of economy for a particular production volume.

The ribs used in the DLR concept described here employ a fibre composite construction method. At 25–50 Euros/kg, the CFRP (carbon reinforced composite material) is certainly expensive; however it possesses the necessary mechanical properties in terms of stiffness, energy absorption and weight.

Using a relatively small amount of CFRP allows components and structures to be optimised according to local requirements and to achieve considerably better results in side impact crash tests than would be possible with conventional materials. This high-performance material is capable of absorbing energy inputs of up to 60 kJ/kg during a side impact, with the minimum amount of travel in the shortest time, **Figure 4**.

The material notwithstanding, the closed hoop structure of the ribs means they are predestined to accommodate the radial loads caused by lateral and roof impacts. The rib construction therefore represents an innovative approach that shows a great deal of promise in providing effective side impact protection.

The connection between the individual CFRP ribs can be implemented using metal profiles with simple geometric shapes. It is precisely this combination of expensive high-performance components in relevant structural areas and cost-effective materials and manufacturing techniques elsewhere that can make a contribution to a safe, lightweight solution at a reasonable cost.

4 Development Methodology for the Rib Construction

The development methodology used covers every stage of the process, from topology optimisation, design, and statistical and dynamic analysis, through to the construction of a prototype, which was tested in trials and serves to validate the results of the simulation process. During the early concept phase, analytical evaluations were conducted in respect of the side impact crash requirements. The result of these deliberations is the use of the mechanical principle illustrated in Figure 2.

When a side impact occurs, the rib in the vicinity of the roof beam acts as a deformable joint. By contrast, the area between the roof beam and the rocker panel is designed to be extremely stiff, so as to provide the best possible protection to the occupants in the head and shoulder region.

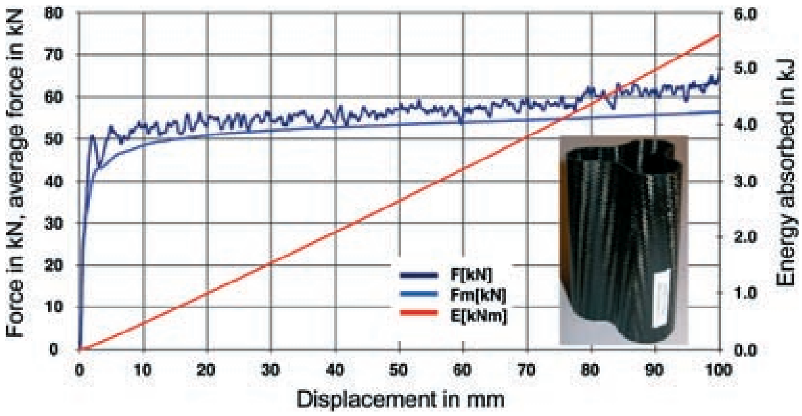

Figure 4: Example of a crash cone made of CFRP (bottom right) with resultant force-displacement curve

The resulting rotation about the deformable joint means that the crash energy is transferred to the crash elements in the floor region. These are located in the area between the rocker panel and the continuous side member and absorb energy through a crushing action.

This mechanical principle is particularly useful in that it allows the engineer to effectively evaluate the energy absorption in the floor, the maximum permissible intrusion and the acceleration of the occupants. The specific aim is to guarantee that there is little or no intrusion during a Euro NCAP side impact test, as well as affording passengers the best possible protection in the new IIHS tests [7, 11].

5 Optimising the Topology of the Rib Cross-section

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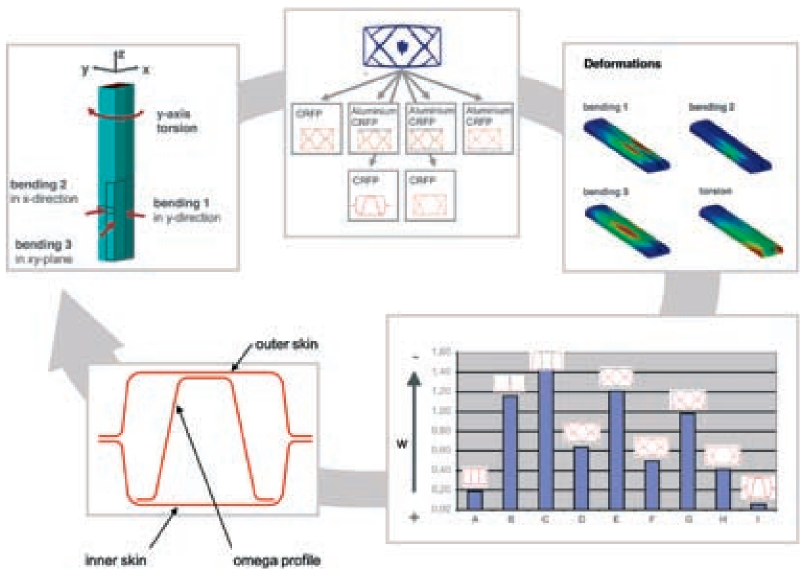


Figure 5: Topology optimisation and flexural strength calculation of the B-pillar structure

ty reasons extends from the corner regions to the centre of the opposite belts.

However, the cost and effort involved in manufacturing and joining this cross-sectional shape with fibre composite construction methods would be prohibitive, so several other types of simplified cross-section were designed and constructed with a view to better manufacturability. The cross-sectional designs that were developed were investigated for flexural strength by means of static analysis and stability calculations. The evaluation took account of a characteristic values w, determined in relation to a reference cross section, which was used to adjust for the relationship between the material utilisation factor and the mass. Two distinct cases were considered: for isotropic materials, the utilisation factor of the material is the ratio of the maximum stress that occurs to the maximum strength of the material; by contrast, for fibre composites the utilisation factor corresponds to the first ply failure index f [9]. The Tsai-Hill criterion was used to calculate the first-ply-failure index. The First-ply-failure index indicates the ratio of the occurring stress in a ply to the materials strength for a given load case. It is therefore the equivalent to the utilisation factor, used for isotropic materials. For isotropic material the Eq. (1) is used:

$$w = \frac{\frac{\sigma}{\sigma_{max}}}{\frac{\sigma_{Ref}}{\sigma_{max,\,Ref}}} \bullet \frac{m}{m_{Ref}} \qquad \text{Eq. (1)}$$

For fibre composites the Eq. (2) is used:

$$w = \frac{f}{\frac{\sigma_{Ref}}{\sigma_{max,\,Ref}}} \bullet \frac{m}{m_{Ref}} \qquad \text{Eq. (2)}$$

The formula signs are described in the **Table**. The cross-section "I", shown in Figure 5 (right, bottom), which has a monolithic construction and is made from CFRP, emerged as the best solution as a result of the investigation. This was therefore examined with regard to its stability, whereby the critical load case was the buckling of the top surfaces. This can be, however, be prevented by using a structural foam. This means that the rib consists of three components: inner skin; outer skin; and an omega profile that serves to stiffen the cross-section, Figure 5 (left, bottom). The three components of the rib are bonded together with adhesive at the contact surfaces of the half-shells.

Table: Meaning of formula signs

Meaning	Formula
First ply failure index	f
Material utilisation factor	$\frac{\sigma}{\sigma_{max}}$
Material utilisation factor for a reference cross-section	$\frac{\sigma_{Ref}}{\sigma_{max,\,Ref}}$
Mass, reference mass	m, $m_{Re\ f}$

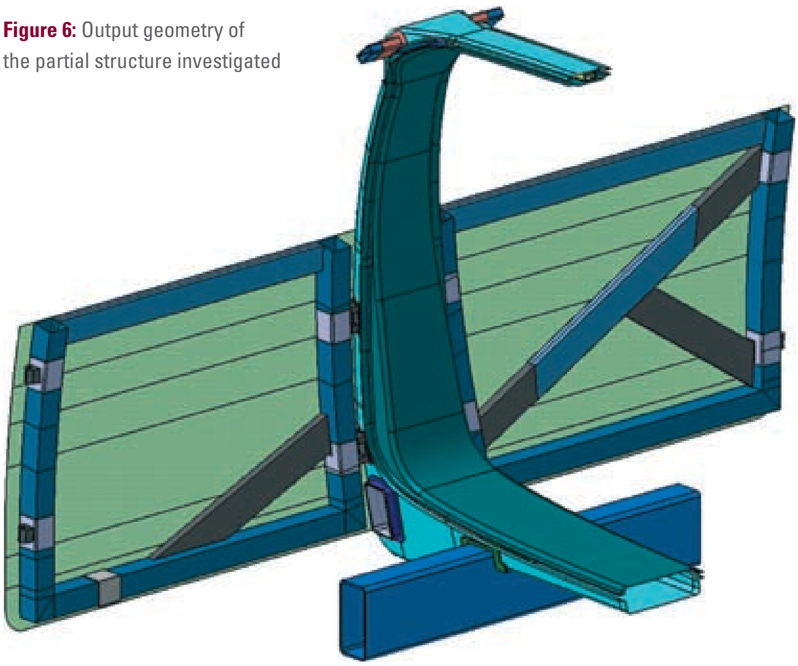
6 Selected Results from the Dynamic Simulation and Validation

In order to minimise the effort and expense required to implement a prototype, while still creating a realistic structure for the investigation, a numerical vehicle replacement model was produced and compared to the side impact behaviour of the reference structure. The plane load of the side crash barrier was realized by using simplified doors in the simulation as well as in the prototype structure. The output model for the dimensioning of the rib is represented by a rigid connection between the centre of the rib and the outside of the doors, with the additional definition of discrete mass points in the centre of the vehicle to achieve the crash weight of 1250 kg, **Figure 6**.

In order to identify weak points, each CFRP component was initially assigned a constant, but low, wall thickness, so that failure zones occurred in the dynamic simulation with LS Dyna. Based on these results, the structure was improved over several iterations, in order to exploit the advantage offered by fibre composite construction of easily implementing variable wall thickness.

Trials were carried out on physical parts in order to validate these simulation results. To this end, the simulated structure was built at a 1:1 scale and impacted with a barrier of the type used for Euro NCAP test at a speed of 50 km/h. The rib structure was fitted with acceleration sensors in order to obtain better values for comparison than those provided by

Figure 6: Output geometry of the partial structure investigated



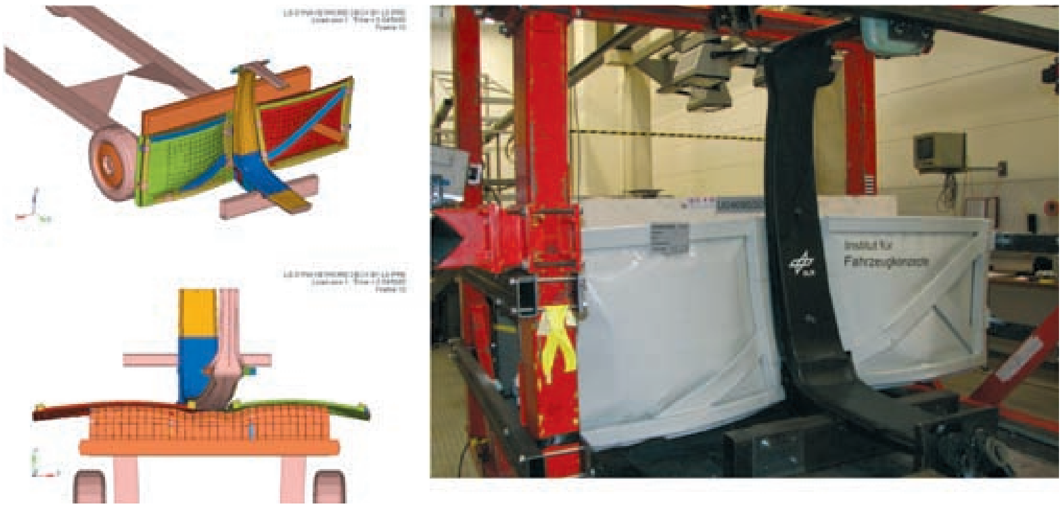


Figure 7: Comparison between the CFRP simulation (left) and a physical crash trial (right)

just the deformation of the barrier and the intrusion of the doors. **Figure 7** shows a comparison between the simulation and the physical trial.

Comparison of the acceleration sensors demonstrated that the FE model provides a good prediction of the experimental test curves, even without validation measures. Using the simulation results, however, it was established that the FE model had a greater stiffness than in reality. To date, the following major changes have been made to the FE model for the validation process, which is not yet fully complete.

- decoupling of the rigid connection between the lock panel and the door frame
- definition of the failure points for the hinge joints on the frame side
- improved modelling of the beading and reinforcement plate along the outside of the door frames
- flexible connection of the upper rib through modelling the frame carrier.

The acceleration curve demonstrated an oscillatory behaviour and was predominant in the upper region due to the flexibility modelled for the frame carrier. Furthermore, reducing the stiffness in the doors also improved the values obtained in the centre of the vehicle, **Figure 8**.

7 Results and Outlook

An innovative vehicle structure concept was developed using a geometric design, a rib structure, crash cones, an innovative door attachment and specially selected materials. This concept promises higher levels of safety for vehicle occupants, lower weight and more cost-effective options for modularisation than can be achieved with conventional construction methods. The first key component to be developed was the rib which forms the centrepiece of the rib and space frame construction. This was completed within 9 months and boasts a weight reduction of up to 50 % compared to a steel reference structure [8]. Physical testing, which followed the concept phase, the successful implementation of the structural design and the manufacturing of a prototype, demonstrated that during the trial implementation in line with Euro NCAP no component failure occurred and would therefore also be able to address the IIHS requirements with higher speed and barrier weight [7].

Areas that have not yet been investigated, but show great potential for further optimisation, include, for example, the use of self-adhesive foam for improved transmission of shear stresses through the rib structure. The material parameters for the FEM simulation also need to be optimised. In the case of the

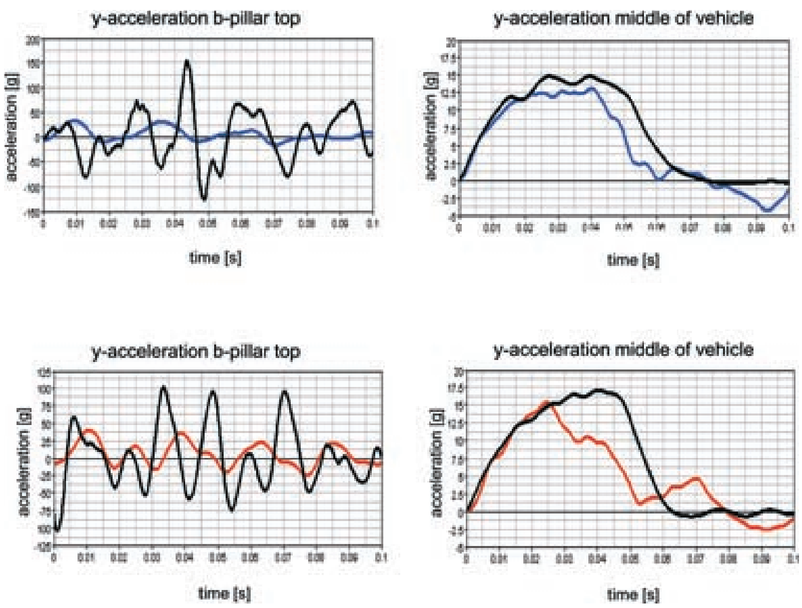


Figure 8: Acceleration values: blue line = test 1, red line = test 2, black line = FEM (with validation measures for two test procedures)

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joints, this is being done by means of additional trials on generic samples.

A significant phase in the future development will be the integration of the rib into the space frame environment. It is also planned to implement and test this complete theoretical model as a prototype, with the goal of achieving a weight reduction of between 20 and 25 %, whilst also improving safety and various other factors. It is hoped that the possibilities in terms of creating derivatives and functional containment solutions for alternative drives offered by the concept described will set a trend for low-emission mobility in the future.

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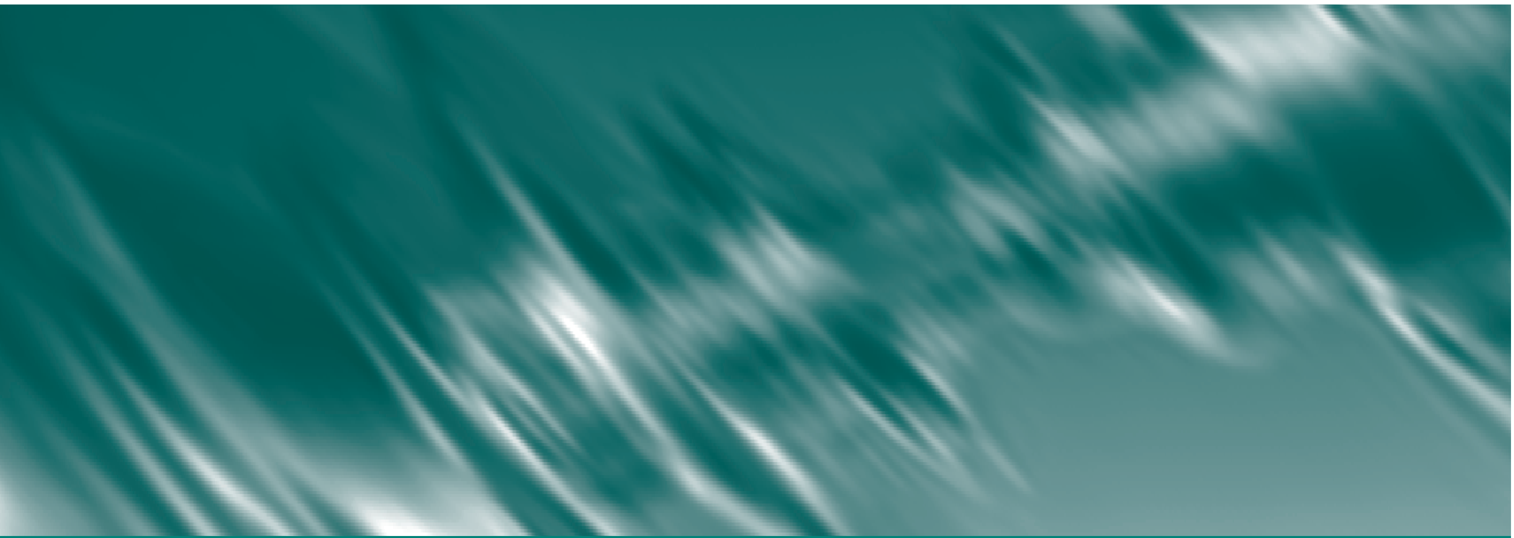

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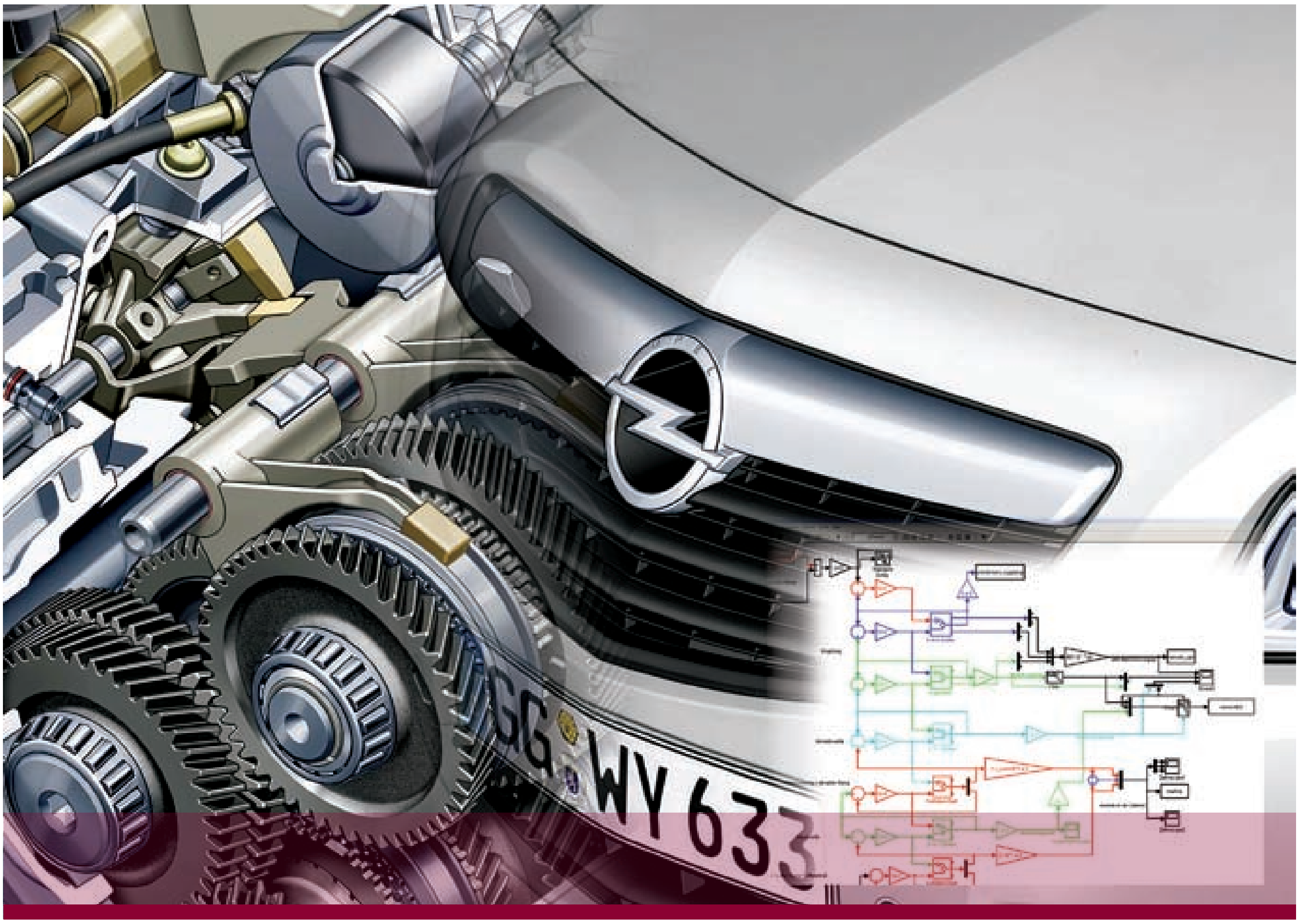

Assessment Method for Passenger Car Transmissions under Abusive Loads

A multi-body simulation method is established in collaboration with the institute for machine elements, gears and transmissions of the Technical University of Kaiserslautern (Germany) and General Motors Powertrain Europe to optimise development and validation of the abuse-safety of manual transmissions. This computational approach provides a better understanding of the transmission dynamics even on component level which is not achievable by measurements on the car.

1 Introduction

In manual transmission development context idioms such as snap- or race-start as well as the journalist test are often used. These are usually load profiles with extreme mechanical loads but limited numbers of cycles. Potential scenarios to achieve these extreme load peaks are for example a sidewise slip from the clutch pedal while running the engine at high speed with a gear engaged; the vehicle is at rest. The sudden frictional engagement of the clutch causes peak loads exceeding the nominal engine torque by a factor of approximately two to three. Due to the continuous increase of vehicle curb weight and wheel diameter the abuse loads on the driveline increase as well, even if the nominal engine torque does not change. High development costs and limited development time require the precise determination of the dynamic abuse loads as a solid basis for further virtual transmission development [1 to 8].

When loading the transmission with these exceptional loads the system shows a highly dynamic and complex load-deformation behaviour that is hardly accessible by analysis. The real loads and component deformations, for instance of shafts and housings and of their respective joints – usually roller bearings –, are unknown to a large extend; simulation is the only key to increased understanding of the processes.

The presented simulation model of a passenger car transmission enables the analysis of the highly dynamic processes in the transmission and the verification against experimental data. In the following the hybrid simulation model is explained and selected simulation results including the comparison of simulation and physical test are being presented. The focus is on the deflections and axial displacements of the shafts, on gear tilting, housing deformation etc. The results presented demonstrate the potential of such a hybrid simulation model.

2 Goal of the Project

A three-dimensional hybrid simulation model was developed by General Motors (GM) Powertrain Europe in collaboration with the institute for machine elements, gears and transmissions (MEGT) of the University of Kaiserslautern to gain improved insight into the dynamic processes inside a passenger car transmission for example when loaded in a so-called snap

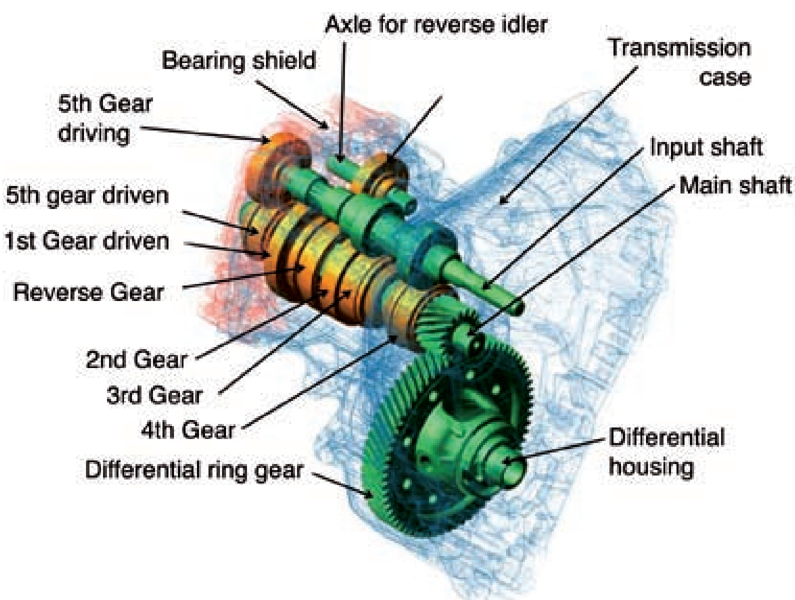

Figure 1: Transmission components of the rigid MBS model

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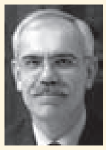

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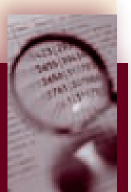

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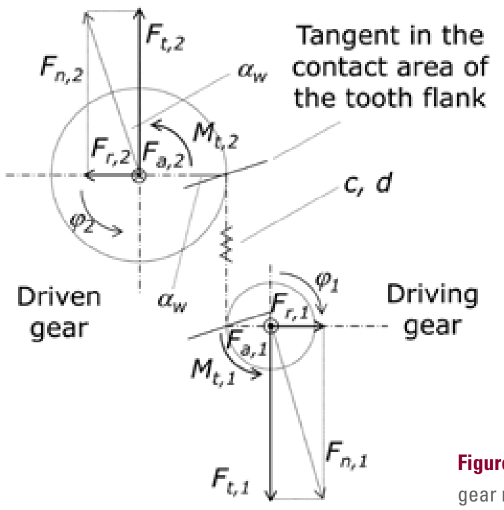

Figure 2: Parameters for a simplified gear mesh evaluation

start. The presented model was built up in Adams. All imaged torque transferring components of the transmission excluding the gears themselves are modelled as flexible bodies to capture the effects of highly complex loads realistically. Since the measurement of the stresses in the gear teeth are not in the focus of this project it is sufficient to model the elasticity of the gear teeth with simplified force-displacement relations. The discretisation of the transmission components and the bearings in the model can be easily adapted to the appropriate level of model accuracy and complexity by the modular setup and the parametric structure of the simulation method. In addition to the external mechanical loads it is also possible to capture thermally induced deformations of the system "steel shaft in aluminium housing" and the respective influence on bearing supports etc.

To solve the postulated task a method was developed by GM Powertrain Europe and MEGT, which is described in the sequel. The method enables its user to

- gain information about the dynamic behaviour of the transmission components and their respective interactions
- locate potential weak points with regard to abuse loads in the overall system „transmission“
- estimate and evaluate the influence of various components on the system behaviour
- further optimise the ultimate dynamic strength of the transmission.

The following shows selected results achieved with a fully flexible three-dimensional simulation model and compares those with quasi-static and dynamic experimental results.

3 Modelling Strategy

This chapter describes the modelling strategy. In a first step a rigid body model of the transmission is set up. Afterwards most of its components are subsequently replaced by flexible bodies. In the last step the model is expanded in order to consider sensors and respective supports.

3.1 Rigid Transmission Model

The rigid body model of the transmission is the basis for the following refined models. The rigid model allows testing single components and enables simplified studies on sub-assemblies.

3.1.1 Imported Rigid Transmission Components

For setting up the rigid body model it is most convenient to import the relevant geometries directly from Computer-aided design (CAD) data into the pre-processing system. **Figure 1** shows the content of a typical model. Material parameters are assigned to all components to calculate mass and inertia of these bodies. Usage of the original CAD geometries enables illustrative visualisation of the processes later on. The integration of the original geometry facilitates the replacement of the initially rigid bodies by elastic, flexible components when upgrading the overall model.

3.1.2 Simplified Rigid Bearing Models

After geometry import and material parameter assignment the user has to define interface points to apply loads to the structure – so-called markers. These markers are used to define joints of different types to tie components to each other and thus reduce the relative degrees of freedom. A slider joint for instance disables three rotations and two translations and it only allows a one-dimensional movement. The support of the shafts inside the transmission case can be realised in a similar way. The simple bearing models only allow a single rotational degree of freedom. The support of the speed gears on the respective shafts is realised in the same way.

3.1.3 Gear Mesh Description

The gear mesh connects the input shaft with the main shaft and differential pinion with the final drive ring gear, respec-

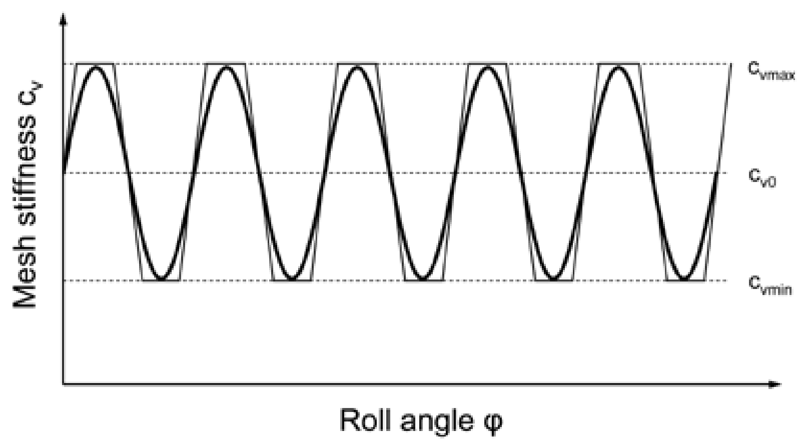

Figure 3: „Real" (trapezoid) and approximated (sinusoidal) characteristic of the torsional mesh stiffness c_v versus roll angle φ [9]

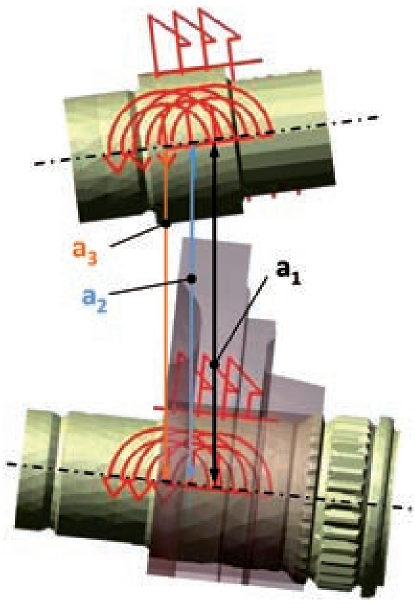

Figure 4: Discretisation of the mesh description along the tooth width

tively. Since non-standard load cases of bearings, shafts and housings are of main interest here, only a simplifying modelling approach is chosen to describe torque transfer and gear mesh stiffness.

Starting from the roll angles $\varphi 1$ and $\varphi 2$ at the gear wheels, **Figure 2**, a differential twist angle can be calculated which is proportional to the provided torque. Applying the standard equations from the theory of gears the forces at the active gear mesh for the individual gear parameters can be derived. The forces and the tilting torque due the helical gearing are included in the model acting at the geometrical gear centre point.

In the first step the torsional stiffness c of the gear mesh is assumed constant, when refining the model the stiffness c is more realistically assumed variable along the active gear mesh reflecting the changing number of load carrying teeth. This is achieved in the model by using the static stiffness c_{v0} combined with a sinusoidal variation along the pitch line. **Figure 3** shows the comparison between the „real" trapezoidal stiffness and the „sinusoidal" approximation versus the roll angle φ. Furthermore, the model contains the angular back lash of the gears leading to further nonlinear effects in the model.

An even more sophisticated method of gear modelling is to discretise a gear with a slice-model, the slices being coupled at the shaft axis by rigid connectors. For each individual slice the forces of the gear mesh can be calculated considering the radial deflection of the slice element; **Figure 4** shows as an example a three-slice model of a gear pair. In case of the flexible transmission model the same methods of modelling the gear mesh are used.

3.2 Flexible Transmission Model

The stepwise replacement of rigid elements by flexible bodies is the next step. The modelling of the flexible components can be done in a Finite Element (FE) environment – as for example in this paper – as well as in the pre-processor of the multi-body simulation (MBS) system.

3.2.1 Imported Flexible Transmission Components

The flexible components are modelled using the Craig-Bampton method which reduces a full-scale FE model to only a few modal degrees of freedom without loss of major elastic properties of the components. Adams/flex is used as interface between the FE environment and the MBS environment. The procedure facilitates the determination for example of the component deformation in the MBS model with comparatively little numerical effort.

Load and boundary conditions on the flexible bodies are defined on so-called Interface Nodes. **Figure 5** depicts an example for a bearing shield with interface nodes in the bearing seat centre point and at threads and bosses which are connected to the surrounding structure by rigid elements. The bolt connections can be represented realistically using non-linear spring-damper elements. This procedure is applied to replace all required rigid components by their flexible counterparts.

3.2.2 Enhanced Flexible Bearing Models

A first upgrade of the rigid bearing models is achieved by modelling the bearings as linear Voigt-Kelvin-elements to include their stiffness in the simplest way. The enhanced bearing models allow by the definition of non-linear force-deformation relations a more realistic image of the radial, axial and rotational stiffness. Since locating bearings prevent axial displacements in both directions it is necessary to determine the reaction forces based on a suitable definition of compressive and tensile forces. Within the axial bearing clearance the bearing can move without resistant forces. The tilting stiffness of the bearings is defined in analogy. The operational bearing clearance can be parametrically varied in the model. Furthermore, it is necessary to include axial pre-tension of tapered roller bearing systems in the model. The thermal expansions of the transmission components are specified by a separate FE analysis and are taken into account regarding the bearing pre-tension as for instance for the differential shaft being supported in a tapered roller bearing arrangement in the housing. Locating-floating bearing arrangements are used for input and main shaft carrying the driven gears.

3.2.3 Flexible Coupling of Gear Body and Shaft

Spring-damper elements can be used to connect the gear bodies to the shaft. The

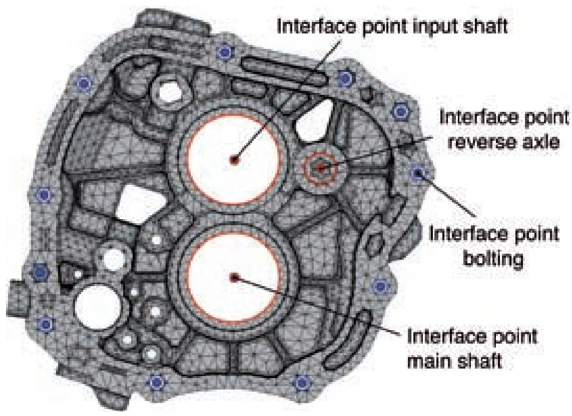

Figure 5: Meshed structure of the bearing shield and interface points

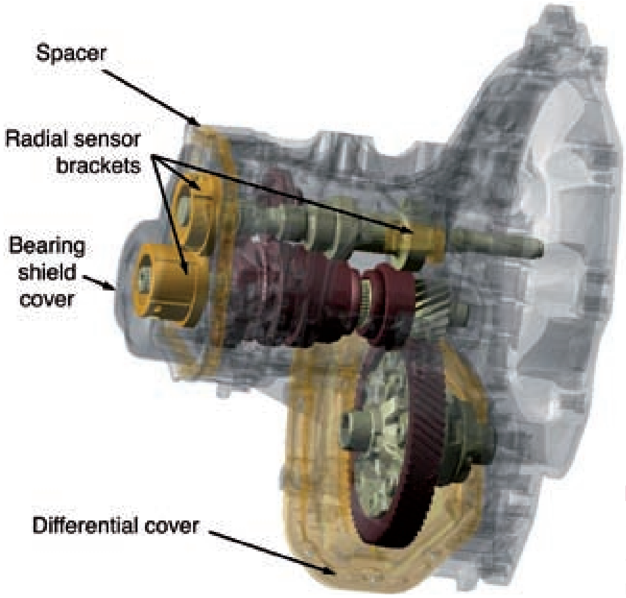

Figure 6: Additional components to adapt to the experimental setup

spring stiffness of the coupling in circumferential direction can be chosen to model a joint of hub and shaft as well as the engagement or disengagement of the drive gears.

3.3 Adjustment of Flexible Transmission Model to Experimental Setup

To better adjust the computational model to the experimental setup some additional elements representing the measurement instrumentation are included into the flexible MBS model. **Figure 6** depicts this extension; the additional flexible elements – supports for the sensors at differential, bearing shield cover and transmission case – realistically represent the influence of the instrumentation on the overall system behaviour. The spacer between the bearing shield and its cover serves as a guide for the electric instrumentation.

4 Adjusting Measurement and Simulation

This chapter contains selected results from quasi-static as well as dynamic simulations in direct comparison with the respective experiments. Displacement and torque level are recorded during a verification experiment to understand the governing processes in the transmission. Various gears, load levels and time dependencies are analysed in experiment.

A specifically modified transmission is used for the experimental quasi-static and transient test. **Figure 7** shows the respective measuring planes in radial direction and the locations of the axial sensors. Two measuring planes are defined for the input shaft: One is located between the pinions of third and fourth gear, the other in the area of the fifth gear. At the measuring points the radial displacements are recorded by two perpendicular sensors to determine the trajectory of the shaft during loading. Furthermore, the axial movement of the shaft end is recorded relatively to the bearing shield cover. At the differential shaft the procedure is applied in analogy. Besides, the tilting movement of the differential ring gear is determined.

In addition to the displacements and angular movements the torque level is recorded. Calibrated strain gauges are used to measure the torque acting on input and output shaft, the strain gage voltage of the rotating sensors is transferred to a stationary receiver by telemetric submission.

4.1 Quasi-static Analyses

During the first tests performed for method and model validation the transmission is driven by an electric engine with constant torque at low number of revolutions, approximately 40 rpm. In this operational mode the eccentricity is recorded. Seizing is prevented by the experimental and operational setup. At the output a second electric engine is used as a generator break. The differential is mechanically locked to enforce a symmetric loading. The quasi-static experiments covers operations in first and second gear as well as in reverse gear, torque is applied in coast and drive conditions.

The following selected results compare each simulation and measurement.

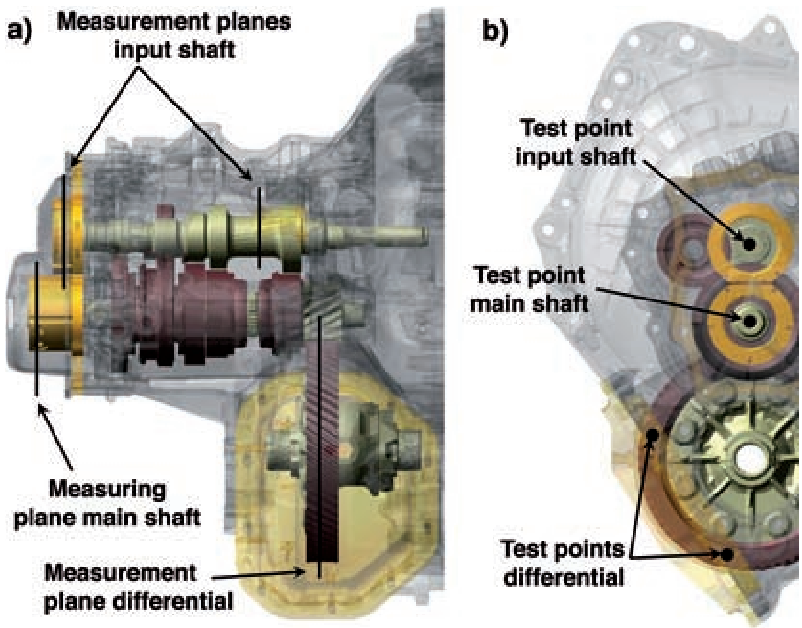

Figure 7: Measuring planes and points of the examined transmission: left radial, right axial

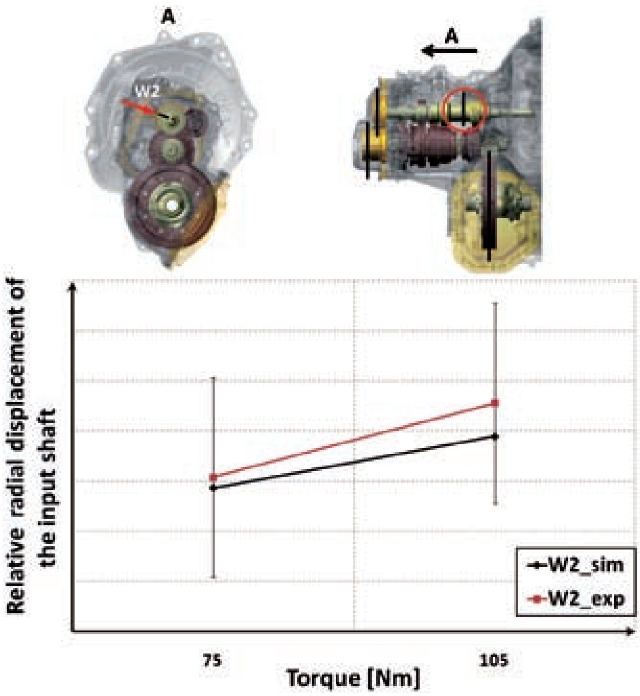

Figure 8: Radial displacement of the input shaft at measuring point W2 for load case "second gear coast"

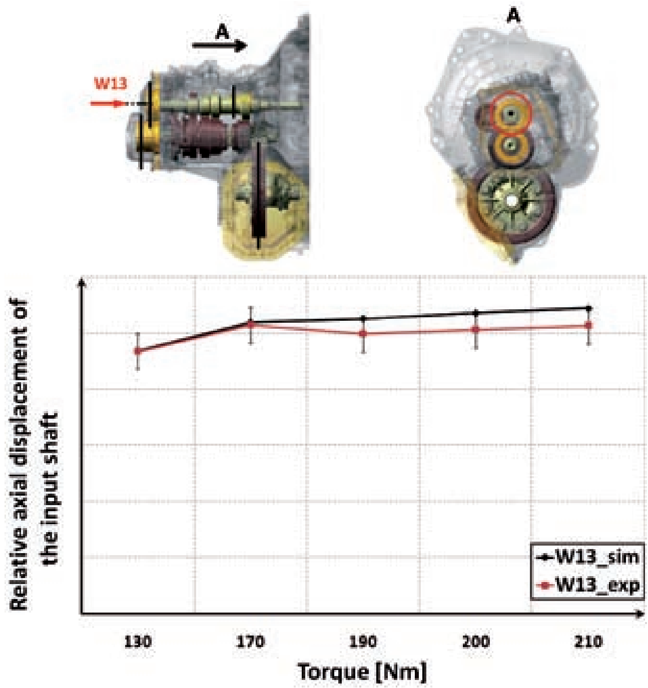

Figure 9: Axial displacement of the input shaft at point W13 for load case "reverse drive"

All results refer to the unloaded state which was experimentally determined and which serves as a reference for the calculation of relative displacements for each load condition – drive or coast, respectively. Each operational point was repeatedly analysed to be able to examine some statistically relevant data and to identify potentially malfunctions of experiment or measurement.

Figure 8 shows the relative axial displacement of the input shaft at measuring point W2 versus drive torque for the second gear coast. The simulation results are well within the confidence range of the experimental observations. As expected the displacement increases with the applied torque. Results of similar quality are recorded for the radial displacements at point W13 for load case reverse drive, **Figure 9**. The simulation results are also covered by the confidence range of the experimental observations.

Figure 10 sheds light on the behaviour of the differential in drive operation at point W6 and W7 for the first gear. The orthogonal arrangement of the sensors allows in addition to display a trajectory of the differential in the measuring plane indicated. The dependency of the changing relative displacement on input torque matches fairly well for simulation and experiment.

The shown results form the basis for a detailed comparative analysis. They allow for calibration of the MBS model for further investigations. Respecting the fact that tolerances and unsteady thermal loads are excluded from the analysis presented the overall agreement can be rated as good if not even better.

4.2 Dynamic Analyses

For transient testing the transmission is examined on a test rig with a combustion engine. The vehicle mass is represented by rotating momentum wheels to

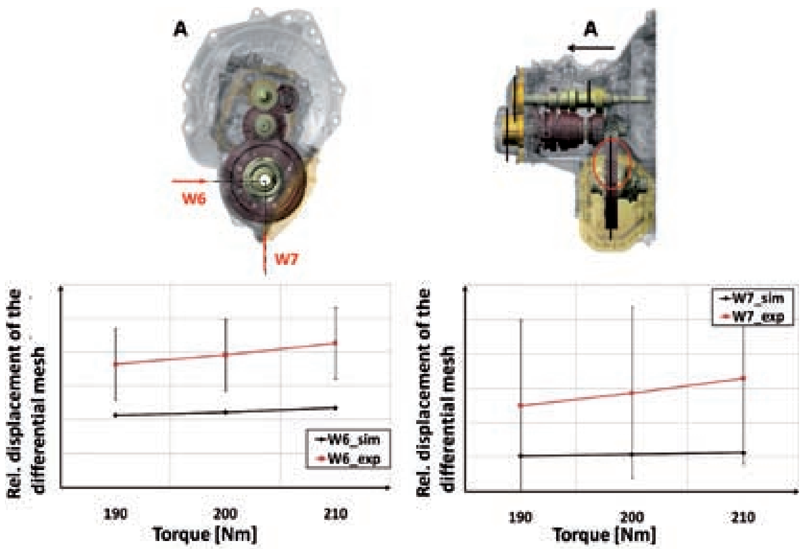

Figure 10: Radial displacement of the differential at points W6 and W7 for load case "first gear drive"

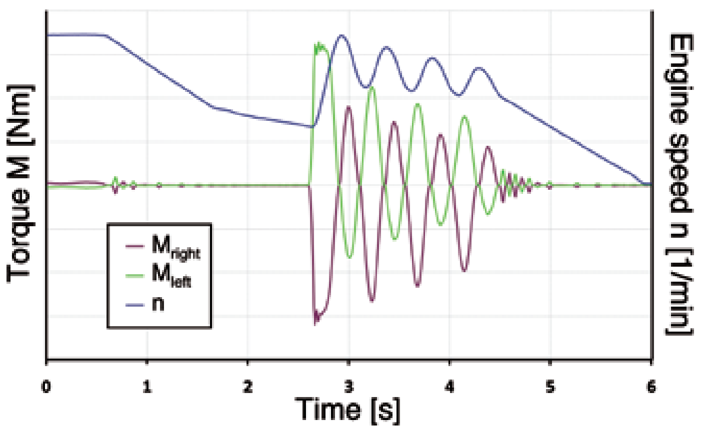

Figure 11: Measured engine speed and drive shaft torque right and left versus time

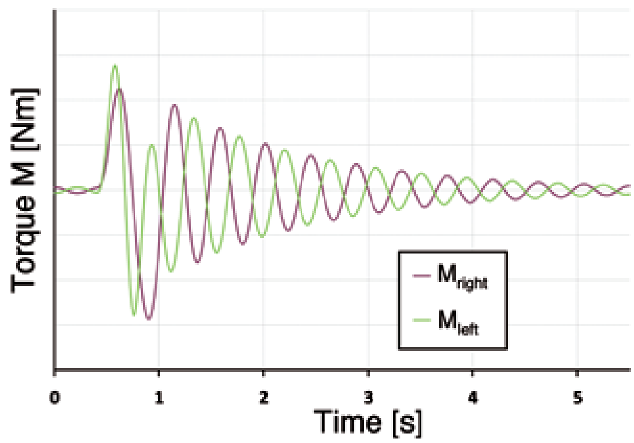

Figure 12: Calculated drive shaft torque right and left versus time

simplify the experimental setup, the measurements cover events during coast and drive load.

For the experimental analysis of the first gear drive load case the clutch is closed spontaneously while the engine is running, the driveline stands still. The clutch engagement is performed by a controlled pneumatic actor which guarantees fast and repeatable dynamic events.

For the measurements in coast operation in first and reverse gear the complete driveline is accelerated to a defined speed to prepare for the test. Afterwards, the clutch is disengaged and the engine speed is reduced to idle, thereafter the clutch is engaged rapidly simulating a sidewise slip from the clutch pedal by the driver. Highly dynamic scenarios take place which severely load the driveline in coast conditions.

As an example of verifying the simulation model the torque levels at the axle shaft are displayed for the first gear coast. **Figure 11** shows the experimental findings: After deceleration of the combustion engine the clutch is closed instantaneously causing a torque peak. After the force fit at the clutch the drivetrain inertia speeds up the engine. Consecutively, the torsional vibrations caused by the sudden clutch engagement are damped out quickly at a natural frequency of approximately 3 Hz.

Figure 12 shows the simulated torque level versus time. Peak torque and natural frequency of the torsional vibrations match very well. The differences in fade-out characteristics result from the engine speed: On the test bench the engine controller helps to pull the driveline speed again towards idle, a phenomenon which is not covered by the simulation model.

The level of congruence between simulation and experiment is good for both the static as well as the transient tests analysed in this study. After further verifications the next steps are to analyse component behaviour in detail for example shafts, housings and bearings when being submitted to these extreme load cases.

5 Summary

The dynamic processes inside manual transmissions during abusive load conditions are to date almost unknown and it is hard to assess these events either experimentally or analytically. But knowledge about dynamic quantities such as loads and deformations are required in early development stages due to cost and time requirements. This challenge motivates the development of a hybrid MBS-based simulation method as depicted providing the following advantages:

- simplified model creation utilising imported CAD data in the MBS environment
- visualisation of dynamic processes in the MBS Post Processor
- derivation of dynamic load data for three-dimensional structural analysis
- reduced calculation times by the modular structure and the analytic description of boundary conditions, body interactions etc.

The results obtained with Adams are experimentally validated and allow for time and cost effective application of the simulation method at GM Powertrain Europe.

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