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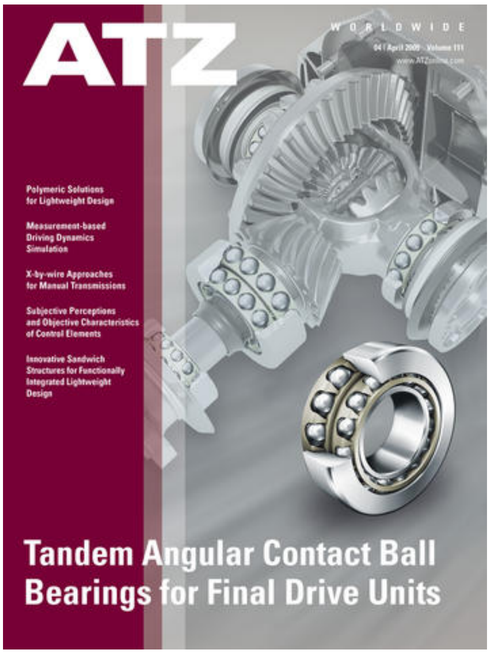

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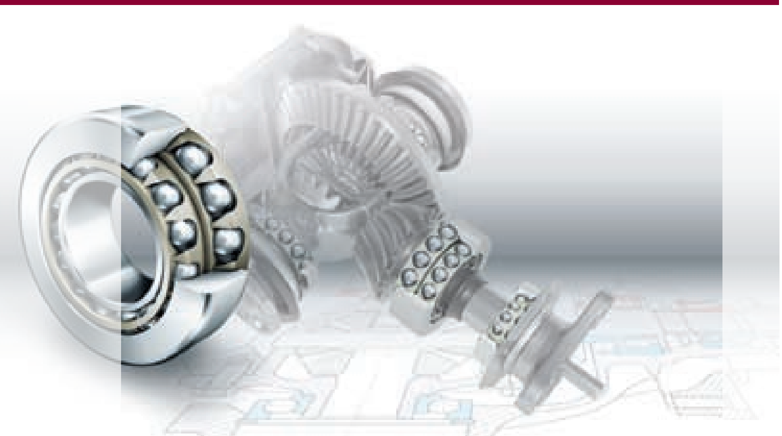
Tandem Angular Contact Ball Bearings for Final Drive Units



4
Schaeffler demonstrates that efficiency can be significantly improved through the use of **Tandem Angular Contact Ball Bearings**. The improvement is an effect of the low-friction and low-temperature running of the tandem angular contact ball bearings.

No More Fear

Dear Reader,

"Stop all this talk about the crisis and focus on the fascination of the car in your reporting," is what the chairman of a major automotive supplier said to a group of journalists he had invited to an informal dinner at the Geneva Motor Show. By saying this, he was expressing what many members of the worldwide automotive family would dearly like to see in the current situation: that, at long last, people once again start discussing sophisticated body designs and innovative engines and put an end to scare-mongering.

And there are certainly enough opportunities. The attractive, convertible versions of the Audi A5 and Fiat 500 alone provide much better advertising for the pleasure of driving than all of the gloomy economic prophesies can ever destroy. And the fact that an eco-friendly car doesn't necessarily have to be a shoebox with four electric motors in its wheel hubs is demonstrated by the new Polo Blue Motion. With its three-cylinder diesel engine and mild hybridisation, it achieves CO_2 emissions of just 87 grams per kilometre.

But nevertheless: all that is no consolation for the fact that projects and orders are being cancelled throughout the industry. Or for the fact that, in spite of all the show of confidence, many in the automotive sector fear for their very survival – and that too was unfortunately the bottom line of many of my conversations at the show. Fear, however, is a bad adviser. It numbs our minds. It was "invented" by nature to make the maximum amount of energy available for a short time to enable us to escape from a predator. But it is not there to help us solve complex tasks with a sense of proportion. Franklin D. Roosevelt was right when he said, at a critical time in the country's history: "The only thing we have to fear is fear itself."

Whatever you can do to allay the fears of your employees, suppliers, colleagues and customers, then do it. We know that we can build good cars. Now we need to show that we can also deal with a significant fall in production volumes – without tearing apart Germany's unique network of manufacturers, suppliers, universities and media.

Johannes Winterhagen
Geneva, 6 March 2009

Johannes Winterhagen
Editor-in-Chief

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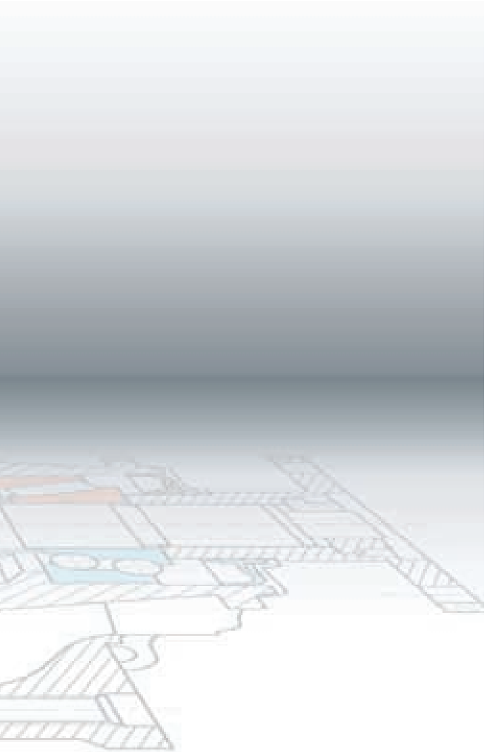
Organ of the Wissenschaftliche Gesellschaft für Kraftfahrzeug- und Motorentechnik e. V. (WKM)

Tandem Angular Contact Ball Bearings as Innovative for Final Drive Units

The final drive units designed today contain almost exclusively tapered roller bearings for supporting securely the highly loaded pinion shafts. Only recently, an innovative type of bearing support, the tandem angular contact ball bearing, has been developed. It allows the combination of high rigidity and long rating life with low friction. Schaeffler demonstrates that efficiency can be significantly improved through the use of tandem angular contact ball bearings. The improvement is an effect of the low-friction and low-temperature running of the tandem angular contact ball bearings. These characteristics become ultimately noticeable in reduced CO_2 emission and fuel consumption in the classification of vehicles types. Easy mounting and safe installation preload round out the product characteristics.



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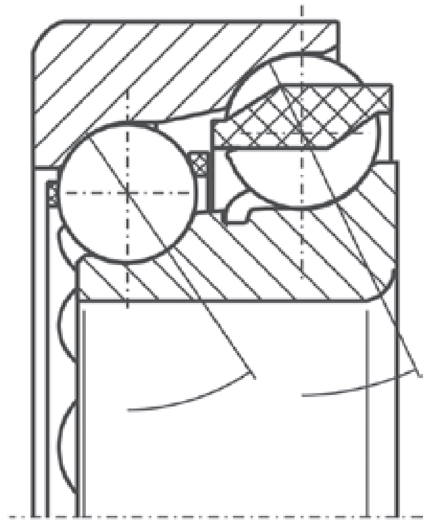

Figure 1: Principle of a tandem angular contact ball bearings

1 Introduction

Vehicles with rear wheel drive or four-wheel drive use final drive units to distribute the engine torques to the wheels. Final drive units usually also contain the differential, which ensures that the wheels can run independently. The main bearings of final drive units are subjected to very high loads as these bearing positions transmit the entire vehicle power within a very small space. Pinion bearings must be extremely rigid to keep deflection in the tooth mesh of the gear set as low as possible. Therefore, the bearing support in nearly all cases consists of bearing types that can be set clearance-free and are operated with preload. A widely used design is the bearing support by means of tapered roller bearings.

A car manufacturer can benefit relatively quickly from improvements in final drive units in the entire vehicle fleet due to the modular system of these final drive units [1]. Schaeffler KG has developed a new and innovative bearing, the tandem angular contact ball bearing, which first OEMs have successfully used in volume production [5, 7]. Tandem angular contact ball bearings are characterized by low-friction and low-temperature running. These characteristics become ultimately noticeable in reduced CO_2 emissions in the classification of vehicle types.

2 Product Presentation of the Tandem Angular Contact Ball Bearing

Tandem angular contact ball bearings are double row angular contact ball bearings whose two rows can be designed differently in terms of contact angle, pitch circle diameter, and number of balls, **Figure 1**. The design of the bearings is matched to the specific application in order to achieve low friction as well as optimum load support and load distribution in the bearing, and highest rigidity at the same time. The outer ring can usually be mounted separately from the inner ring which forms a complete unit with the balls and cages. This characteristic is often important during the mounting of final drive units when tight fits between outer ring and housing as well as inner ring and shaft are required. The bearings can be fitted on existing machines that include a nut tightening process with integrated friction measurement as required for the fitting of tapered roller bearings. A higher level of accuracy is achieved in the setting of the bearing preload in the volume production process as the frictional torques of the angular contact ball bearings vary within a very small range compared to tapered roller bearings. Unlike tapered roller bearings, tandem angular contact ball bearings are also suitable for alternative setting methods based on force-displacement measurement without rotating the shaft. Tapered roller bearings must be rotated beforehand to ensure clearance-free contact of the rollers with the inner ring rib.

Tandem angular contact ball bearings support axial and radial forces while dispensing with the unfavorable friction contact between rollers and ribs present in tapered roller bearings. Thanks to their design, tandem angular contact ball bearings do not lose preload due to running-in effects in the roller-rib contact. Only small quantities of oil are required for sufficiently lubricating the raceways, which is particularly advantageous for the pinion flange bearing during uphill driving. Tandem angular contact ball bearings are thus less sensitive to a short-term insufficient oil supply on the pinion flange compared to tapered roller bearings [4].

3 Experimental Testing

Schaeffler KG has already published results about the advantages of tandem angular contact ball bearings with regard to friction values [3]. Customer measurements have confirmed the very low friction of tandem angular contact ball bearings in comparison with other bearing designs [6, 9].

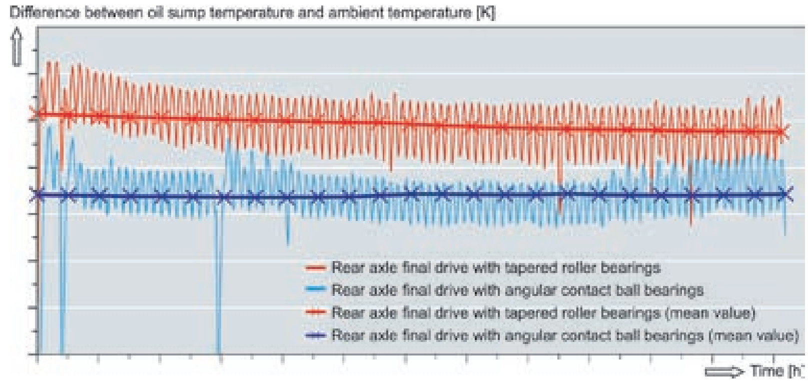

Figure 2: Difference between oil sump temperature and ambient temperature in K during the running-in cycle

Schaeffler KG has conducted efficiency measurements on complete final drive units to investigate the effect of bearing friction on final drive efficiency. For these tests, rear axle final drives of a premium car manufacturer were equipped with the innovative tandem angular contact ball bearing concept. The differential carrier bearing support of these final drive units was fitted with single row angular contact ball bearings. The volume-produced final drive units were completely equipped with low-frictional torque tapered roller bearings.

4 Bearing Preload and Running-in Cycle

The preload of the bearing support is particularly important for correct running of the teeth. It is a proven fact that tapered roller bearings are subject to a noticeable running-in effect that is accompanied by a reduction in preload. Even specially machined ribs and raceways cannot completely eliminate this effect [2]. Measurement of the preload in addition to the determination of final drive efficiency was therefore an important part of the tests conducted and commissioned by Schaeffler.

The pinion shafts of all final drive units to be tested were fitted with a special strain gauge application for measuring the elongation of the shaft. This allowed very accurate setting of the preload of the pinion bearings during preparation of the tests as well as highly precise measurement of the preload after running-in of the final drive units. An additional temperature sensor for measuring the inner ring temperature was installed on the shaft.

Prior to the actual efficiency measurement, the final drive units underwent a highly dynamic running-in program involving pinion torques of up to 600 Nm under drive conditions and 200 Nm under coast conditions as well as different speeds on the output shafts. This ensured running-in of the gear set and the bevel gears in the differential and a steady state at the measurement point. The running-in cycle was repeated over a period of 24 h. Preload, speeds, torques as well as various temperatures were continuously monitored during this time.

The final drive units with tapered roller bearings were subject to higher temperatures already during the running-in cycle despite an identical installation preload. **Figure 2** shows the development of the temperature difference between ambient temperature and oil sump temperature using one final drive unit each with low-frictional torque tapered roller bearings and tandem angular contact ball bearings as an example. The periodicity in the temperature reflects the repetition of the dynamic running-in cycles. The diagram clearly shows that the final drive units with tapered roller bearings achieve a temperature that is on average 15 to 20 K higher for the duration of the entire running-in cycle.

Preload measurements after completion of the running-in program revealed a reduction in preload by as much as 37 % in the tapered roller bearings. This re-

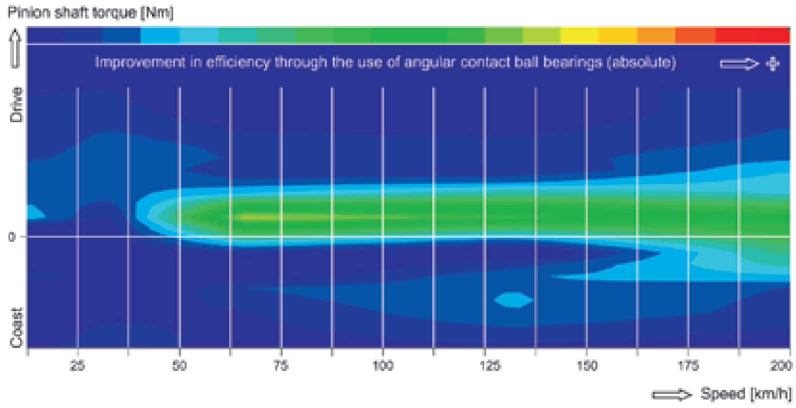

Figure 3: Improvement in efficiency at different pinion shaft torques through the use of angular contact ball bearings depending on the car velocity in km/h

duced preload is presumably mainly the result of running-in in the roller-rib contact area. Setting processes in the joint between pinion flange bearing and flange are also assumed to contribute to the reduction in preload. This effect caused by the joint cannot be eliminated by using a different bearing design. This is why a reduced preload was also observed in the final drive units with tandem angular contact ball bearings. However, this reduction of 14 % was significantly lower than that of the tapered roller bearings. During the subsequent efficiency measurement, the final drive units with tandem angular contact ball bearings did not show any further reductions in preload, which was not the case for the final drive units with tapered roller bearings.

5 Measurements

In the following efficiency some of the measurements shall be explained more in detail. This concerns the Measurement of the final drive efficiency, the measurement of the energy loss in the consumption cycle and the measurement of drag torque.

5.1 Measurement of the Final Drive Efficiency

The efficiency measurements were carried out on an electrically driven dynamometer test stand at an independent, certified test laboratory. The oil sump temperature of the different final drive units was held constant for the measurements by means of external hot/cool air fans. This way, the efficiencies could be compared under identical external conditions. The final drive units were supported by the original final drive suspension. All elastic elements were replaced by rigid connections so as to eliminate sources of parasitic energy consumption. The torque measuring equipment was adapted to the side shaft flanges and the pinion shaft flange via rigid shafts to prevent distortion of the measurement results caused by energy dissipation in the input and output shafts. A proven measurement accuracy of 0.4 % could be achieved with the special measuring equipment and the chosen measurement setup.

It is not permissible to refer to only one singular point when evaluating efficiencies. Instead, an efficiency matrix representing the efficiency for a large number of operating points (speed, torque, temperature) must be drawn up. A representation of the efficiencies over the input torque and pinion shaft speed at a constant oil sump temperature is advisable. The efficiency gain resulting from the tandem angular contact ball bearings is determined from the difference of the measured efficiency matrices. **Figure 3** shows the improvement in final drive efficiency through the use of tandem angular contact ball bearings at a constant oil sump temperature of 80 °C. Advantages are apparent over the entire speed range and are most significant at pinion torques below 200 Nm. The use of tandem angular contact ball bearings on the pinion shaft and single row angular contact ball bearings on the differential carrier side resulted in an efficiency gain of up to 9 % (absolute value) in the measurements conducted. The advantage in efficiency was also confirmed at different oil sump temperatures as shown in **Figure 4**.

5.2 Measurement of the Energy Loss in the Consumption Cycle

The significant improvements in the partial load range verified by the measurements are expected to lead to a reduction in CO_2 emissions during vehicle classification in the New European Driving Cycle (NEDC) as the NEDC mainly covers the partial load range. The final drive units thus operate exactly in the range where the largest efficiency advantage in percentage terms is achieved through the use of tandem angular contact ball bearings. Measurements on the rear axle

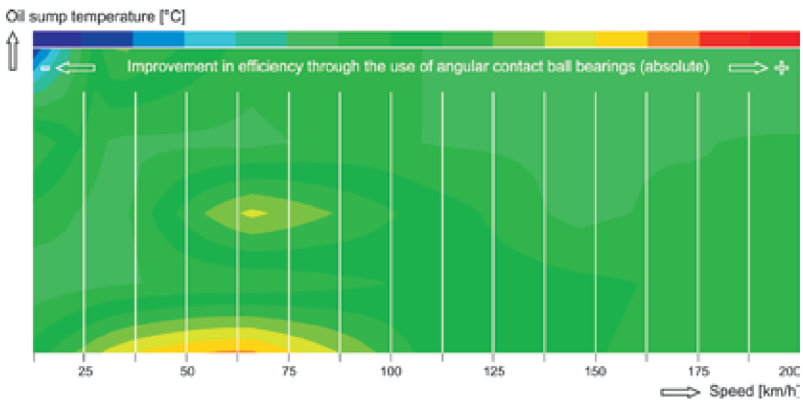

Figure 4: Improvement in efficiency at different oil sump temperatures depending on the car velocity in km/h

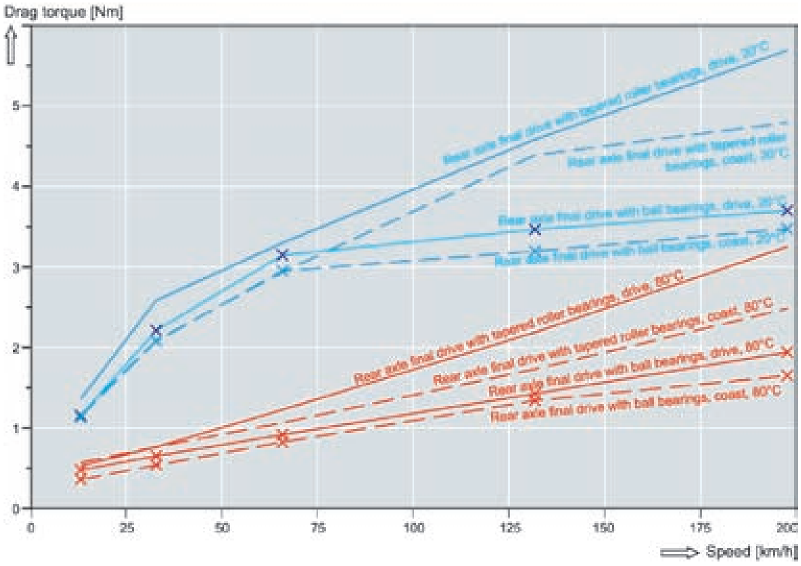

Figure 5: Drag torque measurements on rear axle final drives in Nm

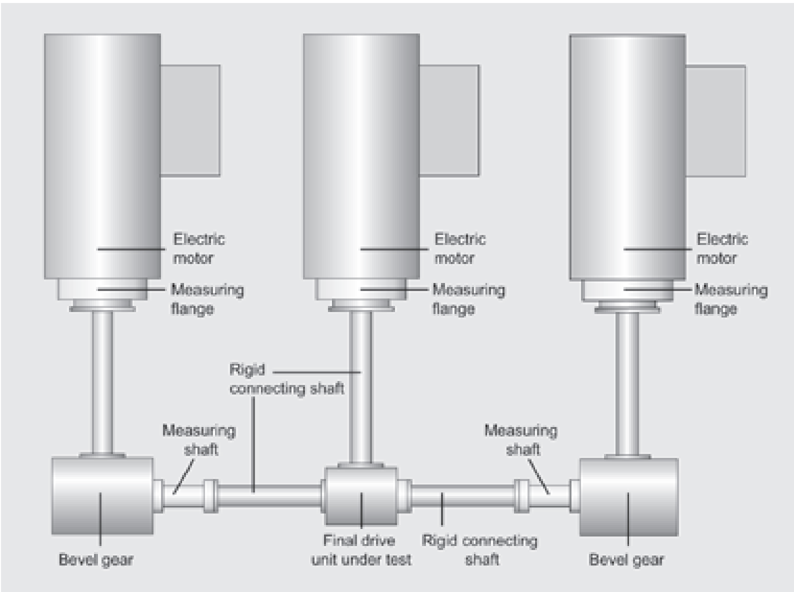

Figure 6: Electric dynamometer test bench

final drives showed a reduction in energy loss in the NEDC by 20 % versus the final drive units with tapered roller bearings. The oil sump temperature was allowed to adjust freely during the NEDC.

5.3 Measurement of Drag Torque

Drag torque measurements in the normal speed range of pinion shafts clearly demonstrated the potential offered by tandem angular contact ball bearings. The final drive units with tandem angular contact ball bearings showed, **Figure 5**, significantly lower drag torques. **Figure 6** presents a scheme of the electric dynamometer test bench.

Customer measurements during high-speed driving cycles found differences in oil sump temperature of up to 35 K between final drive units with tapered roller bearings and tandem angular contact ball bearings. In sports car applications, the new bearing support concept can therefore offer the possibility of dispensing with separate oil coolers on the final drive unit in addition to the efficiency improvement relevant to consumption.

6 Summary

The very favorable friction properties of the tandem angular contact ball bearings were confirmed in experimental testing. Replacing the tapered roller bearings with tandem angular contact ball bearings led to a significant improvement in final drive efficiencies in the final drive units tested. The improvement in efficiencies was particularly noticeable in the torque range of the vehicle classification. Tandem angular contact ball bearings of Schaeffler KG are therefore a design measure for optimizing final drive units in terms of efficiency. It is with good reason that several manufacturers have already placed their trust in the innovative angular contact ball bearing concept [7, 8, 9, 10].

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An Innovative DCT System for Sports Car Applications

Hoerbiger Drivetrain Mechatronics has developed the innovative dual clutch system for the Ferrari California. In the project, Hoerbiger was responsible for the development of the dual clutch, hydraulics, transmission control and related software. The dual clutch technology redefines the complexity of automotive transmissions. It also opens the possibility to combine fun-to-drive in sporty mode with limousine shift quality in the automatic mode. This article will guide the reader through the technology of a dual clutch, on which design requirements like shift performance and comfort, power and efficiency, torque and layout limitations are resolved without compromises. It allows the OEM to tune every manoeuvre to perfection.

1 Layout

The main objective of the overall DCT layout is to make the transmission as light and compact as possible. To this aim, all hydraulics, sensors and the clutch are fully integrated into the part of the gearbox that is carrying the differential. Due to the high input rpm of the sports car engines it is mandatory that the clutch runs in a dry sump area. The valve plate carrying the dual clutch, integrates also pump, hydraulics and sensors needed to control them. In the rear of the differential housing a second valve plate is assembled having all hydraulics and sensors to control the synchronizers and the park-lock disengagement. These two valve plates including the clutch, pump and suction tube are delivered as complete tested units to the gearbox assembly line, **Figure 1**.

An interlock system improves the safety level of the overall system, directly acting on the shift forks and causing no delay during shifting. The speed of both input shafts and the output shaft is measured using a fully integrated sensor module.

2 Control System

The DCT is controlled using a 32-bit control unit running multi-tasking OSEK. It controls all functions of the gearbox using electro-hydraulic valves. These functions are mainly the hydraulic power system, the two clutches, the eight gears

(first till seventh gear and reverse) and the park lock.

The shift paddles and other control buttons on the dashboard are permanently checked by the Auto/Manual module. In one operating mode, the system acts as a manual operated gearbox, but with a push on a button, the system is converted into an automatic transmission.

The driver requests are passed from the Auto/Manual module to the DCT manager which transforms them into individual commands for the clutch, gear and hydraulic manager modules. For instance in case you want to start driving, the Auto/Manual module asks for 1st gear. This is translated into a request to the hydraulic manager to provide the required system pressure, a request to the gear manager to engage 1st gear and then a request to the clutch manager to start a drive-away manoeuvre on the odd clutch. In parallel, the DCT manager can already pre-select 2nd gear on the other transmission halve in order to save time. If then somewhat later, the Auto/Manual module requests 2nd gear, the gear is already pre-selected and the DCT manager commands an odd to even upshift manoeuvre to the clutch manager, **Figure 2**.

The gear manager can be compared with conventional AMT systems except that the DCT has to control two separate gearboxes – one for the odd gears and one for the even gears.

The clutch manager autonomously controls both clutches as well as the engine. A tight synchronization between engine and transmission control is essential to meet the tough performance requirements of a sports car application. It generates torque commands which are converted to clutch target pressures using the torque map. The resulting target pressure is then converted to a valve current by the clutch pressure control module – a model based component capable of correctly predicting the required current for the most demanding dynamic conditions. The thermal protection module, which permanently optimizes clutch cooling (lube) and limits power dissipation in case of misuse, makes the clutch virtually indestructible.

Besides the functional and performance related requirements a great deal of attention was paid to optimizing availability and maximizing system safety. To this aim three levels of diagnosis are provided. Level 1 monitors sensors and actuators and provides backup and limp-home functions in case something goes wrong. Level 2 independently monitors all safety-critical functions (see also unintended acceleration, park-lock malfunction etc.) and reacts if faults are not detected in time by the Level 1 monitor. Level 3 monitors the correct operation of the main 32-bit CPU and uses a redundant 16-bit CPU to accomplish this. If there's any doubt about the integrity of the CPU or the signals it acquires or generates, the system is brought into a safe state.

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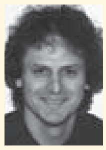

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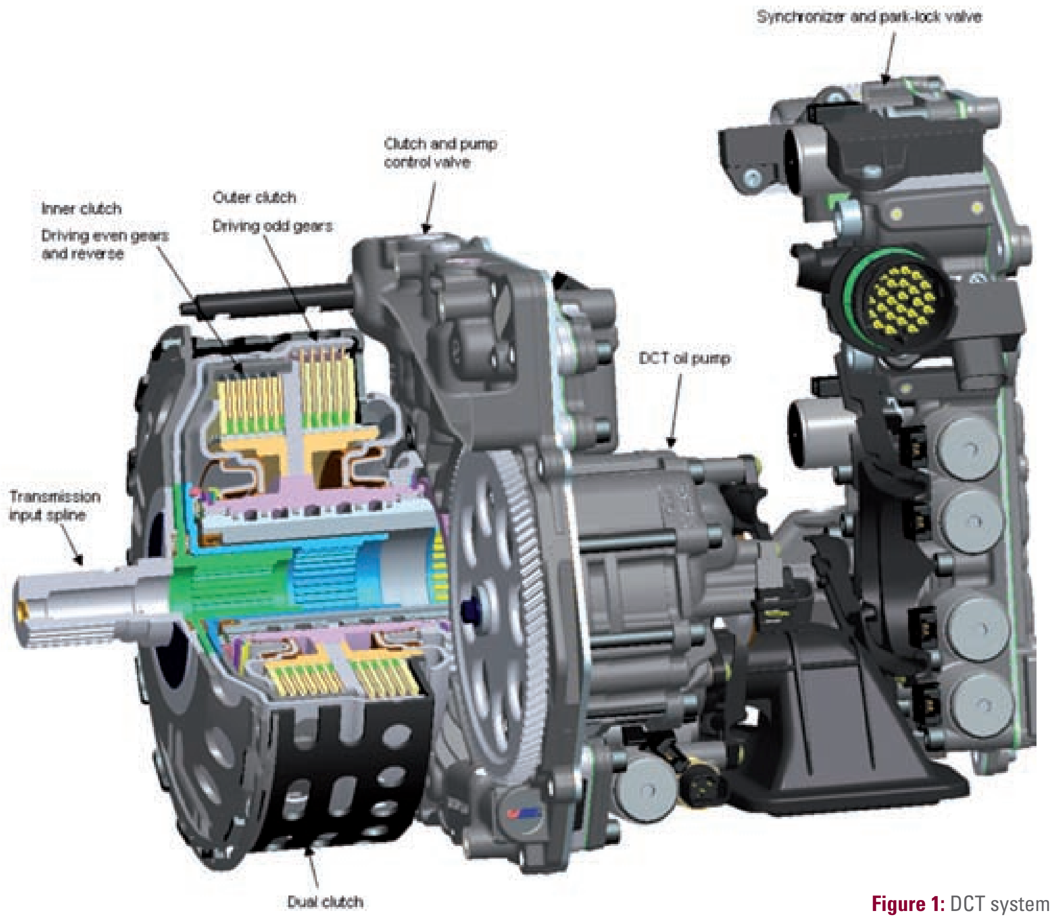


Figure 1: DCT system layout

3 Clutch Design

Capabilities of up to 750 Nm, speed of 10000 rpm are challenges, which have been brought to success but a dual clutch is much more than torque and speed. In order to obtain a clutch for a sports car with outstanding performance, it is essential to design it in such a way that the total system including software and hydraulics is optimized, taking into account controllability requirements. The most demanding are the extremely short shift times. Fast shifting requires also fast opening of the clutch. More precisely, it defines the need for a repeatable, fast and controllable reduction of the clutch torque by the clutch pressure. This is realized by strong springs which are positioned between each plate rotating with engine speed, **Figure 3**.

The clutch design is hydraulically 100 % balanced which means that the pressure-to-torque relation is basically independent from the engine speed as the balancing circuit is completely independent from the two lube circuits. While the clutch is slipping for a sporty launch, the thermal dissipated power is exceeding 100 kW. During parking manoeuvres, the car switches fluently from one driving sense to the other, only by controlling the clutches. This is thermally possible up to steep slopes since the clutch for reverse (and even gears) has almost the same thermal capacity as the clutch for

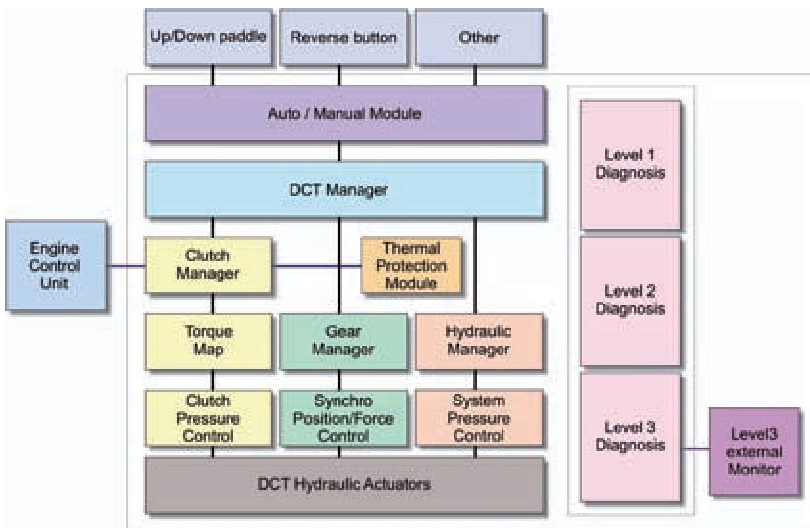
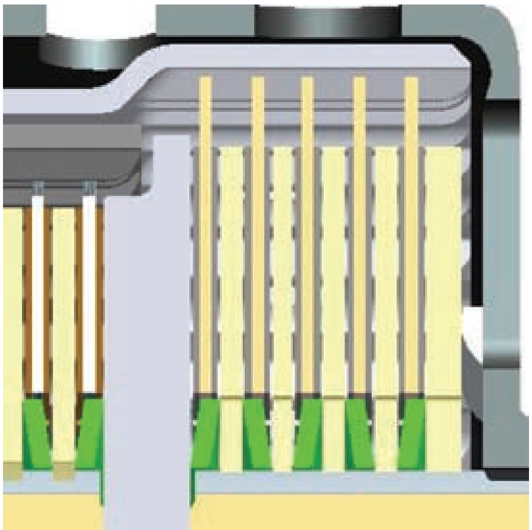


Figure 2: DCT software functional architecture

1st gear (and odd gears). The durability of the thermal challenges is validated by 4.9 GJ spent on pre-defined manoeuvres simulating different drive-away or shifts events using an appropriate control approach.

Both clutches are positioned side by side so that each clutch gets separate, fresh oil for an effective cooling. As soon as the thermal model predicts that the clutch temperature is low enough, the oil flow on that clutch is cut. The closed or open clutch can run for an unlimited time without any lubrication, thus consuming almost no power anymore and enhancing the system efficiency especially during constant driving. An intelligent split of the oil flow to the clutches limits the total needed flow, reducing the pump size and the pump driving power. The total system (clutch output to clutch input shaft) with all hydraulics, seals, bearings, splash losses and the pump included can reach efficiency values at high speed of 98.7 %.

Figure 3: Detail of the springs inside the clutch



4 Electro-hydraulic Design

The electro-hydraulic unit of the DCT system consists of two main blocks: the clutch control valve plate (CCP) and the synchro actuation valve plate (SAP). A block diagram, **Figure 4**, shows the basic layout of the system. Minimal system response times are reached by installing the main actuators (clutch and gear actu-

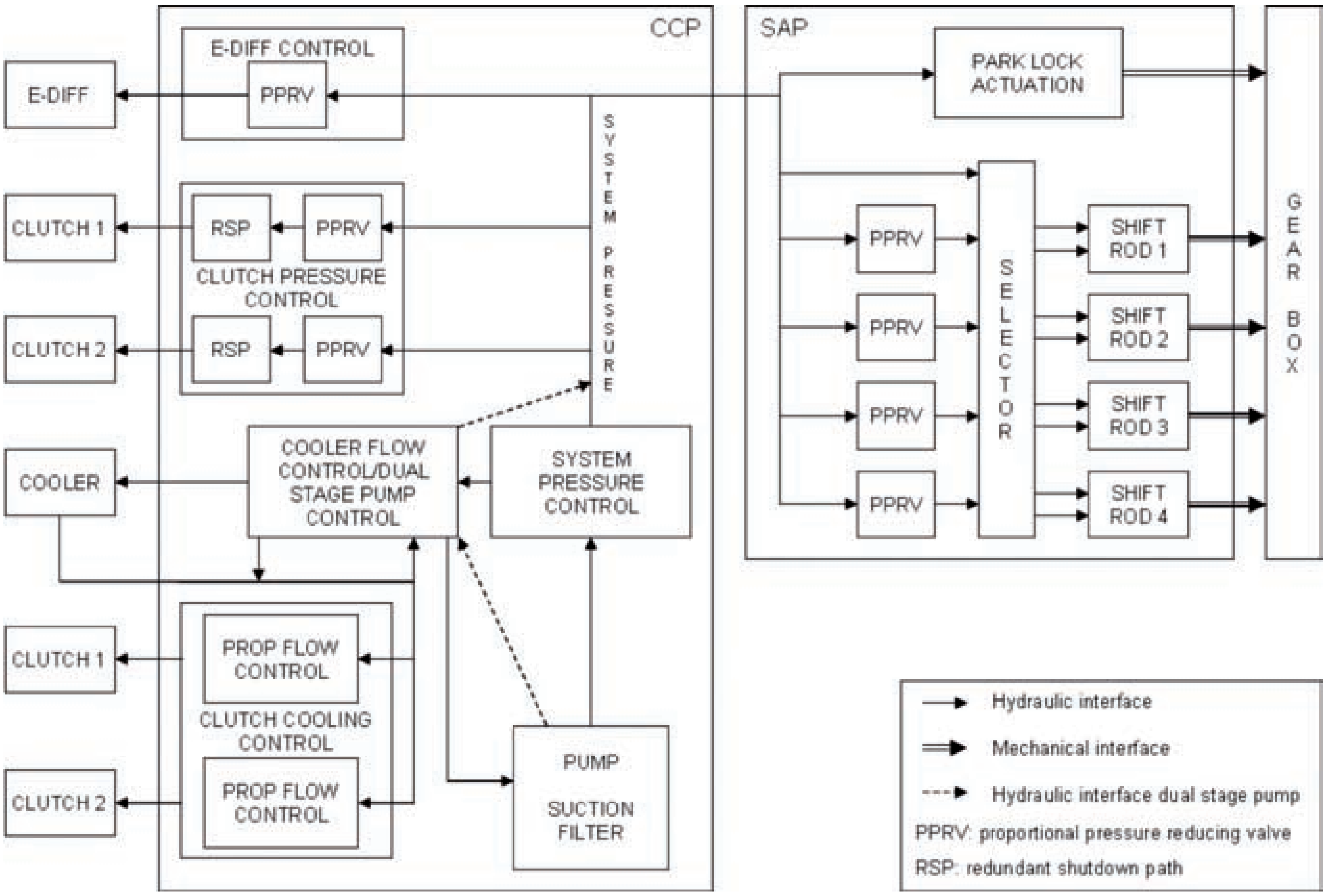


Figure 4: Block diagram of the basic layout of the system

ation systems) directly on the hydraulic unit itself, Figure 1.

The primary CCP function is to control the dual clutch: it handles the clutch pressure and cools down the clutches. To control the torque, each clutch has a proportional pressure control valve in series with a redundant shutdown path (RSP). The latter ensures an open clutch without electric activation. Both clutch pressures are measured, enabling electronic (via software) as well as mechanical closed loop (by the PPRV). The clutch cooling is also handled separately for both clutches, via two proportional flow control valves. For accurate thermal protection, the clutch cooling oil inlet temperature is monitored, as well as the temperatures of both cooling oil splash flows exiting the clutches.

The CCP carries the oil pump and system pressure regulation block. Finally, the cooler flow is also controlled by the CCP, both passively and actively by the software. When an electronic differential (E-diff) is present in the vehicle, its pressure is controlled by the CCP as well, with a proportional pressure reducing valve and pressure sensor.

The main functionality of the SAP is to perform the gear shifting, by means of four pistons. The shifting process and force are controlled by four proportional pressure control valves. The position of the four shift rods is monitored with PLCD sensors. The multiplexer valve (selector) enables the system to shift either odd or even gears, and can be moved in less than 50 ms. As secondary function, the SAP also actuates the park lock piston and measures its position.

Three major energy losses are influenced by the hydraulic system:
- the oil pump and its continuous power consumption
- the splash losses created by the pump effect of the clutch on its cooling flow
- drag torque of the open clutch(es), dependent on the cooling oil flow.

The layout and design decisions taken in the concept of clutch, hydraulic system and software enable significant reduction of these losses.

The requirements for the engine driven oil pump are very demanding: high volumetric efficiency at low speed for excellent performance at low engine revs, combined with high mechanical efficiency over the widest possible speed range. The planetary rotor (P-rotor) pump solution was selected, and developed in close cooperation with supplier GKN, enabling a volumetric efficiency of 80 % at engine idle speed.

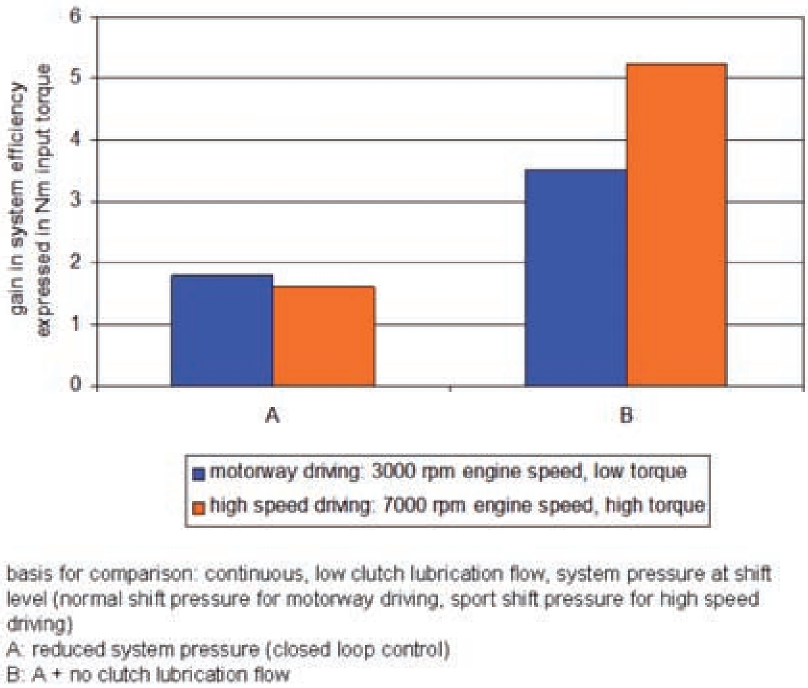


Figure 5: Effect of various system elements on power losses

The targets for the system pressure regulation are equally demanding: enable high performance shifting and maximize system efficiency while providing broad calibration possibilities for the trade-off between the two. This can be realized with a mechanical and electronic closed loop system pressure control circuit, ensuring a usable pressure range from 7 to more than 30 bar. Being able to operate at low pressure is not enough however, since the driver can request a gear shift any time. By being able to increase system pressure in just 30 ms, the responsiveness of the system remains extremely high.

An example of the above mentioned system efficiency enhancing system elements is given in **Figure 5**. As a guideline: a loss reduction of 7 Nm at 7000 rpm engine speed equals more than 5 kW of power saving.

Solenoids are crucial parts of the hydraulic system. Aiming at global system performance optimization implies that the solenoid design criteria come from an analysis of the (sub)system requirements.

This system approach to solenoid design can be illustrated by the proportional valve that controls the clutch pressure, **Figure 6**. Because of the clutch filling behaviour, this valve does not only have to be able to control the clutch pressure but also the oil flow that is sent to the clutch. Due to the presence of a clutch pressure sensor the current to force gain of the electro-magnetic coil is not the main focus in the design criteria as some dispersion on this can be compensated in the software. The focus is instead put on the predictability of the flow through the prop valve.

5 Basic Functions: Torque Map and CPC

The task of the torque map is to convert a requested torque from the clutch manager into a requested clutch pressure.

For the open clutch, the drag torque model is mainly dependent on the lube flow, the oil temperature and the clutch speeds. For the closed but slipping clutch, a friction torque model is used, which is mainly dependent on the friction characteristics of friction plates, steel separator plates and oil. Characteristic curves contain the steady-state dependency of the torque on pressure and slipping speed at different temperature conditions. Also dynamic effects are taken into account to compensate for sudden changes in requested torque and/or slip.

Lifetime effects such as setting, wear and degradation of the friction coefficient are compensated by adaptive routines in the software.

The task of the CPC (Clutch Pressure Control) module is to calculate the current for the hydraulic proportional clutch control valve so that the requested clutch pressure is realized fast and smoothly. In dynamic situations (fast changing requested pressure), this task is quite challenging due to the clutch filling behaviour: clutch piston travel is needed to close the clutch but once the clutch is closed, there is no more travel of the piston and a high oil flow immediately results in an unacceptable clutch pressure spike.

Through multi-disciplinary modelling and test activities an accurate physical model was built that can predict all important dynamic aspects in clutch pressure control.

The resulting know-how has been used to optimize the hardware, develop a model based clutch pressure control strategy and take a well-founded decision on which control parameters need adaptation.

As can be seen from **Figure 7**, this approach has enabled a proper control to fill the clutch, even if the clutch is not completely open and only needs to be filled partially. Being able to handle these kinds of challenging situations has proven to be important to realize a responsive, sporty reaction of the car in Drive Away situations.

The quality of the model based clutch pressure control strategy is illustrated on a clutch with filling range between 2.3 and 3.5 bar: Figure 7 shows measurements on a Hoerbiger test bench which is, from a clutch pressure control point of view, very similar to a car.

Cpc also contains a dedicated software module to realize high-quality clutch fillings with fill times less than 120 ms to enable minimal reaction time during shifting.

6 Manoeuvres: Drive-away, Shifting and Kickdown

Drive-away is the phase in which the car starts driving from standstill until the clutch is closed. A possible drive-away strategy is the following: as soon as the driver pushes the throttle pedal, the engine torque increases making the engine speed rise. The function of the clutch is then to act as a closed loop controller that tries to regulate the engine speed to a certain engine speed profile target. The acceleration of the car depends on the applied clutch torque, that means the acceleration profile cannot be tuned independently in this strategy.

An optimized drive-away manoeuvre predefines engine speed profile and acceleration profile independently. This can only be achieved with a full integration of the engine and clutch control, enabling the TCU (Transmission Control Unit) to be master during a drive-away manoeuvre. Obviously this requires intelligent combinations of open loop targets and closed loop controls for both the engine and clutch, **Figure 8**. This TCU-ECU (Engine Control Unit) system is also integrated with the ESP. The TCU obeys

Figure 6: Proportional control valves for sports car applications

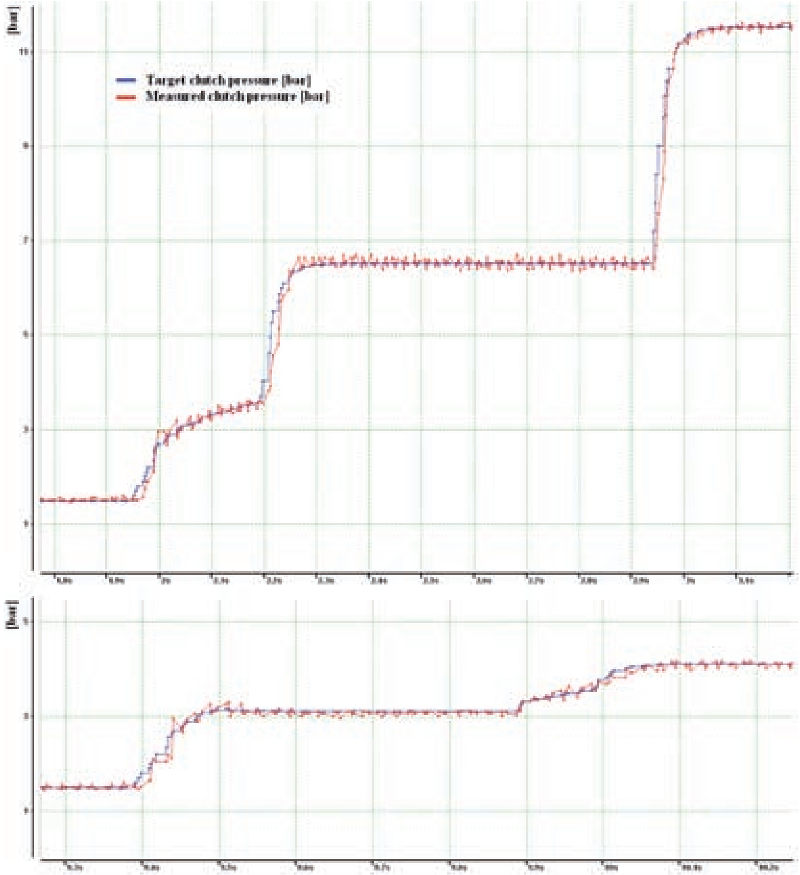


Figure 7: Measurements on test bench

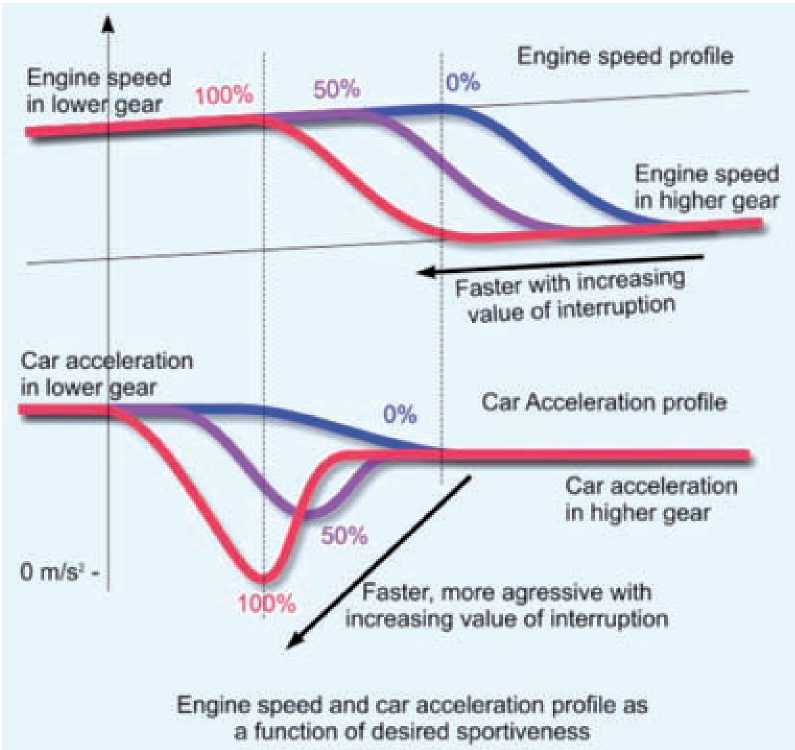


Figure 8: Engine speed and car acceleration profile as a function of desired sportiveness

the ESP requests while still commanding the ECU during a drive-away.

Modern sports cars need to have the possibility to change automatically from a comfort drive mode into a sporty drive mode. To handle this requirement a shift control method is implemented which is able to gradually change from comfortable and nearly imperceptible shifting to very responsive and aggressive shifting. Comfort shifts are performed as power-shifts, meaning there is no torque interruption during the shift. The complete engine torque is brought to the wheels during the torque handover from one clutch to the other. Therefore the software has to delay the torque handover till the incoming clutch is properly filled and ready to transfer torque.

To create a stronger driving experience it is important to make the shift more aggressive above all by reducing reaction timings. This can be done by introducing a torque interrupt by reducing the first clutch torque even before the other clutch is properly filled. During this manoeuvre the DCT is master over the engine to achieve the optimal driveline control.

Result is that in automatic mode, the car reacts very comfortable and in high performance race conditions where the driver is fully focused on being fast, the car reacts very strong and very nervous like it was impatiently waiting for the shift request.

During kickdown shifts the DCT has the ability to accelerate in the active gear while the new gear is (eventually) preselected, the clutch is filled and the engine speed is increased to the level of the new gear. This results in immediate car acceleration from the moment the pedal is pushed.

7 Modelling, Simulation, Testing and Calibration

Target is to achieve global system optimization rather than an individual component or subsystem optimization. This is realized through close interaction and know-how transfer between developers from different disciplines (mechanic, hydraulic, electronic, software etc.). To this purpose, hardware definitions and simulations take into account control strategy

Figure 9: System development testing tools

requirements and model-based control software is developed by adequate multi-disciplinary physical modelling of the (sub-)system(s). This summarizes the approach which was illustrated in previous paragraphs.

Similarly, multi-disciplinary testing of innovations goes beyond applying standard test procedures. It also means developing new testing techniques and benches to characterize the system behaviour in such a way that the results are not only mechanical or hydraulical informations, but physical characterization info suitable for usage in control software, **Figure 9**.

The Hoerbiger Drivetrain Mechatronics' part of the software contains over 5000 tuneable parameters and arrays, partly calibrated upon supply (for example CPC and Torque map), partly calibrated in close relation with the customer (for example Clutch Shifting and Driveaway).

The system development team testing and validation capabilities include Simulink, car(s) with torque measuring device, a hardware in the loop (HIL) test stand and a roller test bench.

8 Outlook

This article from Hoerbiger has shown how conflicting objectives can be solved in a fascinating way. Component and system technologies have been described, a comprehensive look into application challenges and solutions have been given.

Nonetheless, further developments are required to reduce fuel consumption. These developments will focus further on drive line integration including the integration of E-drives. The DCT technologies will not be limited to the car applications. ■

Polymeric Solutions for Automotive Lightweight Design in Body and Interior

Lightweight design calls for low-price and quick solutions to reduce the CO_2 emissions – in comparison to cost and time intensive development steps for new drives like hybrids. In the field of the body and interior design at Dow Automotive material-mix constructions of steel, aluminum, magnesium and plastics are establishing themselves today. They can be joined by crash durable bonds with very high impact peel values. Pumpable PU-based structure foams are injected in component cavities to get thinnest possible pillar wall thicknesses and to avoid heavy metal reinforcements.

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1 Introduction

The proposed EU tailpipe emission legislation defines a limit value curve of permitted emissions of CO_2 for new vehicles related to the mass of the vehicle [1]. The curve is set in such a way that a fleet average for all new cars of 130 g of CO_2 per km is achieved. A penalty per car will be based on the number of grams per kilometre (g/km) that an average vehicle sold by the car manufacturer is above the curve. A premium of 20 € has been proposed for new vehicles sold in 2012, rising to 35 € in 2013, 60 € in 2014, and 95 € per g/km in 2015 and thereafter.

Utilizing the proposed penalty system, in 2015 an OEM producing 500,000 cars with average fleet CO_2 emissions of 159 g/km (2007 European average) would pay a premium of 2755 € per car or 1.37 Billion € in total. According to different sources 100 kg of weight reduction can save 0.3 l per 100 km fuel [2], or 300 l in a 100,000-km vehicle lifetime, which is 750 kg CO_2 reduction in total, or a reduction of 7.5 g/km CO_2.

Taken another example in 2015 with 7.5 g/km CO_2 above the 130 g/km fleet average will result in a penalty of 712 € per vehicle translating into a value over 7 € per kilogram of weight saved to the OEM. At the same time the consumer would have a tangible benefit saving 390 € of gasoline over the lifetime of 100,000 km at current prices around 1.30 €/l.

Therefore, alongside power-train innovation, light weight is an important focus enabling significant reduction of fuel consumption and tailpipe emissions. Generally weight saving can be achieved by clever design of structures in the vehicle allowing down-gauging and by using low density materials. Both adhesive-based and plastic solutions have been proven in these areas. Innovations in the area of adhesives and plastics have already been introduced and adopted in recent years.

This paper of Dow Automotive will present potential weight saving opportunities when using innovative polymer materials, processes and design in the area of body construction and major interior and exterior systems. In the current economic environment, it is also expected that solutions which enable weight reduction may be more interesting to pursue quickly, because they are often very cost effective, addressing the needs to reduce CO_2 emissions without huge cost and time investments in development of new power-train solutions.

2 Body in-white Light Weighting Potential

Steel has dominated Body in-white (BIW) design for as long as cars have been produced. However, the balance of continuous improvement of BIW performance and reduced weight at the same time has become a major industry challenge. In the last ten years we therefore saw a small amount of aluminum and plastic being used for the BIW, which is mainly limited to exterior panels.

It is expected that low density materials will have a growing importance based on the current weight reduction pressure. This trend is reinforced by the fact that the industry has accepted to put a monetary value on weight reduction, which enables new technologies to be introduced a lot faster.

2.1 Joining Technology

Changing from a steel intensive BIW to a broad material mix of different kinds of steel, aluminum, magnesium and plastic will pose new challenges in joining technology as follows:

- Potential galvanic corrosion issues need to be addressed.
- Traditional spot welding joints are impossible for certain material combinations.
- Peak loads need to be prevented, in order to maximize durability.
- "Hot" joining technologies like welding can create unwanted deformation issues.

Structural adhesives have proven to be very effective in complementing traditional and existing joining technologies, to overcome the above mentioned challenges. Although structural adhesives are used in many industrial applications, they have only been introduced in crash sensitive and critical areas of a vehicle since special toughening technologies, like applied in Betamate (made by Dow Automotive) structural adhesives enabled very high impact peel values. This made the structural bond also a "crash durable" bond.

Besides the increasing material mix that is used to reduce the BIW weight, there is another important trend contributing to lightweight solutions: the increasing use of spot-welded high strength steel (Trip steels). Weight saving of up to 10 % (approximately 30 to 35 kg) can be achieved with (advanced) high strength steel contents of 50 % and more of BIW. However, there are also some drawbacks to increased high strength steel usage:

- Spot welding cost and welding investment are considerably higher.
- Thin walls lead to bending stiffness reduction, as the yield strength increase does not compensate for a wall thickness reduction.
- High strength steels are more brittle, which potentially reduces fatigue performance.

Structural adhesives allow less spot welds with increased stiffness and fatigue performance thus compensating for the challenges that we see when using high strength steel, **Figure 1**. This makes high strength steel and structural bonding an ideal and powerful tool for BIW weight reduction.

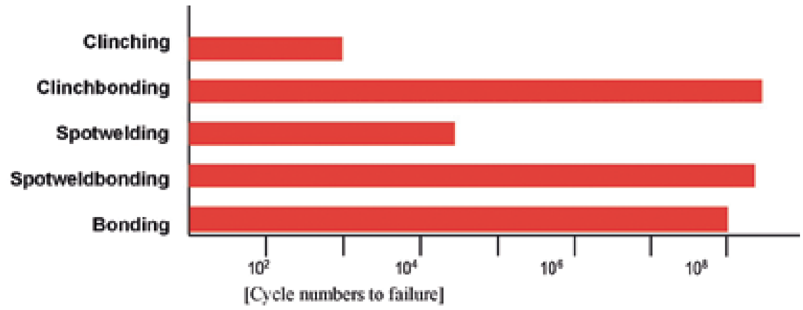

Figure 1: Cycle number to dynamic fatigue performance for different joining technologies with high strength steel

2.2 Structural Foams

A new solution on the market, are pumpable high density foams that also reinforce the BIW for weight reduction. Such a fast reacting polyurethane (PU) based foam is injected in cavities after painting to allow for minimum steel reinforcement and thinnest possible wall. Advantages of this solution are:

- There is no tooling needed, so many application areas can be foamed with a single robot and foam equipment.
- There is a high flexibility in fine-tuning the solution even shortly before start of production.

Also, structural foams allow a more effective use of higher strength steels, by reinforcing only in those areas where needed. Current studies have shown a weight reduction potential of up to 7 kg per vehicle depending on the application. The structural foam solutions are very effective when solving crash performance issues for the lowest cost per kilogram of weight saved.

3 Components and Modules Light Weighting Potential

The use of plastic as metal replacement is not new to the automotive industry. On top of its use for aesthetic, comfort and acoustic parts, polymers have increased their share of the vehicle weight to approximately 14 % in EU cars being used as metal replacement in different components and modules for interior, exterior and under-the-hood applications.

3.1 Currently in Production

In the last decade a lot of material science and application development has led to a successful introduction of new concepts that we will cover in a brief overview, graphically represented in **Figure 2**.

The fully structural plastic instrument panel (IP), introduced in the late 1990s in North America, used on several mainstream SUVs and pick-up trucks. Leveraging the ductile behavior of plastics like PC/ABS, optimizing vibration welding process and engineering greater functional integration (air ducts becoming integral part of the IP structure) saved 2 to 4 kg vs. conventional metal cross car beam structures.

The integrated plastic knee bolsters were produced for the introduction of safety rule FMVSS208 for lower extremities injury protection. Utilizing ductile PC/ABS enabled the elimination of most of the metal structure, yielding 1.5 to 2 kg saving.

Door panel outer skin, a full plastic body panel on the Saturn from GM through newly developed mineral filled PC/ABS plastic with excellent ductility (even at low ambient temperature), enabling 1.5 kg saving and no denting issues. Additionally the controlled coefficient of linear thermal expansion (CLTE) allows off-line painting with attractive material application.

Vertical body panels, patented on-line capable PU RIM for exterior body panels replace steel for large vehicle builds with enhanced design freedom, reduced damage issues and mass of 1 to 3 kg.

Bonded hybrid metal-plastic front end carrier (FEC) are first introduced on the current VW Polo (model year 2009) and then adopted on five additional platforms globally. The highly engineered LGF-PP based concept leverages the innovative Betamate LESA (Low Energy Surface Adhesive) technology, enabling adhesion of coated metal on untreated PP without surface treatment. The 2005 SPE Award and 2006 R&D-100 award recipient, delivers 1 to 2 kg mass reduction compared to a conventional plastic-metal FEC, at the front of the vehicle.

The structural blow molded rear seatbacks have been engineered utilizing specific PC/ABS grades and introduced first on the Audi TT coupe, enabling 2 to 3 kg weight savings and meeting luggage retention safety requirements. Further commercialization is done on three other vehicles in Europe and additionally two vehicles in North America, including a large volume platform model (>150,000 cars/year).

A bonded PC roof saves 4 to 5 kg per vehicle – the weight saving potential of approximately 40 % of polycarbonate compared to glass and the elimination of glass cracking were the main drivers to use a plastic material for the bonded Smart Fortwo panoramic roof. Adhesive technologies enabled the assembly of dissimilar materials dealing with different CLTE and achieving appropriate adhesion on the scratch resistance coating.

The bonded plastic tailgate of the former Mercedes-Benz A-Class in the late 1990s showed the potential of part integration, weight saving of 3 kg and design flexibility. Due to the replacement of steel with PP GMT for the inner structural part and a Polyamide material for the exterior skin part, a bonding system was required which was able to match the different CLTE and achieve robust adhesion on the low energy plastic surfaces, adhere on the integrated rear glass and provide fast curing properties.

In all the cases above the key ingredients of the successful development and validation have been the fundamental understanding of application requirements, the material science knowledge and the ability to customize the polymer technology to match the application needs (reverse engineering). In-depth processing knowledge as well as selected development partners (Tier2s and Tier1s) have been the key to successfully delivering these innovative components and modules to the OEMs.

3.2 Further Developments

Under the increasing pressure to reduce weight, further polymer science based developments are being executed to address modular concepts. For front seats, tailgates and instrument panels but also roofs they are presented in the following more in detail.

Considering that approximately 40 % of the overall interior weight is represented by seating, the seats and specifically their structure are a prime target for lightweight engineering by a plastic intense front seat structure. Developments consider various thermoplastics and thermosets and adhesives. The objective is to achieve functional integration (headrest, airbag housing, cable clips, springs) whilst still meeting all regulations. Target reductions are:

- 30 % weight
- 60 % part count
- 35 % assembly operations
- 15 % estimated overall cost.

With mass ranging between 10 to 20 kg, tailgates offer a great potential for metal replacement with plastics. A basic option for plastic intense tailgates is to utilize a low CLTE TPO, off line painted exterior skin, bonded to an LGF PP inner frame. The ability to mold the structure in color with good aesthetics enables the elimination of interior trims offering weight and cost reductions. For more demanding requirements a hybrid metal-plastic bonded variation of this baseline concept or thermoset interior structures represent the next best alternative to achieve low weight objectives without moving to very expensive lightweight alloys. Overall weight savings of 30 % versus incumbent steel solutions are the objective of these developments.

In specific BIW designs, a rigid A-pillar to A-pillar connection is required. A highly integrated hybrid bonded instrument panel structure made of metal/plastic utilizing a simple, light- and cost-effective metal profile, adhesively bonded to a thermoplastic structure, integrating the air ducts has been validated, yielding:

- 20 % weight reduction
- significantly lower tooling investment
- improved packaging space
- improved lifetime.

Figure 2: Examples of lightweight structures currently in production and further developments

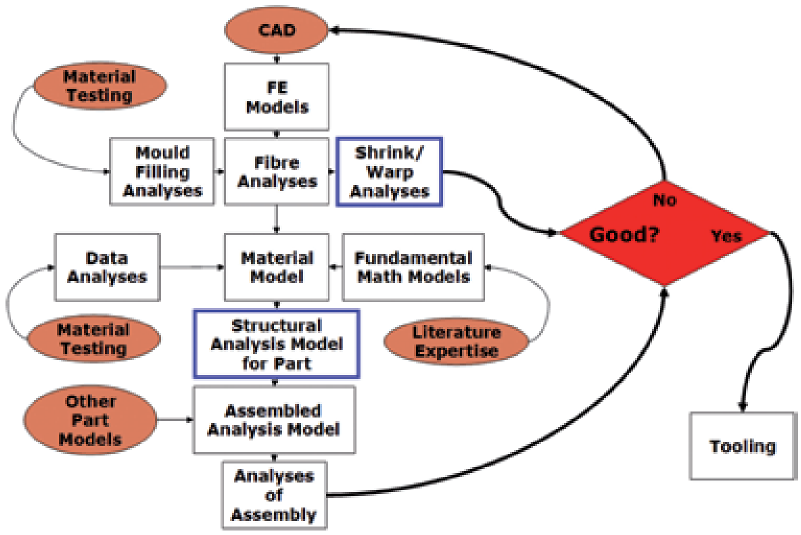

Figure 3: Use of material science to optimize part and system design

The concept of a modular front of a instrument panel is to provide a plastic intense lightweight module to be delivered just in time to the assembly line and subsequently bonded on the BIW giving:
- up to 30 % weight reduction
- improved acoustic performance and noise reduction
- better sealing
- offline assembly and testing
- elimination of blind operations
- functional integration.

Considering that the roof area is one of the biggest surfaces of the passenger car the weight saving potential of a bonded structural roof is expected to be high at 6 to 10 kg. Beside the weight saving opportunities, the design flexibility and increased driving performance (lower center of gravity) are further advantages using lightweight materials like fibre reinforced plastics and aluminum instead of steel. Bonding technology is expected to be the ideal assembly technology as it is able to match different CLTE, ensure structural performance, provide sealing properties and fit with its curing properties to the assembly line cycle.

4 Material Science and Engineering Approach to Lightweight

In all these developments it is essential that the solution is engineered as a complete system. In these developments we combined application engineering and material science with expertise in fabrication, industry knowledge and customer requirements, to achieve:
- weight reduction
- increased functionality
- cost effectiveness
- improved assembly methods
- long-term sustainability.

It is extremely important to master a wide variety of engineering tools, such as CAE capabilities for processing, structural, thermal, fluids and crash simulation and extensive, versatile rapid prototype resources, to reach an optimized solution. Another important contribution is the development of material models for use in application development. These models build on molecular models used by chemists and combine these with engineering-driven process and application modeling. This links the micro- to the macro-scale models and enables further optimization of design, manufacture and assembly to minimize weight.

One such example is the modeling of fibre-filled materials from the injection-moulding process, through the effects of moulding on fibre orientation distributions to final mechanical behaviour in a total assembly, **Figure 3**. This approach makes use of mico-mechanical models to link fibre orientation and length distributions, resulting from the moulding process, with the matrix and fibre material properties to predict behaviour under various time, temperature and loading conditions.

One of the critical breakthroughs in development of plastic-intense structural applications, such as knee bolsters or seat structures, was the understanding of plastics in crash situations. The effects of strain-rate on material properties were measured and studied at an early stage. The conversion of the material properties into validated models for use in crash simulation was critical to the ability to develop new innovative solutions and minimize weight. These validated models were then used in crash simulation to develop solutions virtually before any steel was cut, reducing costs and development time.

5 Conclusions

This paper of Dow Automotive summarized some of the possible applications for polymers and adhesives to substitute and complement metal, delivering lightweight to BIW, components and modules. Most of the innovations presented have been commercialized already demonstrating how metal to plastic conversion, hybrid construction and adhesive bonding could be cost effective even in times when the pressure on weight saving was not as strong as it is today. We expect to see a significant proliferation of some of the technologies already introduced as well as continued development to deliver some of the new concepts we described.

Clearly, in order to maximize the benefits of total system design, it is essential to combine engineering and material science to create effective solutions. Through the use of CAE and advanced material models, combined with innovative application design, material and fabrication process, it is possible to reduce weight and still maintain or improve the performance of the vehicle. If an OEM would apply all of the weight saving technologies described in this paper, overall savings above 50 kg could be achieved, with the consequent positive impact on the bottom line as well as on the environment.

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Measurement-based Driving Dynamics Simulation

The properties of the axles and tyres play a key role in the simulation of driving dynamics. In most cases, the axles are represented as multi-body systems. In the measurement-based driving dynamics model that is used in passenger car chassis development at Volkswagen, the axles and tyres are described completely by measured data. The axle data are gained from measurements taken on the Kinematics-and-Compliance (K-and-C) test rig, while the tyre data are from flat track measurements.

1 Introduction

The familiar features of this type of modelling are its real-time capability and Hardware in the Loop (HiL) suitability. Less attention has so far been paid to its relevance for testing and chassis tuning. Furthermore, this innovative tool is extremely suitable for benchmarking, as data generation on the K-and-C test rig requires the same low amount of effort for the company's own vehicles as for those of competitors. This article describes the efficiency of measurement-based driving dynamics with the aid of extensive and detailed measurements taken during road tests.

Kinematics and compliance are measured on the K-and-C test rig as shown in the **Cover Figure**. Kinematics describes the controlled wheel movements caused by wheel travel and steering. Compliance describes the changes in the wheel position due to forces and moments acting on the wheel. During driving, kinematics and compliance are superimposed.

In order to allow real driving manoeuvres to be simulated even to the limits, numerous special measuring procedures need to be carried out on the K-and-C test rig. With the superimposition of individual load cases, the acquisition of the complete data of a chassis involves approximately 180 measurements and takes between four and five days. As a result, the lateral and longitudinal dynamics can be comprehensively studied. If only partial aspects are to be examined, for example for benchmarking lateral dynamics at a constant speed, the number of measurements and time required are considerably reduced.

2 Tyre Forces and Tyre Positions During Driving

In order to examine the suitability of measurement-based driving dynamics simulation, a test vehicle was fitted with a steering machine, a measuring platform, Correvit sensors, a track measuring system and dynamometric hubs and was subjected to various lateral dynamics tests. The measurements, which were performed for the first time in this depth, can be used to test the calculation results right through to changes in wheel positions and wheel forces.

During steady-state circular driving, the wheel forces acting on the outer wheels increase linearly with the lateral acceleration, while the forces on the inner wheels decrease linearly accordingly, **Figure 1** top left. The lateral forces acting on the tyres increase linearly on all wheels up to approximately 2 m/s², Figure 1 top right. Above that acceleration, the lateral forces increase progressively on the outer wheels and increase degressively on the inner wheels. Above 6 m/s², the lateral forces on the inner wheels even start to decrease. At low lateral accelerations of up to just under 2 m/s², the lateral forces on both wheels of an axle are transmitted in approximately equal amounts. At high lateral accelerations, more than 70 % of the lateral forces act on the outer wheels.

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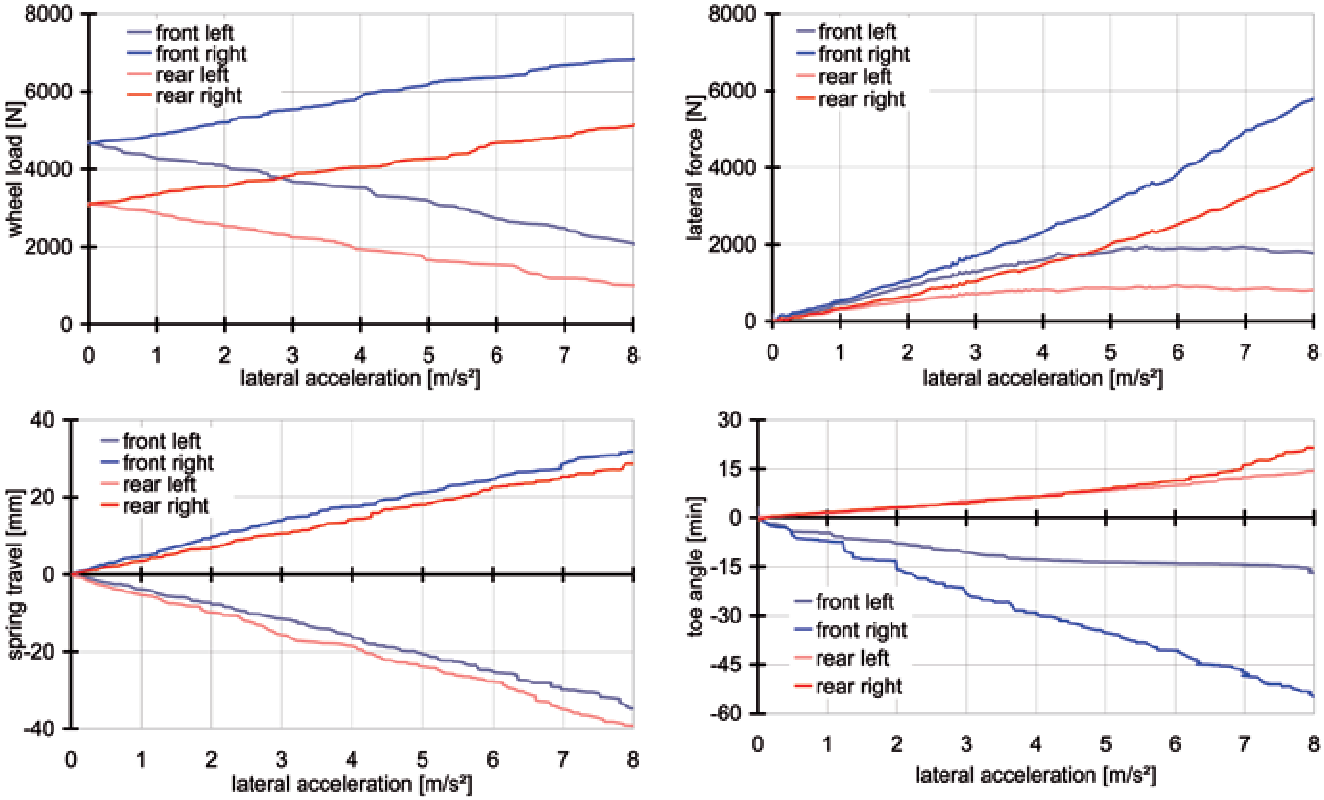


Figure 1: Top left: Wheel loads in the 100 m circular track, anti-clockwise; top right: Lateral forces in the 100 m circular track, anti-clockwise; bottom left: Spring travel distances in the 100 m circular track, anti-clockwise; bottom right: Toe angles in the 100 m circular track, anti-clockwise; the toe angle changes of the front wheels were adjusted by the proportion of steering

The changes in the spring travel show that the vehicle presses down on the suspension and undergoes a slight diagonal motion, Figure 1 bottom left. The spring compression travel on the outer wheels is smaller than the spring extension travel on the inner wheels. The vehicle presses down on the additional springs. The outer front wheel on the right compresses the spring slightly more than the outer rear wheel. The inner rear wheel on the left extends the spring slightly more than the outer rear wheel. Consequently, the vehicle undergoes a diagonal motion.

All four wheels exhibit stable, understeering changes in the toe angle, Figure 1 bottom right. The highest proportion is performed by the outer front wheel, with 54 min toe-out at 0.8 *g* lateral acceleration. The inner wheel then has 15 min toe-in. The two rear wheels steer almost in parallel, the outer wheel with toe-in and the inner wheel with toe-out.

During extreme slalom driving, the transmitted rear axle lateral forces in **Figure 2** alternate in accordance with Fig-

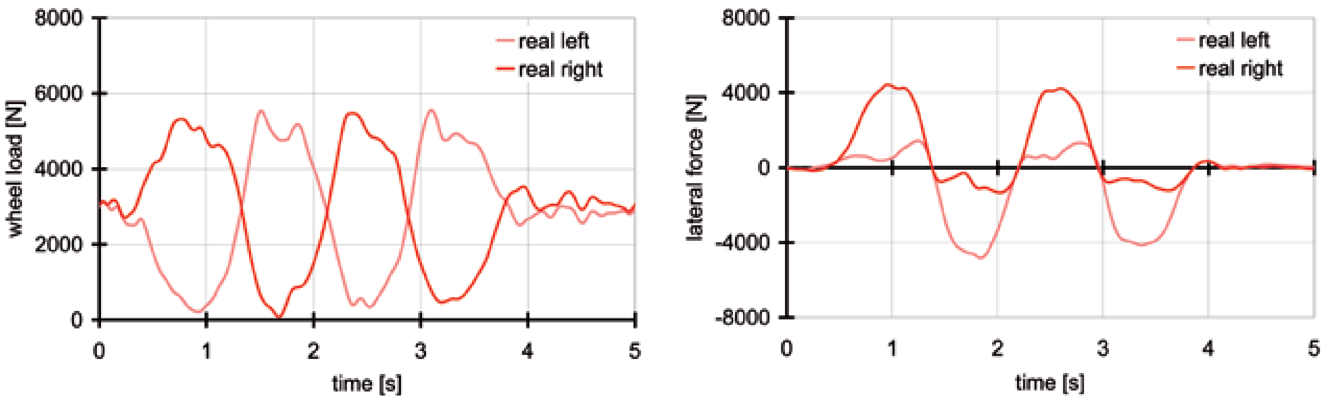


Figure 2: Wheel loads and lateral forces on the rear wheels during the extreme slalom test; driving velocity approximately 60 km/h

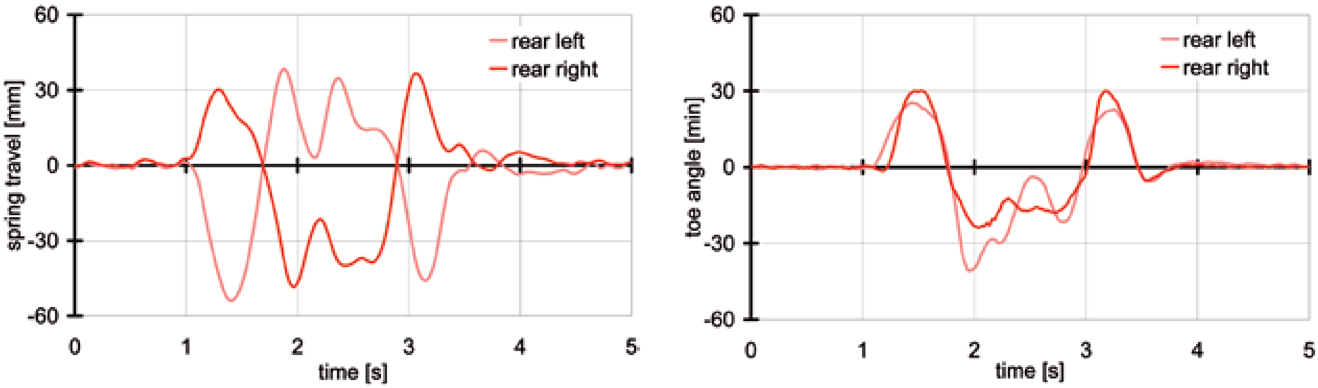


Figure 3: Spring travel distances and toe angles on the rear wheels during the double lane change test; driving velocity approximately 80 km/h

ure 1 top left and right between maximum values at high wheel loads and minimum values when the vertical load on the wheel approaches zero. In detail, the dynamic test shows time displacements between the change in the wheel load and the build-up of the lateral force.

The double lane change shown in **Figure 3** is a further extreme and highly dynamic driving manoeuvre. Here, the spring travel distances are much greater and more asymmetrical than in the steady-state circular driving, Figure 1 bottom left. The understeering toe angles at the rear wheels are almost twice as great as during the steady-state skidpad test at 0.8 g, Figure 1 bottom right.

3 Vehicle Model and Axle Parameterisation

The vehicle model is based on the programme "veDYNA" from the company Tesis Dynaware implemented in Matlab/Simulink. Details can be found, for example, in [1]. For that reason, only two short examples of axle parameterisation will be given here. The procedure for axle kinematics shows alternate wheel travel as an example, **Figure 4**. Both axles are alternately moved with controlled forces in their suspension height states of curb, full and 1.5 times full. The axle load in each case is kept constant. Essentially, the track curve remains the same at the suspension strut front axle. The multi-link rear axle, on the other hand, has a more complicated picture. This axle has a different toe angle at each suspension height. Based on these different values, the axle then develops the respective toe curve.

The lateral force steering at the front and rear axle at different height states can be used as an example of compliance parameterisation, **Figure 5**. As is typical of the suspension strut-type front axle, the toe angle change due to lateral forces is largely independent of the suspension height. In the case of the multi-link rear axle, on the other hand, the toe angle change becomes greater as the suspension gets lower.

4 Results of the Driving Dynamics Simulation

4.1 Comparison Between Measurement and Computation

The comparison between measurement and computation is the touchstone of every simulation model. The steady-state circular driving is the most demonstrative and simplest lateral dynamics driving manoeuvre and is particularly well suited for a comparison. In the comparison, it was found that, for the wheel forces and motions shown in Figure 1, there was a good accordance between measurement and calculation.

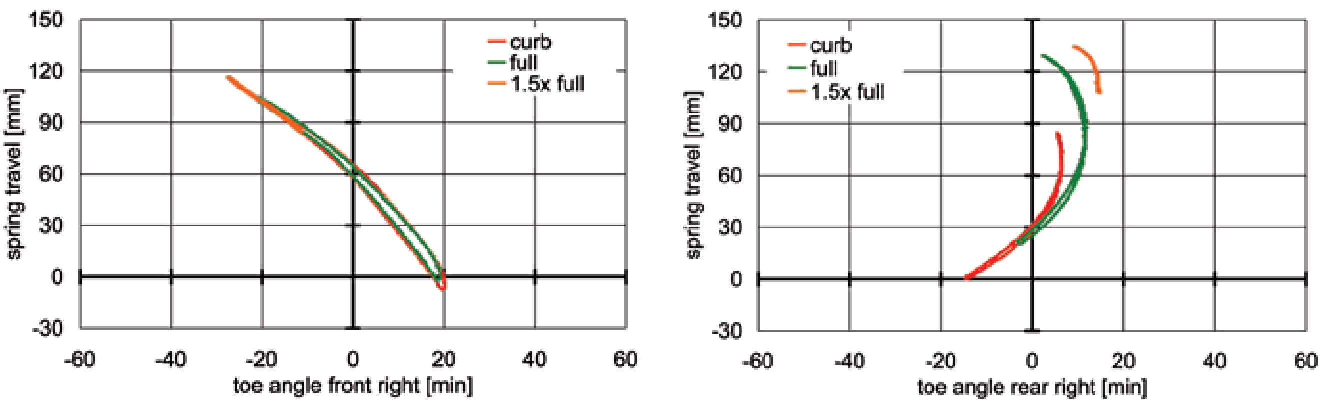


Figure 4: Toe angles during alternate wheel travel at three suspension heights; K-and-C measurement

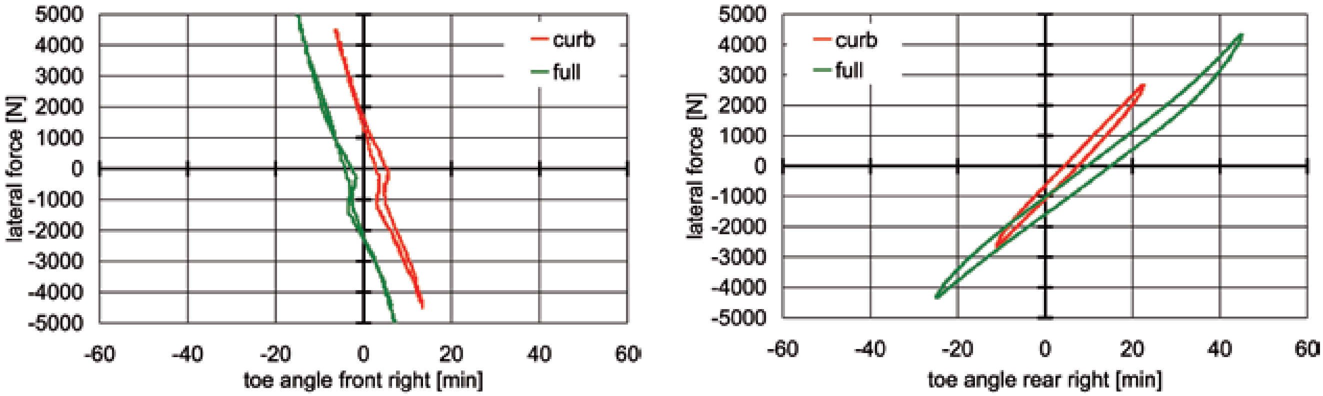


Figure 5: Lateral force steering at two suspension heights; K-and-C measurement

This good accordance was also achieved in dynamic driving manoeuvres, Figure 2 and Figure 3.

While the wheel forces and motions represent a reference value for the quality of the modelling process to a certain extent, the motions of the vehicle body are the actual target values for the simulation. The quality of the body motion simulation is shown on the basis of a demanding driving manoeuvre, the double lane change, **Figure 6**. The measured curves for the steering wheel angle and driving speed are the input signals for the computation.

The accordance between measurement and simulation is very good for this complicated driving manoeuvre. Simulation delivers the same high lateral accelerations as the measurements. The yaw angle velocity shows slight inaccuracies in the amplitudes, while the time curve is correct. The sideslip angle and therefore the rear axle tyres and the rear axle toe behaviour are convincingly represented. The quality of the roll angle is at the same level as yaw angle velocity.

4.2 Further Examples of Results

The simulation model makes it possible to separate the kinematics and compliance components of the wheel motion. In the double lane change test, the rear

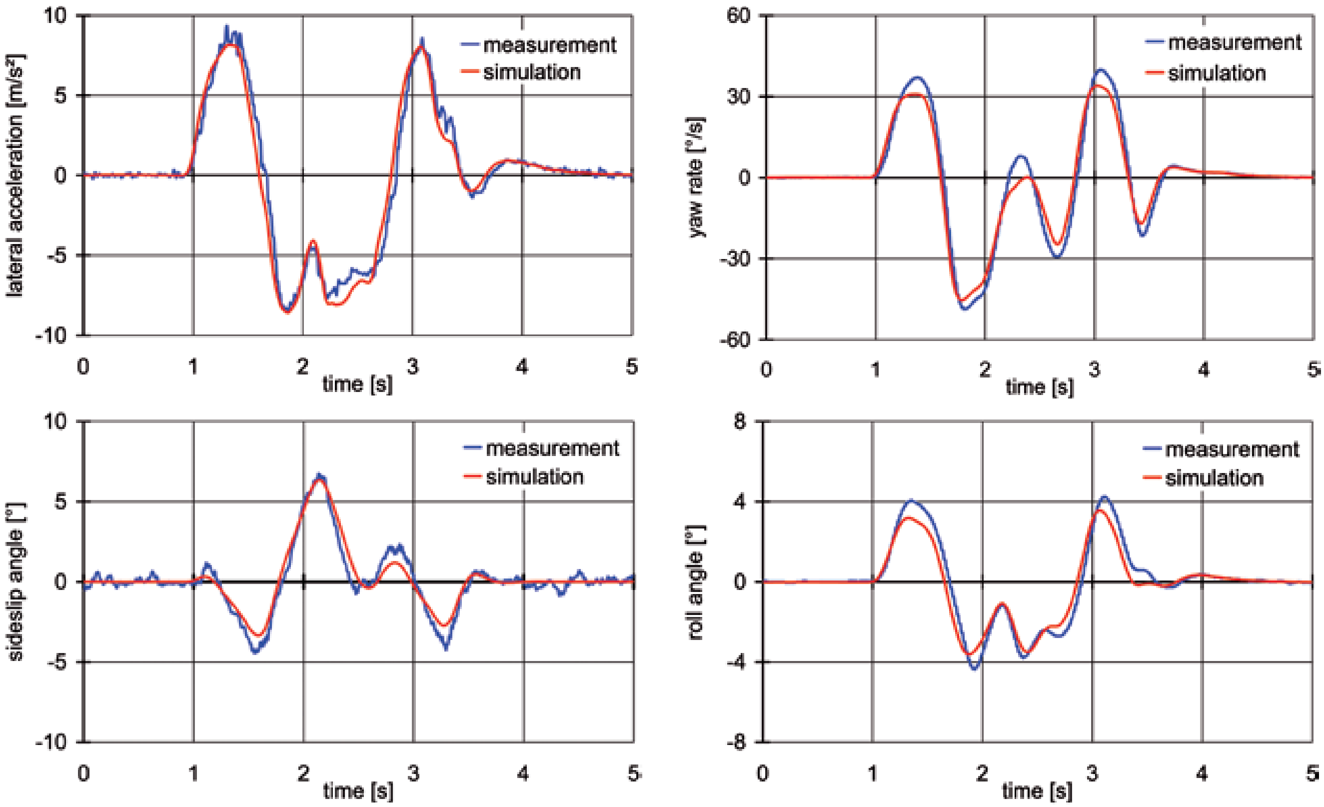


Figure 6: Simulated motion values for the vehicle body compared to measurement; double lane change

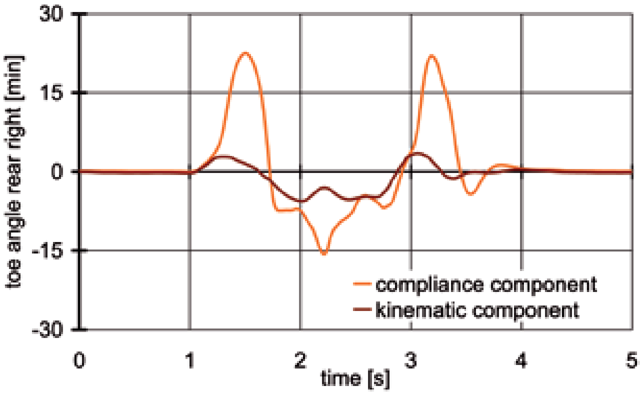


Figure 7: Separation of the rear right-hand toe angle into kinematics and compliance components; double lane change

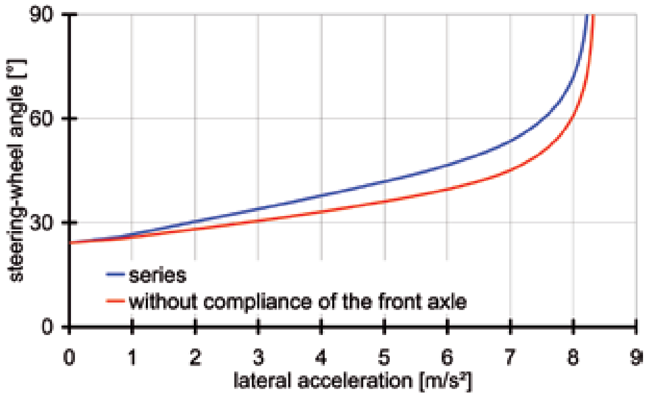


Figure 8: Steady-state circular driving in series trim and without compliance of the front axle

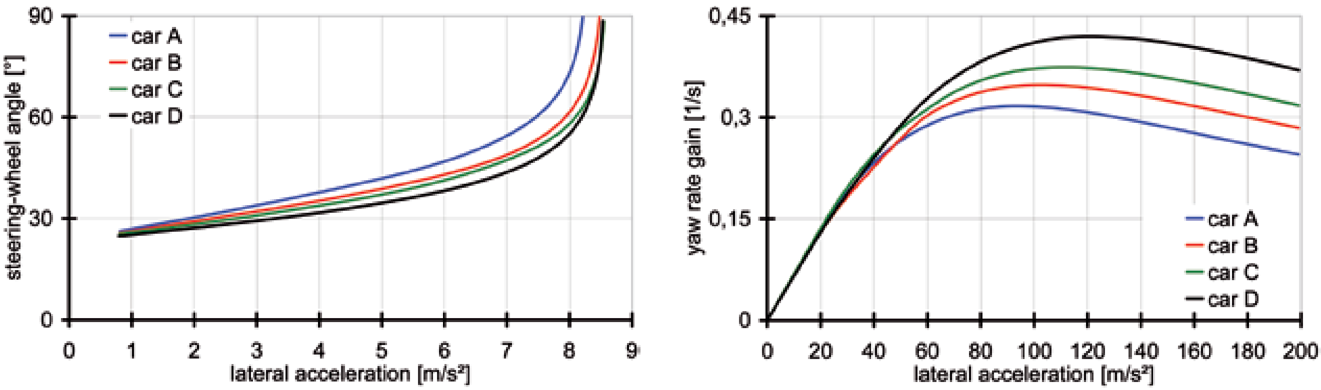


Figure 9: Stationary lateral dynamics of four lower midsize passenger cars; left: 100 m steady-state circular driving; right: 0.4 *g* yaw rate gain

right-hand wheel essentially steers due to compliance, **Figure 7**.

In addition, **Figure 8** shows the steady-state circular driving of the test vehicle optionally without wheel position changes due to compliance at the front axle. This shows the influence of compliance on the steering behaviour.

The simulation model allows the targeted manipulation of characteristic maps. Adapting the measured changes in track width and camber angle, for example, makes it possible to vary the instantaneous centre height of the axle. The other axle properties remain unchanged.

4.3 Benchmarking

The steady-state circular driving on the 100 m circular track and the steady-state yaw rate gain test at 0.4 *g* lateral acceleration with three lower midsize vehicles are examples of the benchmarking capabilities of the simulation model based on K-and-C test rig data, **Figure 9**.

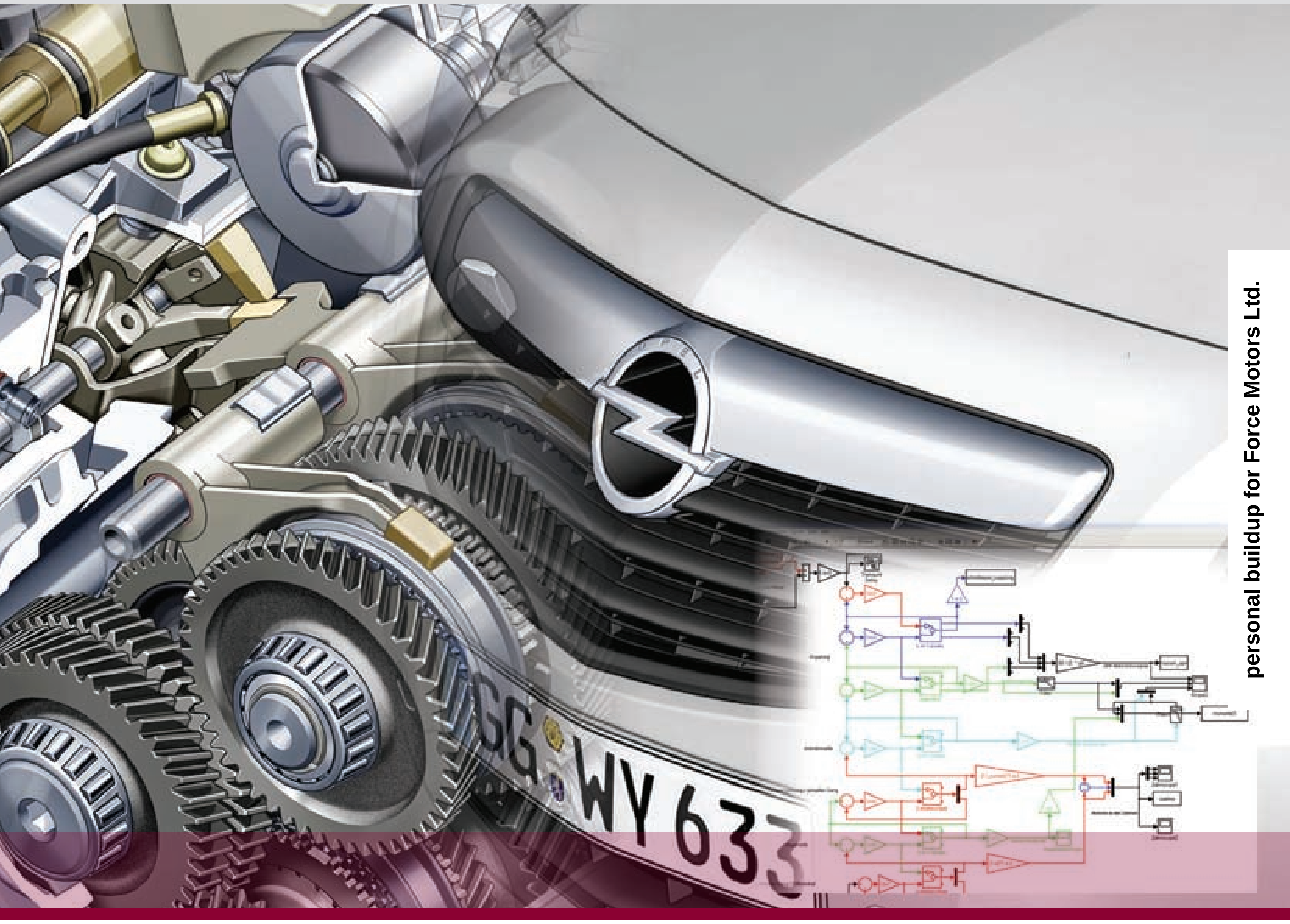
5 Conclusion

This article documents the potential of measurement-based driving dynamics simulation. The vehicle model with precisely parameterised axles based on extensive kinetics and compliance measurements together with a high-quality tyre model provides lateral dynamics results that are close to reality. Verification of this is still required regarding the superimposition of lateral and longitudinal dynamics.

In spite of this, there are plans to use measurement-based driving dynamics simulation as an aid to interpreting K-and-C results in chassis tuning and for comparisons of K-and-C measurements taken on the company's own and competitors' passenger cars. This will provide more accurate benchmark characteristic values and, in the medium term, new knowledge regarding the target areas for chassis set-ups.

Reference

[1] TESIS DYNAware: veDYNA Suspension Generator Manual, Munich, 2008



X-by-wire Approaches for Manual Transmissions

While searching for the ideal actuation concept for manual transmissions a demonstrator vehicle was designed and prototyped by GM Powertrain with a fully by-wire actuated manual transmission. Shift comfort and consistency of the shift events over the whole operational range are very promising. Further opportunities for driver and vehicle manufacturer can be exploited by application of novel safety strategies within a closed loop controls system of the transmission actuation system.

1 Introduction

Since some years the reduction of the operational forces for increased perceived drive comfort is being one of the major challenges in manual transmission development, which is the first of three categories of arguments. This holds especially true for vehicles with stronger engines. Due to the increased demand on torque transfer and the associated increase of rotating inertia also the synchronization effort required from synchronizer, shift system and driver continues to grow. Using – somewhat simplifying – the chain of arguments „Power = Force × Displacement“ makes it obvious, that the increased required synchronizer effort requests either more powerful synchronizer components or a prolonged synchronization time; keeping shift force at the shift lever knob is important to avoid complaints about too high shift forces. Furthermore, an additional challenge arises from the tendency to increased overall ratio spreads to cut fuel consumption while keeping the number of gears the same. The higher ratio progression will enlarge the required synchronizer capacity but more packaging space is hardly available.

By the intrinsic concept of a synchronized manual transmission with purely mechanical actuation their operational comfort is subject to big scatter, as the second category. Oil temperature, rotational speeds when shifting, force and pace of the driver initiated shifting process are of big influence on shift feel. In addition, the locus of initial contact between shift sleeve and dock body under engagement determines by its strong influence the occurrence of double bumps and shift scratch; unfortunately, the driver has no means of influencing this process. In summarizing, the manual transmission concept suffers from a high degree of repeatability and stability of the force-travel relations at the driver interface; the concept driven deviations from the theoretic optimum are high.

In the third category, manual transmissions provide only very limited opportunities to protect vehicle, driver and environment against abusive operation and associated consecutive failure mechanisms. As an example one may think of an unwanted actuation of the reverse gear locking mechanism and reverse gear engagement when driving forward at elevated speed. Another example is the snap type engagement of the launch clutch at high engine speed. The failure mechanisms of such abuse span from partial damage, which is not noticeable to the driver up to incompliance to federal law and danger to safety and security of the car passengers and their environment.

These three categories of arguments motivated GM Powertrain to setup a demonstrator vehicle to understand opportunities and threats of a full x-by-wire system for manually operated transmissions. Clutch pedal position, shift- and select motion are electronically captured and translated into appropriate actions for clutch and transmission. All customer interfaces – clutch pedal and shift lever – still have the same optical appearance for the driver. Furthermore, an important development goal was to implement the by-wire system without any modification to the base transmission; a fact which is of vital importance for later serial applications.

Within this paper of the GM Powertrain project “Full-by-wire” possible by-wire approaches for manual transmissions are discussed and the respective user interfaces as well as actuator concept are reviewed briefly, which were used in the demonstrator vehicle. Some remarks on important safety issues conclude the main part.

2 X-by-wire Approaches

By-wire systems are subject to discussion in automotive industry since a couple of years, some concepts such as for accelerating (e-gas), and braking (brake-by-wire) or steering (steer-by-wire) have been successfully brought into production as lead applications. Within the transmission sector the automatically shifting concepts have achieved most, applications with electronically actuated park systems and electronic mode select are visible on market. Some suppliers of manual transmission integration components have however derived by-wire variants of their products but these stand-alone solutions do not allow for an overall optimization of the envelope system for car manufacturer and vehicle customer. Often safety issues or the limitation to the 12-V on-board electric system are making too high demands or the systems are even prohibitively expensive for mass application.

Figure 1 shows the conventional approaches for shift and select actuation by means of a rod kinematic system or push-pull-stable shift and select cables; in addition, the by-wire technology enables now electronically controlled actuation concepts. A sensor reads the shift lever position; the execution of the target position

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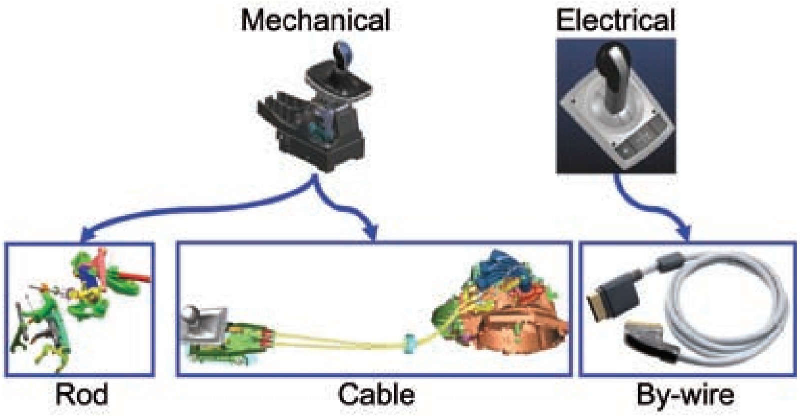

Figure 1: Available technologies for execution of shift- and select processes

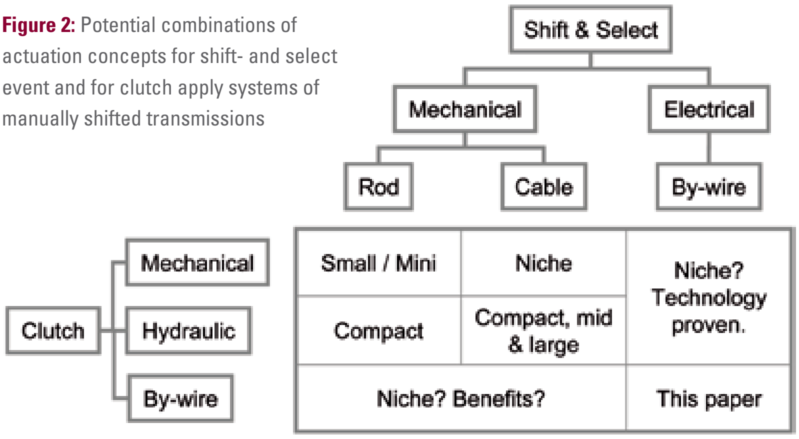

Figure 2: Potential combinations of actuation concepts for shift- and select event and for clutch apply systems of manually shifted transmissions

values for the transmission internal shift system can use electro-mechanical or (electro) hydraulic working principles.

Similarly, the existing clutch actuation approaches using either a purely mechanical guided tensional cable or a hydraulic system with master and slave cylinder are being amended by the by-wire technology. A position pickup monitors the clutch pedal position; the target position is then achieved through an electro-mechanical or a (electro) hydraulic actuator system.

One short remark may be added with regard to commercial vehicles, which use hydraulic or pneumatic actuation concepts for theirs group transmissions since several years. The hydraulic actuation concept may be seen as the same but the details differ totally from commercial vehicles to passenger cars. Due to the large difference in shift comfort targets and overall noise level these systems require clear distinction. Truck group transmissions require actuators to be somewhat user friendly in the gear shift pattern; but shift comfort or shift noise of the actuator system has much lower priorities than for passenger cars.

When arranging the potential combinations of shift and select actuation in horizontal and of the clutch apply system in vertical direction one arrives at the matrix of actuation technology shown in **Figure 2**, which allows for pinpointing of market segments and potential fields of application.

The identification of segments follows mostly driven by cost criteria. For the lowest row and the right hand column the rating reflects GM Powertrain's expertise in this field. A rod shift system will hardly be combined with a clutch-by-wire system; furthermore, shift-by-wire systems with conventionally actuated clutch do exploit only a small fraction of the potential benefits of a full by-wire system.

3 Driver Interface

It has been one of the project targets to develop the demonstrator vehicle with x-by-wire system in such a way that it does not differ from a conventional car with manual transmission at the first glance. Generally, the by-wire technology can be combined with touch-pad, which can be located freely in the vehicle but this approach can lead to a series of unsolved legal type-approval questions but offers great additional design flexibility.

Right at the clutch pedal in the demonstrator vehicle the hydraulic master cylinder of the apply system is being replaced by a mechanically decoupled system with a position pick-up and spring type package. The spring type package consists of proportional- and over center spring to allow for modulation of the pedal characteristic. It is worth to be mentioned that the clutch-by-wire system can be realized with almost no hysteresis due to its mechanical design. Due to the missing mechanical linkage and its associated requirements all tuning efforts for the clutch-by-wire system can be

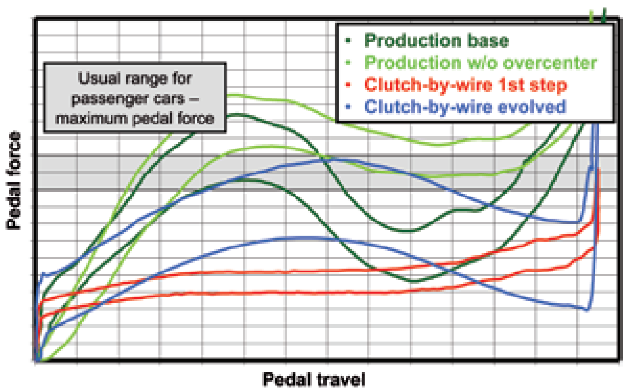

Figure 3: Development of the pedal-force characteristic from the initial system (production basis) to the finally tuned state system (Clutch-by-wire evolved) for the clutch-by-wire system

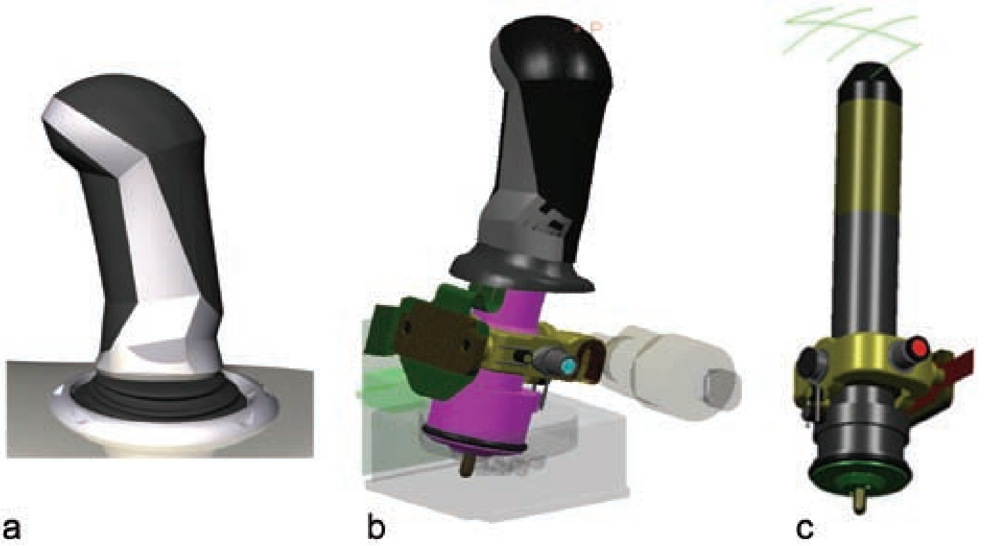

Figure 4: Shift-by-wire shift lever: CAD-view of the lever (a), system section with 3D coulisse-contour and position sensor (b), in-lever spindle motor for detent force adjustment (c)

fully dedicated to comfort optimization. Thinking even about individually adjustable clutch pedal characteristics brings numerous opportunities for comfort and functional optimization measures into reach. **Figure 3** shows the development of the clutch force characteristic for different stages of development.

The by-wire shift lever length in turn can follow design and ergonomics criteria only to arrive at a very short shift travel since only an electronic signal needs to be provided and not a mechanical actuation force for the internal shift system. **Figure 4** shows the shift lever of the demonstrator vehicle. The position sensors and a spindle motor, which can rapidly adjust the lever detent force, are clearly to be seen. By the variation of the detent force the shift force level can be proportionally changed, either statically or in dependency on shift speed. **Figure 5** displays a selected family of force-travel-characteristics in shift direction as an example. Also, the by-wire shift lever shows almost no hysteresis.

The shift speed dependency of shift feel is an important feature of the by-wire system developed by GM Powertrain – the driver receives a feed-back on the synchronization force associated to his shifting speed. Without such a speed dependency the driver feels nevertheless consistent and continuously comfortable shift, but he is suffering information on the transmission; a slow 1-2 gear change feels the same as a rapid 2-1 power shift. GM Powertrain rates this particular feedback as essential for success of a manual transmission with by-wire actuation.

The select force characteristic depends on the central detent force since the detent system for the select movement is also realized through the three-dimensional coulisse contour and the engine driven variable detent spring as shown in **Figure 6**. In contradiction to the conventional external shift system the by-wire shift lever does not use a mechanical reverse engagement locking system but an electric magnet in the lever. Above a forward threshold speed the magnet blocks select movements into the reverse gate.

Due to the missing mechanical connections between clutch/transmission and the customer interfaces clutch pedal and shift lever the by-wire system is able to eliminate all engineering challenges associated to shift scratch, shift lever vibrations, select roughness, pedal vibra-

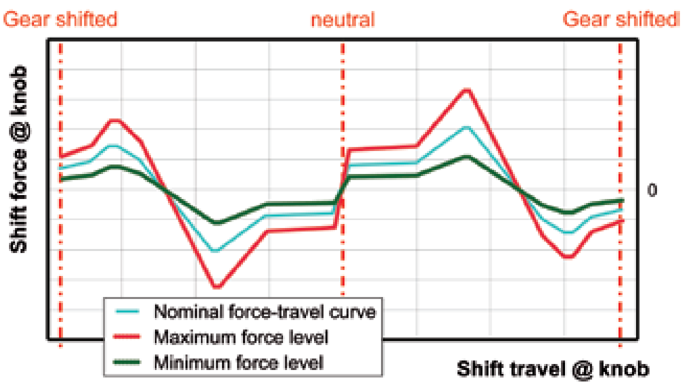

Figure 5: Exemplary force-displacement characteristic in shift-direction of the shift-by-wire lever

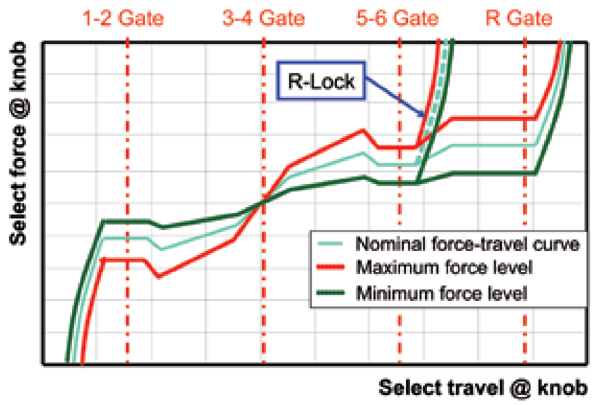

Figure 6: Force-displacement characteristic in select-direction of the shift-by-wire lever

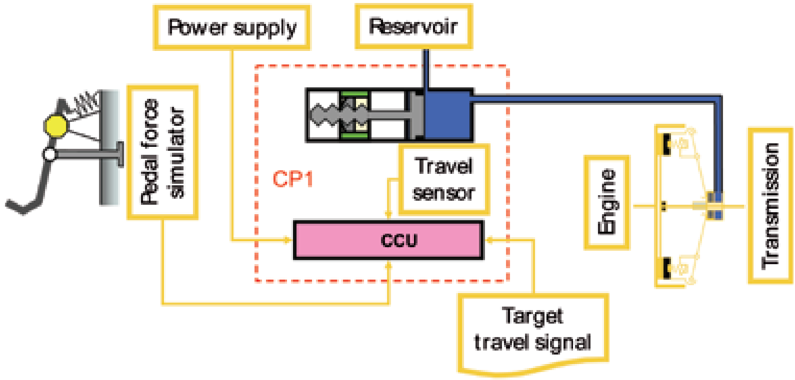

Figure 7: Schematical block diagram of the clutch-by-wire system

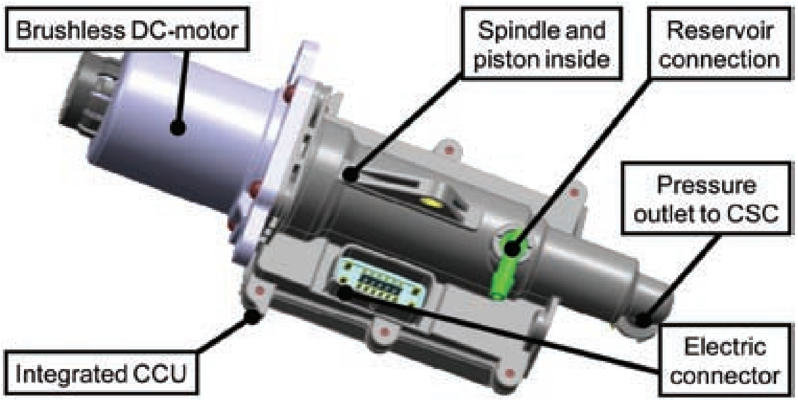

Figure 8: Electro-hydraulic apply master unit for clutch actuation

tions and wear dependent increase of pedal force [1].

4 Actuator Concepts

During concept definition and design of the actuators the limitation to the mechanical interfaces of the base transmission was one key focus besides the functional requirements such as achievable speed, maximum actuator force or torque, controls accuracy and temperature stability. Interface points and working principles of the conventionally actuated base transmission of the demonstrator vehicle were to be maintained.

Figure 7 schematically displays the clutch-by-wire system, the concentric slave cylinder (CSC) of the conventional clutch apply system is preserved. The conversion of the position signal read from the clutch pedal into a hydraulic activity is achieved through the electro-hydraulic master unit shown in **Figure 8** using closed loop controls. The additional travel sensor – which is typical for the by-wire system compared to the conventional base – allows for wear compensation through controls without extra design effort, which also leads to a reduction of rotational inertia. For the implementation of efficient safety strategies it is necessary to closely connect clutch control unit with the controllers for shift and select actuator.

The shift actuator is driven by a high speed spindle motor and is equipped with a linear travel sensor to monitor the shift sleeve position during synchronization, **Figure 9**. The connection to the central shift- and select shaft is designed such that no torsional or bending forces are being transferred to the central shift-shaft to avoid jamming of the shaft or unwanted pre-synchronization.

Regarding the demonstrator car, the same electric drive unit was selected by GM Powertrain for the select actuator, **Figure 10**. The kinematic principle of selecting through a small-scale rotation of the central shift shaft is maintained. The rotation during select of the base transmission shift shaft is driven through a high ratio gear train. An angular position sensor monitors selection of the appropriate gate, in collaboration with the shift actuator linear position sensor the engagement of the correct gear can be checked.

Naturally, alternative actuators are possible as it is the case for the automation of manual transmissions such as a shift drum for a sequential shift pattern or an electro-hydraulic shift- and select system. The actuator concepts can to a big extent just be taken out of the existing tool box for automation components, however, one approaches their limits

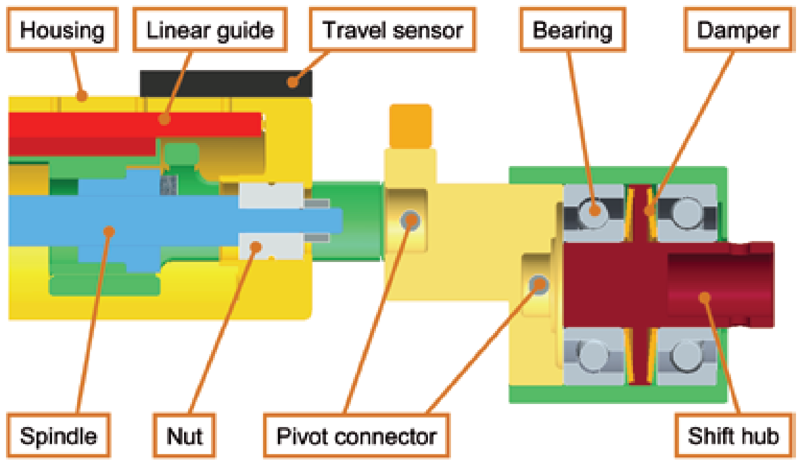

Figure 9: Shift actuator section – connection to the central shift-shaft accomplished again by a bolt connection

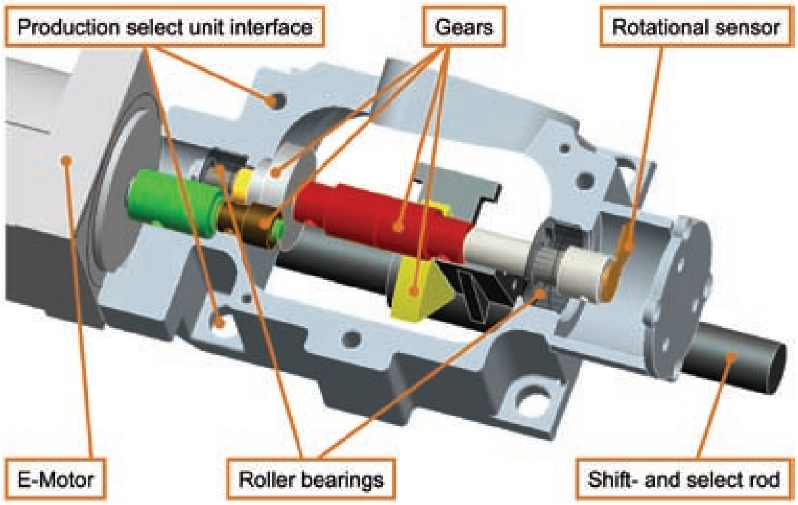

Figure 10: Open view of the select actuator

with respect to power and torque/force fairly quickly. Shooting for very sporty cars with extreme short shift times implies high available power especially for the shift actuator.

5 Aspects of Safety

Having sensors available at the clutch apply and shift system and exploiting the controllability of the actuator system opens opportunities for the fully by-wire actuated manual transmission to increase safety and comfort; the conventional transmission is suffering these. A snap start event causing very high spike loads in the transmission [2], can be identified as being critical; setting a somewhat smaller speed for the closing clutch can successfully reduce the torque peak height. A wrong shift into a too low gear can also be identified and gear engagement or clutch closure can be prevented to avoid over reving the engine.

These two examples show that basically three possibilities exist to cope with up-coming safety-critical or damaging events. The system can simply execute the drivers command and the vehicle behaves as the driver commands, but this may lead from damage of single components up to walk home failure of the vehicle. Alternatively, the transmission controller can execute the drivers command with a small delay or modification such that the safety critical situation is alleviated but the driver may complain about the short term discrepancy between his command and the experienced vehicle response. Finally, the drivers command can be ignored for safety reasons as it is done for instance on automatic transmissions in tip mode to avoid too high or too low engine speeds. The large discrepancy between driver operation and response of the transmission system will then cause complaints since it is still a car with manually operated transmission.

To achieve compatibility with legal type approval regulation it is based on the findings out of this project useful to use for the by-wire actuated manual transmission a normally open clutch system as it is done for dry dual clutch transmissions. In doing so, maneuverability of the vehicle is maintained also in the unlikely case of an electric defect. Unwanted rolling of the car can be prevented either by a conventional park lock system or by an electrically actuated park break system when the car is parked with "engaged" gear, which means that the shift lever is off from neutral position.

6 Conclusions

The by-wire technology of GM Powertrain allows for the optimum integration of manual transmissions into the vehicle with regard to consistency and repeatability of the handling process. The system actuators allow also for further features such as for example an automatic drive mode as for any automatic transmission. Furthermore, the technology opens up opportunities to optimize shift feel and clutch pedal characteristic according to individual customer requirements in a fairly large range. At the same time, the existing boundaries between the classical manual transmission and its automated derivatives fade away; car manufacturers, drivers and legislation will re-consider their positions.

Simultaneously, the safety related deficits of the manual transmission concepts in for example snap start maneuvers or miss-shifts become obvious. The list of failure modes, which can be avoided using a by-wire technology, leads to the appropriate question whether a manual transmission concept could be "invented" again today having all the sensorics available.

Handling comfort and capability of the demonstrator vehicle are very good; GM Powertrain rates the project "Full-by-wire" as an important step towards the proof of possibilities and advantages of the technology. The on-cost of the by-wire system currently seems to be high but potential simplifications of the base transmissions can to a large extent counterbalance the cost disadvantage once the technology is established to a certain market volume. The target of most comfortable manually shifted transmissions for even safer and at the same time more fuel efficient cars can be met with the by-wire technology.

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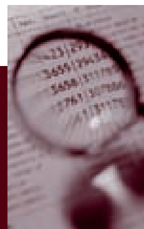

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Subjective Perceptions and Objective Characteristics of Control Elements

Can physical measurements be used to predict how a control element will feel to the user? To answer this question, Battenberg Robotic collaborated with the Philipps University Marburg (Germany) Department of General and Biological Psychology to perform a representative test study, the results of which show how well subjective evaluations by study participants correlate with objective characteristics.

1 Introduction

When constructing control elements for vehicles, it is primarily the objectively measurable characteristics that are optimised for each task. These include, for pushbuttons and rotary control dials for instance, the mechanical resistance to be overcome, the distances to be traversed, and the feedback characteristics (for example audible or inaudible switching sound). These parameters can be directly connected to the ergonomic conditions, for example size optimisation according to Fitts' law [1], force optimisation depending on the location in the driver's working space [2], or optimisation of the haptic or auditory feedback characteristics, etc.

How the objective characteristics are designed results on the one hand from the requirements profile of the task to be carried out, and on the other hand on the physical and psychological limitations of the "average" vehicle driver. The matching of both variable vectors is the goal of an optimum ergonomic design. However, there is another aspect in addition to this which is extremely significant for the layout of the cockpit and the elements it contains: control elements and gears also support the purely subjective, emotional evaluation by the vehicle driver. For identical functionality, distinctively designed elements can have entirely different connotations to the user, for example perceptions such as sporty, conservative, smooth, reliable, or evaluations such as pleasant, pretty, etc. These aspects become even more important as the functionality becomes more ideal and the functional differences become less discernable to the user. Subjective evaluations are the only characteristics that allow for differentiation, thus influencing the decision to purchase.

2 Subjective Evaluation of Control Elements

Until now, very little has been known about how control elements in motor vehicles are subjectively experienced and evaluated. It has also never been explained whether subjective evaluations can systematically be related to the objective characteristics of the control elements. If this were possible, and physical characteristics such as values on the force/distance diagram could be used to estimate whether a control element might be categorised, for instance, as sporty, elegant, or pleasant, then vehicle interiors could be designed with specific characteristics for specific target groups right on the drawing board.

Psychophysics and experimental psychology offer a series of methods by which such subjective evaluations can be determined [3]. By means of suitable analysis, these methods yield metrics which can then be related to the objective measurements, so that subjective characteristics can be predicted from the objective measurements.

There is a basic problem to be solved: subjective evaluations of perceived facts (switches, rotary dials, surfaces, displays) are not one-dimensional. A control element can be experienced as aesthetically pleasing, but at the same time as sporty

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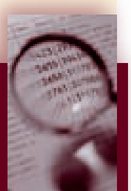

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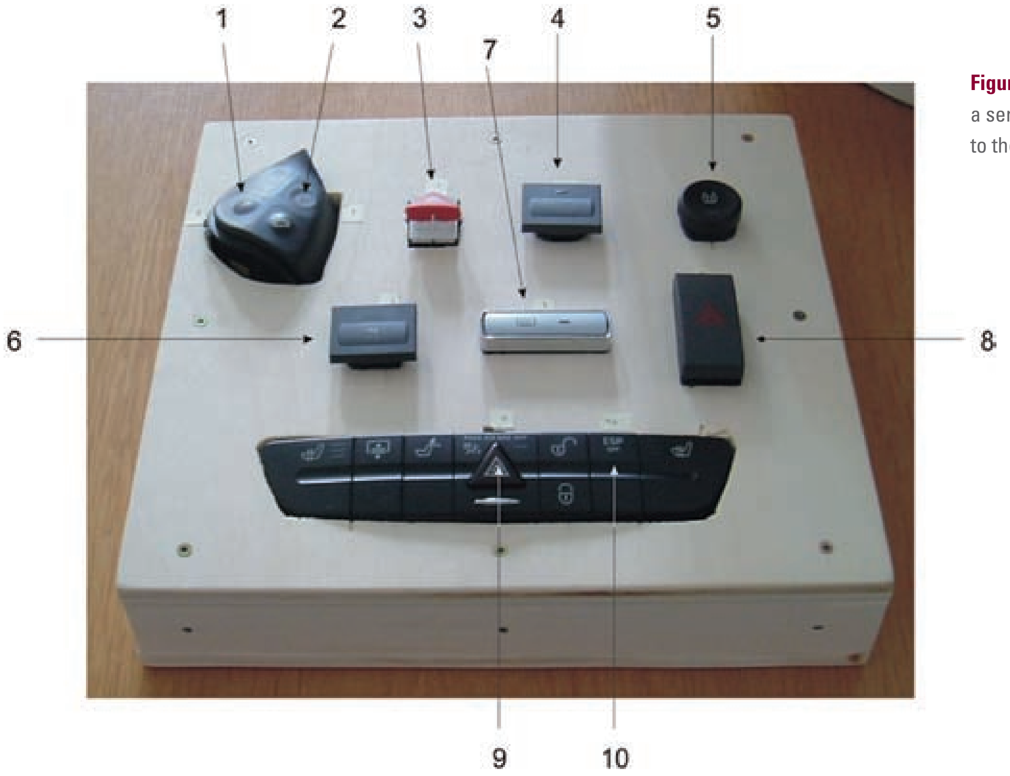

Figure 1: Experimental setup with which a series of different buttons is presented to the test subjects

1	large	-3	-2	~~-1~~	+1	+2	+3	small
2	strong	-3	~~-2~~	-1	+1	+2	+3	weak
3	soft	-3	~~-2~~	-1	+1	+2	+3	hard
4	male	-3	-2	~~-1~~	+1	+2	+3	female
5	solid	-3	-2	-1	~~+1~~	+2	+3	liquid
6	coloured	-3	-2	-1	+1	~~+2~~	+3	pale
7	fast	-3	-2	-1	+1	~~+2~~	+3	slow
8	exciting	-3	-2	~~-1~~	+1	+2	+3	boring
9	modest	-3	-2	-1	+1	~~+2~~	+3	immodest
10	simple	~~-3~~	-2	-1	+1	+2	+3	complex
11	elegant	-3	-2	-1	~~+1~~	+2	+3	crude
12	thick	-3	~~-2~~	-1	+1	+2	+3	thin
13	shaky	-3	-2	-1	~~+1~~	+2	+3	stable
14	snapping	-3	-2	~~-1~~	+1	+2	+3	fluent
15	angular	-3	-2	-1	~~+1~~	+2	+3	round
16	dominant	-3	-2	-1	+1	~~+2~~	+3	submissive
17	durable	-3	-2	~~-1~~	+1	+2	+3	fragile
18	heavy	-3	-2	-1	+1	+2	+3	light
19	ill	-3	-2	-1	+1	+2	+3	healthy
20	warm	-3	-2	-1	+1	+2	+3	cold
21	convincing	-3	-2	-1	+1	+2	+3	questionable

Figure 2: Semantic differential used to evaluate a pushbutton x with respect to the given property pair on a scale of one to six

or not sporty, or as reliable or unreliable. Thus the multidimensionality of the evaluation must be taken into consideration. The goal should be to determine the minimum number of evaluation dimensions necessary to define the judgment. At the same time, each dimension of judgment should model the associatively connected ideas and evaluations in a representative and comprehensive manner.

A robust method already used in market research for some time is the recording of associations with a so-called semantic differential [4-5]. Test subjects are asked to evaluate the objects in question, that is, the pushbuttons or rotary dials, Figure 1, on a scale of one to six for a predefined set of property pairs, Figure 2. For each object there is thus an evaluation profile. Empirical research has shown that with a set of about 30 property pairs the connotative evaluation space can be representatively covered [6]. The selection of property pairs should only partially be oriented towards specific features of the objects to be evaluated, for instance mechanical properties for switches, or haptic features for surfaces. More importantly, the property pairs must represent general qualities of impression and evaluations, that is, indirect associations such as male versus female, humble versus sophisticated, etc. This is of particular importance, if emotional and evaluative associations are to be measured. If, however, research has its focus on the denotative meaning of objects the adjective pairs should be adapted to the object domain and selected on the basis of pilot studies [7].

Judgments are recorded for a certain number of objects, during which both the order of evaluation of the objects and the property pairs should be permutated for each test subject using a random algorithm in order to exclude systematic evaluation errors. The property profiles recorded for the evaluation objects can then be reduced to a minimum non-redundant number of evaluation dimensions [8]. Solutions generally result with three to five independent dimensions which exhaustively describe the system evaluation variance. There are then two results:

1. a matrix of so-called factor loadings E, and
2. a matrix of so-called factor values F, **Figure 3**.

The factor loadings characterise the relationship between the property pairs used for the testing and the derived orthogonal, or non-redundant idealised evaluation dimensions. For instance, in one study of pushbuttons there were four independent dimensions which could be described as follows based on the property pairs which characterised them:

1. hardness versus softness, defined by the property pairs hard or soft, tense or relaxed, firm respectively worn out, loud or quiet, male or female, and strong or weak
2. stiffness versus looseness, with the property pairs heavy or light, thick or thin, rough or fine, slow respectively fast, long or short, sedate or sporty, and large or small
3. unreliability versus reliability, with the characteristics dangerous or safe, unfamiliar or familiar, uncomfortable respectively comfortable, unreliable or reliable, fragile or tough, repellent or attractive
4. originality versus conventionality, defined by the features original or conventional, showy or humble, elegant or simple, expensive respectively cheap, exciting or boring, beautiful or ugly.

The important thing for the interpretation of these derived dimensions is the fact that each dimension represents the entire set of words listed after it. These dimensions are not single, elementary properties, but higher-level evaluation dimensions which represent the semantic field or the features associated with them.

The factor values in matrix F characterise the test subjects and the evaluation objects. By taking the average over the test subjects, for instance, it can be determined how the pushbuttons are evaluated on average on these idealised evaluation dimensions, for example, which of the buttons are estimated high on the factor reliability, or which are seen as conventional and which original. If characteristics of the test subjects are known, for instance personality features such as extraversion or demographic features such as age, status, etc., then these averages can also be examined for corresponding subgroups of test subjects. Likewise, it can be tested whether judgements are affected by the ergonomic context, for example, where the controls are located in the car (in or out of sight, close or far from the body centre) or how densely controls are packed within a dashboard area.

3 Relationships Between Objective and Subjective Features

In the next step, relationships can be determined between the objective physical features of an evaluation object and the subjective evaluation features. The first thing is to define and measure suitable physical characteristics. For pushbuttons that have been examined, significant

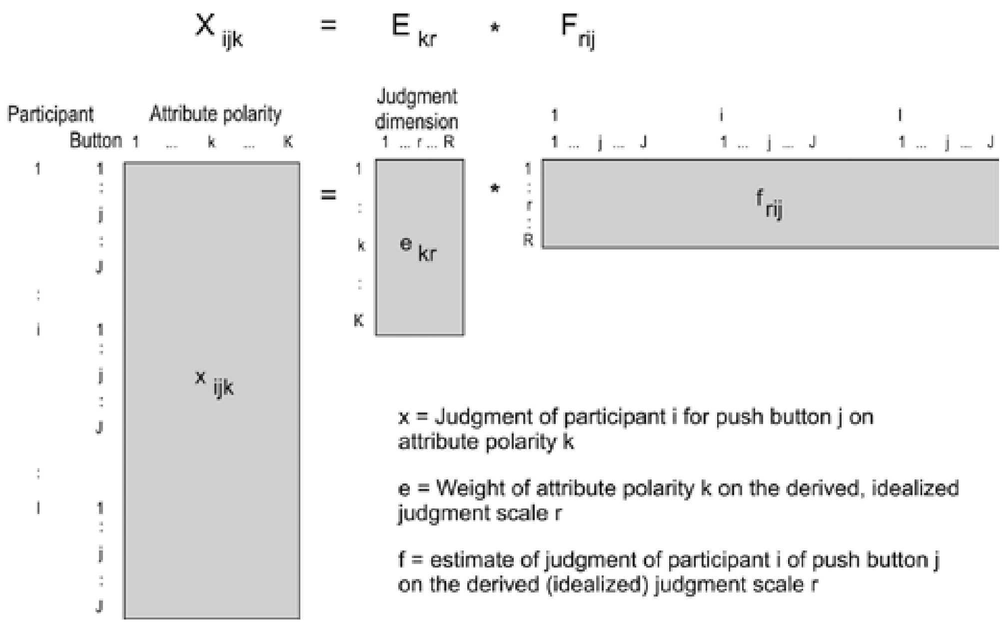

Figure 3: Principle of decomposition of an evaluation matrix using trimodal factor analysis in independent, idealised evaluation dimensions, and the associated estimated factor values characterising the test subjects and evaluation objects

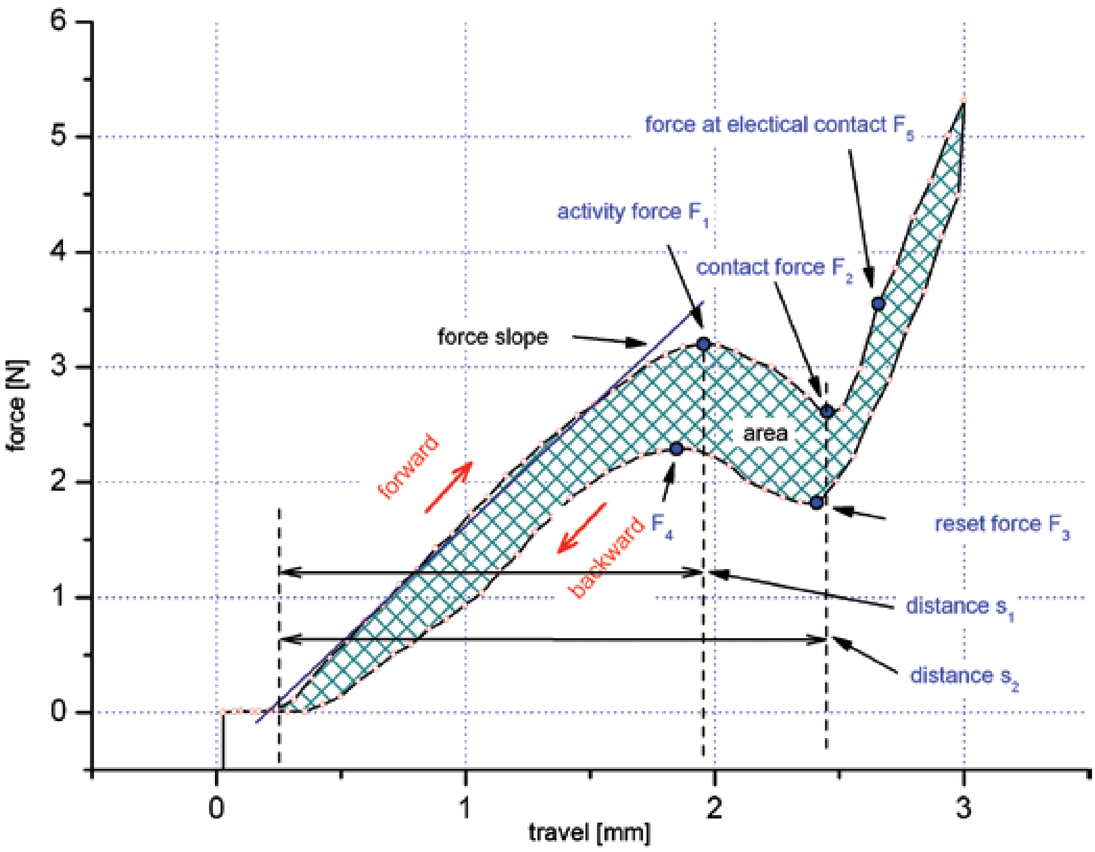

Figure 4: Force/travel diagram of a typical pushbutton, and marking of the objective features of button characteristics derived from it

characteristics could be derived from the force/distance diagram, **Figure 4**, for example maximum and minimum forces, path lengths, force onset and force drop determined for forwards and backwards movements.

Using multiple regression [9], these physical metrics can now be used as predictors of subjective characteristics, Eq. (1):

$$\hat{f}_{rj} = a_{r0}z_{j0} + a_{r1}z_{j1} + ... + a_{rm}z_{jm} + ... + e_{rj} \quad \text{Eq. (1)}$$

In the equation, z_{jm} denotes the objective metrics from the force/distance diagram for button j (that is, for instance, the force onset on the forward path, maximum force, etc.), a_{rm} are weights used to weight these metrics to predict the factor values f_{rj}, and e_{rj} is an error term. For the buttons considered in one study, it was discovered, among other things, that the first evaluation dimension hardness versus softness could reliably be predicted by the force onset of the forward movement or the backward movement. That is, a pushbutton with a relatively fast force onset tends to be experienced as hard, with all the other associations of that concept (tense, firm, loud, male, strong, etc.). The second subjective dimension stiffness versus looseness could reliably be predicted by the length of the button stroke: buttons with a longer movement were experienced as heavy, thick, rough, slow, long, and sedate. A shorter travel distance for a button also strongly predicts the third evaluation dimension reliability, and the fourth dimension originality versus conventionality depends very strongly on a derived value, the so-called snap rate, that is, buttons with a small change in force in relation to the initial force tend to be experienced as original and modern.

4 Perspectives

Connotative evaluations of control elements, that is, subjective preferences and feelings in handling, need not have functional implications. A pushbutton, which feels smooth or sporty, after all, can execute a switching function in exactly the same way as a button perceived as stiff or old-fashioned. However, such connotative characteristics are the deciding factor in whether the driver of a vehicle feels good in a cockpit, whether there is a feeling of safety and comfort or not, and finally, these perceptions determine the purchase decision for or against a particular product. As shown, the method presented makes it possible to measure such subjective features reliably and to reduce them to a minimum number of non-redundant evaluation dimensions. Moreover, between the evaluation dimensions determined in this way and the objective, physical features of control elements, it is possible to determine systematic, calculable relationships. In this manner, it is possible to predict certain impression-type qualities of control element from their physical characteristics. As can easily be seen, this method is not limited to the evaluation of pushbuttons or other control elements. It can just as easily be used for the evaluation of surfaces or even entire vehicle interiors.

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Innovative Sandwich Structures for Functionally Integrated Lightweight Design

Sandwich structures permit the realisation of the optimal combination of material, shape and function in lightweight design. Hence, reduction in vehicle weight and conservation of resources are possible. Implementation of such solutions is however subject to economic efficiency of the overall concept. A number of different institutes of DLR and Fraunhofer are therefore working on new and economical sandwich structures as part of the "Competence Centre for Automotive Light-Weight Solutions (KFL)" in an effort to further develop the potential of this construction method for planar structures.

1 Introduction

Sandwich structures represent a relatively recent lightweight design method, available as an possible completion to classical integral or differential design. The idea to cover a core with two layers is quite old and very common in nature [1], the advantages of this concept first being described by Duleau around 1820, followed by a systematic description by Fairbairn later [2]. The construction method became commercially successful during the 20th century, chiefly in the aviation industry during World War I and II. Since then, further development of adhesive systems has helped to establish sandwich components as an integral part of lightweight design in many different areas including for example aircraft construction, space flight, shipbuilding, as well as the building and packaging industries [3, 4, 5]. 700 different honeycomb core structures for sandwich components were produced in the past 50 years by one long-time manufacturer alone [6].

The principle of sandwich constructions can be considered to be analogous to the bearing properties of an I-beam. Similar to the flanges of the I-beam, the cover layers withstand tension and compression caused by bending, while like the web of the beam, the core structure bears shear forces. The area moment of inertia and thus the bending stiffness of the whole structure can be increased significantly by this type of construction and by increasing the height of the core layer [3]. An advantage is that variation of the shape and material of lightweight constructions can be used to improve bending stiffness significantly and specifically, **Figure 1**. The specific bending stiffness of bending beams with widths of 100 mm is represented in the diagram. It can be seen that an over 50 % increase in the specific bending stiffness can already be achieved in structures with a total height of 10 mm with the sandwich construction method as compared to solid steel or aluminium constructions.

Highly integrative lightweight constructions in a sandwich structure are found in vehicles relatively rarely as yet, since the considerable advantages with regard to weight, bending stiffness and bending strength are outweighed by the higher cost of manufacturing, processing and integration in structural applications. Sandwich structures could be made more attractive for vehicle manufacturing by simplifying sandwich component production methods in conjunction with integrating several functions in one component. This approach could be used to achieve a cost-effective solution. An important requirement here is to make sure that the best suited sandwich

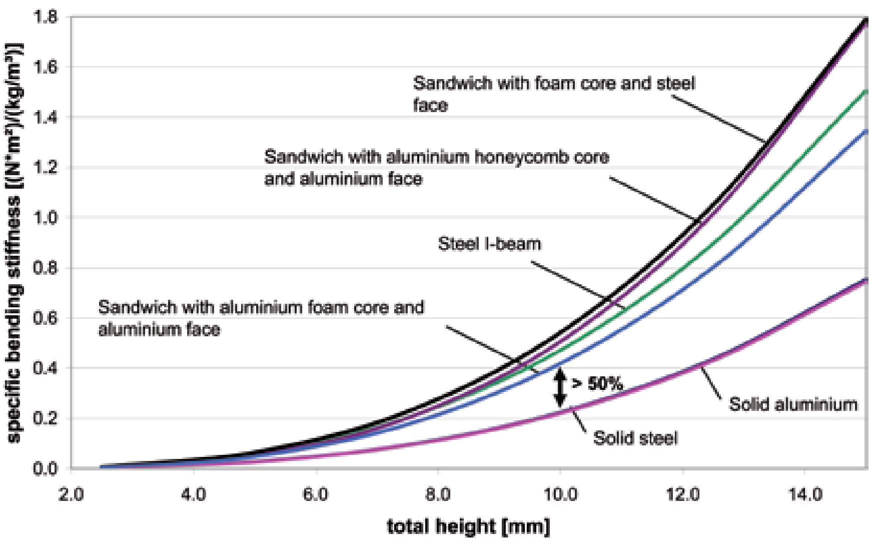

Figure 1: Comparison of specific bending stiffness

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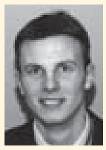

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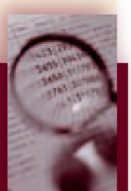

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construction is selected for the particular application requirement.

A study of these challenges is part of the work of a preliminary research project by the Competence Centre for Automotive Light-Weight Solutions (Kompetenzzentrum für Fahrzeugleichtbau, KFL). The KFL is a collaboration of the Fraunhofer Institute for Chemical Technology, two institutes of the German Aerospace Centre (DLR) (the DLR Institute of Vehicle Concepts and the DLR Institute of Structures and Design) and the University of Karlsruhe (TH). The Institute of Aircraft Design at the University of Stuttgart is an associated partner.

2 Core Layer and Cover Layer Materials

As described earlier, technically relevant sandwich structures are normally composed of a low density core structure and two cover layers joined to the core structure. A great variety of sandwich designs and material combinations are currently used in many different areas. **Figure 2** presents an extract of the diversity of core and cover layer materials regarding their elasticity modulus in logarithmic view. Especially core materials are often selected with very low density.

The range of cover layer materials includes simple paper as well as high performance fibre reinforced plastic (FRP) composites. Selection of material is governed by the mechanical properties and other factors such as corrosion, chemical resistance, thermal expansion, behaviour at different temperatures, recycling and manufacturing costs.

One of the more recent developments in fibre reinforced plastic composites currently being studied at the Fraunhofer Institute for Chemical Technology are long-fibre or fabric reinforced plastic cover layers. The objective is to produce sandwich constructions cost-effectively by means of the selected manufacturing method. Polyurethane fibre spraying and LFI (Long Fibre Injection) technology notably facilitate economic production of sandwich structures in a single step process [9, 10, 11]. The material-based advantages of the fibre reinforced polyurethane are combined with a flexible and economic procedure that allows functional integration [12].

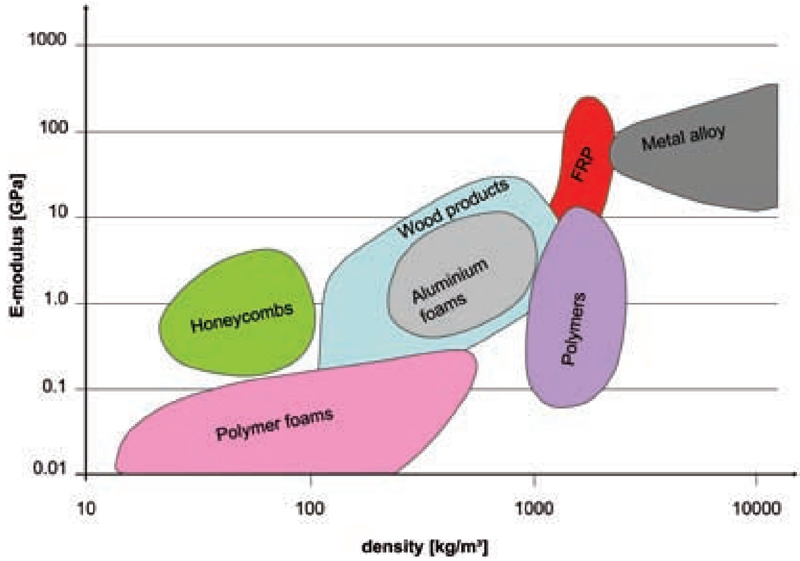

Figure 2: Comparison of the E moduli of various core and cover layer materials in relation to their densities (based on [7, 8])

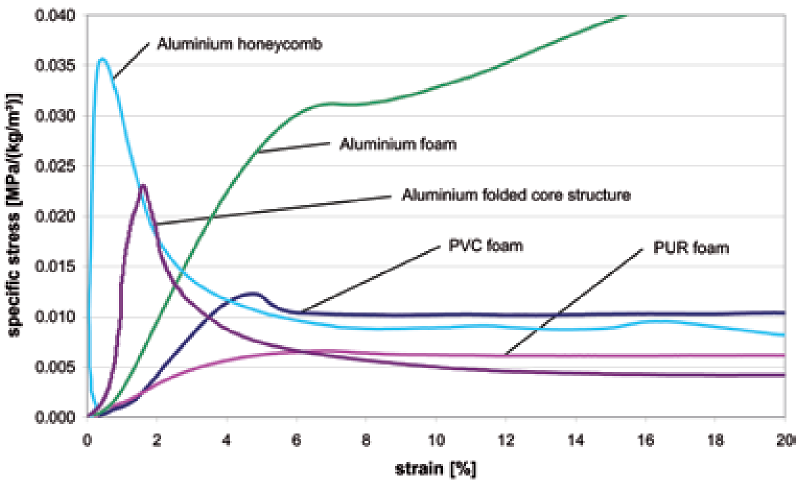

Figure 3: Example of compression tests on various core materials

An even broader range of materials is available for selection of core structures. The numerous options for all sorts of applications include honeycomb structures, familiar from bees, algae or soap bubbles [8], solid cores, wooden and foamed structures made of metal or plastic, as well as folded, profiled and built structures.

'Classic' core materials such as balsa wood or honeycombs are for instance widespread and well-established in aircraft construction. These materials are however characterised by differing disadvantages such as high production costs or closed honeycomb cells that tend to take up condensed water and hence gain weight. Although the mechanical properties of rigid polymer foam materials are inferior to those of honeycomb cores, these materials have considerable potential in the utility vehicle sector on account of the flexible and continuous nature of core and sandwich production, in combination with their heat insulating capacities. One of the most recent developments being studied increasingly at present are metal foams for example made of steel or aluminium [8, 14]. Areas of application of these metal foams include the vehicle floor or front wall, where acoustic properties of the components are important [8].

New core materials, such as for example folded core structures developed

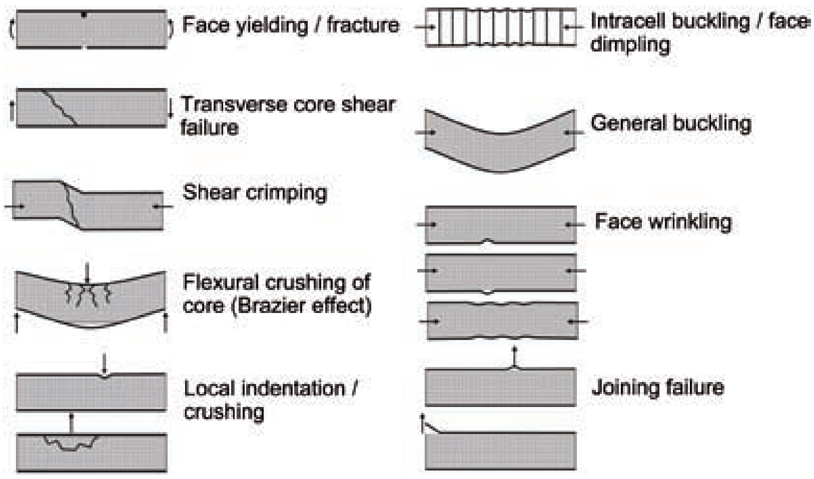

Figure 4: Failure modes of sandwich structures, schematic

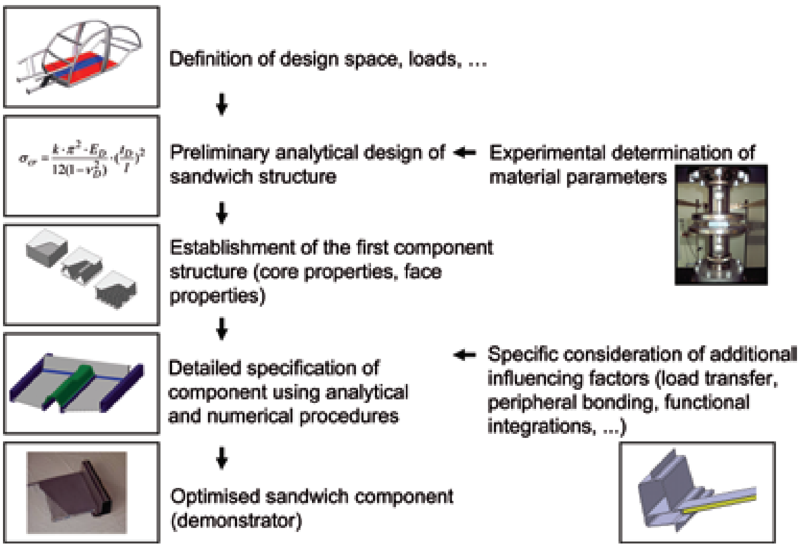

Figure 5: Sandwich design procedure

by the Institute of Aircraft Design at the University of Stuttgart, open up further possibilities for more extensive utilisation of the potential of sandwich structures. Continuous production and implementation of diverse materials could make a less expensive production method available [13]. Core structures with very different properties to suit specific areas of application can moreover be manufactured. **Figure 3** for instance illustrates the relationship of compressive stress and strain for various core materials. The specific compressive strength of the studied folded core structure is clearly superior to that of the foam materials. The former can moreover be manufactured to include additional functions, such as education of condensation, head of process media or sound insulation. Integration of very different functions is therefore possible [15].

3 Design Criteria and Procedure

Sandwich components may fail in many different ways by rigidity or stiffness. These are represented in **Figure 4** on the basis of information provided in the references [2,3,4,5,6,7]. These modes of failure have to be taken into account when designing sandwich structures.

Due to the very different failure mechanisms involved and the option to combine very different materials to make up a sandwich structure, components have to be designed on the basis of a rigorously logical system. The relevant procedure used by the Institute of Vehicle Concepts (FK) of German Aerospace Centre (DLR) is represented in **Figure 5**.

Starting with the definition of design space, loads and other basic conditions, the sandwich construction is defined using the determined material properties. An analytical design instead of an FEM analysis is normally carried out, since simplified loading cases can be conducted relatively easily and with sufficient precision. The comparison of an analytical calculation with the experimental results for an aluminium honeycomb structure under four-point bending test is shown in **Figure 6**. Failure mode and

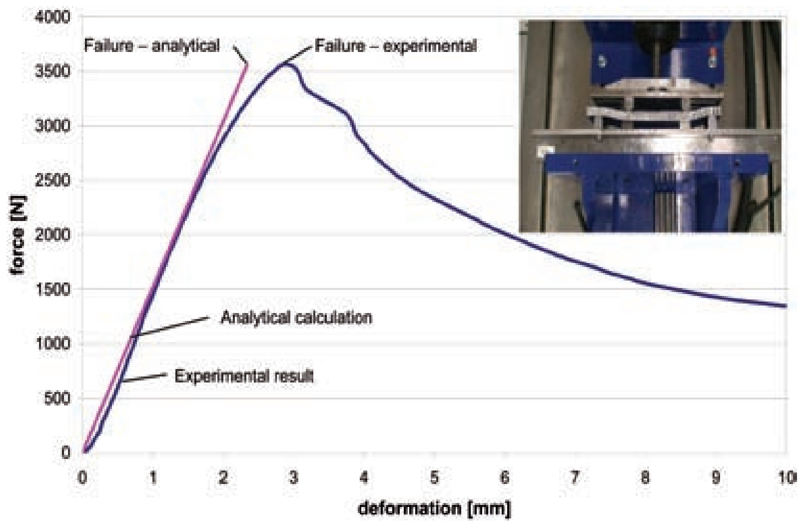

Figure 6: Comparison of four-point bending test to analytical calculation

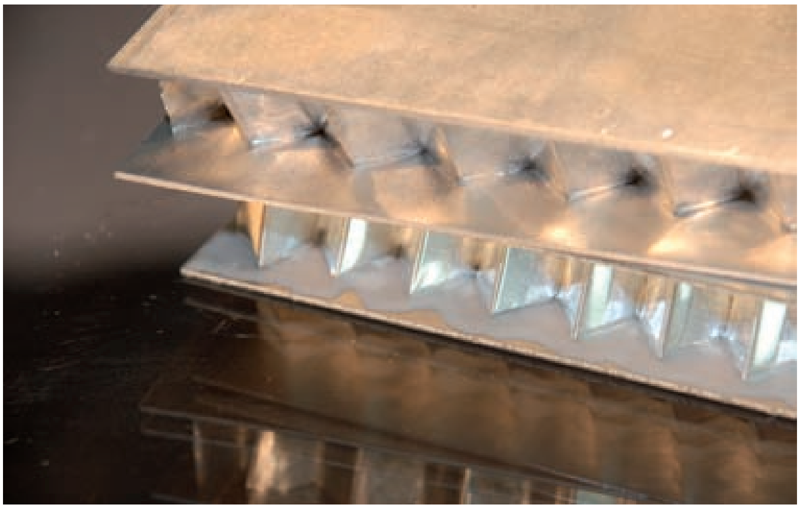

Figure 7: Comparison of a soldered (top) and glued (bottom) aluminium folded core structure sample

strength at the end of the linear behaviour can be predicted relatively well (deviations below 5 %), allowing reliable initial dimensioning.

Detailed specification and optimisation of the component follows, with consideration of other influences including manufacturing methods, load transfer, peripheral bonding or other additional functional integrations. Constructive realisation has to particularly take into account locally transferred loads and moments as well as peeling factors that may lead to failure.

4 Example: Failure of Core-cover Layer Bonding

Challenging factors associated with all sandwich structures include the design, the testing and repair. Optimal bonding of cover layer and core layer is particularly imperative in core structures such as honeycomb or folded core structures, which are only bonded locally.

In addition to various methods such as rivets or screws, fibre composite parts of sandwich structures are also sewed. Soldering and welding can be used for purely metallic sandwich structures, **Figure 7**. A self-adhesive connection of core to cover layers can be achieved by means of further methods such as shape forming foam material or spraying on cover layers.

Today, most sandwich structures are glued. The development and easy application of adhesive films coupled with the good mechanical properties of different adhesives has made this generally most versatile bonding method highly attractive.

Different tests are used to quantify the quality of adhesion of the cover layer to the core structure. The experimental setup and results of a peel test carried out here on different honeycomb cores and adhesive films using a drum with DIN 53295 compliant is presented, **Figure 8**. Such tests can be used for reliable determination of the peel resistance of the adhesive bond between core and cover layers. This depends on the adhesive selected as well as the geometry of the core structure or honeycomb width. Significant influence on the quality of the connection and therefore on the bearing capacity of the component is particularly attributable to geometric influences like the development of fillet welds during adhesion of the honeycomb structures as well as to the cell widths.

5 Lightweight Design Solutions by Functional Integration

Sandwich constructions improve mechanical properties and allow integration of all sorts of secondary functions by appropriate selection of core material. Weight can be saved in this manner. The functions that can be integrated by appropriate selection of a suitable core material include thermal insulation, improvement of acoustic properties, energy absorption in crash load cases or collisions with pedestrians, antenna systems, or media supply lines. Various foam structures for absorption of energy or increasing rigidity are for instance included in vehicle structures at present [7, 15]. At the Institute of Vehicle Concepts, the term 'hybrid' is generally used to describe this strategy for realisation of an integral consideration of materials, design methods and functional effects [15].

A new approach involves integration of media supply lines to support the hospital in the comfort of the passenger

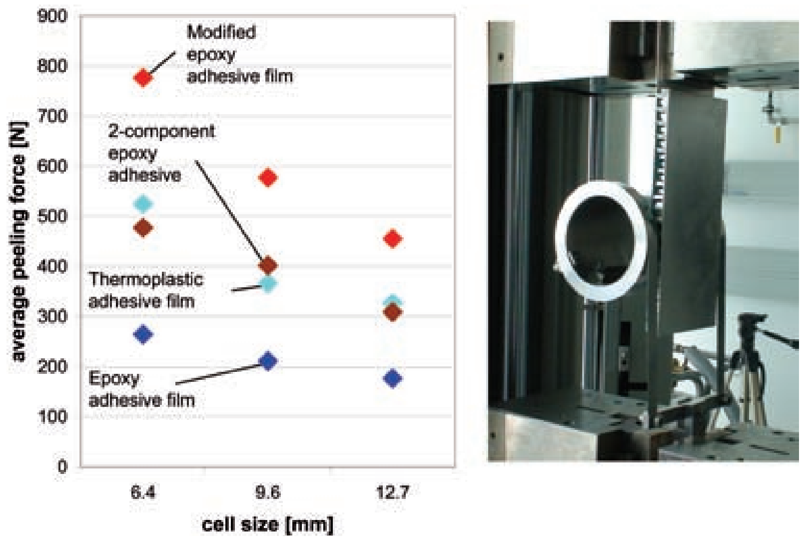

Figure 8: Peel test by means of a drum, DIN 53295 compliant

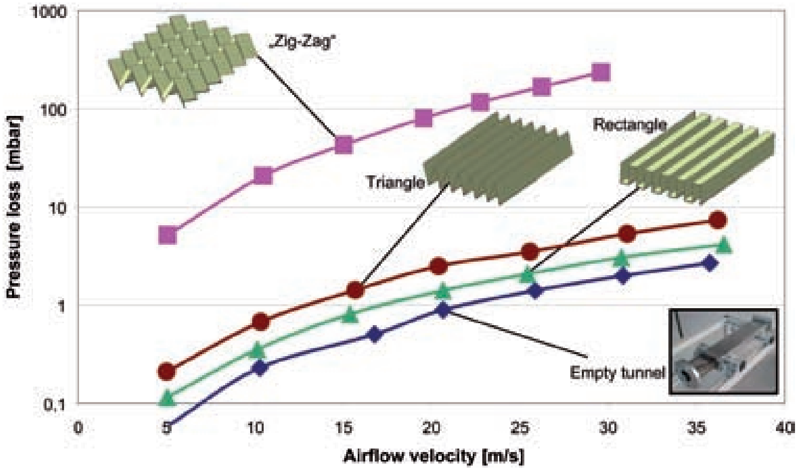

Figure 9: Examination of pressure losses of different core geometries

compartment of cars and trucks. This however requires a core structure that is open on one side at least, which means that honeycomb or closed-cell core structures are ruled out for this application. In an effort to study this possibility, an experiment was set up and systematic tests on different folded and corrugated core structures were carried out, **Figure 9**. The core structures were subjected to various air flow rates and the pressures in front of and behind the 200 mm long core structures were measured. This allowed determination of pressure loss at different flow rates due to the introduced core structure.

Core structures with different geometrical cross-sections, but constant test structure core height and core density, were selected. The cross-sections of the selected core structures corresponded to a triangular, rectangular and zigzag-shaped structure. The determined values could therefore be compared to each other, however with only the zigzag structure possessing better mechanical properties to withstand compression and shear than the simple folded structures.

The tests carried out showed that the pressure loss caused by the complex folded core structure was up to ten times higher than that caused by the simple folded, corrugated-type structures. In order to establish an optimal area of application, an optimum between the required mechanical properties and the pressure losses caused by media supply lines would then have to be determined. It should also be taken into account that vertebration or vibration effects may give rise to undesirable noise that would further limit the area of application.

6 Conceptual Prospects

Reduction of weight and therefore driving resistances can be achieved by utilisation of sandwich structures in future vehicles. This is however subject to realisation of an economically efficient overall solution. Development of new manufacturing methods and the possibility to integrate functions in various sandwich structures can help to achieve this related to the work of DLR and Fraunhofer. Systematic design of sandwich structures is absolutely essential for realisation of optimised components and for tapping the huge potential that lies in this construction method as compared to other construction methods, particularly with regard to large planar components or simply curved floor or roof structures and interior parts.

Various institutes are cooperating as part of the Competence Centre for Automotive Light-Weight Solutions in order to develop solutions for vehicle-specific requirements. Using metal sandwich plates instead of conventional car body structured metal plates for the floor structure for example, can contribute to weight reduction and to support comfort aspects in the passenger cell by means of integrated air or water. For example the transitions can be realized through inserts or plastic connectors. The high capacity of sandwich plates to withstand torsion and shear could also reduce the susceptibility of the underside to drumming and vibration as well as increase vehicle body stiffness or quality of the lightweight structure. Fibre reinforced plastic sandwich composites are also being considered for application in semi-structural components like doors, flaps or covers, since this construction method is more favourable for realisation of an economic and weight-specific optimum.

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