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LIGHTWEIGHT Roof Module with Integrated Solar Cells

FIBRE-REINFORCED Polymer High-pressure Tanks as Load-bearing Structural Elements

VEHICLE SIMULATION by Consistent Component Modelling and Parameterisation

/// INTERVIEW

Klaus Rohde-Brandenburger
Volkswagen

BODY ENGINEERING AND LIGHTWEIGHT DESIGN GO HAND IN HAND

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COVER STORY

BODY ENGINEERING AND LIGHTWEIGHT DESIGN GO HAND IN HAND

4, 12, 16 I Lightweight design and the use of lightweight materials are being given fresh impetus by rising fuel prices and the ongoing climate debate. An important lightweight design project in steel is the FutureSteelVehicle (FSV) from Edag and WorldAutoSteel. Ford resolves the conflict between sporty design and easy access to the vehicle in its B-Max minivan by using a body structure that has no B-pillar. In our interview, Dr. Klaus Rohde-Brandenburger, Functional Manager Vehicle Technique at Volkswagen, discusses the key approaches towards breaking the weight spiral by applying lightweight design.

COVER FIGURE © Daimler
FIGURE ABOVE © Porsche

ZERO TOLERANCE

Dear Reader,

Alcohol is your rescuer in times of need, sang the German rock musician Herbert Grönemeyer back in 1984, commenting on the drink problems of many Germans. And the subject is no less relevant today. There are still far too many deaths on our roads caused by drink-driving.

A sensible way of reducing the number of people being killed or injured would be to introduce a 0.0 mg per ml blood-alcohol limit, as is customary in other European countries (Estonia, Romania, Slovakia, Czech Republic and Hungary). The majority of Germans would be in favour: according to a study by the insurance company DA Direkt, two out of every three people questioned supported an absolute ban on alcohol in road traffic.

Zero means zero, one might say – a clear rule for everyone, which should finally put an end to comments like “the odd pint or two won’t do any harm, surely”. Just how easy it is to underestimate the effects of “the odd pint or two” was shown some time ago when the Bavarian prime minister, Günther Beckstein, erroneously claimed that a “real man” would still be able to drive a car after downing a couple of litre-sized glasses of good Bavarian beer.

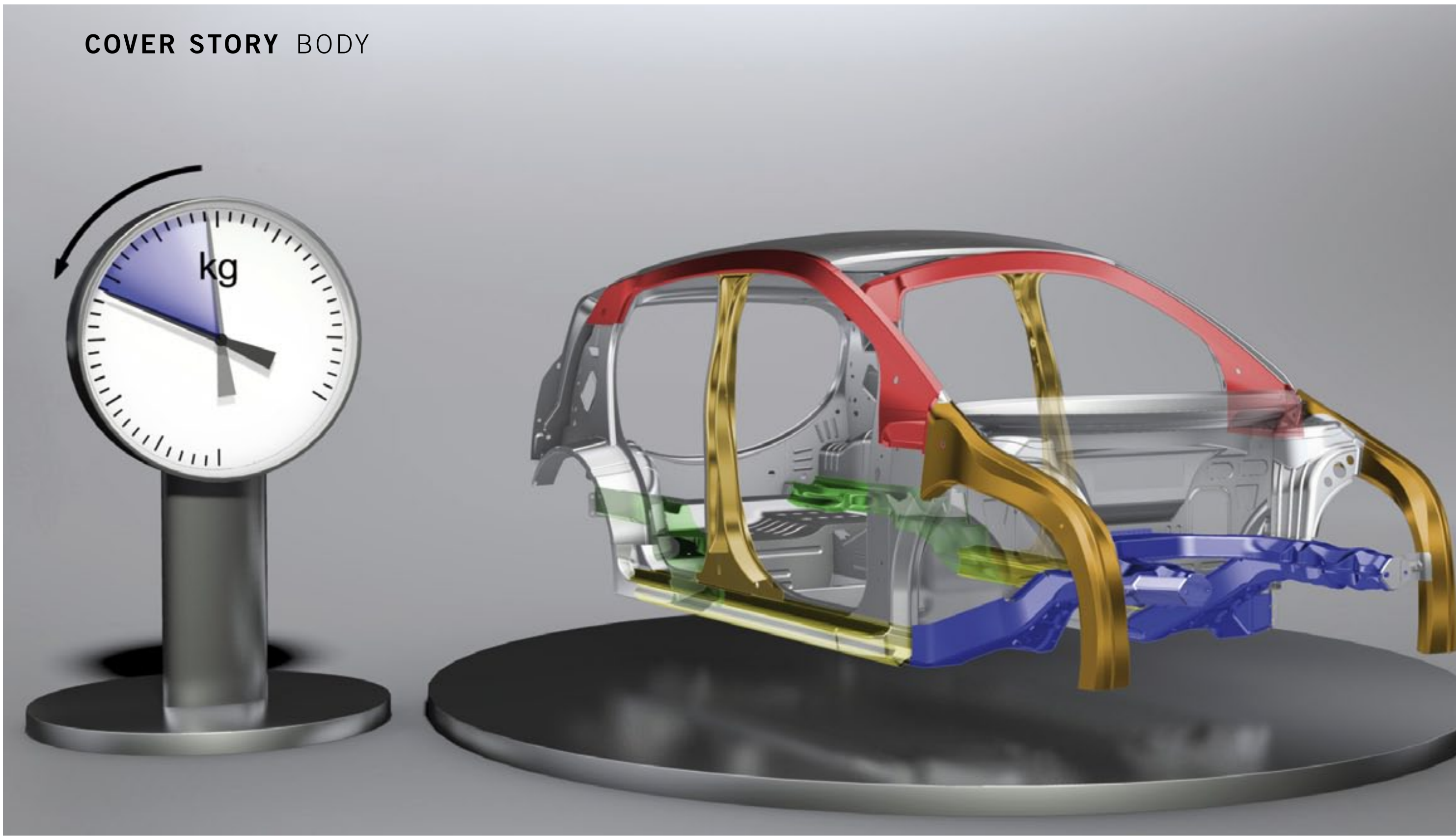
Drink-driving is not a trivial offence. Last year, according to the ACE Auto Club, approximately 58,000 people were killed in accidents on German roads – and around 7100 of these, or 12 percent, were alcohol-related.

But it is doubtful whether the government’s new penalty points system will solve the problem. Germany’s Federal Transport Minister, Peter Ramsauer, is planning a radical reform. Instead of the previous system, which imposed between 1 and 7 points depending on the seriousness of the offence, there are to be only two categories, 1 and 2, for offences in the future. France is taking a different approach, but one that has met with an incredulous shake of the head by most people. From 1 July, every driver will be required by law to carry an unused and fully functional breathalyser kit in the car. As if simply carrying a breathalyser will make people drink less. And what is more: the legal obligation also applies to holidaymakers. A better idea came from Sweden. As long ago as 2010, Volvo launched Alcoguard, a breathalyser system that is completely integrated into the vehicle. If the blood-alcohol limit is exceeded, the engine will not start.

To support zero tolerance, I believe that more intensive police checks are just as important. But that would cost taxpayers more money for personnel and equipment – money that, at the moment, tends to be invested more lucratively in parking wardens. In the long term, there is no alternative to 0.0 mg/ml. Or simply: If you drink, don’t drive.

Michael Reichenbach

DIPL.-ING. MICHAEL REICHENBACH,
Vice-Editor in Chief
Wiesbaden, 28 March 2012



FOURTH STEEL RESEARCH PROJECT AFTER ULSAB

The FutureSteelVehicle (FSV) is an important, ongoing lightweight design project in steel for forward-looking drive concepts in electric mobility. WorldAutoSteel, the automotive group of the World Steel Association, consists of 17 international steel manufacturers. From 2008 to 2011, Edag Inc., the US subsidiary of Edag GmbH & Co. KGaA, was commissioned to carry out development on behalf of WorldAutoSteel. Under the chairmanship of the United States Steel Corporation, Edag Inc. took on the project management, concept, development, calculation and documentation.

The FSV is the fifth automobile steel research project, similar to Ulsab, Ulsac, Ulsas and Ulsab-AVC. These made a major contribution to the development, implementation and processing of new high-strength (HSS) and advanced high-strength steels (AHSS). With the FSV, the global steel industry continues the reinvention process of steel in the automobile. This means that the FSV will bring about the implementation of new steel qualities and innovative semi-finished products for future generations of cars with alternative and conventional powertrains.

The FSV is currently being presented to car manufacturers in a road show. The steel industry communicates extensively with the development departments, to ensure optimum implementation of lightweight steel concepts for current and future vehicles designs.

OBJECTIVES AND MOTIVATION

New advanced powertrains (hybrid, electric and fuel cell drives) are bringing about radical changes in vehicle structure, and thus also in the choice of materials used. When working towards an environmentally friendly car, the entire body concept needs to be re-considered and alternative powertrains integrated in the best way possible, ❶.

The FSV utilises forward-looking CAE methods with an extended portfolio of advanced high-strength (AHSS) and ultra-high-strength steels (UHSS), along with a variety of intelligent semi-finished products and processing technologies, in order to reduce the body-in-white weight of a battery electric vehicle (BEV) to just 188 kg.

The target was to find safe, weight-optimised lightweight steel concepts for future vehicle bodies, which would reduce greenhouse gas (GHG) emissions to a minimum throughout the product life cycle. To achieve this end, the steel manufacturers incorporated the latest steel innovations and processing technologies into the concept. The FSV programme consisted of three phases:

- Phase 1: technology assessment (completed)
- Phase 2: conceptual design (completed)
- Phase 3: demonstration and implementation (2011 to 2012).

Started in 2009, Phase 1 included the evaluation and identification of fully mature powertrains and future vehicle technologies for series production in the period 2015 to 2020. In Phase 2, optimised, high-strength body concepts made of AHSS and UHSS steel materials

LIGHTWEIGHT DESIGN FOR THE FUTURE STEEL VEHICLE

The FutureSteelVehicle (FSV) is the fifth automobile steel research project. These projects made a major contribution to the development, implementation and processing of new high-strength and advanced high-strength steels. Using HSS, AHSS and UHSS steels the body-in-white weighs only 188 instead of 290 kg. The FSV will bring about the implementation of new steel qualities and innovative semi-finished products for future generations of cars with alternative powertrains for hybrid and battery electric vehicles. An article by WorldAutoSteel and Edag.

were developed for the basic vehicle variants, ❷.

Edag implemented the development method of an innovative and consistent CAE-intensive strategy for the systematic selection of materials. It includes the typical work stages analogue to a body development. The points of focus were the topological analysis and optimisation of the overall concept, the development of lightweight steel concepts, and checking crash properties by means of simulation.

❶ Powertrain package layout of the FutureSteelVehicle (FSV)

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FSV 1 A-/B SEGMENT 4-DOOR HATCHBACK 3700 mm LONG	**Plug-in hybrid drive PHEV-20** Electric range: 32 km Total range: 500 km Max. speed: 150 km/h 0 – 100 km/h: 11 – 13 s BIW weight: 176 kg	**Battery electric drive BEV** Total range: 250 km Max. speed: 150 km/h 0 – 100 km/h: 11 – 13 s BIW weight: 188 kg
FSV 2 C-/D SEGMENT 4-DOOR SALOON 4350 mm LONG	**Plug-in hybrid drive PHEV-40** Electric range: 64 km Total range: 500 km Max. speed: 161 km/h 0 – 100 km/h: 10 – 12 s BIW weight: 201 kg	**Fuel cell electric drive FCEV** Total range: 500 km Max. speed: 161 km/h 0 – 100 km/h: 10 – 12 s BIW weight: 201 kg

❷ Vehicle variants and drives of the FutureSteelVehicle (FSV)

CHOICE OF TECHNOLOGIES, PACKAGE, STYLING AND AERODYNAMICS

Benchmark analyses were taken as the basis on which to work out technical requirements and target values. After powertrain packaging, interior occupant space, ingress/egress requirements, vision/obscuration, luggage volume requirements, and ergonomic and reach studies of interior components established the component and passenger package space requirements. An exterior styling was applied to the packaging, followed by several fluid dynamic simulations, resulting in a drag coefficient of cd = 0.25.

TOPOLOGY ANALYSIS, LINEAR-STATIC AND NON-LINEAR-DYNAMIC OPTIMISATION

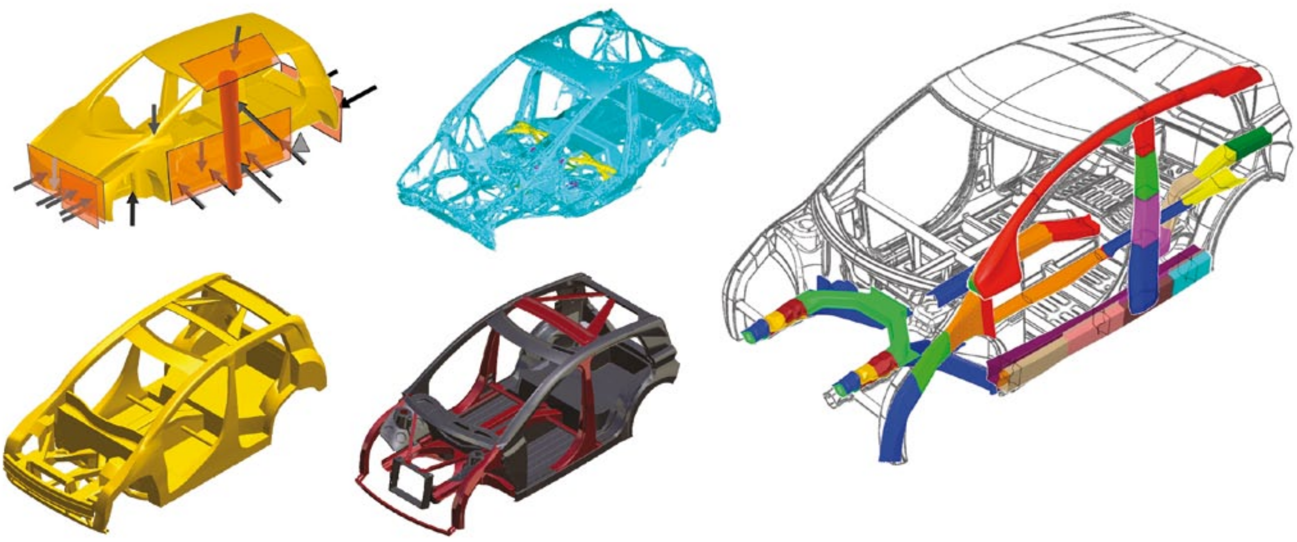
Topology optimisation provides an initial structure based on the available structure package as shown within ①. The FSV programme developed this initial structure by considering three longitudinal load cases, two lateral load cases and one vertical load case as well as investigations about bending and torsional static stiffnesses.

Though the topology optimisation was able to provide an initial starting point for the FSV's geometry, it is limited by its static approximation of dynamic crash loads and does not consider grade variations of the sheet metal within the structure. Therefore, the load path optimisation is moved to the dynamic design domain (using LS-Dyna) combined with a multi-discipline optimisation programme (Heeds), which also addresses a low fidelity optimisation of the major load path cross-sections, grades, and gauges of the body structure. The output is designated the "low fidelity geometry, grade and gauge" (LF3G) optimisation, ❸.

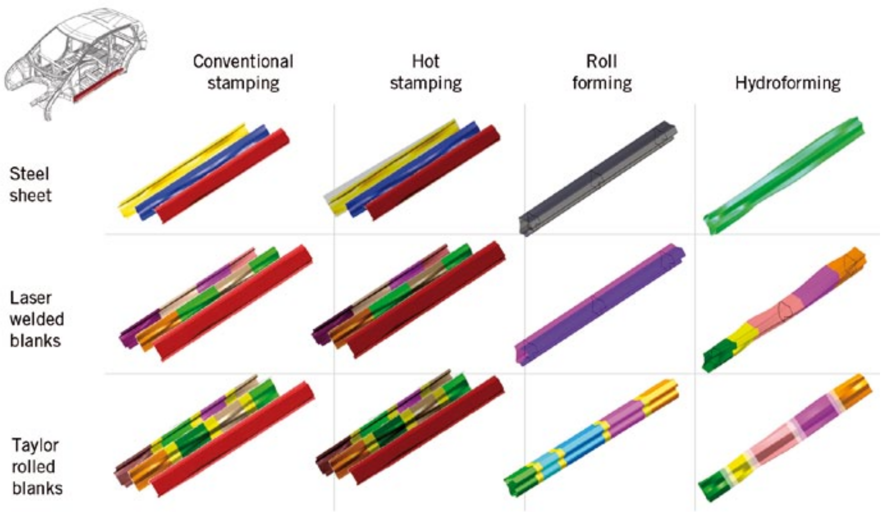
OPTIMISATION OF SUB-SYSTEMS

To create the required reference body structure, the LF3G body structure was combined with engineering judgment of current benchmarked design catalogues. This reference assumes typical manufacturable sections and joint designs combined with extensive use of HSS, AHSS and UHSS achieving a calculated mass for the sheet steel baseline of 218 kg. Based on load path mapping, seven structural sub-systems were selected for further optimisation using a broad bandwidth of manufacturing technologies.

The objective was to minimise the mass of each sub-system and simultaneously maintain the deformation energy in the sub-systems as that in the full LF3G model for each respective load case. The solutions obtained from the structural sub-system multi-discipline optimisation runs had appropriate material strengths and gauges, optimised to



❸ The design methodology shows up the dimensioning load cases, the topology optimisation, the structural optimisation and the generic CAD model as well as the LF3G optimisation of the reference body structure with eight structural sub-systems (from left to right)



❹ Steel solution manufacturing catalogue for the rocker sub-system

give a low mass solution, that met the structural performance targets. These solutions were assessed considering different possible manufacturing technology to ensure manufacturability of the sub-system.

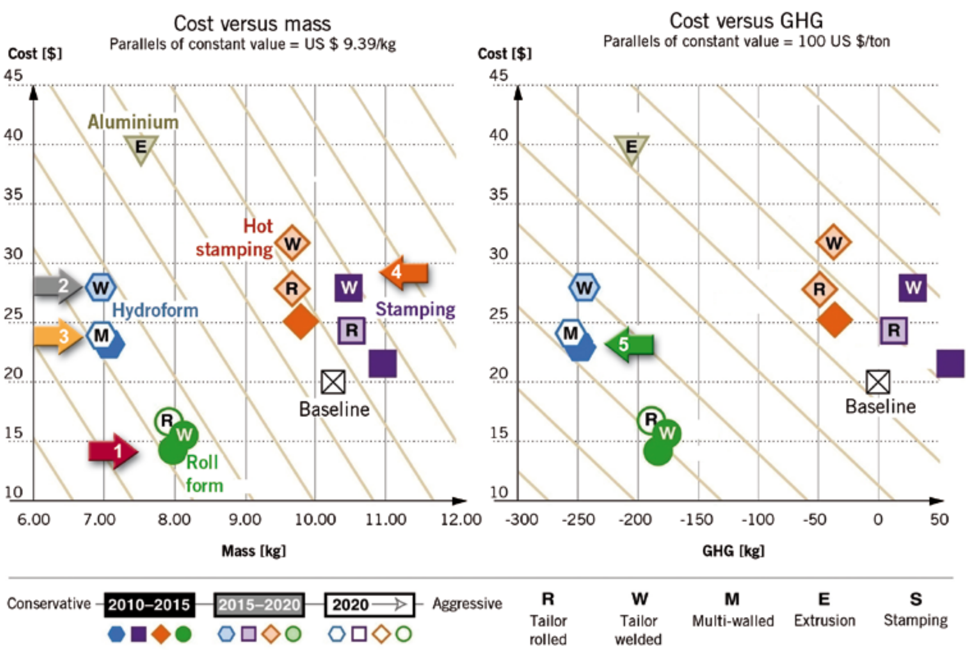
For example, the rocker sub-system model was optimised with three different AHSS steel solutions in combination with four manufacturing processes stamping, hot-stamping, rollforming, and hydroforming, ❹. Each of the twelve combinations for the rocker structure have equivalent in-vehicle performance. These parameters were used later as the input for the technical cost model to determine the sub-systems' manufacturing costs.

EVALUATION CRITERIA: WEIGHT, COST AND LIFE CYCLE

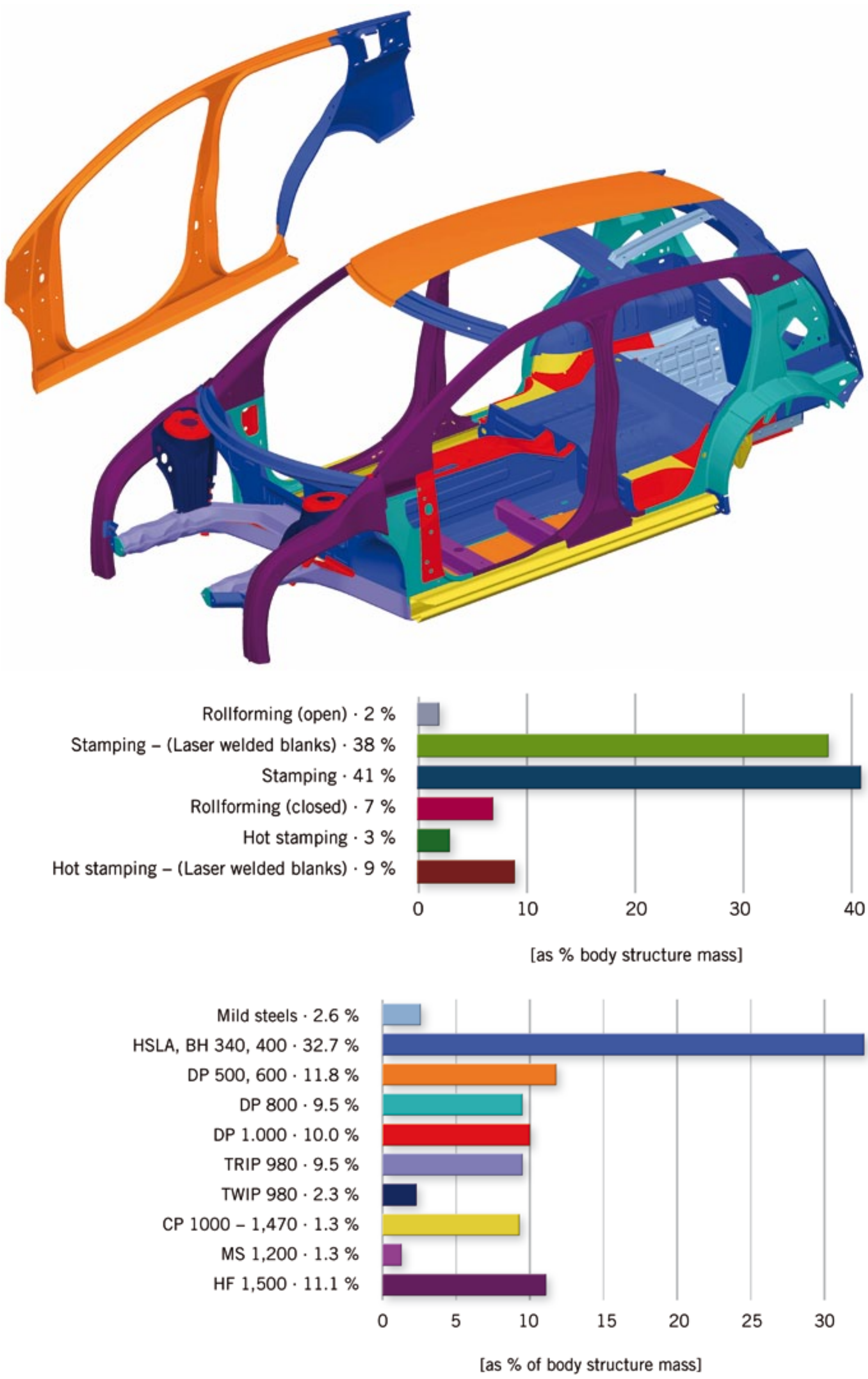
A Life Cycle Assessment (LCA) approach assists automakers in evaluating and reducing the total energy consumed and the GHG emissions of their products. The engineering team used this approach of the so-called UCSB GHG Comparison Model to ensure that FSV's carbon footprint was reduced over its life cycle.

The optimisation process resulted in a portfolio of solutions that demonstrate dramatically reduced mass and reduced GHG emissions in the seven optimised sub-system structures at lower or comparable costs to conventional solutions. The next step for the FSV was to select the most appropriate sub-system options from those developed through the design methodology based on the following factors: mass, cost, and LCA for GHG emissions.

❺ shows the rocker sub-system where the twelve solutions described in ④ are compared on a mass and cost basis,



❺ Rocker solution comparison: cost versus mass (left); cost versus greenhouse gas emission (right)



❻ Top: locations of the steel qualities used in the body-in-white of the FutureSteelVehicle (BEV); middle: manufacturing processes of the FutureSteelVehicle (BEV); bottom: steel qualities of the FutureSteelVehicle (BEV)

including a set of Iso-value lines, enabling evaluation of solutions relative to each other on a total vehicle manufacturing cost basis. Any solutions that fall within the same iso lines result in the same total manufacturing cost due to the off-setting reduction in powertrain costs.

Solid colouring in ⑤ refers to technology that is already in place today (2010 – 2015), shaded colouring indicates technology that is in development today and ready for implementation in the 2015 to 2020 time frame. And open shapes indicate technology that will not be ready until after 2020. By using this type of data, the engineering team can extrapolate solutions based on a range of design drivers, such as:

: lowest cost (rollforming, red arrow 1 in ⑤)
: lightest weight and therefore best fuel economy (laser welded hydroformed tubes, grey arrow 2)
: lowest total manufacturing cost and best fuel economy (hydroforming or hydroformed tailor-made tubes, yellowish brown arrow 3)
: reflects the existing manufacturer infrastructure (stamped laser welded blanks, orange arrow 4)
: contributes to the lowest carbon footprint (hydroformed tailor-made tubes, green arrow 5).

In the case of the rocker sub-system, there are a number of attractive steel rollformed options that are achievable, cost effective and excellent in terms of carbon footprint reduction. In addition, looking at the Iso lines, there also are hydroformed solutions that would meet the design targets. The diagrams in ⑤ are useful tools to allow comparison among the varieties of steel solutions which are given by the FSV.

BODY-IN-WHITE

The next step of the engineering team was to select the most appropriate sub-system options from those developed through the design methodology. The engineering team made these decisions based on the following factors:

: mass reduction
: cost: a technical cost modelling was applied to estimate the manufacturing cost of the sub-systems with the production variants
: LCA: an analysis of each sub-system's impact on the total LCA of the vehicle was conducted with the UCSB GHG Comparison Model.

The body-in-white of the FSV contains more than 20 different UHSS, AHSS and HSS but also highly ductile steel qualities, which will be entering the market from 2015 to 2020. The weight reduction was achieved by means of a rigorous load path design, material selection of the steel qualities, and design optimisation, ❻.

The following processing technologies, which have great lightweight design and cost potential, were utilised in the body-in-white, in the form of intelligent semi-finished products:

: conventional deep drawing (stamping)
: laser welded blanks
: flexible rolling (tailor rolled blanks)

❼ Expanded steel portfolio – white: steel qualities applied for Ulsab-AVC; grey: steel qualities added for the FSV

Mild 140/270	DP 350/600	Trip 600/980
BH 210/340	Trip 350/600	Twip 500/980
BH 260/370	SF 570/640	DP 700/1000
BH 280/400	HSLA 550/650	CP 800/1000
IF 260 /410	Trip 400/700	MS 950/1200
IF 300/420	SF 600/780	CP 1000/1200
DP 300/500	CP 500/800	DP 1150/1270
FB 330/450	DP 500/800	MS 1150/1400
HSLA 350/450	Trip 450/800	CP 1050/1470
HSLA 420/500	CP 600/900	HF 1050/1500
FB 450/600	CP 750/900	MS 1250/1500

DESCRIPTION	CLASSIFICATION	DESCRIPTION	CLASSIFICATION
Mild	Mild steel	HSLA	High strength low alloy
BH	Bake hardening	IF	Interstitial free
CP	Complex phase	MS	Martensitic
DP	Dual phase	SF	Stretch fangeable
FB	Ferritic bainitic	Trip	Transformation induced plasticity
HF	Hot formed	Twip	Twinning induced plasticity

: induction and/or laser welded hydroformed tubes
: hydroforming, including flexible rolled tubes (tailor rolled tubes)
: hot forming with direct and indirect press hardening
: roll forming.

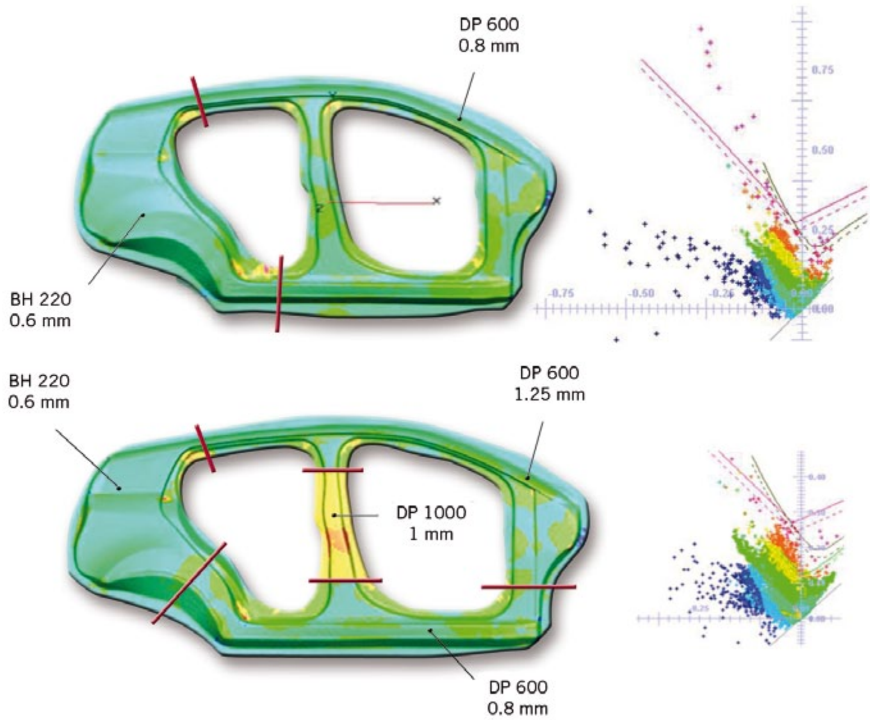
❼ shows the steel qualities selected and used in the FSV, and the changes brought about by new steel qualities (grey) which have been added since Ulsab-AVC (white).

One step simulation was done for all the parts of the body-in-white. Parts that play an important role in crashworthiness like B-pillars, shotguns and roof rails are made through a hot forming process. In that case, a one-step simulation with IF steel parameters was used. A total of nine very sophisticated forming parts were also incremental simulated, optimised and protected, ❽.

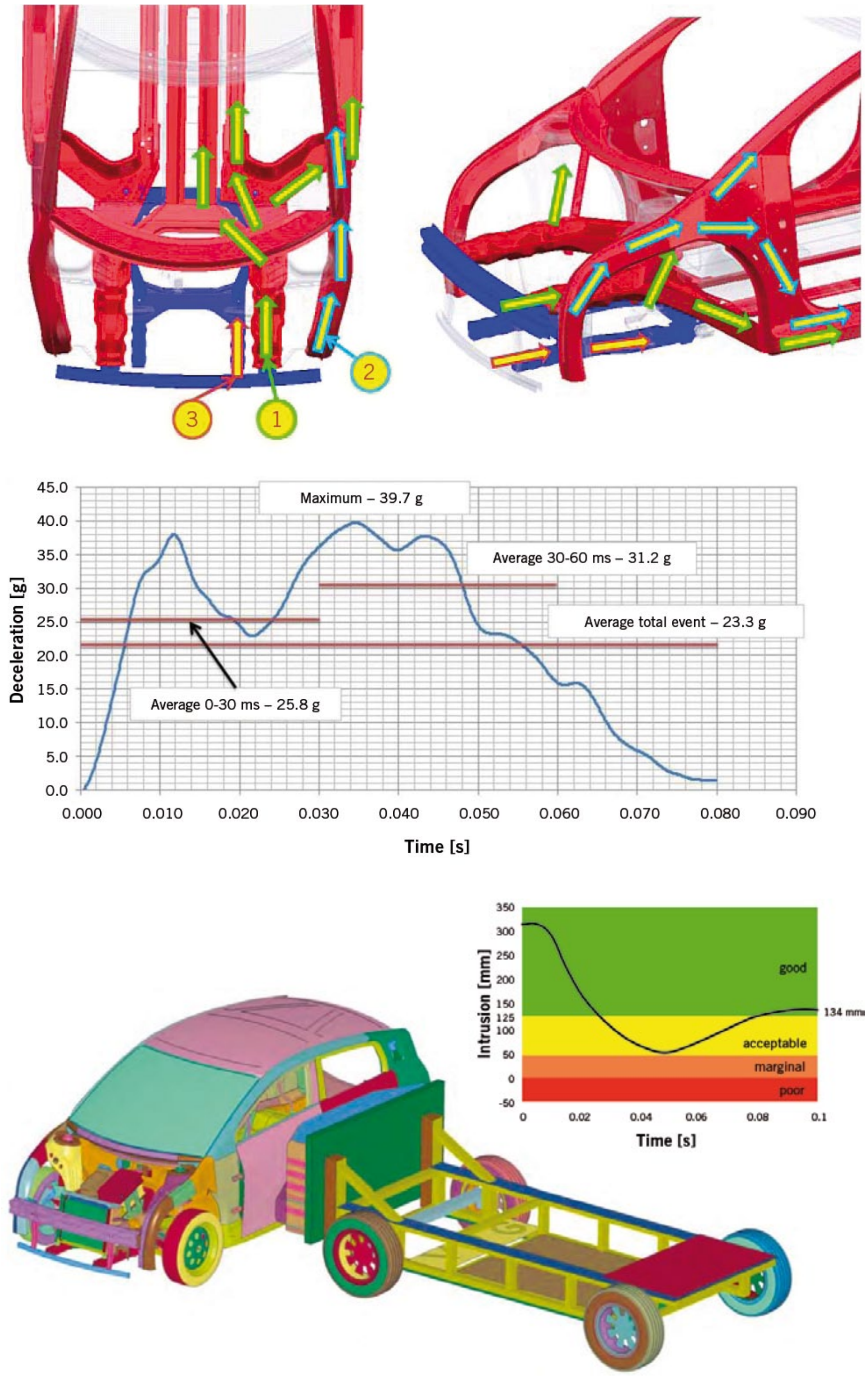
The joining technology is geared to today's state of the art technology:
: number of spot welds: 1023
: laser welding length: 83.6 m
: laser soldering length: 3.4 m
: roller hemming length: 2 m
: structural bonding length: 9.8 m.

Using the BEV as an example, data measured with the help of CAE are:
: body weight: 188 kg
: torsional stiffness: 19,604 Nm/°
: flexural stiffness: 15,552 N/mm
: 1st torsional eigen frequency: 54.8 Hz.

The highest development requirements were met by taking into account the selected crash load cases, which include the most stringent regulations in the world. In addition, five load cases from service strength and vehicle dynamics also were rated successfully.
: fish hook test, based on SAE 2003-01-1008 (extreme driving maneuver carried out to determine the tendency of a vehicle to overturn)
: double lane change
: 3 *g* pothole test
: 0.7 *g* constant radius turn test
: braking manoeuvre 0.8 *g* delay (forward braking test).



❽ Incremental forming simulations using the example of body side panels; comparison of body outer side: two blank solution (top) with 11.6 kg weight and 39 US Dollar manufacturing cost versus an four blank solution (bottom) with 13.9 kg weight und 61 US Dollar manufacturing cost



❾ Top and middle: load paths (BEV): front rails (1), shotguns (2) and motor cradle (3) and its US NCAP 35 mph front rigid barrier pulse in B-pillar region; bottom: IIHS side impact crash analysis and B-pillar intrusion diagram

The BEV front end takes full advantage of the smaller package space required for the electric drive motor as compared to a typical internal combustion engine and transmission package. The additional packaging space allows for straighter, fully optimised front rails with larger sections. The front rails (load path No. 1), shotguns (load path No. 2) and the engine cradle (load path No. 3) work together to manage frontal crash events with minimal intrusions into the passenger compartment. The front rail loads are managed by the V-shaped construction through the rocker section, base and top of the tunnel. To stabilise the rear of the front rails, an additional load path is introduced behind the shock tower to direct the loads into the base of the A-pillar. The deceleration pulse for the BEV (US NCAP 35 mph Rigid Barrier Impact), is shown in ❾. Crash-load cases and CAE results are shown in ❿.

The FSV side structure's design incorporates several load paths that take advantage of AHSS very high-strength levels. The B-pillar inner and outer as load path 1 are constructed from hot-stamped HF1050/1500 steel. Load path 2, which is the roof rail inner and outer, is also hot stamped. Through the use of hot stamping, complex shapes can be manufactured with very high tensile strengths (1500 to 1600 MPa). This level of strength is highly effective in achieving low intrusions into the occupant compartment and strengthening the upper body structure for rollover protection. The rocker, load path 3, with its unique cross section and CP1050/1470 roll formed steel, plays a major role in side impact protection, in particular for side pole impact.

WEIGHT REDUCTION AND LOW MANUFACTURING COSTS

Extensive use of high-strength and ultra-high-strength steels in the FSV means great stability and high energy absorption at low weight. To achieve this end, the 17 international steel producers involved have contributed their latest steel innovations. The effect is intensified by the use of processing technologies such as roll forming, flexible rolling or hot forming with press hardening, and further intelligent semi-finished products. All these measures reduce the weight of the BEV body to just 188 kg.

The FSV illustrates the fact that steel is the most economical material for car bodies. It can be made into sub-systems and a body-in-white at relatively low cost. The cost estimate of 1115 US Dollar for the body-in-white confirms this statement, and presents no cost disadvantage compared to today's vehicle bodies.

VEHICLE SAFETY

The FSV concept is geared to crash safety standards, which will become increasingly stringent over the next

LOAD CASE	TARGET	FSV RESULT
US NCAP	peak pulse < 35 *g*, footwell intrusion < 100 mm	Peak pulse 36.6 *g*, footwell intrusion 32.3 mm
Euro NCAP	Peak pulse (driver side) < 35 *g*, footwell intrusion < 100 mm	Peak pulse 32.2 *g*, footwell intrusion 90 mm
FMVSS 301R	Battery should remain protected and should not contact other parts, after the crash	Battery is protected and there is no contact with other parts, after crash
ECE-R32	Battery should remain protected and should not contact other parts, after the crash	Battery is protected and there is no contact with other parts, after crash
IIHS Side impact	B-pillar intrusion with respect to driver seat centreline ≥ 125 mm	136 mm
US SINCAP Side impact	B-pillar intrusion with respect to driver seat centreline ≥ 125 mm	215 mm
FMVSS 214 Pole impact	Door inner intrusion with respect to driver seat centreline ≥ 125 mm	173 mm
Euro NCAP Pole impact	Door inner intrusion with respect to driver seat centreline ≥ 125 mm	169 mm
FMVSS 216a and IIHS Roof	Driver and passenger side roof structure should sustain load > 28.2 kN within the plate movement of 127 mm (FMVSS 216a), > 37.5 kN (IIHS)	Sustains load = 45 kN for driver side = 43 kN for passenger side
RCAR/IIHS Low speed impact	Damage is limited to the bumper and crash box	There is no damage in components other than the bumper and crashbox

⑩ Crash-load cases and CAE results

THANKS

Edag would like to express sincere thanks to WorldAutoSteel and its member companies for financing the FSV project and for the years of reliable cooperation we have enjoyed. Also we credit the Bren School of Environmental Management at UCSB University of California Santa Barbara (USA) for their contributions in Life Cycle Assessment research and UCSB Greenhouse Gas Comparison Model. Further, our thanks are extended to the subcontractor partners LMS International, Inc. and Engineering Technology Associates, Inc.

The members of WorldAutoSteel are: Anshan Iron and Steel Group Corporation, Arcelor Mittal, Baoshan Iron & Steel Co. Ltd, China Steel Corporation, Hyundai-Steel Company, JFE Steel Corporation, Kobe Steel, Ltd., Nippon Steel Corporation, Nucor Corporation, Posco, SeverStal, Sumitomo Metal Industries, Ltd., Tata Steel, ThyssenKrupp Steel Europe AG, United States Steel Corporation, Usiminas, Voestalpine Stahl GmbH.

decade. It meets both European and US five-star safety performance requirements. Latest CAE methods permit the use of the most up to date optimisation tools, and with these, the FSV is more than able to meet even the most stringent crash safety requirements in the world.

LOWEST TOTAL LIFETIME EMISSIONS

Besides weight optimisation, cost reduction and functionality, the FSV produces lower CO_2 emissions (evaluated as CO_2 equivalent in kg CO_2), so 10 % less during vehicle production compared to benchmark, and significantly less when the vehicle is being driven, depending on the electricity source in use. In addition, steel is the world's most recycled material, with recycling infrastructures around the globe. The FSV project underscores the importance of a LCA in the vehicle development process.

"WE HAVE BROKEN THE WEIGHT SPIRAL"

A growing awareness of the finite nature of resources and the urgent debate about reducing CO_2 emissions are giving new impetus to the issue of lightweight design in vehicle development. In addition to the vehicle body, all other technologies are also being re-appraised. In an interview with ATZ, Dr.-Ing. Klaus Rohde-Brandenburger, Head of Passenger Car Performance/Fuel Consumption and Weight at Volkswagen AG, described the most important approaches.

Dr.-Ing. Klaus Rohde-Brandenburger (born in 1953) studied mechanical engineering at the TU Braunschweig. He gained his doctorate (Dr.-Ing.) in 1986 at the Institute of Internal Combustion Engines at the TU Braunschweig with his dissertation on "The Influence of the Ignition Angle on the Efficiency of a Multi-Cylinder Spark-Ignition Engine in the Transient Range". Rohde-Brandenburger began his career at Volkswagen in 1987 in diesel engine development. From 1989 to 1996 he was an employee and later sub-department head in spark-ignition engine development, before coordinating type support and test drives as head of the Complete Vehicle Development department from 1996 to 2001. Since 2001, Rohde-Brandenburger has been head of the Vehicle Technology department, where he is in charge of Vehicle Performance/Fuel Consumption and Weight for VW cars. His main responsibility is reducing the CO_2 emissions of the vehicle fleet.

ATZ _ Dr. Rohde-Brandenburger, with regard to the potential of lightweight design for reducing fuel consumption, in the 25 years that I have been reporting on the subject, figures of between 0.3 and 0.7 l/100 kg and 100 km have been mentioned. The figures for reductions in emissions also vary accordingly. Do you have an explanation for these different figures?

ROHDE-BRANDENBURGER _ We have also come across figures of up to 1 l/100 kg per 100 km. Unfortunately, such statements are often made without further explanations. It remains unclear, therefore, which driving profile they are being applied to or whether they refer to diesel or petrol vehicles. Furthermore, there is frequently a lack of a physically correct consideration of this issue, with the result is that it is very uncertain which figure is correct. In my opinion, one reason is that, in the case of approximate estimates, the overall fuel consumption of a vehicle is divided up precisely in proportion to the driving resistance components. For a compact-segment vehicle with an unladen weight of around 1200 kg, the driving energy requirements in the NEDC are divided up very roughly as follows: 35 % for aerodynamic drag, 30 % for rolling resistance and 35 % for acceleration resistance. In other words, approximately 65 % of the driving energy required is proportional to weight. This leads some authors to erroneously believe that 65 % of fuel consumption is also dependent on weight. For an NEDC fuel consumption of 6.2 l/100 km for this sample vehicle, this leads to the false assumption that approximately 4 l/100 km is caused by weight, thus resulting in 4.0:12 = 0.33 l/100 km per 100 kg. It is also supposed that real-world driving situations must necessarily cause significantly higher values due to stronger acceleration phases, and that larger engines and vehicles must have higher fuel consumption values. This then leads mistakenly to the “supposed” range of 0.3 to 1.0 l/100 kg per 100 km.

In your paper at this year’s Hamburg Conference for Body Engineering organised by ATZlive, you want to present a figure that is generally valid. What is your approach towards achieving this?

In the simple but incorrect approach mentioned above, two facts are left out of the equation. Firstly, when the vehicle is decelerating, the driving energy initially required to overcome drag and rolling resistance is recovered from the kinetic energy of the vehicle mass by a kind of natural recuperation. Secondly, the energy consumption of the powertrain itself is neglected. If, in the case of the sample vehicle mentioned above, we set all driving resistances theoretically at zero and allowed the engine alone – with an open clutch and in idling operation – to drive through the speed profile specified in the NEDC, its fuel consumption would still be approximately 2.8 l/100 km. That is still 45 % of 6.2 l. The remaining 55 %

“The main levers are the engine and vehicle weight.”

can be roughly divided up into approximately 5 % (0.3 l/100 km) for the electrics and other components and around 50 % (3.1 l/100 km) to overcome the typical driving resistances. Only these 3.1 l/100 km can also be converted into the partial fuel consumptions for driving resistance in accordance with the percentage driving resistance division. The fuel consumption for the 65 % weight proportion of the driving resistance is therefore around 2.0 l/100km (65 % of 3.1). If one then also takes into account that the NEDC specifies a reference mass of “unladen weight + 100 kg” (1200 + 100 = 1300 kg) for the cycle, this results approximately in a fuel consumption of about 2.0:13 = 0.15 l/100 kg per 100 km. This approach is based on the model of constant differential engine fuel consumption efficiency, and will be explained in more detail in my paper.

What are the main levers and their effects?

There are three main levers, which altogether make up around 90 % of the entire fuel consumption in the NEDC. The first is the engine itself, which, as I mentioned earlier, already accounts for 45 % of the fuel consumption even without the vehicle and without driving resistance. The second main cause of fuel consumption is the vehicle mass, which, in our sample vehicle, makes up around 33 %. The third main cause is drag, accounting for approximately 16 %. The remainder is made up of electricity generation and other factors. Therefore, the main levers are the engine and vehicle weight, if one takes the NEDC as the basis. For a high-mileage driver on the motorway, aerodynamic drag plays a much more important role, of course. Above 130 km/h, drag is already responsible for more than 50 % of fuel consumption.

How cost-effective is lightweight construction compared to measures in the fields of aerodynamics, powertrain technology or reductions in rolling resistance?

The cost-effectiveness of lightweight construction is greatly dependent on the type of lightweight construction applied and the production volumes. For example, high investment in high-strength steel, for example, may certainly be economically viable if production volumes are high enough. There are also examples of intelligent lightweight design that

Rohde-Brandenburger (left) talking to ATZ chief correspondent Stefan Schlott

can even cut costs. That is the case when components can be made lighter due to optimised calculation methods or when several functions can be integrated into one component. Today, all measures to reduce fuel consumption are subjected to a strict comparative evaluation. This is done by using fuel consumption characteristic figures that enable the design engineer, for example, to precisely evaluate a reduction in drag compared to a reduction in weight. Nevertheless, there is no generally valid answer to the question of which costs of reducing fuel consumption are the lowest for which factors that cause fuel consumption.

Against this background, is it true to say that lightweight design is overrated?
Quite the contrary: lightweight design is highly rated, and justifiably so, for two reasons. Firstly, because it has a positive influence on fuel consumption. Secondly, weight has a direct influence on driving performance, thus enabling further savings in fuel consumption to be achieved by secondary effects such as longer gear ratios or the use of smaller engines. Lightweight design is therefore the key to reversing the weight spiral and to further downsizing.

How much is a weight saving of 1 kg allowed to cost today and to what extent has this figure changed?
There is no clear answer to that question. Different projects always have different cost targets. The commonly cited figures for the automotive industry range from 2 to 10 Euro/kg depending on the vehicle segment and vehicle performance requirements. As a comparison, such figures can be calculated from the penalties specified by EU legislation from 2015, which can amount to as much as 95 Euro for every 1 g/km of CO_2 in excess of the limit. If the Fuel Reduction Value (FRV) of 0.15 l/100 km per 100 kg is applied, that would equal around 3.30 Euro/kg. In the case of greater savings in weight in the vehicle, the FRV can rise to 0.35 l/100 km per 100 kg as a result of secondary effects, which is equivalent to approximately 7.80 Euro/kg. It is important to note that these are overall costs, including all expenditure. The pure material cost must be correspondingly lower.

What has become of Volkswagen's plans to use magnesium?

Klaus Rohde-Brandenburger: "There are examples of intelligent lightweight design that can even cut costs."

Volkswagen has been familiar with the use of magnesium as a lightweight material for a long time, although it is often the additional costs that are the limiting factor. Currently, around 2300 gearboxes with a magnesium housing are produced every day. Various other parts, such as the steering wheel skeleton, are also used in large-volume series production. And the trend is continuing upwards.

"Lightweight design is justifiably highly rated."

On the subject of carbon fibre: where do you see opportunities and limitations, also with regard to costs?
In addition to aluminium and ultra-high-strength steel, the greatest potentials for lightweight construction lie in carbon fibre composites and magnesium. Carbon is a very interesting material for lightweight construction. However, its use in large-volume series production depends on the development of cost-effective production processes and a reduction in the cost of the raw material. Volkswagen is advancing the development of these technologies. Currently, carbon only plays a special role in niche projects such as the XL1 and in motor sport due to the high material and processing costs.

What possibilities do you see to break the weight spiral in the introduction of new models?
Today's vehicle bodies are already significantly lighter than those of the predecessor model in each case. The sustainable objective of the Volkswagen Group is to reverse the weight spiral with no impairment of safety or comfort in the vehicle. For the new Golf on the Modular Transverse Matrix (MQB) platform, we have achieved the same weight level as the Golf IV with comparable engines while meeting higher demands regarding safety, emissions and equipment levels. We have therefore already broken the weight spiral.

Dr. Rohde-Brandenburger, thank you very much for this interview.

INTERVIEW: Stefan Schlott
PHOTOS: Stefanie Löhr/Volkswagen

INTEGRATION OF B-PILLARS INTO THE SIDE DOORS OF THE FORD B-MAX

It is especially difficult to combine modern, sporty styling with easy and flexible entry along with a high standard of utility specification in a small car. Ford is losing this knot for his minivan B-Max with a body which needs no B-pillar and has useful sliding doors. Thus, the car is uncomplicated to use without neglecting vehicle safety and body stiffness.

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FOR URBAN USE

By many utility optimised cars there is a real compromise made with the external styling. The positioning of the rear door opening line is limited due to vehicle dimensions such as wheelbase, reducing egress ingress comfort of the passenger, who needs to sidestep the lower B-pillar. Additionally, access is restricted when loading from the side of the vehicle – for example when installing a child seat. It is worth noting that these cars are designed primarily for urban use, where narrow parking places are the norm. That the second row doors cannot be fully opened, reduces the access even further.

The objective from the beginning of the Ford B-Max project was to prioritise the entry/exit as well as general vehicle access without compromising the requirements of a modern small car. The analysis of this concept is detailed below. Solutions for the essential vehicle basics such safety, rigidity, locking, sealing and weight will be reviewed. Following the solutions for essential vehicle attributes as safety, stiffness, latching, sealing and weight are considered. As you can see there are varied creative and innovative challenges.

MARKET ANALYSIS AND CHOICE OF CONCEPT

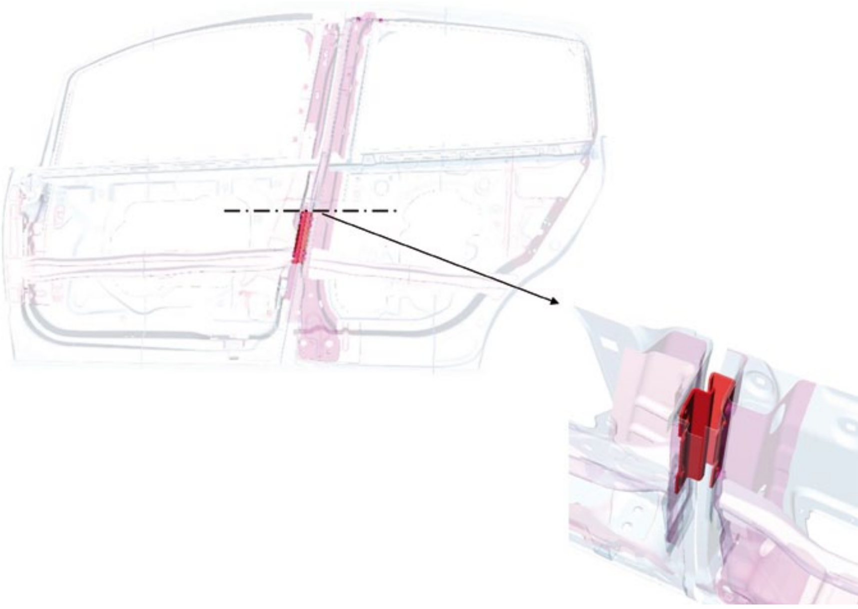
It was immediately obvious that we should focus on side vehicle access, primarily not having a B-pillar in the body side. The necessary functionality of it should be integrated into the door design, without accepting any functional compromises.

Investigating available solutions can be summarised by two main latching systems. On the one hand there are existing vehicles having no B-pillars with the doors front and rear oppositely hinged and therefore with opposing opening swing. One significant characteristic is that the doors do not operate independently of each other. In other words, the rear door cannot be opened or closed, while the front doors are closed. The doors are connected with a conventional latch system to prevent wide opening margin gaps of the doors in the event of a crash. Additionally some vehicle rear side doors using this concept are narrow in width. The rear door uses a twin latch system, which engages it to the roof frame and the side sill. This concept is commonly used in various European and North American market segments.

On the other hand there are vehicles without a B-pillar on one side only in combination with rear sliding doors. An alternative concept contrary to that already mentioned is to operate both doors on one side independently of each other. In order to achieve this, additional to the conventional sliding door C-pillar latch, a complicated latch connecting rear and front door is required. This would need to be activated independently from both doors or it needs to lock automatically at the start of the journey and unlock automatically when the journey ends. This concept is only available in the Japanese market at present. The basic challenge is to avoid the disadvantages of the currently available production solutions. These lack independent operating ability and have a complicated latch system but in doing so identify and maintain function along with a smooth opening position and operation of a rear sliding door.

BODY WITHOUT B-PILLAR

For solving the challenge, crash catchers, ❶, are added to a conventional sliding door latch system. The doors are coupled



❶ The vertical crash catcher system delivers an safe coupling of the doors in crash events to each other and the sliding door is coupled to the body of the vehicle. The front door latch system uses a latch at the side sill and another at the roof frame, which operate independently from the sliding door. To prevent separation of front and rear doors, hooked vertical crash catchers of length 300 mm are used. This concept robustly fulfils various side crash load cases and sliding door ECE requirements.

To prevent the crash catchers from warping during crash, they have a large surface and are mounted to large ultra-high-strength door structure reinforcements. These reinforcements bridge the midway mounted crash catcher with the lower and upper area of each door, and thereby create an integrated B-pillar. Additional hooks are used in the lower and upper track of the sliding door and are supported by the body structure. These hooks are designed for handling interior and exterior loads also.

A further essential element of the concept is to use the front seats with integrated seatbelts. It is known from some convertible vehicles that this is a key enabler to provide unhindered access to the second seat row.

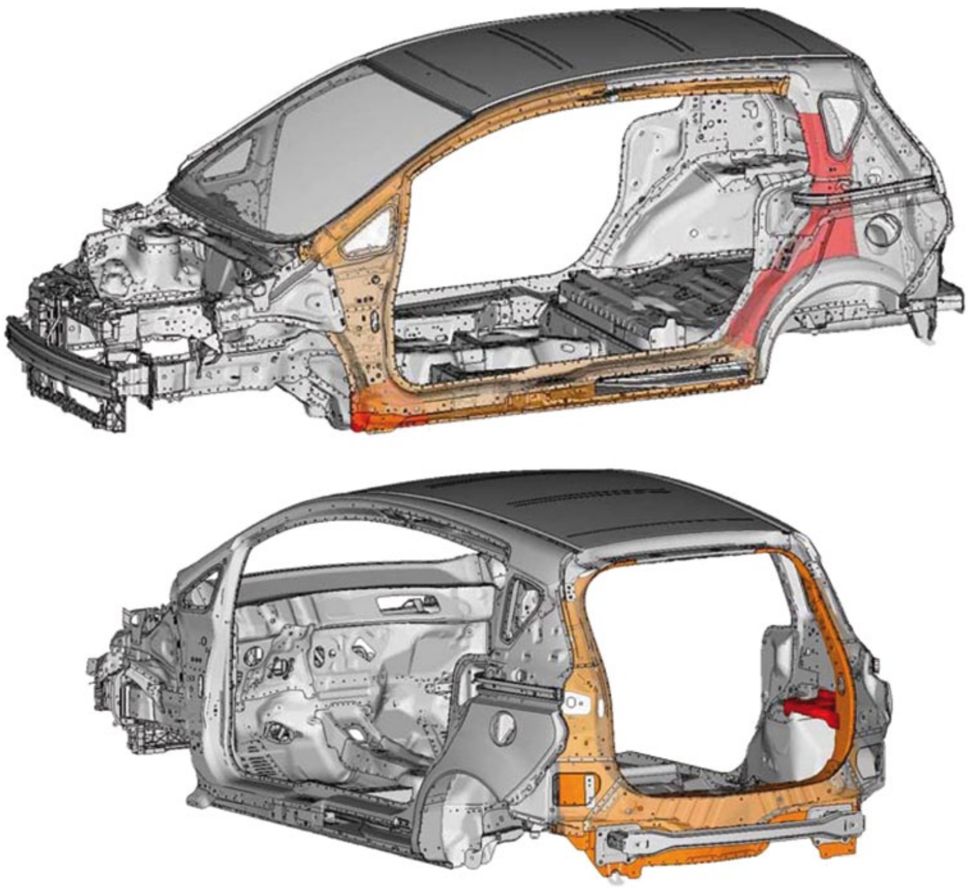
PASSIVE VEHICLE SAFETY

The choice of body side and door system is a complete new design with significant functional improvement for the vehicle side door access. Nevertheless all requirements regarding occupant safety are fulfilled or over achieved. Targeting a five star Euro NCAP test results is the logic consequence of this project. During the development process detailed CAE models were set up and permanently updated. Component, system and vehicle level tests were performed during the whole development process to achieve robust predictions based on the virtual simulations.

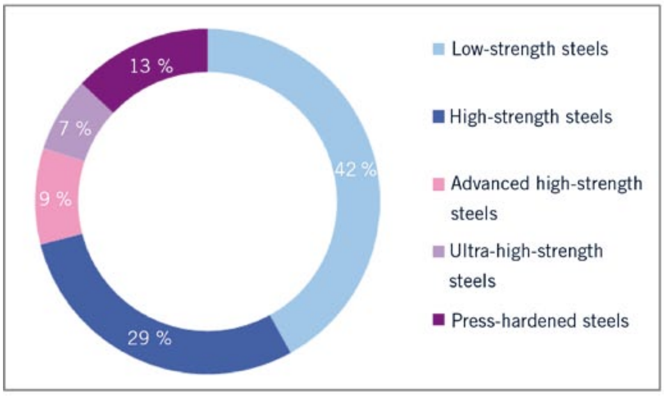
To ensure dependable test results several load cases varied in parameters like velocity, mass or crash angle based on the standard load case simulation. In particular, it is imperative to integrate the rigidity of a conventional B-pillar in the doors during testing of side crash conditions without having to accept any decrease in functional performance. Especially for side crash it is a challenge to integrate the rigidity of an conventional B-pillar in the doors without having to accept functional compromises. It is very important that the coupling of the doors to the body structure in order to minimise intrusion on depth and speed.

The required space for passenger safety is then achieved, along with sufficient space and time for airbag deployment. Airbag deployment activation is desired as early as possible. The use of an air pressure sensor in the front doors ensures quick detection of a side impact is supported effectively.

Just to substitute the state-of-the-art high-strength B-pillar reinforcements with similar reinforcements in the doors is not sufficient, because the doors would be pushed to far into the occupant safety space. The front door needs to be braced to the rear door in addition to its roof frame and side sill latches to prevent this happening. Above that any loads need to be transferred into roof frame and side sill. This bracing between the doors and between door



❷ Good joint reinforcements around door and liftgate ring combined with two additional parts deliver an excellent torsional stiffness



❸ Steel grade mix in the structure of front and sliding door

and body crash catchers, would have a more strength structure than conventional latch claws. To evenly distribute the crash energy within the big doors and to prevent accidental actuation of the lower latch cables, the side impact reinforcements are made out of Ultra-High-Strength Steel (UHSS).

At the point of pole impact the special focus was on the latch system, because it is packaged in an area that is subject to substantial deformation. To join front door and side sill a crash catcher is used in the area of the lower latch with which a homogenous deformation between door and side sill is assured to allow vice versa for the latch functionality.

The rising shape of the upper A-pillar, creates a vertical force component which needs to be handled by the high-strength reinforcements in the doors and in the latch system as well. This will ensure that an unacceptable increase of the margin gap between doors and body is created.

BODY STIFFNESS

A good balance between comfort and handling is a key ingredient of the distinctive brand behaviour of every Ford vehicle. Bearing this in mind, it was essential to ensure the required body structure performance as well for this new concept. Investigation during the concept phase of the project demonstrated that by simple deletion of the B-pillar the static torsion stiffness is reduced by 15 % as the vertical bending "eigenfrequency" is reduced by 20 %.

Several simulations identified and ensured that the load paths and joint design of door and liftgate ring were optimised, ❷. It should be underlined, that the negative effect of a missing B-pillar regarding body stiffness could have been compensated without adding numerous additional parts. Just at the lower A-pillar and C-pillar two additional reinforcements are dedicated for this function. Finally it was achieved that the Ford B-Max is even a bit more rigid/stronger than the current Ford Fiesta (model year 2012).

DOOR DESIGN

It is intended to position the B-Max as a premium small car in the market, the sliding doors are equipped with permanent power supply, speaker system, full-size door trim with integrated armrest and powered window regulators.

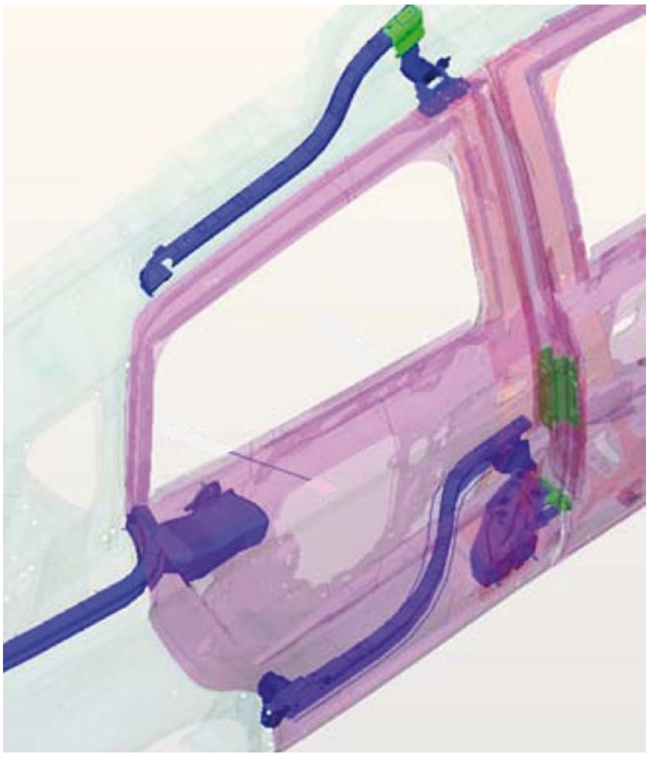
To achieve the required load levels from the side impact simulation within the doors, the B-pillar was cut midway and the created parts were added to the front end of the sliding door respectively to the rear end of the front door. Likewise state-of-the-art B-pillar reinforcement's hot formed Boron UHSS is used. ❸ shows the steel grade mix in the structure of front and sliding door with 20 % UHSS fraction in the doors (7 % UHSS and 13 % press-hardened Boron steel).

In the front door the main reinforcement is backed up by another reinforcement made out of high-strength DP600 steel. This reinforced design ensures that the lower latch system is well protected in case of a pole impact. For laminar support of the outer panels the side impact beams are made out of martensitic UHSS. Due to the increase door weight of approximately 9 kg the rigidity/strength needs to be increased significantly as well. For this reason all hinge and roller reinforcements are made out of high-strength DP600 steel. In addition the joint design of these reinforcements is optimised for every applicable load case and enables the same outstanding stability and durability as every Ford vehicle delivers.

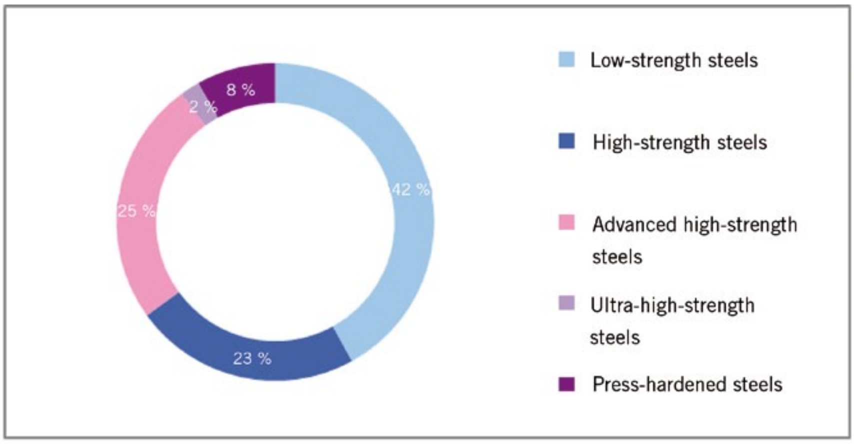
SLIDING DOOR KINEMATICS

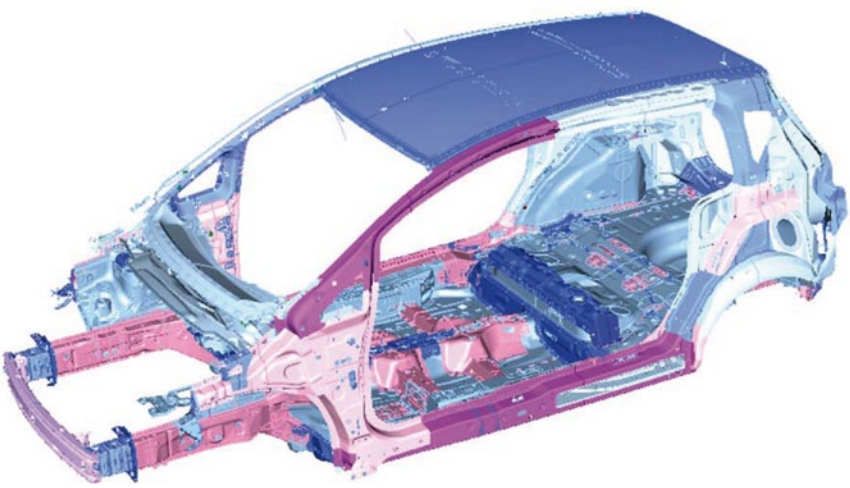
In order to achieve optimum overlapping conditions of the chosen vertical crash catcher concept, during either front or side impact, the S-shape design was developed for the slide mounting of the sliding door, ❹. At this the sliding door already reaches its lateral position 80 mm before final closure. Hence an optimum overlapping combined with minimum distance of the hooks is achieved.

The door function is designed in a way, that by opening the front and rear doors a total opening width of 1,5 m is offered to the customer. Therefore excellent vehicle access is possible in the first as in the second seat row.



❹ The S-shape slide mounting supports independent operation of the doors and enables the body system of the integrated B-pillar





❺ Steel grade mix in the body structure

STYLING

To ensure key Ford Kinetic styling features are met, proud wheel arches and a lowering roofline along with an upper roller with additional degree of freedom by an integrated hinge pivot is used. However, the roofline could be lowered an additional around 25 mm and the door can be opened more in the lower area than in the upper area to pass the wheel arch. So the upper track is curved not only in the top view but also in the side view.

The side view tilting of the tracks and the lateral travel differences in top view are designed in a way so as to achieve the best balance between opening and closing operations. To avoid uneven force peaks in the opening operation the S-shape is designed to be as smooth as possible so the inclination of the door centre of gravity is even and level.

SEALING SYSTEM

A unique challenge of this concept is the achievement of a robust sealing function between one fixed and two movable elements. This function is mainly covered by circular margin gap sealing. The gap between both doors is closed by a double circular sealing ring mounted at the front face of the sliding door.

To achieve maximum strength despite non avoidable manufacturing tolerances a smart drainage system behind the first and the second water barrier was installed. Therefore water collection in the area between the sealing paths is counteracted. In this way the water is then routed outboards.

Due to the S-shape kinematics of the sliding door an increase of friction at the sealing is created. The related effect regarding closing function and durability is balanced by the utilisation of friction optimised coatings.

WEIGHT

The chosen concept requires significant more efforts for the sliding door passage along with door reinforcements similar to reinforcements of the under body and seat structure. By using a particular very high-strength material mix all over the body components the incremental weight for the entire vehicle was limited to 95 kg, ❺.

❻ In addition to the well-known attributes as driveability and safety the new Ford B-Max offers an new kind regarding passenger access

In total 58 % of the structural weight of the body was of high-strength steel, only 42 % are low-strength steels. Due to the key role of the side doors for crash energy management a portion of 20 % ultra-high-strength steel with a tensile strength above 1000 MPa is used. More than the half of it is hot formed Boron steel. Both are of an excellent standard in this segment and they are fundamental to the efforts to create an efficient body.

The new Ford B-Max, ❻, offers in addition to the well-known positive attributes as versatility, driveability and safety an new dimension regarding vehicle access. This unique combination of vehicle features has been achieved by the development for a body and door system by a variety of innovative solutions described here.

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THANKS

The development of a new passenger car like the Ford B-Max with an innovative door system is only possible as a team work. The author says thank you to the following persons at Ford-Werke GmbH for the assistance during creating this trade article: Thomas Müller, Christian Morgenstern, Dr. Robert Spahl, Jörgen Hilmann, Thomas Benderoth, Sönke Kleinert and Uwe Bülow as well as Peter Maher for his help of the English translation.

USER-CENTRED DESIGN APPROACH FOR AN INNOVATIVE HMI CONCEPT

The OEM Seat has designed and developed a new concept of an in-vehicle Human-Machine Interface (HMI) which features multiple systems, integrated in a single unified interface, and allows for tactile interaction. In cooperation with the Spanish Research and Development Centre in Transport & Energy, Cidaut, a user-centred design (UCD) approach has been applied for the evaluation of the concept's usability and its re-design according to the user needs.

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CONCEPT

A new concept of an in-vehicle Human-Machine Interface (HMI) has been designed and evaluated with the joint effort of Seat and the Human Factors Group within Cidaut Research Centre (Fundación para la Investigación y Desarrollo en Transporte y Energía). This new HMI concept is relying on two main features: the integration of multiple systems in a single unified interface, and the use of a tactile modality for driver-system interaction. This innovative interface integrates diverse functionalities ranging from entertainment to communication, such as radio, multimedia, air-conditioning, phone, car and set-up. In addition, it makes use of a tactile interface to allow for introducing information and acting on the system by touch, thus availing itself of the growing presence of tactile solutions in a wide range of everyday interactive products such as smartphones, cash dispensers and navigation systems.

In particular, this article is focused on the assessment process conducted on a functional prototype of the system by both, involved experts and a sample of users in a set of static and dynamic tests, aimed at exploring usability concepts and identifying improvement opportunities regarding the interface design.

RATIONALE

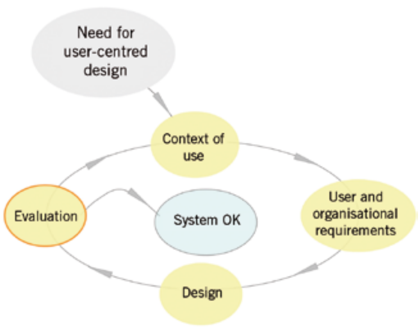
The transfer of modern information and communication technologies to on-road transport gives drivers the option to access various functions and services that have a strong potential to enhance driver safety, mobility, enjoyment, and comfort. Touch screens allow for controlling much information within a limited space. The high immediacy in the interaction between the finger and the device leads to better user acceptance, ease of use and a faster input rate [1].

As the use of the most diverse forms of technologies in cars increases, the Human-Machine Interaction is becoming more and more complex in today's vehicles. In-vehicle systems are thus visually, cognitively, and physically challenging which may lead to driver distraction, especially if the information is not properly managed or not presented to the driver in an adequate manner. The integration of multiple functions in a single system has already been evaluated in the past, for example in the European project Aide (Adaptive Integrated Driver-Vehicle Interface) [2, 3].

Touch screens are limited in several respects. One of these is that finger and hand partially cover the screen and may cause problems to detect relevant information. Another drawback is the lack of tactile feedback when interacting with touch screen which, however, can be balanced otherwise (acoustically, visually). Distraction and workload, though, are the key aspects to be taken into consideration. An easy-to-use interface combined with a high-end sensitive touch screen minimises potentially distracting situations.

In the system development, a User-Centred Design (UCD) methodology was followed that focuses on the users and involves them at various stages of the product development cycle. This means that potential system users were selected for design and evaluation purposes by applying different techniques such as questionnaires, laboratory, and on-road tests.

As the ergonomics of Human-System Interaction are defined in ISO 9241-210: 2010, [4], the Cidaut team referred to these data



❶ UCD process according to ISO 9241-210

in the UCD process, thus creating an iterative process in order to find the optimum solution that satisfies user needs and system requirements most appropriately at the same time, ❶. According to this standard, usability can be defined as the effectiveness, efficiency, and satisfaction with which specified users achieve specified goals in particular environments. So, usability can be understood as the measure of quality of a user's experience when interacting with a product or system. This offers great advantage to developers as it may result in increased productivity or higher customer satisfaction.

There are several methods to study usability. Usability testing is a technique used in UCD that involves user tests evaluating how easily the product can be handled and whether it meets its intended purpose [5]. In contrast to that, there are usability inspection methods carried out by experts evaluating a user interface without the participation of any users.

NEW HMI CONCEPT AND ASSESSMENT OBJECTIVES

The new HMI concept consists of a touch screen and a number of buttons grouped around it, some give access to certain functions and there is, of course, also an on/off switch. This concept was analysed in an iterative process based on expert and user evaluations in order to find out how the system can be further improved. The usability study conducted by Cidaut paid special attention to the system's understandability and ease of use since any driver's cognitive and physical resources need to be focused on driving as the primary task.

In particular, the main areas of interest were:
: the philosophy, usability and intuitiveness of the new concept
: interferences with the driving activity (perceived safety and distraction)
: its aesthetics and graphic design.

METHODOLOGY

The UCD method that was applied to test and evaluate the system and its design has four stages:
1. definition of purpose and context of use, first design
2. expert analysis (by heuristic principles)
   : design (paper prototype)
   : navigation (functional prototype)
3. redesign and first evaluation by users
4. system fine-tuning and second evaluation by users.

The initial phase, carried out by Seat, defined the purpose and context of use. Based on that, the concept design and first prototypes were generated, ❷ and ❸. It is to mention that in this scenario, there were some additional particular requirements: on the one hand, the HMI design should follow the OEM's guidelines and on the other hand, the selected technology required specific technologies and functions. These fix limitations needed to be respected when the proposed improvements were to be implemented.

Next, an expert analysis of the preliminary design (paper version and PC software) was carried out by a multidisciplinary team in Cidaut. In this evaluation, heuristic principles were applied in order to identify system interaction problems.

In stage three, a group of 41 drivers tested the system in a real environment. Specifically, the users recruited for the tests were selected by the driver profile of a defined vehicle model taking into account the distribution according to gender (both male and female) and age (< 40 and ≥ 40).

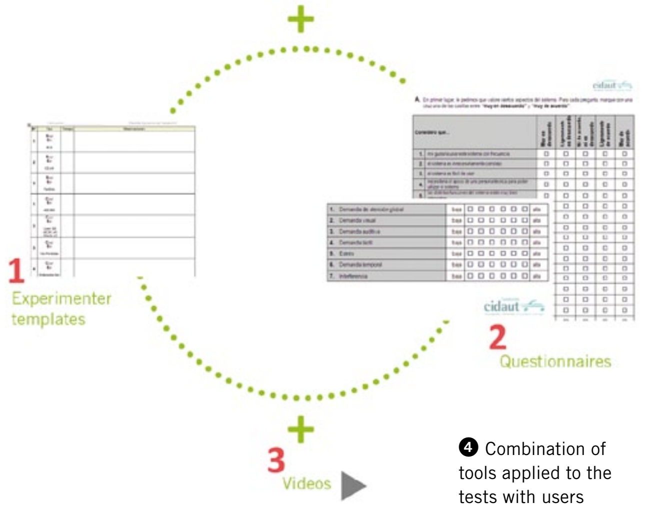
As there were time and resource constraints in stage four, only 20 users were recruited. They were a selection from the participants in stage three who were supposed to analyse the interface improvements that had been implemented after the first user evaluation.

In both user evaluations, the system had been installed in a real vehicle that was driven by the test persons. Furthermore, considering the particular objectives and characteristics of the study, the need of having different types of tools was highlighted, ❹:
: The experimenters had to take down their observations about the interaction with the system as well as task completion times and error rates in templates.
: Users had to answer questionnaires about their opinions and feelings at different test situations, ❺.
: The testing procedure was filmed in order to register any system interaction during the execution of different tasks along the trial together with what the participants verbally uttered.

The combination of these three information sources (ranging from subjective to objective variables) provided a rich data set that allowed for drawing relevant conclusions.

❷ Logical interface development

❸ Graphic interface development



❹ Combination of tools applied to the tests with users

❺ Example of a questionnaire filled in by users during the tests

RESULTS AND COMMENTS

The summarised results of the first and second evaluation focus on perceived usability, user uptake, workload, task completion times and error rates.

Firstly, the users perceived the system as easy to handle, easy to learn and to remember, and they had a positive opinion on the system itself. The visual interface was seen as adequate and nice, and the fact that there was haptic interaction was also considered appropriate.

As for user uptake, the participants rated the system as useful and satisfactory; although they indicated that they would not use the system so frequently and, thus, did not show a strong willingness to have it in their cars.

Concerning workload, the system was considered increasing mental workload, perceptual workload (visual and tactile), and stress in comparison to normal driving activity when no system is used. This result was understood as normal since any system used while driving is considered to be a task secondary to driving itself, somewhat increasing the workload level.

When it comes to task completion times, the radio subsystem was the one that required longer task durations as the users, who had to search for a particular station, had some difficulties in finding out how this worked. Also, high task completion times occurred in the car sub-system. However, a learning effect was observed when completion times were diminishing over time, after the task had been carried out several times; the same happened with the other sub-systems. The multimedia and phone sub-systems did not require long interaction and were simpler to use. A similar distribution occurred in the number of errors per task, thus turning out the radio and car sub-systems the most error-prone ones.

In total, the new HMI concept was assessed positively and its evaluation under real circumstances made it possible to identify specific interface areas that required revision or special attention. There was, however, already observed a significant improvement in the second round of usability tests, after some modifications had been implemented in the system. At the end of the process, HMI recommendations and guidelines were formulated as improvement opportunities to be taken into account in the next phases of system development. This way, the applied UCD method demonstrates how a system can be optimised by following an iterative improvement process and involving potential users at the various stages of product development.

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DAIMLER'S NEW EXTERIOR NOISE TEST RIG FOR ALL-WHEEL-DRIVE VEHICLES

The legally prescribed exterior noise measurements for passenger vehicles can be simulated on exterior noise test rigs, which are among the most complex and sophisticated pieces of test equipment due to the dimensions of the necessary noise measuring room. At the start of 2012 in the Mercedes Technology Centre of Daimler AG in Stuttgart-Untertürkheim, an acoustics test rig like this was modernised and re-commissioned, and the newly developed wedge lining is setting new standards in appearance and acoustics.

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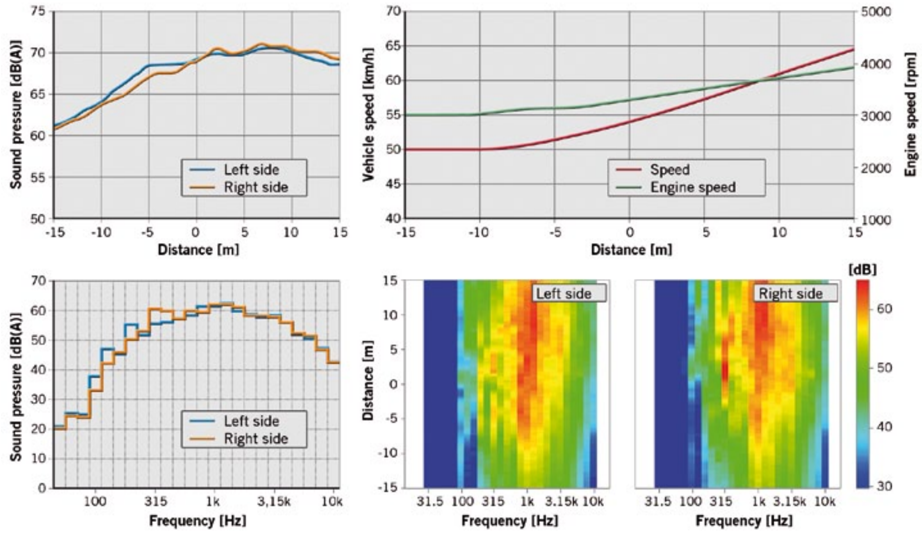
❶ Close-up of the wedge in the measuring room



❷ Sample measurement as per ECE 51.02

HISTORY AND REDEVELOPMENT

As part of its development processes, Daimler AG has for the past 30 years been using exterior noise test rigs to tune the acoustics of its Mercedes-Benz passenger cars. The first exterior noise test rig for Mercedes passenger cars entered service in 1981. This test rig was used in the development of the 190 D, which was dubbed the "whispering diesel" and featured the world's first passenger car with a completely enclosed engine. Even back then, environmental compatibility was at the forefront of this vehicle development offering low fuel consumption, low emissions and low noise levels without compromising performance and comfort.

This first exterior noise test rig was specially designed for vehicles with just one driven axle. This concept, however, no longer meets today's requirements. The test rig needed to be modernised – without altering the existing room geometry – to meet the requirements of two-wheel-drive and four-wheel-drive passenger cars and alternative drive concepts. Given the spatial restrictions, a highly innovative solution was required for the sound absorbers. Even at first glance, you can tell that a whole new approach has been taken. The sound absorption wedges for the ceiling and walls now feature a meshed metal surface, ❶, an innovation developed by Akustikzentrum GmbH.

This solution proved ideal for the following three reasons:

: acoustics: larger absorbing surface than with perforated plates
: appearance: precision design without visible structural components
: long-term effects: easy to change glass-fibre fleece.

Development initially focused on the acoustics. Wedges consisting solely of insulating material (glass or mineral wool) without cladding would be ideal, but for practical reasons (protection on contact) and to prevent fibres from escaping, the wedges were, up to about 15 years ago, placed inside cladding made of glass-fibre fleece. Due to contamination and insufficient wedge stability, however, the rooms that were developed in those days and are still in service no longer meet modern functional or visual expectations. Wedges with a depth of well over 1 m and which are no longer stable without structural reinforcements were not possible either. This led to the now common solution of using perforated plates to encase the wedges. Perforated plates with a perforated surface area of 50 % are pervious to low-frequency sound waves, but the loss of exposed absorbent surface is a disadvantage in the high frequency range, particularly when the sound incidence is at an angle. The search for supportive cladding with the largest possible absorbing surface resulted in the meshed metal solution.

In addition to the interior and exterior noise testing capacity installed in Sindelfingen, the exterior noise test rig modernised in its existing environment opens up further opportunities for Daimler AG in designing the driving noise of all-wheel-drive vehicles. This is particularly important for meeting high, segment-specific customer expectations, particularly in the growing market for all-wheel-drive vehicles. Thanks to a clever design, all the main conditions for measuring drive-by noise can be met in the compact area of around 20×16 m².

DEVELOPMENT OF THE EXTERIOR NOISE

The current ECE 51.02 specifications (sample measurement, ❷) regulates drive-by noise and will in the next few years be replaced by the successor regulation ECE 51.03. This means that the low and medium engine speed range will be more stringently regulated in future, which better reflects actual vehicle usage in urban traffic, ❸ and ❹.

A vehicle's total exterior noise can be roughly divided into drivetrain noise and tyre/road noise. When exterior noise is assessed on a roller test rig, the measured tyre/road noise does not correspond exactly to the actual noise out on the roads. This is due to the different surface conditions and the difference in the deformation of the wheel contact patch on the roller in comparison to a level road. An ISO working group is currently working on an arithmetic correction of the tyre/road noise, which means that we can expect to see a good correlation in the future.

In contrast, drivetrain noise is mapped very well. In conjunction with a quiet road/tyre combination, drive-based measures can be optimally verified, which means that the effectiveness of insulation measures can be accurately reproduced and, in turn, assessed. Any necessary mounting work can be carried out in the pit with a removable cover, giving access to the underside of the test car. This avoids the need to drive to the nearest lifting platform, which is necessary for road-based measurements. Once development work is complete, the final measurements are performed on an ISO test section and then certified with the testing authorities.

The exterior noise development process focuses on two areas: in addition to development work to ensure compliance with legal restrictions regarding noise pollution in traffic, the subjective perception of the vehicle in traffic is also taken into account. Within the framework of existing legislation, development also involves designing a noise that is unique to the brand and gives a sense of perceived value. To achieve these development objectives, optimisation measures – particularly for direct-injection diesel and petrol engines with stringent acoustics requirements – need to be safeguarded. The measured values for the structure variants are recorded on the test rig without any interfering or fluctuating secondary noise. The variants are assessed directly in the acoustics studio objectively and/or subjectively. The lack of secondary noise in the studio is extremely useful for the direct listening comparison, particularly when transient signals are present.

INTERIOR NOISE

One of Mercedes-Benz's brand claims is excellent noise-related comfort, which is achieved by quiet, easy-on-the-ear vehicles. The specific requirements are divided into measured values regarding acoustics and vibrations, which are supplemented by individual criteria. Tonal amplifications such as the howling component in engines or transmissions must not be perceived as annoying. Furthermore, noise from secondary equipment (for example, the A/C compressor, vacuum pump, transmission oil pump or power electronics) must not under any circumstances be perceived as annoying interior noise.

A key part of the development process with regard to interior noise is optimising the vehicle in such a way that the targets can be achieved. Just like the equipment installed at the MTC in Sindelfingen, this test rig allows variant measurements to be performed quickly, flexibly and with a high level of repeatability.

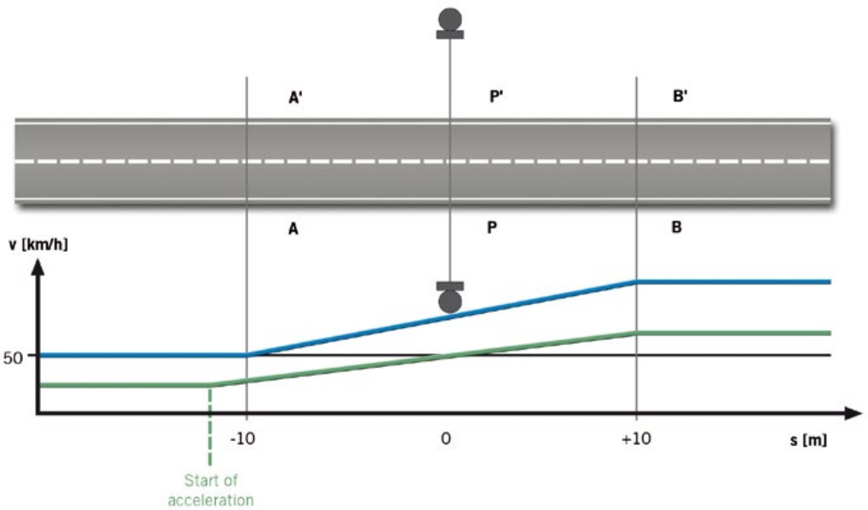
Mercedes-Benz is increasingly using alternative drive technologies such as electric or hybrid drive systems. With hybrid drive systems, the transitions between the hybrid-specific operating modes (internal combustion and electromotive driving, recuperation) have to be virtually silent and imperceptible to customers. A specially modified operating strategy on roller test rigs is required for this development work. The drive assembly operating status to be assessed must be specifically adjustable.

REQUIREMENTS REGARDING ROOM ACOUSTICS

During acoustic measurements on roller test rigs, the room acoustics should generally match the free-field conditions out on the roads. The requirements, however, differ greatly depending on whether a test cell is set up for "normal" acoustic development work or a huge semi-anechoic room for testing exterior noise emissions.

In interior noise measurements, residual, reflected sound waves with low levels compared with structure-borne and direct air-borne noise on the vehicle are of no relevance. In a room of this size, exterior noise measurements (for example, for capsule developments) are normally performed in the near field, where the levels from the emitting sources are also dominant.

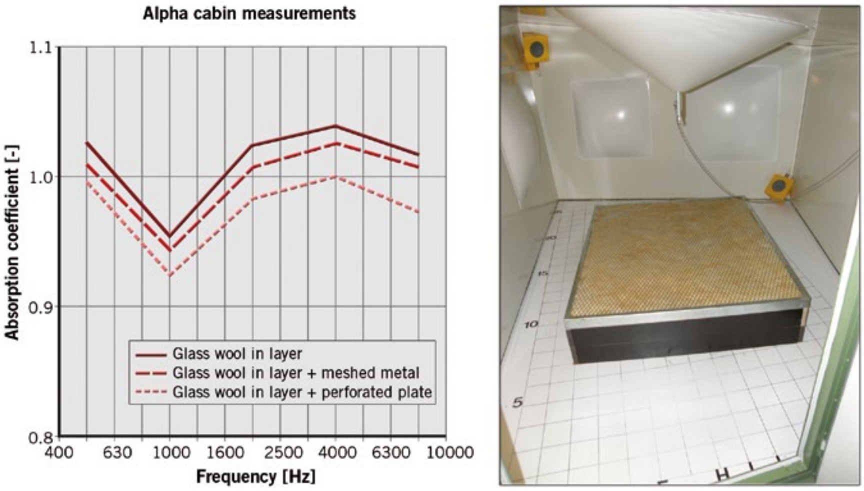
The microphone configuration in exterior noise test rigs, on which drive-by measurements according to ISO 362 are performed, is completely different. The



❸ Drive-by as per current (blue) and future (green) standard (ECE 51.02 vs. ECE 51.03)

	ECE 51.02	ECE 51.03
Measured value formation	Arithmetic mean level	Interpolation
Engine load	Full load	Full load and constant speed
Gear (manual)	2nd and 3rd	2 gears below/above reference acceleration
Gear (automatic)	D (downshift > 3)	2 gears below/above reference accerleration (or D)
Start of measurement	Front of car	Depending on engine position
Speed	50 km/h at entrance	50 km/h at microphone (10 m after entrance)
Acceleration	From entrance	From entrance at the latest

❹ Differences between the old and new exterior noise measurement procedure



❺ Absorption coefficient measurement performed by Rieter Automotive; the absorption coefficient calculated from the reverberation time can have values of >1 due to edge effects and is suitable for relative comparisons

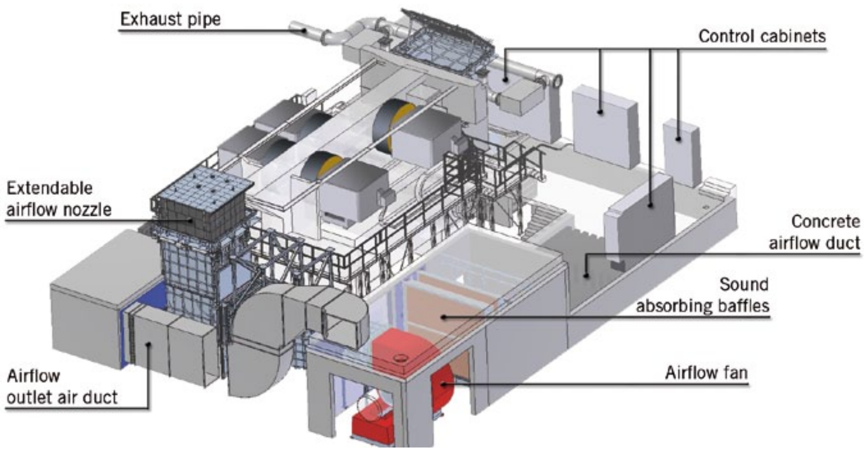
array of microphones along the 20-m test section is positioned at a lateral distance of 7.5 m from the centre of the vehicle, close to the longitudinal wall of the room. This means that practically the entire measuring room must be free of wall reflections if the expected accuracy of 0.1 dB is to be achieved. Furthermore, the process involves not only measuring relative level differences between individual sound sources, but also ensuring that the absolute level correlates with on-road measurements. These should be replaced by measurements on the exterior noise test rig in the near future.

For measuring the absorption quality of the room, ISO 3745 prescribes excitation with a standard sound source in the centre of the room and, as the signal type, defines either white noise or sinusoidal signals with a third-octave mid-band frequency. With the random noise signal, the subsequently calculated third-octave spectrum represents an average level across the third-octave bandwidth; with the sinusoidal single-tone signal, it contains just one frequency, as is the case with narrow-band filtering. Standing waves are much more evident in narrow-band testing, which is why it represents the more stringent specification.

In measuring rooms with a surface area of around 100 m², flat absorbers on the side walls have become the favoured alternative due to their minimal space requirements and practical surfaces. In an exterior noise test rig with a surface area of 500 m², space economy is no longer the primary consideration, which is why wedge lining is the favoured alternative here. Regarding the performance of broadband compact absorbers, opinions diverge, but some general experiences are commonly agreed upon. Plate absorbers can be tuned to low frequencies and can, therefore, be superior to wedge lining below 50 Hz. In the third-octave bands above (between 80 and 250 Hz), however, they often exhibit weaknesses and overshoot requirements when exposed to sinusoidal excitation. Sound absorption wedges with a depth that is one quarter of the maximum wavelength are therefore the "safe" solution for all frequencies above the corresponding limit frequency. At lower frequencies and with longer wavelengths, the damping effect drops continuously.

Although the legally prescribed drive-by measurement involves evaluating the dB(A) level, that means a weighted average value across all frequencies, compliance with ISO 3745 with sinusoidal single-tone excitation was required for this measuring room. This was decided in order to cope with the emission of fixed frequencies by electric vehicles and evaluate analyses at a constant engine speed. As a result, a decision was made to use wedge lining. However, the dimensions of the elastically supported concrete shell of the measuring room, which was designed for a wedge depth of 1 m, had to be maintained. To achieve a limit frequency of 50 Hz, a solution had to be found that would allow for a greater wedge depth without the tips of the wedges on the longitudinal wall being too close to the row of microphones. For this reason, the lining depth on the ceiling and on the side walls was 1.6 m above a height of 2.5 m and 1.4 m below this. The wedge depth (distance between the tip and the base) of 1.3 m remained the same in both cases, which means that the change in lining depth is hardly visible.

To prevent sound reflections, the inlet nozzles for ventilation in the corners of the room are not designed as sheet metal air ducts, but are formed as specially arranged wedges. A core of wedges inside this duct means that the duct cross-section tapers downward, thereby ensuring maximum absorption at the height of the microphones. The appearance of the room is characterised by a new wedge design, which is filed with the German Patent and Trademark office



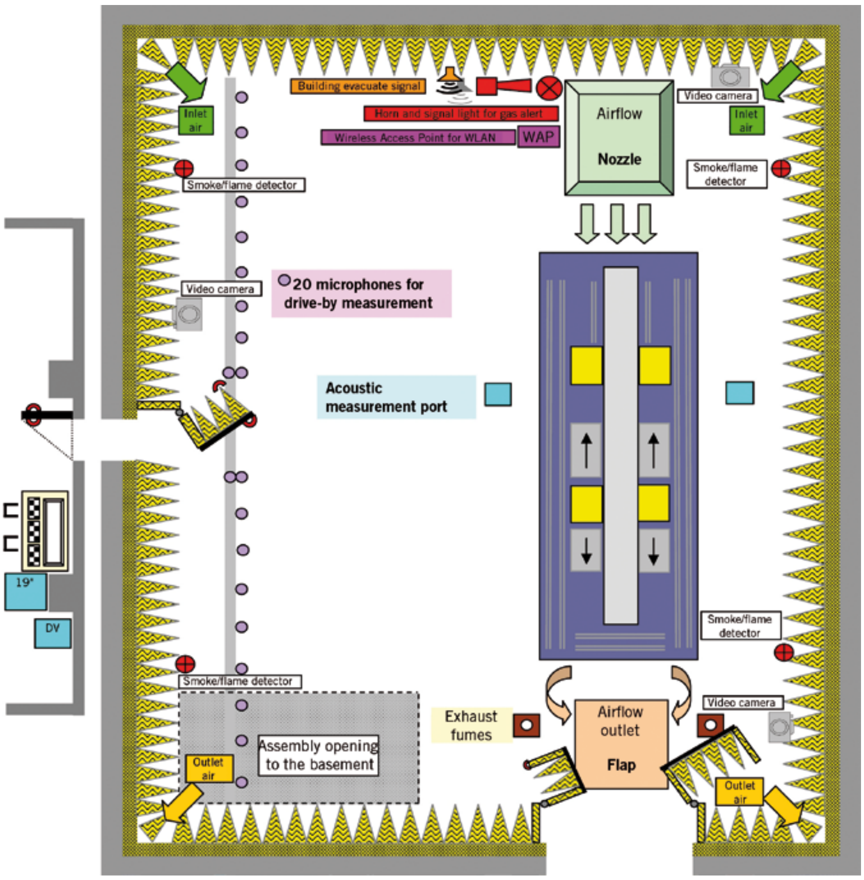
❻ Cut-away view of the acoustics roller test rig, decoupled from the building by a sprung foundation under No. 20 2011 001 443.6, ①. The acoustic benefits compared with perforated plates were verified by an absorption factor measurement performed by Rieter Automotive (now Autoneum), ❺.

The development of a manufacturing process that goes without any visible reinforcements or border strips led to an unrivalled aesthetic appeal. For the stable value of the investment, the unavoidable contamination of the wedges by dust, tyre abrasion and soot is a well-known issue. This is why the more economical design involving double wedges, that means two (or more) wedges in a single housing, was not used.

The glass-fibre fleece used for the cladding acts as a dust filter and filters out the most dust at those points where the air currents are strongest. These are the areas between the two wedges, behind which there is no material. Different material densities under the glass-fibre fleece should therefore be avoided, which means that single wedges with a uniformly dense backfilling behind the fleece are the most cost-effective long-term solution. Due to the transparency of the stretched metal, the visual impression is mainly determined by the fleece. To refurbish severely contaminated parts, only the fleece has to be changed. A mounting system on which the wedges are hung, one by one, on a rail and can easily be removed, allows an easy exchange of the glass-fibre fleece. Total replacement and disposal of the absorption material will therefore not be necessary in the future.

DIMENSIONS (LENGTH × WIDTH × HEIGHT)	
Body dimensions	20.0 m × 16.0 m × 7.0 m
With wedges (below 2.5 m height)	16.8 m × 12.8 m × 5.4 m (17.2 m × 13.2 m × 5.4 m)
ALL-WHEEL ROLLER TEST RIG PERFORMANCE DATA	
Max. speed	260 km/h
Rated power output per wheel	330 kW
All-wheel rated power output (short-term up to)	660 kW (840 kW)
Tractive force per wheel	10,000 N
All-wheel tractive force (short-term up to)	20,000 N (25,000 N)
Min./max. basic inertia mass per axle	1650 kg
Roller diameter	1910 mm (75°)
Min./max. wheelbase	2200 – 4400 mm
Axle load	3500 kg
Ambient air	18 – 25 °C, 42,000 m³/h
Airflow system	0 – 120 km/h, 144,000 m³/h
Exhaust system	8000 m³/h

❼ Functional characteristics of the exterior noise test rig



❽ Layout of the measuring room

ALL-WHEEL ROLLER AND TECHNICAL BUILDING EQUIPMENT

In the concept and design phase, it was considered important to use not only stable, torsionally rigid and vibration-free structural steelwork for the test rig frame, ❻, but also noise-reduced, water-cooled drive machines. These are additionally fitted with a sound enclosure designed under consideration of all noise-reducing aspects with regard to airflow. Other individual measures to minimise noise, such as an acoustically optimised roller design for preventing the so-called bell effect or sound-insulated sliding covers for the wheelbase setting, result in a low overall noise level on the test rig of just 41 dB(A) at 50 km/h (measured at a height of 1 m above the test rig).

An output of 330 kW per motor and a tractive force of 10,000 N per roller ensure that the dynamic acceleration requirements for full-load tests can be met, and allow vehicle mass simulations of up to 3500 kg. Particularly worthy of mention is the patented, highly dynamic load regulation during acceleration with a similarly high level of roller synchronism at the same time.

The main contractor for the acoustics and technical building equipment with ventilation technology and lighting was Kristl, Seibt & Co GmbH from Graz, Austria. The temperature of the anechoic room is regulated by means of an air supply, extraction and recirculation system with heating and cooling elements. The temperature of the airflow is regulated by means of one two-part heating element and one two-part cooling element. The outlet air duct of the airflow system can be raised out of the floor within 15 s.

Given the excellent properties of the all-wheel noise test rig, plans are being made to use it for performing certification measurements too. This offers further potential due to its isolation from climatic conditions and the increased speed of development.

SUMMARY

With this new all-wheel noise test rig, Mercedes-Benz benefits from yet another highly efficient and flexible tool for ensuring that vehicles fulfil the brand claim of maximum noise-related comfort, ❼ and ❽. The use of innovative meshed metal wedges results in exceptionally high acoustic quality in rooms of this size.

Different drive concepts and traction variants can be tested to a very high quality. The test room is, as a result, increasingly becoming the heart of an efficient acoustic development process in which sound quality is being improved step by step as part of the concept configuration process and target conformity is safeguarded as part of the development process.

THANKS

The authors would like to thank Dipl.-Ing. Heinz-Dieter Klein (test engineer for acoustics) and Ing. Johann Mainz (test engineer for acoustic test rigs), both Daimler AG, for their commitment and dedication in setting up the test rig.

AVOIDING JUDDER BY MEANS OF AN **EFFECTIVE TRANSMISSION DESIGN**

Fuel economy has become a vital issue for the brand image of vehicles these days. In response to this challenge, engine developers use downsizing as well as higher torques at lower engine speeds. This development effects also transmissions. Here, new engine trends are to be implemented as efficiently as possible at a minimum of losses. Therefore, LMS analysed the usage of the torque converter and the lock-up clutch at an early stage to optimise performance and losses.

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MOTIVATION

During the engagement of the lock-up clutch in automatic transmissions, judder may occur. Predicting the occurrence of this vibration is essential in the development process of a new transmission especially since the nowadays trends lead to early closing of the lock-up clutch and higher torques at lower engine speeds [1].

This is when the typical self-induced judder occurs. The influence of the clutch friction characteristics has been evaluated in detail. However, in addition to the clutch friction characteristics, the whole drive's damping characteristics do considerably influence the system's stability margin. Transmission manufacturers thus firstly focus on the transmission itself and then on the damping characteristics of components such as the lock-up damper and the torque converter.

This is, however, not a sufficient analysis. It is not uncommon that the same transmission leads to judder problems in one car when there are no problems of the kind in another type of car. So, the vehicle needs to be analysed in total as a second step, after the integration of the transmission in a vehicle model. This gives the transmission manufacturer the opportunity to prevent potential transmission judder problems in different types of cars as early as possible in the design cycle.

Once the full vehicle model is available, a sensitivity study will be performed. This third step will pinpoint, for a specific vehicle, the powertrain mode that will most probably lead to judder and the component that contributes most to the modal damping. This component will then be first in line to be adapted to increase the stability margin. Since it is a full system analysis, the transmission manufacturer is able to check on further components that may cause judder (for example driveshafts and tyres).

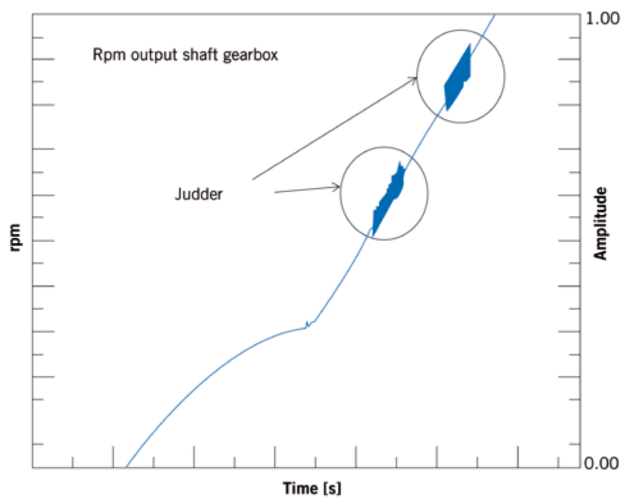
CLUTCH JUDDER

The engagement of the clutch can be responsible for drivetrain oscillations. These torsional vibrations are transmitted to the vehicle body via suspension and engine mounts leading to accelerations of the vehicle body that can be felt by the driver [2]. These clutch-induced drivetrain oscillations are called clutch judder by definition. ❶ displays an exemplary judder event measured at the transmission output shaft.

Judder can be classified as self-excited and forced excited judder depending on how it is caused. Self-excited vibrations can occur in systems where considerable forces are transferred through friction (for example clutches). The basic model to describe this effect is the frictional oscillator, ❷. Linearised at a certain working point, the following equation describes that system by Eq. 1:

EQ. 1	$m\ddot{x}+(c+F_c\mu')\dot{x}+kx=0$

Eq. 1 shows that the friction force reduces the system damping when the gradient of the friction coefficient with respect to the



❶ Measurement of clutch judder

slip angular velocity is negative. ❸ shows a typical friction coefficient curve of a lock-up clutch. The curve makes visible that the risk of judder is situated at lower slip velocities where the gradient of the friction coefficient is the most negative. If the system damping is negative ($\mu' < -c/F_c$), a self-excited oscillation, that is judder, will occur. This judder's distinctive feature is a constant frequency that is independent of the engine's or transmission's rotational speed.

TRANSMISSION MODELLING

By using the simulation software Imagine.Lab Amesim, LMS created a transmission model. ❹ shows an example of such a transmission model. The parts that essentially need to be modelled for the simulation of judder are:
: internal transmission parts
: torque converter
: lock-up clutch
: lock-up clutch control.

INTERNAL TRANSMISSION PARTS

The focus in modelling the internal transmission parts is on the correct implementation of the gear stages using clutches and brakes. This will allow for simulating the gearshifts during a run-up simulation. An accurate inertia representation is required in order to ascertain the powertrain's modal behaviour. The flexibilities are less important for judder simulations as other components have typically lower flexibilities (lock-up damper, driveshafts etc.) [3]. Nevertheless it is recommendable to include the transmission's input and output shaft flexibility margins in the model.

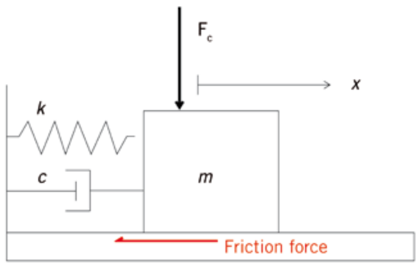
TORQUE CONVERTER

The two typical curves displaying the torque ratio *TR* and the capacity factor *K* as functions of the speed ratio *SR* determine the torque converter's characteristics. These three values are defined as described in Eq. 2:

EQ. 2

$$SR = \frac{\omega_{turbine}}{\omega_{impeller}},\ K = \frac{\omega_{impeller}}{\sqrt{T_{impeller}}},$$
$$TR = \frac{T_{turbine}}{T_{impeller}}$$

For the judder analysis, especially the capacity factor plays an important role



❷ Physical combination at the frictional oscillator

as it determines the torque converter's equivalent damping which also to a considerable degree depends from temperature. At certain speed ratios, damping can easily change by 50 % going from the lowest to the highest operating temperature. When the speed ratio approaches the value 1, the damping decreases. This means that for small slip rotational speeds, the torque converter provides less damping and judder may occur more easily.

LOCK-UP CLUTCH

The lock-up clutch is by definition the most important part for lock-up judder not only because its friction characteristics, ③, originate judder but also because it is attached to the lock-up damper whose features are especially crucial for the judder analysis. Exact modelling of these damping characteristics (hysteresis, viscous damping etc.) is really important here as this component could serve exactly the positive damping that is needed to counter the negative damping caused by the friction characteristics.

LOCK-UP CLUTCH CONTROL

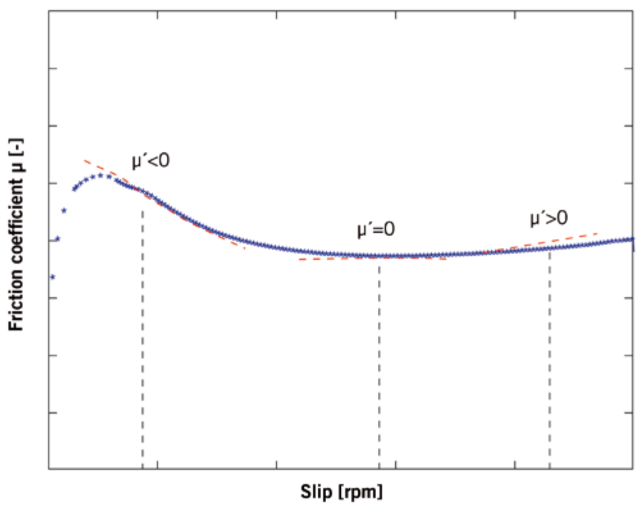
Even if the clutch is designed perfectly, poor control software or hardware design can still cause instability. So-called control judder occurs. Designing an anti-judder control is an option here. This task will not always turn out feasible, especially not at higher frequencies. There will certainly be considerable delays in the hydraulic system when the transmission fluid is cold and, thus, limits the controller's practical bandwidth [4]. All models that have been described previously allow for an accurate transmission simulation.

FULL VEHICLE MODELLING

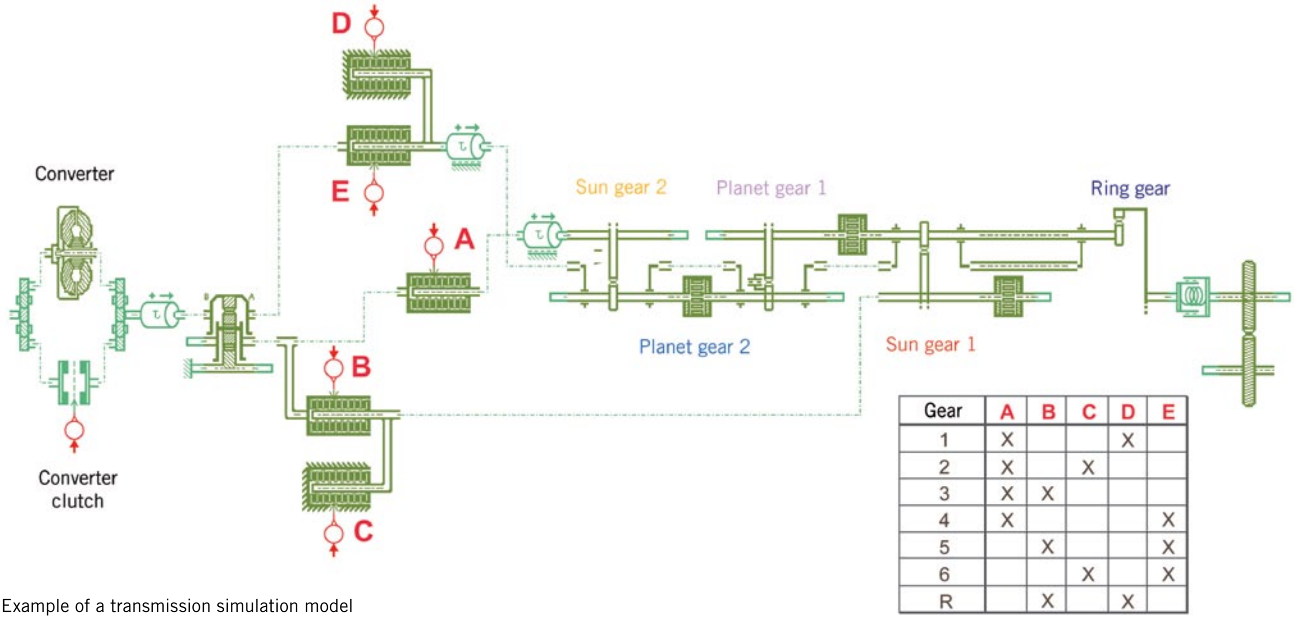
Even if the transmission has not caused any judder on the test bench, that does not mean that there will be not be judder once the transmission has been installed in the vehicle. The vehicle's characteristics such as driveshaft stiffness or tyre damping also determine the powertrain's resonance frequencies and damping. In order to forestall potential judder problems already at an early development stage, full vehicle simulations are absolutely necessary. To do that, the model from the previous step is extended by an engine model and a vehicle model, ❺.

ENGINE MODEL

The engine model can be quite basic for a judder analysis [5]. It typically comprises a map that characterises the torque as a function of engine speed and throttle. This is sufficient to perform sev-

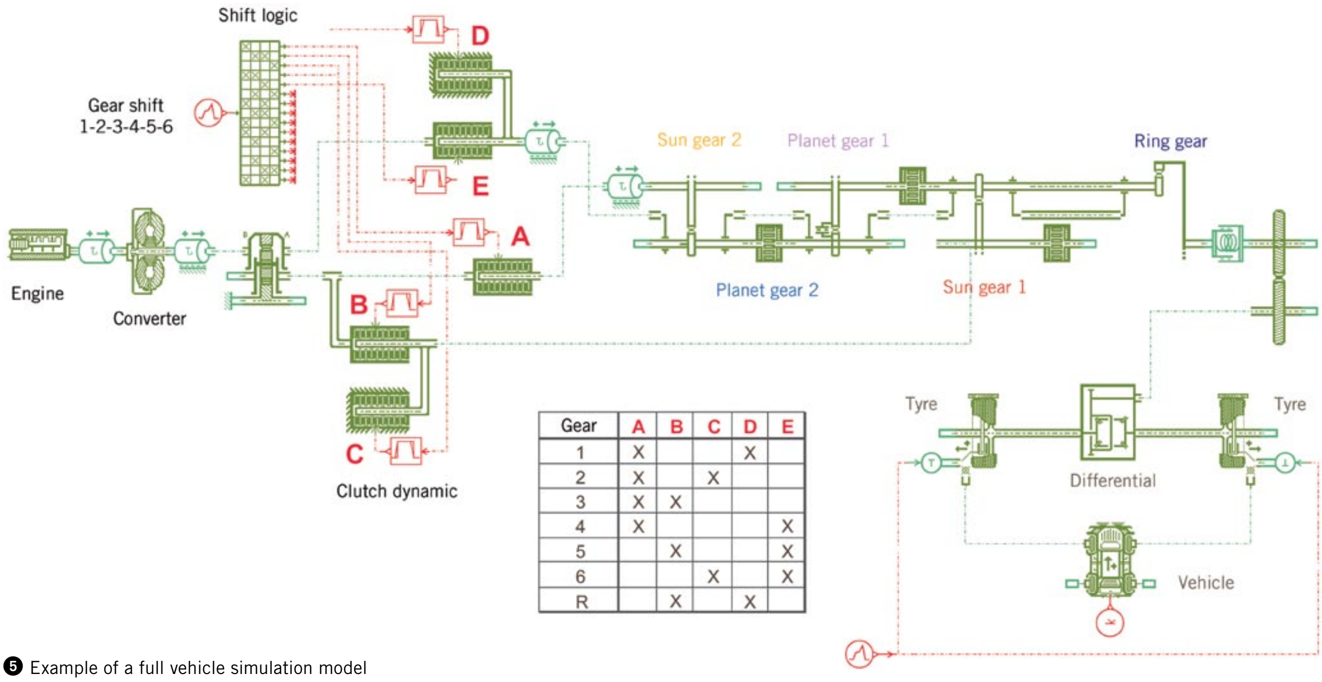


❸ Curve of friction coefficient in relation to slip angular velocity



Gear	A	B	C	D	E
1	X			X	
2	X		X		
3	X	X			
4	X				X
5		X			X
6			X		X
R		X		X	

❹ Example of a transmission simulation model



Gear	A	B	C	D	E
1	X			X	
2	X		X		
3	X	X			
4	X				X
5		X			X
6			X		X
R		X		X	

❺ Example of a full vehicle simulation model

eral drive cycle simulations. For self-excited judder, there is no special need to accurately describe the torque fluctuations. The negative system damping will induce the judder itself, so there is no need for external excitations such as the engine harmonics.

VEHICLE MODEL

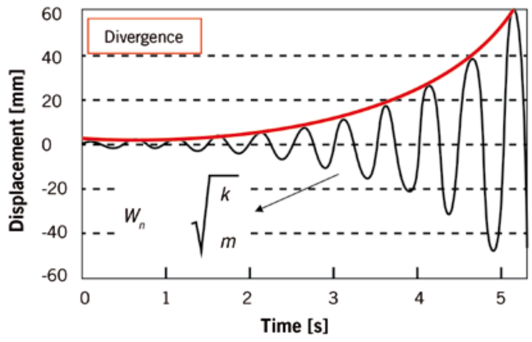
Mainly, the vehicle model has to supply the correct impedance and damping to the transmission. The impedance is typically determined by the driveshafts' stiffness [5]. Other important parameters are wheel inertia, tyre stiffness and damping.

A transmission manufacturer cannot always straightforwardly obtain all these parameters from the car manufacturer. However, it is possible to measure them in some rather simple test setups.

Even though it is recommended to use FRF (frequency response function) based methods to determine the inertia properties for complex structures, the tyre structure is simple enough to use the old-fashioned pendulum test. [3] comprises a whole range of tyre inertia that have been measured in a pendulum test and offers typical values.

SENSITIVITY STUDY

The clutch friction characteristics do, as mentioned above, influence the judder, ❻. In addition to that, the different



❻ Judder: unstable oscillation velocity, the displacement is building up

driveline components' damping coefficients play a vital role as well. The simulation model has various components to include damping, such as the lock-up clutch, the lock-up damper and the tyre. Moreover, losses occur in the prop shaft, the differential and the internal gears of the transmission. For the engineer, it is interesting to quantify how much every driveline component contributes to a particular driveline mode's total modal damping. He will thus be able to assess how much any contributor in fact damps in each mode. This information is crucial for an adequate troubleshooting, but it is nevertheless valuable in an early development phase as well. The relevant contribution to damping is assessed as follows:

: put all damping in the model to zero
: attribute to one component its nominal damping
: run the simulation and linearise the system at a certain working point
: extract the eigenmodes and modal damping
: repeat previous steps for all components.

CONCLUSIONS

For a transmission manufacturer it is important to forestall judder problems as early as possible in the development phase. Therefore, LMS used a three step approach to optimise the transmission adjustment.

After creating and validating a model of the transmission only, this is extended by an engine model and a vehicle model to allow for a full vehicle analysis. As a next step, a sensitivity study helped to pinpoint the components that damp the best. Engineers can thus focus on the relevant parts immediately.

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INITIAL SITUATION

Roof systems such as panorama roofs and sliding panels are ordered by many customers as attractive optional extras. In addition to the increased comfort due to more air and light in the vehicle, roof systems can also be used to produce energy if they are equipped with solar cells. For example, they can reduce the fuel consumption of vehicles with internal combustion engines or feed the batteries of electric cars, thus increasing their range.

Alternatively, for instance, a parking ventilation system in the vehicle can be supplied with power so that this additional function does not draw power from the battery. Last but not least, the generation of electricity using a solar roof is attractive to many customers on an emotional level, since they can visibly contribute to energy savings in these times of growing environmental awareness.

After Webasto had produced around 300,000 lightweight roof systems for the small car Smart Fortwo with a fixed, 1.2 m² large polycarbonate panel, an opening panoramic roof made of polycarbonate was to be developed for the first time. The resulting concept car, which was presented at the 2011 IAA Frankfurt Motor Show, weighs only about 20 kg, has an area of 1.8 m², and has two additional features: solar cells for power generation and a parking ventilation system that is powered by the photovoltaic panels on the roof, ❶.

TASK

Polycarbonate has become established as a lightweight material for fixed glazing elements. Its density is 1.2 g/cm³, only about half that of glass (2.5 g/cm³). Previously, however, its lower modulus of elasticity in particular (at 2.4 MPa) has prevented its use in moving parts in the roof area. Significant suction occurs there; this can "pull" on the roof module with 1500 N or more at high speeds. Even in the case of sliding panels with glass elements (50 to 90 MPa modulus of elasticity), steel reinforcements are therefore required.

To keep the deformation of the polycarbonate panel – caused by this suction – within acceptable limits for classic sliding roof kinematics, either the panel would have to be made much thicker or a heavy reinforcing frame would need to be used. Either of these would reduce the weight advantage compared to a glass design from about 50 % to only 20 to 30 %.

NEW OPENING MECHANISM

To create the new panorama roof module using the lightweight construction method, Webasto thus developed a new type of opening mechanism that is oriented specifically to the properties of polycarbonate. Therefore, reinforcements can be dispensed with almost entirely, and the weight advantage of the plastic is retained.

The movable, transparent polycarbonate panel, which has an area of 0.6 m²,

LIGHTWEIGHT ROOF MODULE WITH INTEGRATED SOLAR CELLS

Multifunctional roofs can be very light. Webasto shows this in a concept car in which polycarbonate is used for the first time as the material for an opening roof system. Using novel kinematics that make reinforcement elements largely redundant, the full weight advantage of around 50 % compared to glass is preserved. Thanks to the integrated solar cells, the roof module also helps to generate energy.

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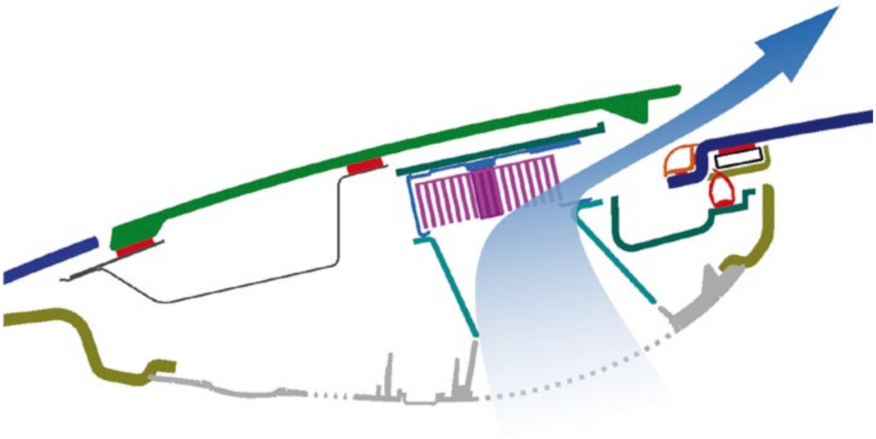
weighs 5.5 kg. If it were made of glass, its weight would be about 10 kg.

The opening mechanism is designed in such a way that the movable panel “presses” from below into the frame and the seals when it is pulled upwards by the suction that occurs while driving. In conventional glass sunroofs, the rear end is first lifted to open the roof. For the concept car, in contrast, Webasto used the so-called “front vent” principle. The front part of the polycarbonate panel lowers first. This reveals vent openings in the front part of the roof module, ❷. While driving, the suction of the passing air provides ventilation. To open the roof further, the rear portion of the panel is also lowered and the element is moved under the roof covering.

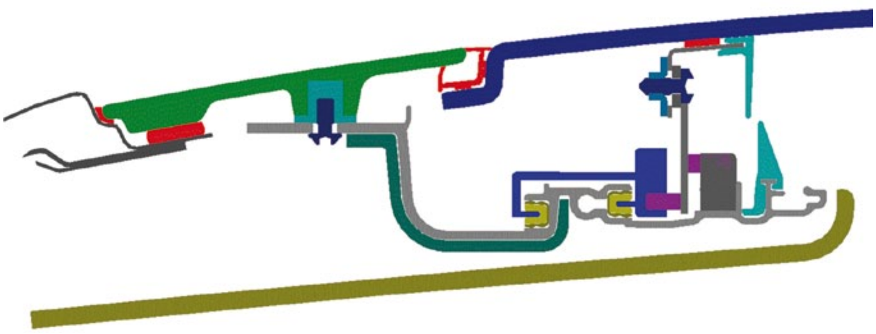
For ventilation when parked, the radial fans integrated into the roof liner promote air exchange when the front of the panel is lowered. A rain sensor detects incipient precipitation and triggers the automatic closing of the roof.

The relatively large thermal expansion of polycarbonate (nearly ten times that of glass) can be controlled using appropriate seals. In the concept car’s roof module, the panel would shrink or expand on either side by a maximum of

❶ The lightweight design roof module combines comfort with solar power generation



❷ Sectional view of the front system: venting through lowered panel



❸ Sectional view of the lateral mechanism: panel seal

1.5 mm between -30 and +90 °C. This change is compensated for by appropriate hollow seals, ❸.

PRODUCTION AND COLOUR

Lightweight design is also used for the roof module's frame. The entire frame is created in just one mould and in a single process step. A paper honeycomb core is reinforced on both sides with glass fibres. Polyurethane functions as a connecting layer between the glass fibres and the honeycomb core, ❹. For the outer skin, either a coloured foil, a paintable polyurethane layer or an aluminium plate – less than 1 mm thick – is used; this is inserted into the mould and pressed together with the frame.

Dyed or printed films provide varied styling options. If the roof module is to have the same colour as the car body, either polyurethane or an aluminium sheet that is subsequently painted is used. Aluminium is suitable for bigger production runs because higher temperatures and shorter cycle times are possible with it.

In addition to the variants that differ due to the functions, appearance-based variations can be implemented at low cost for the OEM. For example, the roof can be a different colour from the rest of the vehicle. Effects such as a carbon fibre appearance can also be realised. Even the colour of the solar cells can be varied.

SOLAR PANEL

The solar cells for energy production are located in the rear of the roof surface. Webasto has integrated a total of 30 monocrystalline solar cells. They are initially laminated onto a thin glass plate, which is then bonded to the roof module.

In the study, the glass plate has a thickness of 1 mm. Work is currently being carried out to further reduce this. Also under consideration is a process in which the solar panel is not bonded subsequently, but is inserted into the mould for the module frame and is directly bonded to it during its production.

VARIANTS DURING ASSEMBLY

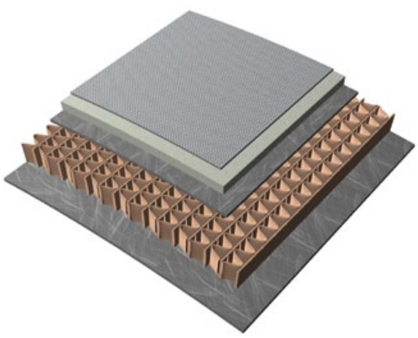
At the OEM, all the variants of the roof module can be easily installed because it is delivered as a single unit to the assembly line and is inserted there in place of the normal roof. Important advantages of this approach for the OEM are that the variant requires no changes to the car shell and that attachment occurs only in a very late stage of the assembly process. Various versions of the roof module can be realised, each offering efficiency, comfort or a combination of both:

: focus on comfort: panorama roof module with two transparent polycarbonate panels; the front panel can be opened, thus providing fresh air
: focus on comfort and efficiency: combination of transparency and power generation; the front polycarbonate panel can be opened, while the rear has a solar module that generates the necessary power
: focus on efficiency: power generation on the entire roof by covering the complete surface with solar cells, ❺.

As part of the cooperation with Konarka Technologies, a US-based manufacturer of organic solar cells, Webasto is also working to integrate semi-transparent solar cells into vehicle roofs.

SOLAR POWER FOR ADDITIONAL FUNCTIONS

In the concept car, the 30 high-performance monocrystalline solar cells provide a combined peak power of 93 W



❹ A paper honeycomb core is reinforced on both sides with glass fibres; polyurethane functions as a connecting layer between the glass fibres and the honeycomb core

❺ Solar power generation over the whole roof surface

(when the roof is partially covered with solar cells). The power can be fed into the battery using intelligent battery management or used immediately for the operation of additional equipment. The performance of the solar roof is enough to operate a parking ventilation system, a seat ventilation system or a compressor cooler.

All three functions are on display in the concept vehicle, a Suzuki Splash converted to electric power, ❻. The OEM can freely specify the logic for the use of electricity from the solar cells and let the end user make the actual decision in individual cases. When used in an electric vehicle, according to calculations by Webasto, the solar roof can provide power for a distance of about 500 km/year.

ECO INNOVATION

In July 2011, the European Union recognised ecological innovations to set off the CO_2 emissions of vehicles [1]. If the solar function is used in a vehicle with an internal combustion engine, the OEM would be able to reduce CO_2 emissions by 2 to 4 g/km. As an energy supplier for additional consumers that are continuously active, solar roofs already proved their worth in the Daimler Car2Go mobility project.

In this flexible renting system, the rental vehicles constantly inform the rental centre where they are parked. The Smart Fortwo vehicles obtain the power for this position signal from solar roofs made by Webasto. If the power is required neither for this function nor for charging the battery, a parking ventilation system starts.

CONCLUSION AND OUTLOOK

OEMs are showing great interest in the potential of the lightweight construction demonstrated in the concept car. This will enable them to offer the end user considerable benefits with only a slight increase in the vehicle's weight. Moreover, the appearance and functions of the vehicles can be customised with little effort because the finished roof module is attached in place of the normal sheet metal roof at a late stage during the assembly process. This concept is particularly suitable for modern space-frame architectures that are designed for plastic planking because the rigidity requirements for the roof module can be fulfilled more easily in this area.

With rising energy costs and increasingly stringent CO_2 limits, the significance of lightweight construction and efficiency is rising. This concept study demonstrates ways in which the dichotomy between comfort requirements and energy savings can be resolved.

❻ Concept study of a lightweight roof module with solar functionality in an electric car based on a Suzuki Splash

REFERENCE

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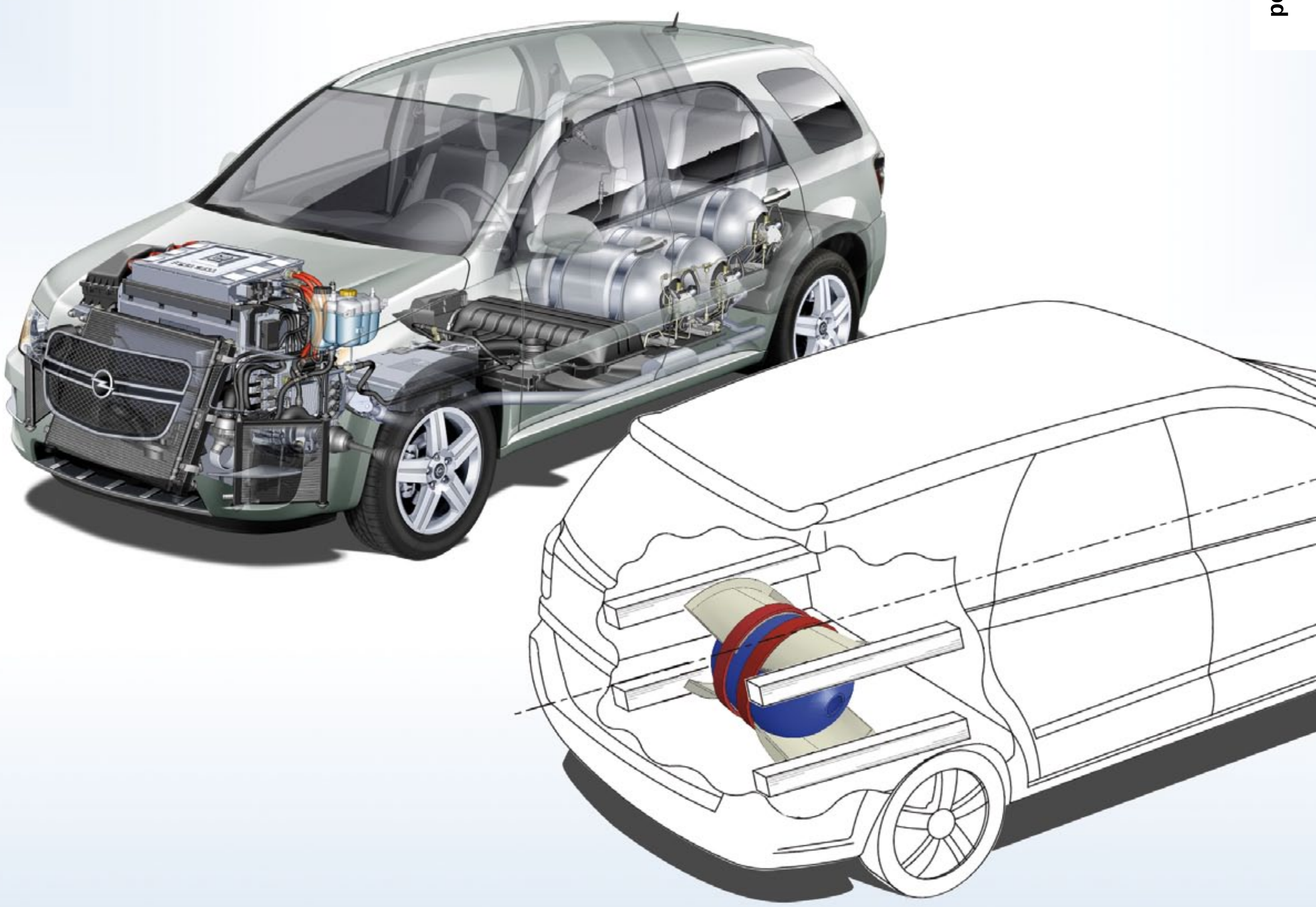
FIBRE-REINFORCED POLYMER HIGH-PRESSURE TANKS AS LOAD-BEARING STRUCTURAL ELEMENTS

A new attachment of the high-pressure tank of a fuel cell electric vehicle has been developed in a joint project between Opel and the Institute of Lightweight Design and Structures at the TU Darmstadt. The aim of the project was to integrate the tank into the car body as a load-bearing structure. This lowers the required torsional stiffness of the car body itself, thus reducing weight and costs. Additional installation space can be regained by a favourable package.



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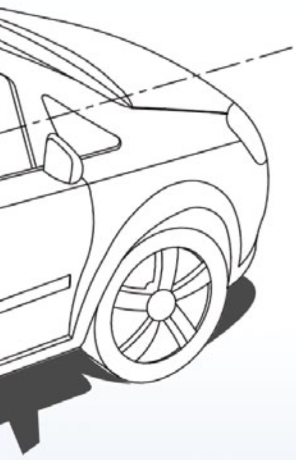
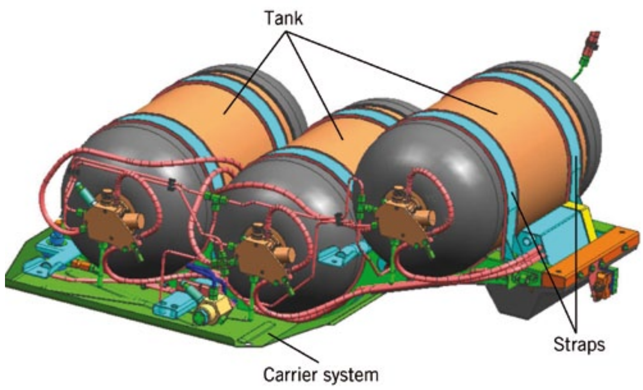


CONCEPT AND DESIGN

The CFRP tank offers high stiffness and strength. Due to the closed circular cross-section, it makes sense to utilise the torsional stiffness of the tank in order to increase the torsional stiffness of the car body. In the HydroGen4 vehicle from Opel, the tanks are attached by straps to a carrier system, ❶. The straps are not suitable for transmitting a torsional moment into the tank. Therefore, a multiple shell concept was developed to integrate the tank into the vehicle structure. ❷ (a) shows the principle of the multiple shells. The shells are fixed to the tank by a longitudinal interference fit, which is familiar from load inputs into drive shafts. An undersized clamping ring presses the shells onto the tank. The interference fit generates sufficient friction for force introduction [1].

The load introduction is obtained by a pair of forces to the shells on two sides of the tank, ② (b). The torsional stiffness of the tank is utilised to reduce the warping of the car body. The optimum position of the tank within the car body and the alignment of the shells is determined by an FE simulation of a car body, abstracted as an open U-shaped cross-section under torsional load, ② (c). A horizontal alignment of the pair of forces and thus of the shells showed the highest warping resistance. Additionally, this position of the shells permits free access to the valve of the tank and can protect it from stone impacts.

The aim of the joint project between Opel and the TU Darmstadt was to increase the local torsional stiffness of the car rear end by a factor of 3 and 6 by using an integrated tank. That means that the shells require a shear stiffness of 20 and 40 kN/mm. Two basic shell shapes were investigated: a straight shell and a cylindrical shell. The straight shell has a lower weight and the straight flange is easier to assemble to a longitudinal beam of the car body. Made of CFRP, the shell has a shear stiffness within the desired range. Due to its geometry, the shear stiffness of the cylindrical shell is four times higher. This permits the use of a low-cost material – for example, Sheet Moulding Compound (SMC) – instead of the much more expensive CFRP. After a deeper investigation of free-shaped shells with different flange diameters, the advantages of the basic shell shapes were combined and an optimum shell was determined under the given boundary conditions. A shell with a flange radius of r_F = 400 mm is an optimum between low weight, easy assem-



❶ Current state of technology: strap fixture in the HydroGen4 bly and high shear stiffness. This shell was chosen for further investigations, ❸.

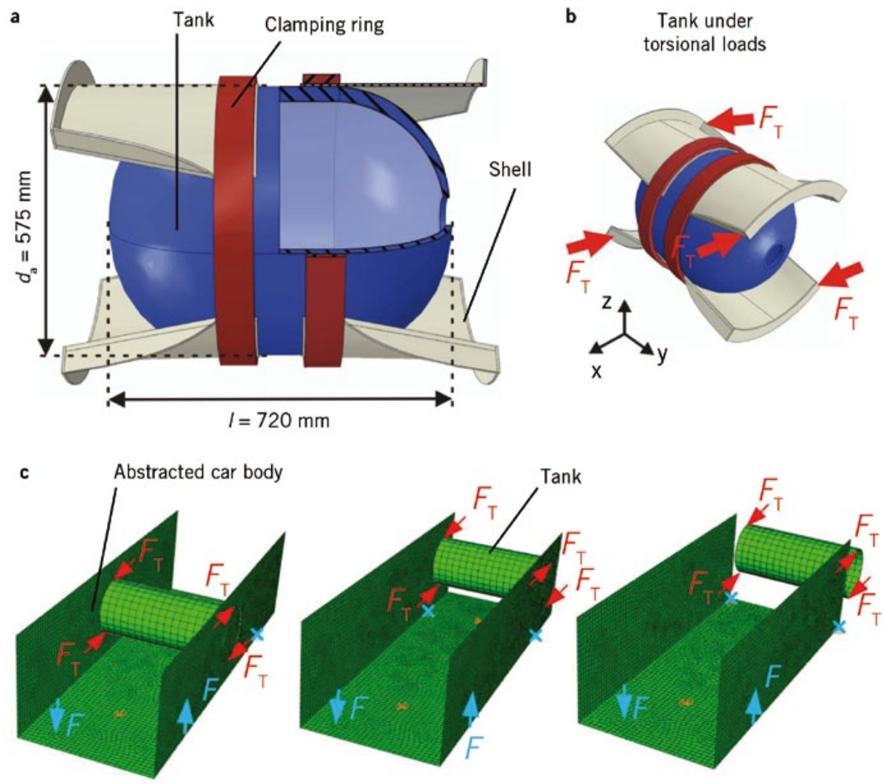
RESULTS OF THE FE ANALYSIS

The shell shape and the clamping forces were optimised for different load cases by FE analysis. For the calculations, the randomly orientated glass fibres of the SMC material were simulated by a $(0/\pm45/90)_s$ laminate and the material properties were adjusted by the fibre volume fraction to values from the literature [2]. The simulation of the isotropic material SMC by a quasi-isotropic laminate permits the usage of the failure criteria for inter-fibre failure [3]. During driving operation, different loads act on the shells:

- axial, tangential and radial strain of the tank due to internal pressure
- acceleration of the tank in the three spatial directions
- torsional warping of the car body.

The tank internal pressure and the acceleration cases apply only minor forces to the shells. Therefore, the clamping and attachment of the shells have no negative effects on the tank. A tank loaded by internal pressure and torsional warping of the car body has to bear only about 1 % higher loads than a load-free tank under internal pressure.

Car body warping is identified as the most critical load case, with shear forces of up to 100 kN applied to each shell. This value is an assumption and is confirmed by the experience of car body constructors. Inter-fibre fractures occur in the shell under the critical load. This failure can be avoided by additional glass fibre reinforcements or by increasing the local wall thickness. The strain of the tank caused by the internal pressure increases the clamping forces. The quality of the clamping was analysed by FE analysis by evaluating the contact shear between the shell and the tank. Complete avoidance of any mobility of the shells on the tank is assumed as long as a radial pressure exists between the shell and the tank and the contact shear remains at zero. The result of the FE analysis was that the clamping is sufficient to prevent relative motion between the shells and the tank, either with or without internal pressure. As a recommendation for the detailed design, sharp shell edges and deviations of the clamping ring from the exact circular shape cause stress peaks. This also is confirmed by FE analysis. The step from shell to tank should have a smooth transition zone to avoid local bending [4].



❷ Dimensions of the tank (a); load case by car body warping (b); variation of the tank position (c); F = tyre forces into the car body, F_T = pair of forces; increase in the torsional stiffness: tank in the middle: factor 68, tank in the rear: factor 15.7, tank in the rear by longitudinal bars: factor 16.1

EXPERIMENTAL VALIDATION OF THE FE ANALYSES

In order to experimentally reproduce the critical load case for dimensioning – torsional warping of the car body and the resulting load F_T – a quasi-static four-point bending test was used. As a test sample, a generic tank with clamped shells was manufactured on a reduced scale, ❹. This four-point bending test with F_x causes the same shear loads on a single shell as the load case with F_T.

The material of the test shells differed from the material used for the calculations and had a lower stiffness. Therefore, two shells buckled during the first test. In one of these shells, a crack formed on top of a buckle due to local bending. Considering the difference of the stiffness, the buckling load in test and FE analysis are in good accordance. For the second test, the wall thickness of the shells was increased by an additional randomly oriented fibre layer to avoid buckling. In the following test, the sample withstood a load of 17 kN without any noticeable failure. This load is equivalent to 111 kN for the real-sized load transmission. Buckling could not be observed in the second test. The final failure developed perpendicular to the principal normal stress and penetrated to the clamping, ❺.

USE OF THE SHELLS AS ENERGY ABSORBERS

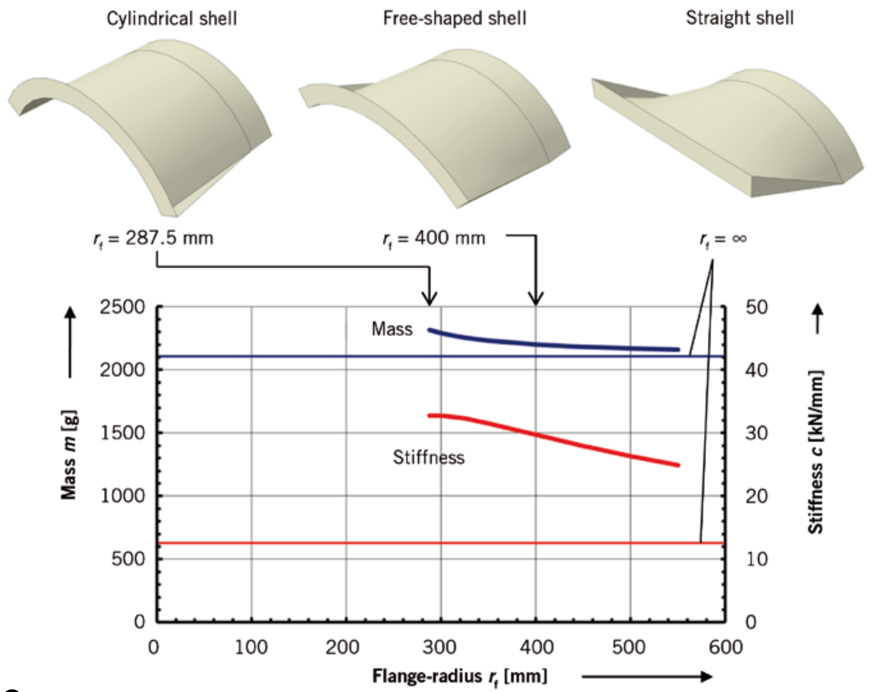
In the case of a side impact – in other words, a side crash in the region of the tank – the shells are used to act as energy absorbers. For that purpose, impact elements (IE) were integrated into the shells to generate controlled damage of the shell, whereas the joint between the tank and the car body should always remain intact. Each of the four shells was divided into two half-shells and assembled at the IE by bolts. The different loads on the shell were translated into bolt loads to use the bolted joint as an energy absorber. It was the aim to generate bearing failure and to draw the bolt continuously through the laminate, thus absorbing the impact energy.

Next to every bolt hole, an area with reduced wall thickness was designed as predetermined trigger point to initiate the failure and to generate a continuous failure evolution in the case of an impact. This trigger area is aligned in the impact direction to ease the cut-through of the bolt through the material. In directions perpendicular and opposite to the impact direction, the IE should bear the load of a conventional bolted joint. The trigger area is a weak point for loads perpendicular to the impact direction due to the reduced wall thickness. To compensate for this, a high-strength material layer – for example unidirectional glass fibre oriented in load direction – replaces the SMC material in the trigger area.

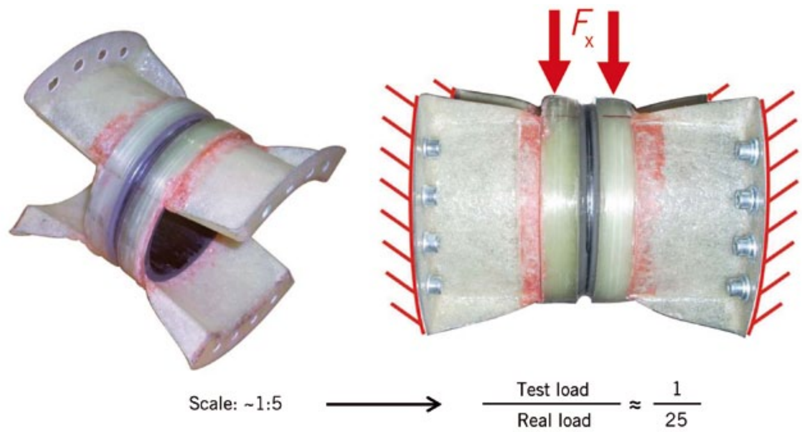
Both, FE calculations and experiments showed that the IE meet the designated aims of one-directional failure. Bearing failure and thus fail-safe behaviour is adjustable, with corresponding edge distances for each load direction. During the failure process, a stable load plateau develops and a substantial degree of impact energy is absorbed, ❻.

SUMMARY

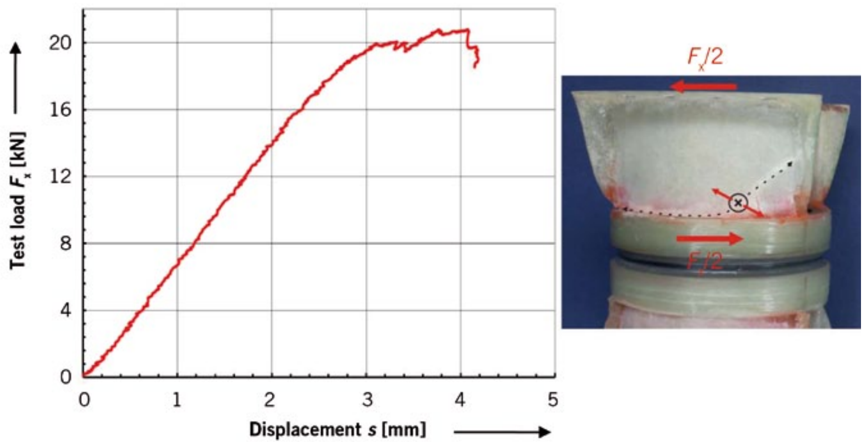
FRP high-pressure tanks have a high torsional stiffness, which can be used to increase the torsional stiffness of the car body. The SMC multiple shell design has been developed to transmit high torsional moments into the tank, while costs and weight remain low. The shells were clamped to the tank by an interference fit. The shape of the shell was optimised for weight, ease of assembly and stiffness. The most critical load case is car body warping in combination with a tank without internal pressure. These and all other load cases were investigated by FE analysis. The shell clamping withstood all occurring loads without any sliding. The FE results of the critical load case were validated by experiment. In none of the load cases could any alarming retroactive loads onto the tank be observed. Hence, the multiple shell design has proven to be suitable for integrating FRP tanks into the car body and



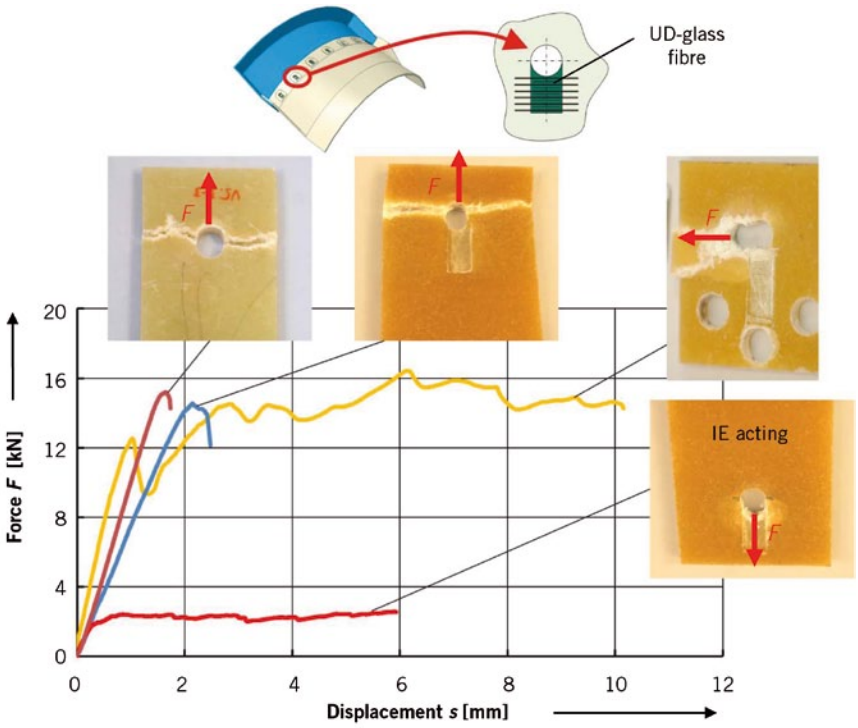
❸ The optimum shape of the shell according to stiffness and weight is between the cylindrical shell and the straight shell; for the straight shell ($r_F = \infty$), the mass and the stiffness reach the marked limits



❹ The generic tank was manufactured in a reduced scale of 1:5 compared to the real tank; thus, the failure load of the real tank is about 25 times higher than the failure load of the generic tank



❺ Load displacement characteristic of the generic tank: from F_x = 17 kN the first damage occurred until the structure failed at F_x = 20 kN; the photo shows the point of the failure initiation, the failure propagation and the direction of the principal normal stress



❻ Anisotropic design of the impact element: load displacement graphs of the IE in different directions; the failure load perpendicular and contrary to the trigger area are in the same range; in the direction of the trigger area, the failure load is deliberately lower

that a considerable increase in torsional stiffness is generated.

Additionally, the shells could act as an energy absorber in the case of a side impact. Impact elements were designed for a continuous failure evolution of the shells without losing the joint between the tank and the car body. All requirements on the impact elements are fulfilled by the design of bearing strength. The concept of using multiple shells and impact elements is patent pending.

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PEEK POLYMERS FOR MORE EFFICIENT TRANSMISSIONS AND OPTIMISED GEAR SHIFTING

Manufacturers of transmissions are faced day by day with the challenge of having to develop more efficient transmission components to achieve ever higher torques and greater facility in gear shifting. Consequently, the polymer manufacturer Victrex has developed innovative high-performance polymers to significantly increase wear resistance and to optimise efficiency by reducing friction losses. Based on these advantages, PEEK polymers are used, for instance, for sealing rings, thrust washers and the friction linings of gearshift forks.

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SIZE REDUCTION

The development and production of automotive transmissions is following the general industry trend towards reduction in the size of components, while aiming to increase performance. State-of-the-art automatic transmissions, such as automatic six- to eight-speed torque-converter automatic transmissions, must be able to apply large torques and cope with higher engine speeds.

This requires considerable optimisation that extends right down to the smallest transmission components, including thrust washers. A thrust washer serves as a support bearing, in order to adjust axial loads on a shaft that is usually rotating against a fixed body. Typical axial forces in passenger cars are in the order of 3000 N at speeds of up to 6500 rpm and higher. The resulting PV values (specific pressure × velocity), based on oil lubrication, can reach up to 100 MPa m/s, depending on the diameter selected. As a result of the highly compact, tight construction of such transmissions, the oil supply is only ensured by defined channels. The design of the supply lines limits the oil feed.

THERMAL PROPERTIES

A complicated system of conduits and hollow shafts in state-of-the-art automatic transmissions assures an effective supply of oil in all driving situations. The oil-sump temperatures within the torque converter area can easily reach 150 °C and higher. These factors place high demands on the materials used for the adjacent components.

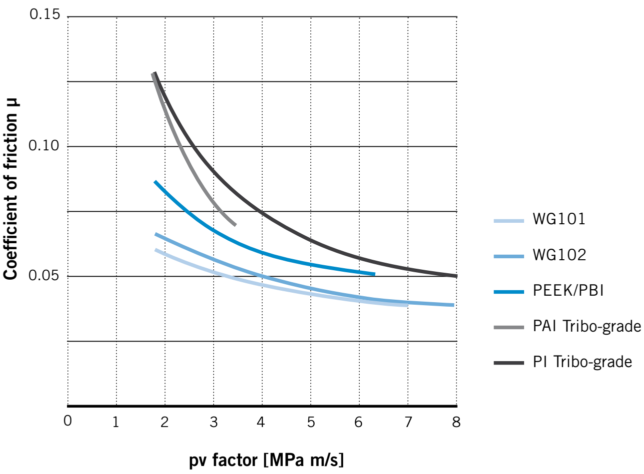
Consequently, thrust washers made of high-performance polymers are being used with ever-increasing frequency in automatic transmissions, to achieve an "integrated" safety factor. High-temperature plastics can typically be exposed to continuous operating temperatures of up to 260 °C. The polymer polyetheretherketone (PEEK) is a thermoplastic and has a melting temperature of 343 °C [1]. PEEK polymer from Victrex is not significantly changed by the temperatures prevailing inside transmissions, even after more than 5000 h. This is particularly important for the compression strength and stability of a component. This means Victrex polymers provide all necessary properties for modern transmisson applications.

AVOIDING FRICTION

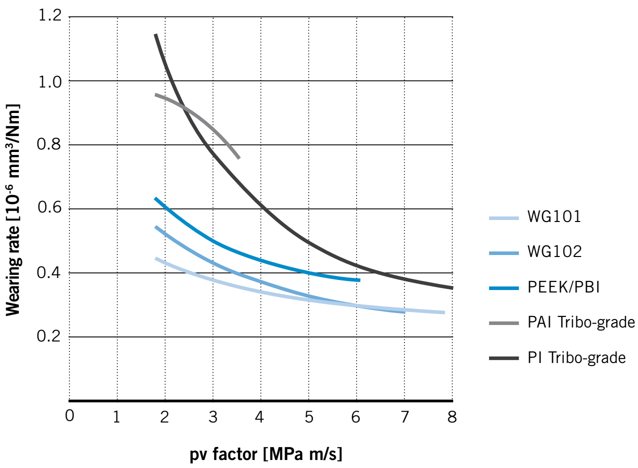
Friction between components generates heat and causes abrasion, with both effects resulting in a loss of energy. Consequently, the aim is always to minimise such losses. The problem can be resolved by selecting materials that have a low coefficient of friction. While adding Victrex polymers, it is managed to create the premium wear grades WG101 and WG102 with remarkable tribological properties. ❶ shows the behaviour of the coefficient of friction in relation to the PV value for WG polymer, PEEK and other plastics. ❷ shows a diagram of the wear rate in relation to the PV value for these polymers.

Plastics such as fluoropolymers (e.g. PTFE) are, however, too soft for use in state-of-the-art transmissions because they cannot withstand the high load pressure-creep requirements – especially under high temperature.

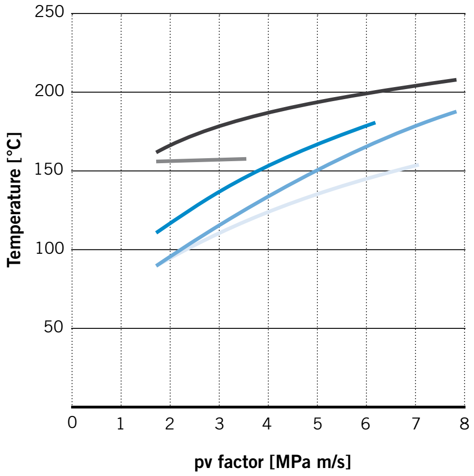
But these are the very requirements that are decisive for the efficiency of the system. The key factor for realising reliable products is the combination of wear resistance, pressure-creep resistance and a low coefficient of friction.



❶ Behaviour of the coefficient of friction in relation to the PV value for WG polymer, PEEK and other plastics (according to ASTM D7302)



❷ Diagram of the wear rate in relation to the PV value for WG polymer, PEEK and other plastics



❸ PV value as a dimension for wear resistance relative to the temperature of the WG polymer, PEEK and other plastics

The plastic Victrex PEEK fulfils these criteria.

RESISTANCE TO AGGRESSIVE TRANSMISSION FLUIDS

State-of-the-art lubricants for automatic transmissions contain numerous chemical components that conform to OEM specifications; however, they can be very aggressive towards the materials used. This has culminated in new challenges, with regard to the chemical resistance of the employed materials.

Most transmission fluids consist of combinations of a multitude of petroleum-based chemical additives and dyes. The latest studies have confirmed that PEEK does not show any significant change in performance and is not influenced in any significant manner during tests with new transmission oils, after 1000 h at 150 °C, so there is nothing to prevent its adoption.

STIFFNESS AND WEIGHT

It makes sense to manufacture components out of polymers instead of metals especially if the aim is to reduce weight at comparable application strength and rigidity. For instance, among the technical plastics, the carbon-fibre reinforced PEEK 90HMF40, from Victrex, offers the highest specific strength, and thus significant potential for weight reduction, where lightweight design is the goal. The substitution of metal components with the polymer achieves a weight reduction in the order of 80 %, while retaining a focus on high strength.

REINFORCED, SIMPLE AND COST-EFFECTIVE

Thermal stability and stiffness can be improved by combining high-performance polymers with special reinforcing materials, such as carbon fibres. The resulting material is known within the industry as CFRP (carbon-fibre-reinforced polymer). Additives such as PTFE and graphite then improve the frictional properties. Victrex has developed wear-resistant materials with less mass, on account of a lower density, compared to solutions with metal based bearings. Furthermore, the cost of transmission components can be reduced by the thermoplastic manufacturing process. Materials such as the WG wear-grade polymer from Victrex offer excellent resistance to wear, as well as very low and stable coefficients of friction. All this contributes to high energy efficiency and a long service life.

Components made of polyaryletherketones (PAEK) can be produced at a much more cost-effective rate than the same parts made of metal or other high-performance plastics such as polyimide (PI) and polyamide-imide (PAI). Such products can be produced to within very close tolerances by standard injection molding processes and do not require any thermal post-treatment or mechanical finishing. The parts produced in this manner are off-tool parts. ❸ shows the temperature in the mating surface in relation to the PV value. The temperature of the premium wear grades WG is the lowest.

APPLICATION: COMPOSITE BEARINGS

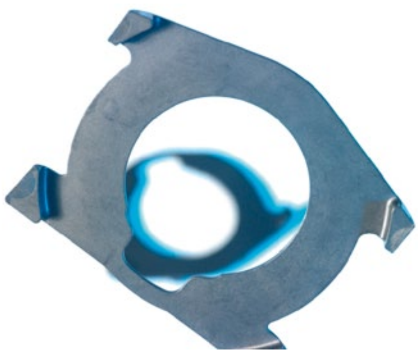
As a result of the previously mentioned properties, PEEK can be used in a multitude of applications. The product portfolio at Victrex ranges from composite bearings to moving parts, lead-throughs and even gear wheels.

Composite bearings have been specifically designed to operate with marginal lubrication. They generally consist of three composite layers: a steel supporting strip and a sintered bronze matrix, impregnated and superimposed with a polymer bearing material, such as PEEK, that contains fillers such as carbon fibres, graphite and other fillers that optimise resistance to wear. The steel support, the first layer, provides strength. The intermediate bronze layer ensures a strong mechanical bond with the cover and functions as an emergency service layer. The third layer consists of PEEK, which ensures tribological suitability. This composition guarantees high dimensional stability and more effective thermal conduction, to reduce the temperature at the bearing surface.

APPLICATION: THRUST WASHERS

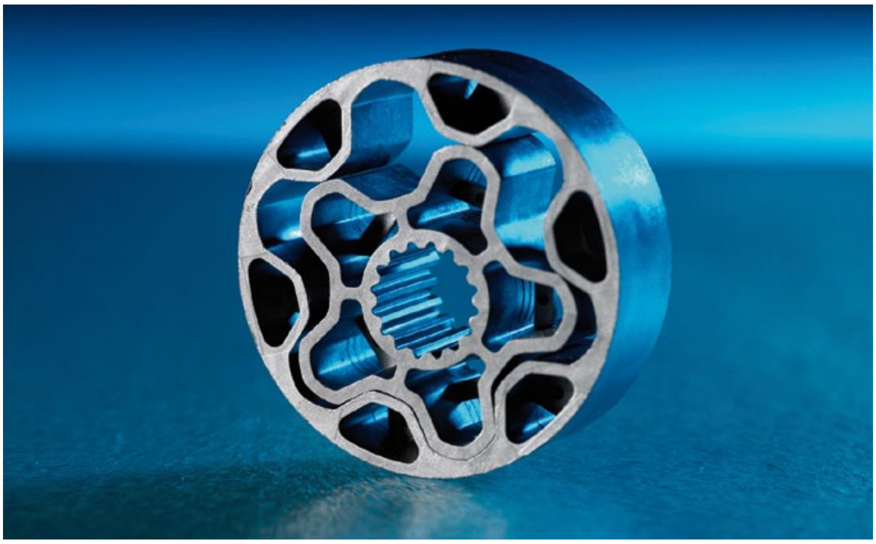
Axial thrust washers, ❹, are important components in automatic transmissions. They are often subject to wear and are made of solid plastic. Since they are used to separate the different transmission stages, they are invariably exposed to high axial forces and extreme tribological stress. The compact design of the transmission casing and the high loads result in higher oil temperatures.

As a result of their balanced properties profile and thanks to the possibility of



❹ Thrust washer made of solid plastic

❺ Rectangular sealing ring made of PEEK



❻ Gerotor pump

producing parts by injection molding, without having to resort to extensive finishing processes, PEEK polymers enable designers to develop ever more economical and efficient applications. These new areas of application now include their role as the preferred material for transmission thrust washers.

APPLICATION: LEAD-THROUGHS

Special lead-throughs with rectangular sealing rings are used for rotating shafts in infinitely variable automatic transmissions with six to eight speeds, as well as in Double-Clutch Transmissions (DCT) and Continuously Variable Transmissions (CVT). They are rated as having zero leakage and therefore ensure the oil pressure that is necessary for the rotating control and actuating elements.

Friction losses may arise within the sealing system, significantly impairing efficiency. The continuous reduction of the displacement of combustion engines (that means, their downsizing) results in ever higher rotational speeds and even increased oil pressure within the transmission. The rise in temperature caused by friction in the contact area between the seal and the shaft may result in decomposition of the oil, which can negatively impact sealing. As a consequence, optimisation of the interfacing between shaft and seal is of primary importance.

For optimised cost effectiveness and to ensure design freedom, gray cast rings have for many years been replaced by new temperature- and oil-resistant high-performance thermoplastic materials, such as PEEK. These plastics feature excellent friction and wear properties that can be further optimised by incorporating additives. A positive side effect is a simplified installation procedure for a rectangular sealing ring, ❺, as a result of their higher elasticity and reduced stiffness, compared to components made of metal.

APPLICATION: GEAR WHEELS

Gear wheels that drive auxiliary units such as Gerotor pumps must fulfil numerous requirements and utilise the advantages of polymers because various loads always arise in different combinations, ❻. The correct performance of a gear wheel in such a pump is dependent on dynamic flexural fatigue strength, and resistance to oils, greases and aging, as well as dimensional stability. In particular, gear wheels profit from having a defined thermal expansion and the fact that their dimensions remain stable within the humid environment of the lubricants. Up to now, no swelling has been detected.

The service life of gear wheels made from new grades of PEEK polymer developed for this specific application area is, according to rig testing, decades longer than that of gear wheels made from other engineering polymers (with lubrication and at 120 °C). This means that PEEK gear wheels and their adjoining components can be downsized, to reduce their mass, to a greater extent than is possible with metals, thus increasing energy efficiency and, furthermore, reducing cost, when compared with mechanically produced metal gear wheels.

CONCLUSION

Transmission components made of PEEK high-performance plastics by Victrex fulfil or even exceed customer requirements and enable cost-efficient production since no secondary processes are required. Moreover, the functions of several metal parts can be integrated in a single injection-molded component. To summarise, suppliers and producers in the automotive industry are able to meet the most discerning demands of their customers, while at the same time fulfilling all the legal terms and conditions of production. These include reduced weight, a higher power density and longer service life, and, ultimately, significantly improved energy efficiency, which then contributes to a reduction in CO_2 emissions.

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DEVELOPMENT OF A CFRP LIGHTWEIGHT DESIGN WHEEL
WITH AN INTEGRAL ELECTRIC MOTOR

Consistent lightweight design is an essential means of extending the driving range of electric vehicles. For this purpose, a carbon fibre-reinforced lightweight wheel has been developed at the Fraunhofer LBF Institute for Structural Durability and System Reliability as part of the joint project “System Research for Electromobility”.

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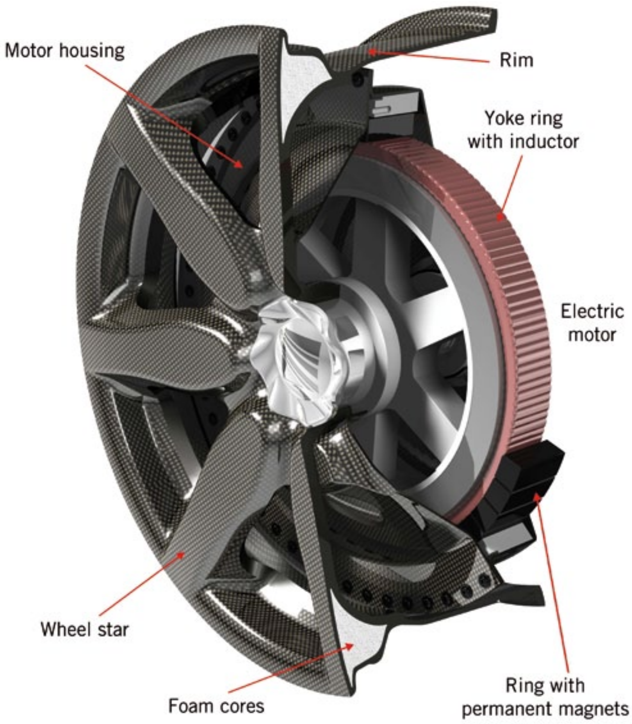
MOTIVATION

The generally well-known causalities between moving masses, energy consumption and pollutant emissions make lightweight design a key technology in automotive development. Furthermore, the heavy battery weight in hybrid and electric vehicles makes the lightweight design concept essential [1]. Maximising weight saving helps to achieve an acceptable driving range in traffic. Another important aspect is the structural durability of the safety components.

The use of fibre-reinforced plastics (FRP), if correctly designed, achieves greater rigidity, higher failure strains, higher damage tolerance and higher material damping with a lower weight compared to metals. A higher natural frequency with higher damping can be obtained by using high modulus fibres in the area where the electric motor and FRP wheel are connected (motor housing, ❶), enabling great potential for lightweight design and low noise emission.

In addition to the excellent design freedom available, another advantage of lightweight design with FRP is functional integration. Fibre-reinforced plastics are on the way to being used in high-volume automotive production. Safety-relevant components in particular, such as the chassis and wheels, are potential fields of application for various fibre-reinforced plastics due to the possibilities that they offer. Material systems exist for almost any application, from those that are very easy to process to those that are particularly effective. It is therefore not only the weight-saving potential itself that makes these materials so attractive. In contrast to metallic lightweight materials, fibre-reinforced plastics also offer an additional degree of freedom through which the material and component properties can be influenced. The type of matrix and fibre used influences the properties, as does their ratio to each other and the orientation of the fibres in the component. Weight-specific factors, such as durability and rigidity, as well as production costs can be ideally balanced depending on the requirements profile.

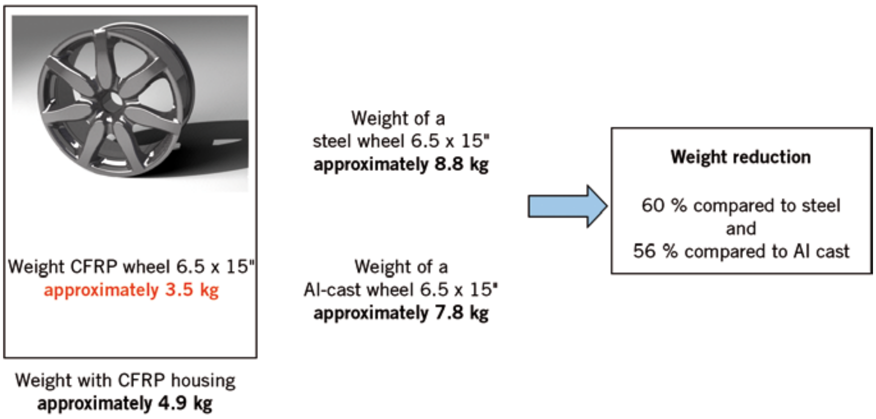
The conventional internal combustion engine is replaced by one or more electric motors in electric vehicles. A fibre-reinforced lightweight wheel with an integral electric motor was developed and constructed at the Fraunhofer Institute for Structural Durability and System Reliability as part of the joint project ”System Research for Electromobility“ in order to illustrate the possibilities of this alternative vehicle drive. The operational demands of the wheel and the influencing factors had to initially be determined in preparation for its construction. Furthermore, a design study was carried out based on existing plastic wheels that determined a variety of solutions for possible designs to mount and attach the electric motor and to make optimum use of the positive material properties. The rough concept for the carbon fibre-reinforced plastic (CFRP) lightweight wheel with an integral electric motor was developed from these solutions.



❶ Section of the lightweight design wheel with CFRP rim and motor housing as well as wheel hub electric motor

CFRP LIGHTWEIGHT WHEEL WITH AN INTEGRAL ELECTRIC DRIVE

The CFRP lightweight wheel even with a wheel size of 6.5×15" – without a CFRP housing to integrate the electric motor, without metal parts, such as sleeves for bearing and screws and without motor components – weighs approximately 3.5 kg, thus resulting in a weight saving up to 60 % compared to a steel wheel of



❷ Weight comparison of the lightweight design potential of the carbon fibre-reinforced plastic wheel

the same size, depending on the wheel design. The weight saving compared to a cast aluminium wheel is depending on the considered design up to 56 %, ❷. The overall weight of the CFRP lightweight wheel with the integral electric motor is approximately 18 kg.

The motor housing is not directly connected to the rim well. This prevents any forces acting radially or laterally, especially shocks caused by rough roads or impacts with the kerbs being transferred directly to the electric motor. Material-specific radii and flowing transitions were used in the component for continuous fibre routing in line with the flow of forces and to avoid peak stresses due to sharp edges or variations in rigidity. The motor housing is connected to the inner area of the wheel axle. Foam cores were inserted into the spokes to reduce weight and increase flexural rigidity. A smaller, commercially available wheel hub motor was used as the electric motor. The motor, consisting of a ring with permanent magnets (external rotor) and a yoke ring with electromagnets (stator), has a motor capacity of 4 kW and a drive voltage of 2×24.5 V, ②.

DESIGN

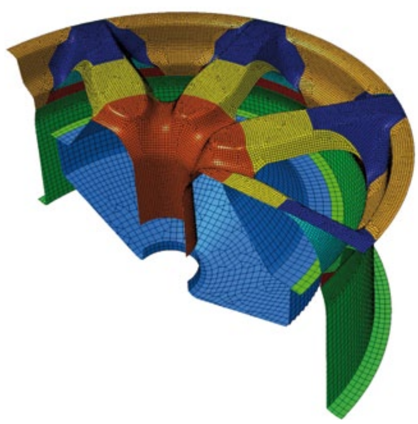
A surface model was created with the aid of a CAD system as a basis for the virtual steps of laminate definition, laminate optimisation and tool design. This already contains the final rim design, including the mould release and draft angles, but as yet no information on wall thicknesses. The laminate design was also completely created with the existing CAD software tools. The zone design was selected from the different approaches for laminate definition that the CAD system offers. A surface model is subdivided here into zones with different laminates, ❸. This method is particularly suitable for subsequently calculating and optimising the laminate using the finite element method (FEM). The laminate of the individual zones can easily be predetermined and adjusted using a table. An FEM network of shell elements was produced on the basis of the surface model with the zone geometries. The layer structure of the respective zone is transferred to the individual elements during crosslinking. The laminate of the individual element is based on a coordinate system that is specific to each zone.

Simulations were carried out for load cases applied during driving on rough roads and around bends. Based on the assumption that the components are thin-walled, the calculation is based on the shell theory, and only the stresses on the plane are considered. The stresses and the shear force on the plane were evaluated for each layer in the direction of the fibres and perpendicular to the fibres. The utilisation of the material could therefore be optimised by using several loops. The required wall thicknesses of the individual zones are determined with laminate optimisation, ❹. The prototype CFRP wheel was calculated under simplified conditions, as the programme LBF.WheelStrength, which was developed specifically for wheel design, is based on the evaluation of isotropic materials and does not yet take into consideration the existing anisotropic material properties of the CFRP wheel [2]. The identified need of action is actually subject of further work. A volume model was created from the surface model with the wall thickness information. The mould cores for the tool could then be derived from this volume model. The last design step was to create the shape of the cuts of the individual layers with a drape simulation.

MANUFACTURING

A carbon fibre prepreg system was used in order to generate the required volume content of fibres, as well as to guarantee a uniform resin-fibre distribution and to keep the equipment costs lower than infusion processes, for example. A standard system from Hexcel was selected: M49 as the epoxy resin base with a high strength carbon fibre as the technical fibre. A 2×2 twill weave with sufficient draping properties was selected as the weave of the fibre mat

❸ Structure of the zones for layer definition, storage conditions and the application of force for the load case “braking”



❹ FE model of the wheel, thickness distribution when load is applied when driving around bends for the wheel rim and spokes. By using twill weave fabric with 50 % of fibres both in a 0° direction and 90° direction, the two main directions are covered at the same time [3]. UD fibres were selected for the rim cylinder due to their geometric simplicity. The wheel, which displays a complex three-dimensional geometry, is produced in one piece. For cost reasons, a two-part mould made of closed cell rigid foam with a polyurethane base was used for the manufacturing the prototype with a low production volume, ❺. The heat conductivity of the mould, which is decisive during autoclave production, was taken into consideration. The mould was sealed and wetted with release agent before the fibre mats were introduced to prevent the epoxy resin matrix adhering to the mould surface.

A corresponding ply book listing the number of layers with their respective orientation and any other relevant manufacturing data was drawn up for the structure of the woven fabric layers of the individual sections. The layer structure was a result of the simulation carried out in the design of the rim of the wheel in which the percentage of the 0 °, 45 °, 90 ° and -45 ° layers was defined in accordance with the stresses and strains. A vacuum was built up over the mould during the application after a defined number of layers in order to press the layers together intermittently by the negative pressure produced following the evacuation of the build-up, thereby achieving a higher quality of component. The final vacuum build-up with the corresponding sequence of layers of membrane was cured for 2 h in an autoclave at a temperature of 120 °C and a pressure of 3.5 bar and tempered at a temperature of 50 °C for 16 h. The finished component was extracted, separated from both halves of the mould and then finished. The wheel, housing and wheel hub were bonded to each other after mounting the valve and fitting the tyre. Finally, the electric motor components were attached and the whole system was put into operation.

❺ Moulds for manufacturing

SUMMARY AND OUTLOOK

Fibre-reinforced wheels have great potential for lightweight designs and high damage tolerance, which is why they are ideal for use as car wheels. As vehicle wheels are highly stressed safety components, their structural durability must be proven and, similarly, the characteristic material, manufacturing and component properties must be taken into consideration. The same level of safety must be proven for fibre-reinforced wheels when subjected to the same stresses (situations with special and improper loads and environmental effects, such as temperature, humidity and ageing) as for conventional, usually metal wheels, taking into consideration where they are used.

Plastics have different failure mechanisms compared with metallic materials; this results in an individual material depending on the laminate structure (number of layers, fibre orientation and volume content of fibres) and the fibre and matrix material. The resilience of the material and stress distribution are stronger in structures with FRP than with metallic materials, regardless of the manufacturing quality and design [4]. The resulting damage mechanisms, which are very different from those of metals, can lead to radically different failure mechanisms. The damage-tolerant behaviour of FRP could be applied to a variety of applications, and even to wheels that had previously been tested [5]. At present, there are no definitive test guidelines for the approval of fibre-reinforced wheels on German roads, but work on this is currently in progress.

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EFFICIENT VEHICLE SIMULATION BY CONSISTENT COMPONENT MODELLING AND PARAMETERISATION

Due to the limited accuracy achievable in full vehicle simulation, the levels of sophistication of all the component models should be well-balanced to save expensive and time-consuming parameterisation and computation efforts: each model should be as complex as necessary but only as complex as meaningful. Hence, this study is concerned with the influence of different component-models on the simulation results in vehicle vertical dynamics as well as with the influence of their parameterisation.
It emerged from a cooperation of the Institute of Mechanics and Mechatronics of Vienna University of Technology and Magna Steyr Graz.

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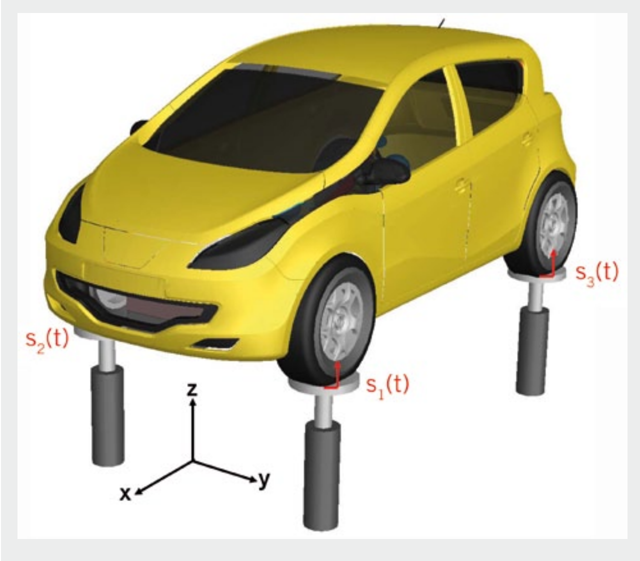


❶ Vehicle model on four-poster-test-rig

1 INTRODUCTION

The necessity to perform each step at the design of a new vehicle or during the derivative development as efficient as possible also applies to the computer simulation of the expected behaviour. This predictive purpose generally is more important than the simulation of already existing prototypes. For such calculations it is characteristic that by appropriate component models the vehicle behaviour should be forecasted without the possibility of validating the results by test data of the full vehicle. Moreover, even if good coincidence of calculated and measured data of an already built prototype has been achieved, one should be aware that in course of the series production scattering will occur. Indeed, the influence of tolerances, thermal effects, ageing and other non-controllable factors can be seen clearly already at single components [1] and may have cumulative effects in the full vehicle. Hence, the minimisation of the differences between computational and test-rig results by more and more complex component-models with high parameterisation and computation effort is meaningful only up to a certain degree. In view of the necessary efficiency it is therefore important that every component model exhibits just the appropriate level of sophistication together with a reasonable parameterisation input; this in turn is however based on a well-balanced choice of the component models and on the knowledge of the effects of the parameters occurring therein.

To this aim, using a hybrid MBS-system, in the present paper the influences of different types of component-modelling of the body, the exhaust system, and the engine- and gearbox-mounting as well as those of different parameter values of the mass geometry, the suspensions and mountings are investigated. In particular, the study focuses on the vertical dynamics, and a single obstacle, a rough road, and an expressway are each considered separately. Both the influences at significant measuring positions and global characteristic values are given [2]. Thus, not only the most sensitive components with respect to their modelling, but also the parameters to be determined with the highest accuracy may be seen.

2 APPROACH

2.1 BASIC MODEL

The investigation is based on a Four-WD vehicle with a mass of 1480 kg, which has been modelled at Magna Steyr Graz using MSC.Adams/Car 2005 r2. The 191 DOF-MBS basic model is composed of the main groups: tyres, front axle, rear axle, engine/gearbox unit, drivetrain and body. The front axle is a joint spring strut axle, the rear axle is designed as an independent suspension. The transversely mounted four-cylinder engine is connected to the body by the engine mount, the gearbox mount, and a pendulum rod.

As the very specific problems of tyre-modelling and -parameterisation are out of the scope of this study, for all the simulations the same tyre-model, namely version 5.2 by Pacejka, is applied [3]; this choice is motivated primarily by the aspect of computational efficiency. In the basic model, the engine hydromount is represented by a linear and thus only frequency-dependent lever model [4], whereas the elastomeric gearbox and pendulum rod mounts are modelled by Kelvin-Voigt elements. The trimmed body is considered to be rigid. Subsequently, the model described above serves as reference for the comparison with all simulation results obtained by other component-models and modified parameter values.

2.2 SIMULATION

The vibrations in the vehicle are simulated by an excitation on a four-poster-test-rig. There, the vertical displacements of the tyre-road contact points are prescribed functions $s_i = s_i(t)$, corresponding to the wheel base and the speed at the respective road surface, ❶. For the single obstacle and the expressway the left and right tracks are identical, for the rough road the tracks are different. Based on these excitations, for each of the modelling and parameterisation variants the accelerations are calculated at the measuring positions given in ❷. Then, the deviations from the reference model are characterised at each measuring position by a (modified) Error Coefficient of Variation (ECOV) [5] according to Eq. 1. This characteristic value represents a very sensitive measure of the difference of two signals in a time interval $[t_0, t_1]$ and is the basis of the following dis-

MEASURING POSITIONS	CHANNEL
Wheel carrier	front left, front right, rear left, rear right
Damper dome	front left, front right, rear left, rear right
Aggregate mounting	Engine mount
	Gearbox mount
	Pendulum rod (bodyside)
Seat rail	Driverside (outer rail)

❷ Measuring positions at the vehicle model

cussions; an evaluation of further characteristic values like extreme values, mean value and root mean square deviation as well as of level crossing diagrams and PSD-diagrams [6] may be found in [2].

EQ. 1

$$ECOV=\sqrt{\frac{\int_{t_0}^{t_1}[(x(t)-x_m(t))-(y(t)-x_m(t))]^2\mathrm{d}t}{\int_{t_0}^{t_1}[x(t)-x_m(t)]^2\mathrm{d}t}}*100(\%)$$

where
$x(t)$: reference signal
$x_m(t)$: mean value of reference signal
$y(t)$: actual signal

For all the simulations, the same solver-settings are applied. They have been chosen in a preliminary study to guarantee both numerical accuracy and robustness. The time increments are adaptive with a maximum value of 0.001 s.

2.3 ROAD EXCITATIONS

Three qualitatively different excitations are considered: a single obstacle, a rough road, and – corresponding to the comfort range – an expressway. All the data come from field measurements of existing roads and thus represent significant extensions to the related studies [7, 8], which are based on sinusoidal excitations. ❸ shows the geometry of the single obstacle, and the power spectral densities of the rough road, with a length of 550 m, and of the expressway with a length of 1.5 km. The single obstacle is crossed under a right angle at 40 km/h, the rough road and expressway speeds are 22 km/h and 100 km/h, respectively.

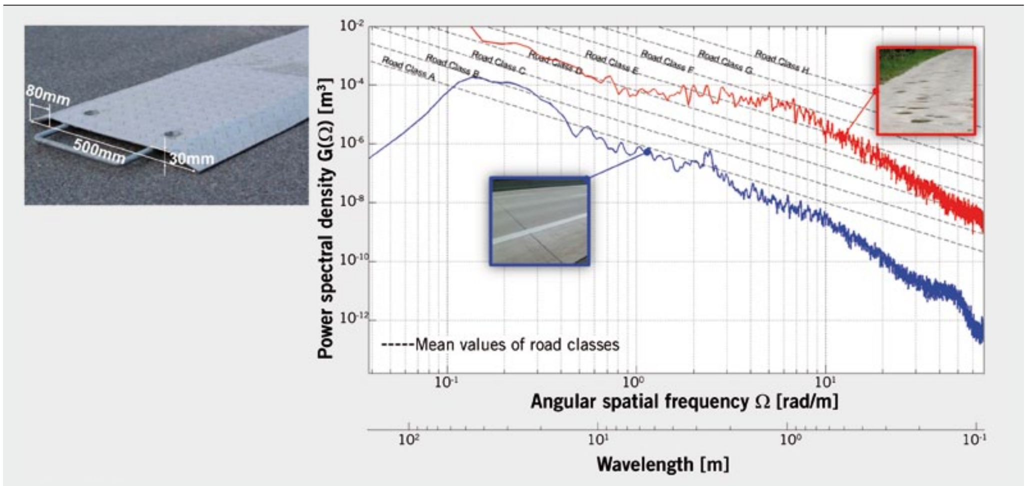
2.4 PRESENTATION OF THE RESULTS

An essential feature of this paper is the summarisation of many individual simulation results for a specific component model or parameter variation in a concise form which is applicable for practical purposes. In particular, comparisons for the different road types should be possible easily. Since primarily for different component models their effects on each of the measuring positions at the entire transfer path – from the wheel carrier up to the seat rail – are of interest, the respective ECOV values are given individually, there (see also [2]). However, at the wheel carriers and damper domes the presented ECOVs are mean values of the four wheels. On the contrary, the diagrams for the parameterisation effects always show the sums of all the ECOVs in the vehicle (with respective mean values for the wheel carriers and damper domes); note that particularly the relations of the ECOVs to each other are significant for the assessment of the effects of a variant. It shall be emphasised that these sums do not differ essentially from the corresponding (specifically comfort-relevant) relations at the seat rail.

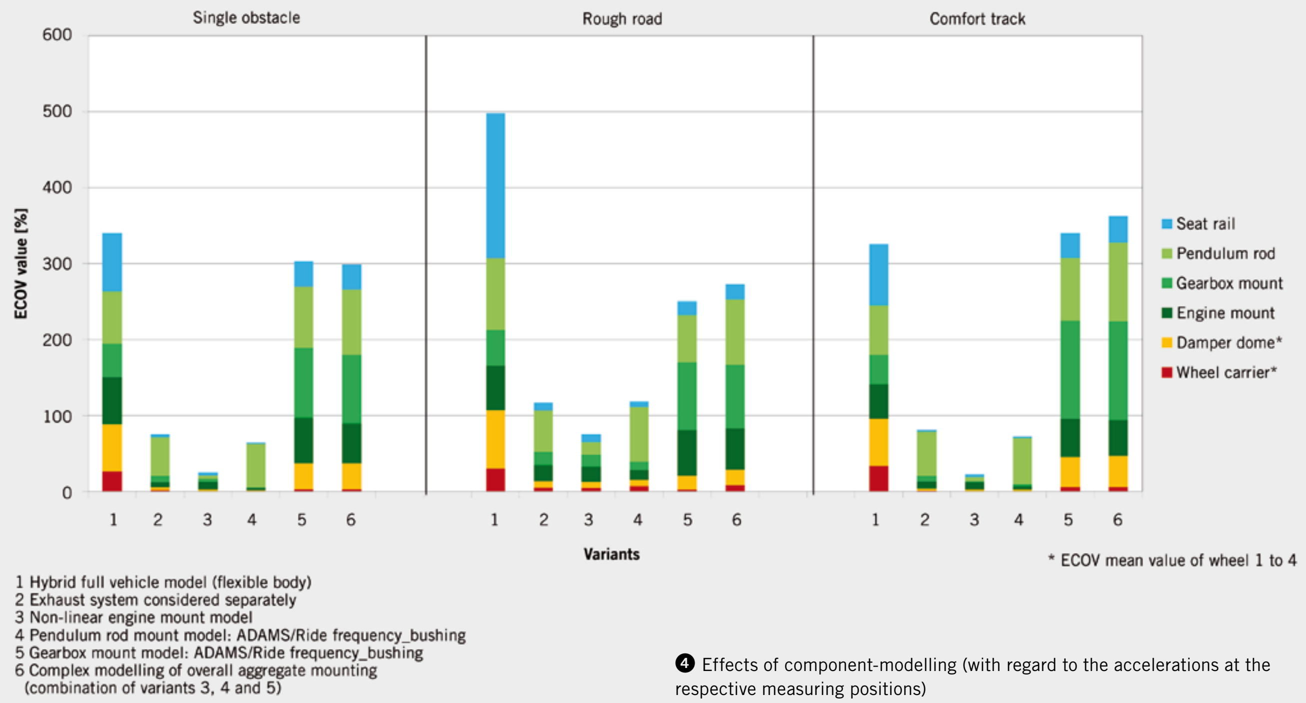
3 EFFECTS OF MODELLING AND PARAMETERISATION

3.1 EFFECTS OF COMPONENT-MODELLING

In ❹ the characteristic values for six variants with different component models are depicted, where the effect at a specific measuring position can be seen from a certain colour within the bars. For a flexible body (represented by the bars with number 1) both the body and the wheel carriers with the wishbones, which in the reference model all are considered rigid, are replaced by flexible parts. Considering modes up to 50 Hz, these elastic sub-systems are derived from FE-models and implemented into the MBS-model. The second bar represents the effects of modelling the exhaust system as a single flexible part, which is connected with the engine by a decoupling element and is elastically linked to the body by mounts. The effects of a non-linear, both frequency- and amplitude-dependent engine mount are investigated by the application of the "Adams/Ride hydro_bushing"-component (third bar). The fourth and fifth bar represent the effects of the "Adams/Ride frequency_bushing" for modelling the pendulum rod mount and the gearbox mount, respectively, and the sixth bar shows the consequences of complex modelling of all the aggregate mounts. It shall be mentioned that the parameterisation of the "Adams/Ride fre-



❸ Geometry of single obstacle, and power spectral densities of rough road and expressway



❹ Effects of component-modelling (with regard to the accelerations at the respective measuring positions)

quency_bushing" gives at 15 Hz rise to the same loss angle as in the basic version.

As expected, a flexible body model significantly influences the simulation results [7]. For the particularly comfort-relevant accelerations at the seat rail the dominating effect of variant 1 is clearly visible. Of almost the same significance is the effect of capturing the behaviour of the gearbox elastomeric mount more precisely, especially for the single obstacle and the expressway. Both of these model variants also show a noticeable feedback effect on the calculated damper dome accelerations, and the flexible body model shows one on the wheel carrier accelerations, too. On the contrary, the consequences of a more complex modelling of the pendulum rod elastomeric mount and of taking the amplitude-dependence of the engine hydromount into account are less pronounced [9]. The same applies to the separate modelling of the exhaust system, although some minor effect occurs at the pendulum rod mount due to the additional support in longitudinal direction (even in case of only vertical excitation which nevertheless causes a pitch motion). A comparison of the different excitations shows essentially analogous results for all of the three road geometries, where however at a rough road the otherwise small effect of variant 3 is somewhat more pronounced – this may be explained by the larger amplitudes and hence the more significant effects of the non-linearities.

3.2 EFFECTS OF COMPONENT-PARAMETERISATION

The effects of modified parameter values – in each case referred to the values of the basic model – may be seen from ❺. The values were increased and decreased respectively by 15 % as compared to the initial values. As one observes, the effects of variations of the masses of body and engine are significantly more pronounced than those of variations of their moments of inertia. An exception thereof is however the moment of inertia of the body with respect to the longitudinal axis at a rough road, since in this case noticeable roll motions occur; the also slightly larger influence of the engine's moment of inertia, with respect to the direction of the crank shaft, on the expressway can be explained by the excitation of an eigenfrequency. A similar significance as a variation of the masses of body and engine show modified stiffnesses of the gearbox mount and pendulum rod mount, respectively; in contrast to this, the results are less sensitive with respect to the damping of the mounts. All the parameters of the suspension also show considerable influence: as expected, the respective effects are particularly pronounced at a single obstacle and a rough road due to larger amplitudes, there. Especially variations of the wheel masses show significant effects at crossing a single obstacle, and the relevance of careful tuning of the damper parameters becomes evident.

4 CONSISTENT COMPONENT-MODELLING AND -PARAMETERISATION

The results of the previous section clearly show the main possibilities to achieve a consistent modelling of the components. On the one hand a detailed flexible model of the body proves to be important, on the other hand the way of modelling the elastomeric mounts – particularly the gearbox mount – exhibits a pronounced influence on the simulation results, too. Therefore, the application of more complex models of elastomeric mounts seems essential, whereas the consideration of the amplitude-dependence of the engine hydromount has minor effects. A separate modelling of the exhaust system may offer some advantages at rough roads, but is of less significance as compared to the previously mentioned modelling variants.

Regarding the interpretation of the effects of the component parameterisation it must be emphasised that for the sake of clarity the results displayed in ⑤ are based on parameter variations by ± 15 %, but uncertainties in measurement occurring in practice are much smaller. Evidently, the (pretty easy) exact determination of the masses as well as of the stiffness values of the mounts is very important. Moreover, all the parameters of the suspension should be determined thoroughly, and utmost care is appropriate particularly for vehicles to be used at rough roads. On the contrary, the comparatively involved measurements of the moments of inertia – except of those with respect to the longitudinal axis – need not be performed with extreme accuracy, and the same applies to the damping parameters of the mounts.

5 SUMMARY AND OUTLOOK

For the development and structural improvement of individual vehicle components highly detailed and therefore complex models will be necessary, of course. In contrast to this – because of the reasons already mentioned in the Introduction – the meaning of very sophisticated component models in full vehicle simulation is limited. Hence, the present analysis of the effects of different model types on vehicle vertical dynamics allows for the identification of both avoidable efforts and the necessity of a more complex modelling of some components. The study moreover enables one to estimate the necessary accuracy in the parameterisation of the individual components, and hence to achieve quite generally a more efficient and cost-saving simulation of the vehicle behaviour. Further advances in this regard are to be expected by future additional consideration of the longitudinal and lateral vehicle dynamics, and by taking into account different model types of additional vehicle components. Finally, it shall be mentioned that similar investigations currently are performed also at other vehicles to recognise possibly different accuracy requirements in component-modelling and -parameterisation, there.

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		Single obstacle	Rough road	Comfort track
Engine parameters	Mass_engine -15 %	175	147	158
	Mass_engine +15 %	150	127	134
	Inertia_Ixx_engine -15 %	21	23	13
	Inertia_Ixx_engine + 15 %	21	22	14
	Inertia_Iyy_engine -15 %	13	29	66
	Inertia_Iyy_engine +15 %	13	28	43
	Inertia_Izz_engine -15 %	23	32	22
	Inertia_Izz_engine +15 %	23	30	22
Gearbox/ pendulum rod mount parameters	Elast.mount_damping -15 %	32	50	29
	Elast.mount_damping +15 %	25	38	23
	Elast.mount_stiffness -15 %	159	172	182
	Elast.mount_stiffness +15 %	155	164	157
Body parameters	Mass_body -15 %	172	162	71
	Mass_body +15 %	161	161	50
	Inertia_Ixx_body -15 %	8	147	6
	Inertia_Ixx_body + 15 %	12	105	7
	Inertia_Iyy_body -15 %	43	63	38
	Inertia_Iyy_body +15 %	36	58	31
	Inertia_Izz_body -15 %	13	63	7
	Inertia_Izz_body +15 %	14	43	5
Chassis parameters	Mass_wheel -15 %	127	89	37
	Mass_wheel +15 %	115	91	36
	Front_axle_damping -15 %	109	97	43
	Front_axle_damping +15 %	94	85	37
	Rear_axle_damping -15 %	74	98	36
	Rear_axle_damping +15 %	60	80	26
	Front_axle_spring -15 %	46	109	34
	Front_axle_spring +15 %	33	88	44
	Rear_axle_spring -15 %	57	103	27
	Rear_axle_spring + 15 %	27	71	24

❺ Effects of component-parameterisation (ECOV-sums of characteristic values for the full vehicle model with regard to the accelerations)