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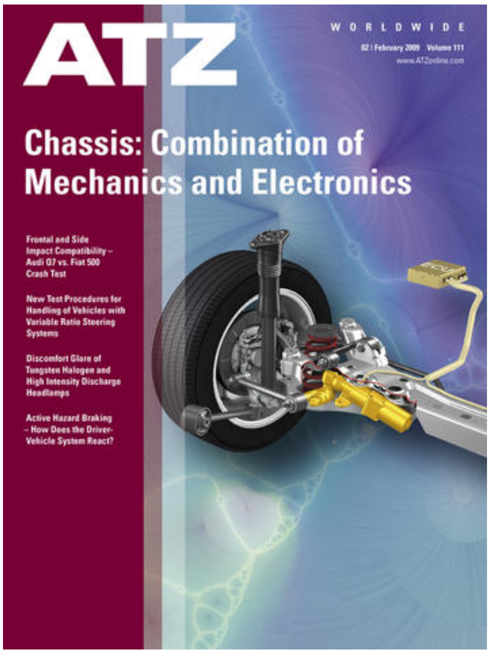

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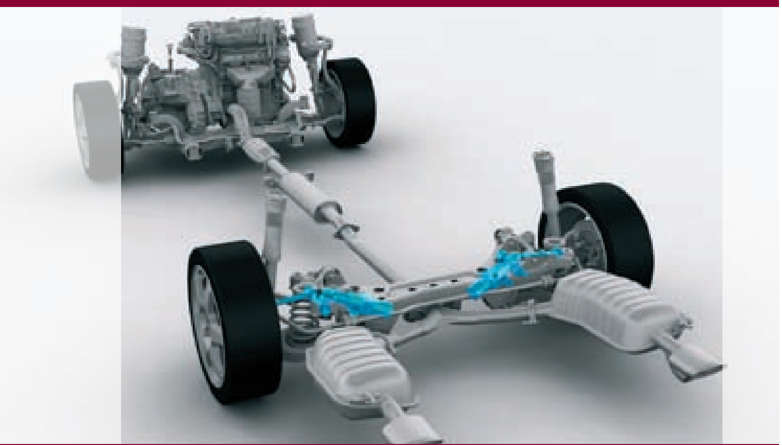
Chassis: Combination of Mechanics and Electronics



4
AGCS is a **Chassis** control technology developed and patented by Hyundai which is different from conventional active suspension control systems. Benteler presents a compound crank rear axle with a compact design and good axle kinematics.

Reading Helps

Dear Reader,

For a year now, we have been publishing the English-language version of ATZ as an e-magazine. We have been very pleased with the positive response and grateful for your loyalty and for any suggestions on how we can make the magazine even better for you.

I often find that top managers say that they have hardly any time to read. As understandable as this may be in times of crisis, I would maintain that now, of all times, it is more important than ever to take some time out to read. Speed is essential for putting out fires, but it is less helpful for developing a strategy to reduce the risk of a fire. On the contrary, many fires are caused by carelessness, almost always a side-effect of hastiness.

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Johannes Winterhagen
Wiesbaden, 8 January 2009

Johannes Winterhagen
Editor-in-Chief

ATZ WORLDWIDE
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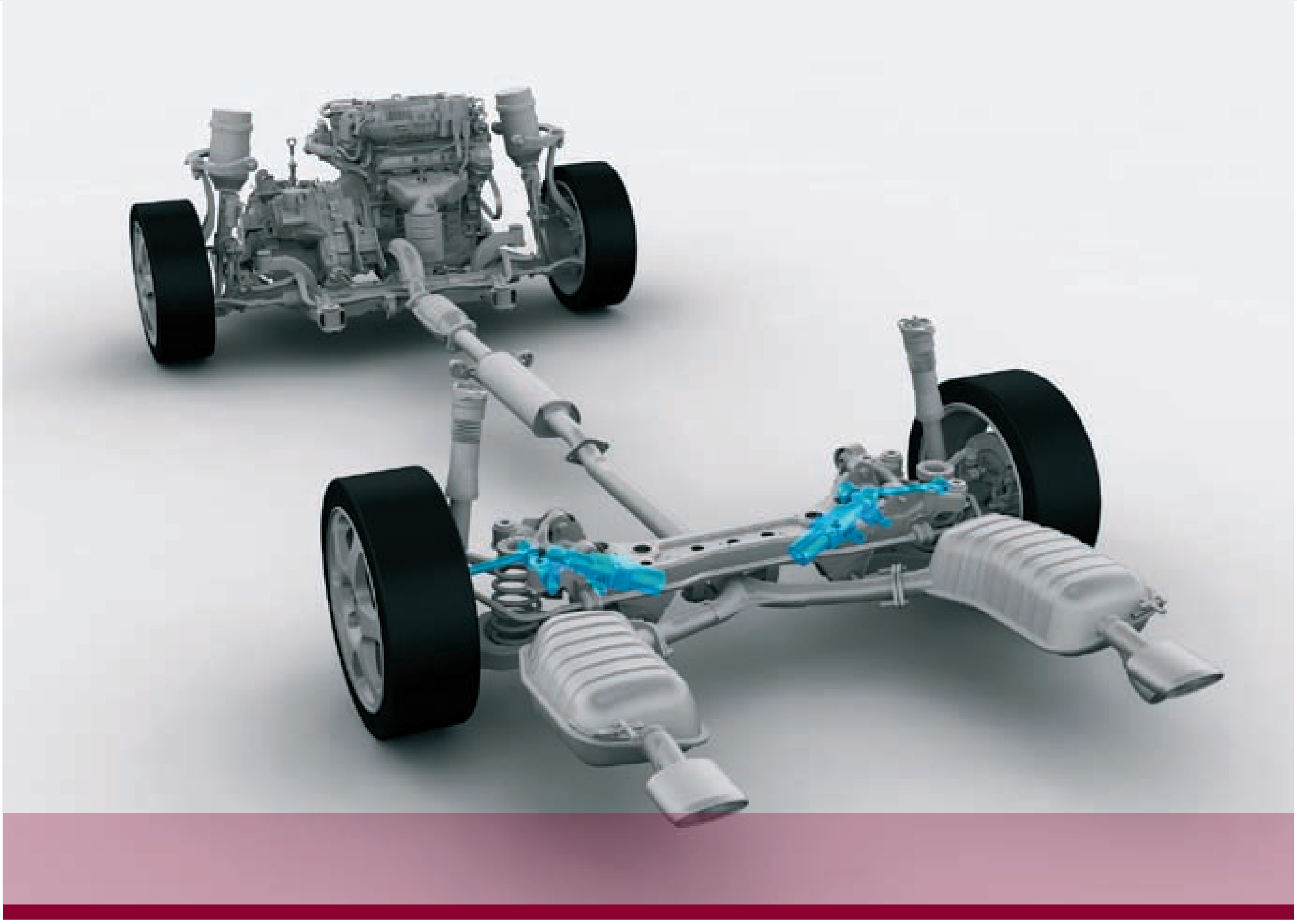

Active Geometry Control Suspension

Improvement of Vehicle Stability in the Hyundai Sonata

The Active Geometry Control Suspension (AGCS) is a chassis control technology developed and patented by Hyundai which is different from conventional active suspension control systems. Conventional systems such as four wheel steering (4WS) [1] and active suspension are designed to directly control the phenomena such as roll and pitch. AGCS controls the cause. It controls the inboard suspension link mounting position to generate optimal suspension geometry characteristics for driving conditions.

1 Introduction

Extensive effort has been made by automotive chassis engineers to overcome the performance limit of passive suspensions by developing active control systems. Recently, some automotive companies have developed effective systems such as 4WS, semi-active, slow-active (recently applied to luxury class vehicles) and full-active (controls up to 20 Hz) based on their accumulated engineering technologies. However, these systems require a number of sensors, excessive energy for the control, and thus they are expensive. The fundamental reason for these matters of the conventional control systems could be their control concept of "Phenomenon control". The term "Phenomenon control" is a method to improve performance by detecting improper or undesired motions of the vehicle by sensors, and then mitigating or absorbing them promptly. These motions of the vehicle would be caused by input from external disturbances. Unfortunately, since the phenomena occur simultaneously in various shapes in response to one input, it is difficult to handle and improve all the phenomena at the same time. However, if there exist any control systems that can control the cause of the phenomena, it can be much more efficient and the result of vehicle motion would be more natural. AGCS is an example of a cause control system.

2 AGCS System Overview

The AGCS varies the inboard mounting position of suspension link in the vertical direction to change toe and camber characteristics for optimal suspension geometry under various running conditions. The AGCS applies control force in the perpendicular direction to the applied load to minimise energy consumption. The differences in fundamental concepts between AGCS system and the conventional control systems are summarised in **Table 1**. As to be seen in **Figure 1**, the AGCS system does not change merely toe and camber angles but controls the migration characteristics of toe and camber. The system is now under further improvement to downsize motors, package layout and so on. In the near future, AGCS will evolve as an enhanced system to optimise caster, steering axis and roll characteristics together with toe and camber angles.

However, the AGCS system in the rear suspension of the current Sonata is focused on the control of bump toe change characteristics. The AGCS is different from the conventional 4WS system because it minimises bump toe during high speed straight driving and slow cornering for straight forward stability, but increases bump toe to increase understeer tendency. As a result, cornering stability can be improved under large lateral acceleration due to high speed cornering, side wind, and abrupt steering.

As shown in **Figure 2**, the AGCS system consists of an electrical actuator, a control lever, and an Electronic Control Unit (ECU). The ECU controls the actuator stroke based on the vehicle speed and steering wheel angle. The actuator rotates the control lever about the hinge, which

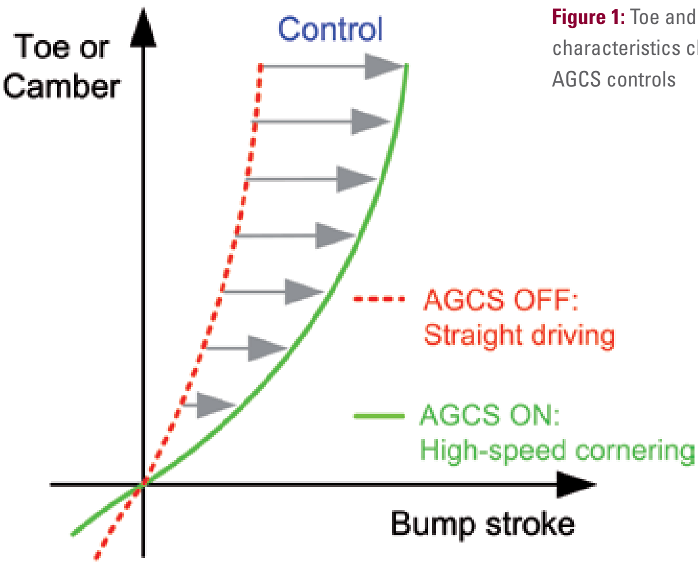

Figure 1: Toe and camber characteristics change with AGCS controls

Table 1: Conventional active control systems versus AGCS

Item	Conventional active control system	AGCS
Control concept	controls the phenomenon (roll, yaw, etc.)	controls the cause of phenomenon
Direction of control and energy consumption	– in the same direction of the actuating load – requires excessive energy – weight and cost disadvantage	– in the perpendicular direction of the actuating load – minimum energy is required – advantageous in weight and cost
On fail	vehicle might be unstable	independent from vehicle stability (only suspension geometry changes)
Driver's reeling	might be unnatural	natural

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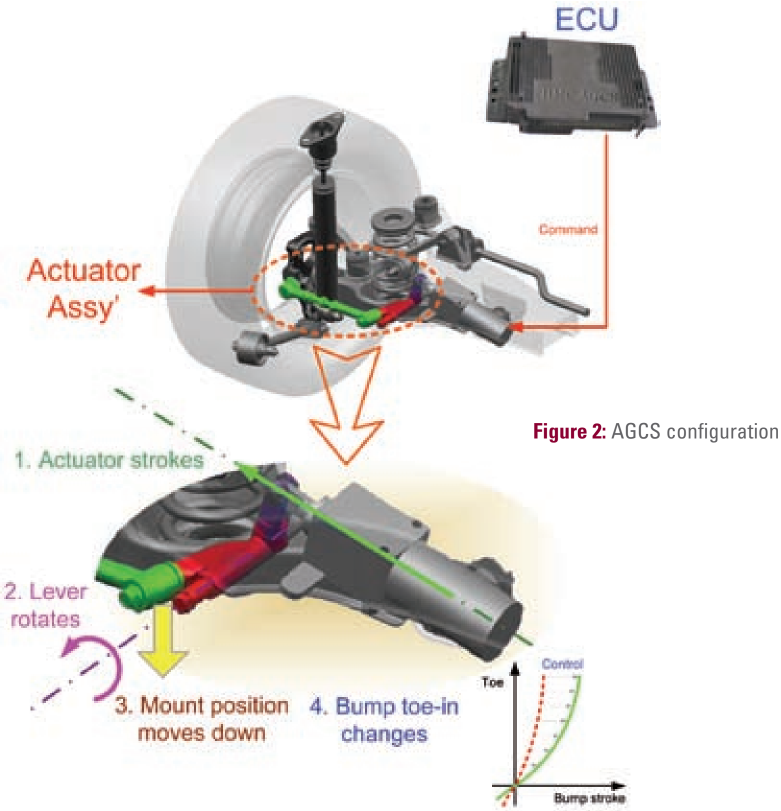

Figure 2: AGCS configuration

is fixed on the rear subframe. Then the control lever moves inboard mounting position of the assist link of the rear suspension downward to change suspension geometry for optimal toe in value. This procedure is illustrated in **Figure 3**.

Since the load applied to assist link and the control force is almost perpendicular, the energy required for the control is nearly zero ($W = F \cdot S \approx COS(90°) \approx 0$). Therefore, the AGCS is basically much more efficient than the conventional active control systems which have the control force and applied load in the same direction.

Figure 4 shows the trajectories of vehicles with AGCS on and off. The vehicle with AGCS on shows much more stable behaviour during cornering because the toe angle of rear-outer wheel is increased, thus generates more tyre force.

3 Control Logic

The AGCS consists of sensor part, control part and actuator part. The sensor part consists of vehicle speed sensor and steering angle sensor. The control part estimates vehicle lateral acceleration based on the signals from the sensor part, and determines control command to the actuator part. The actuator part carries out the control of inboard mounting position of the rear outer suspension link to generate optimal toe angle based on the command from the control part. **Figure 5** shows the flow of AGCS control. The control logic of the AGCS system is simple,

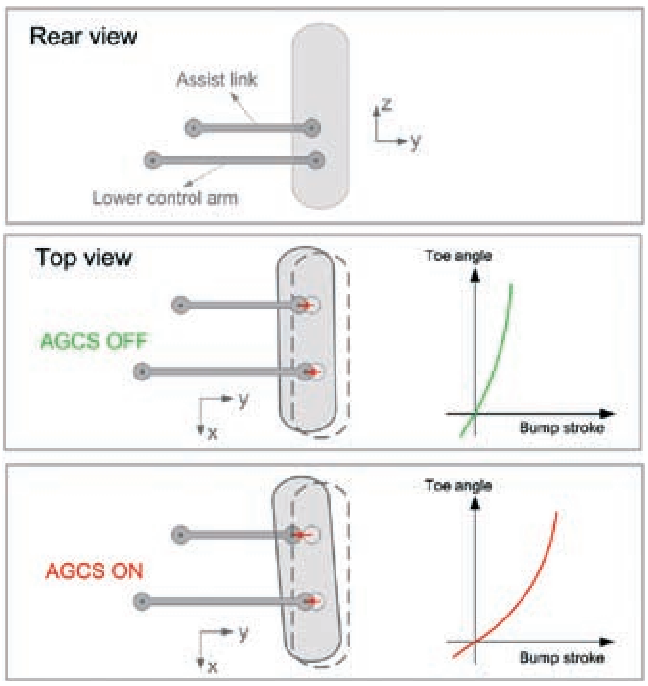

Figure 3: Bump toe characteristics

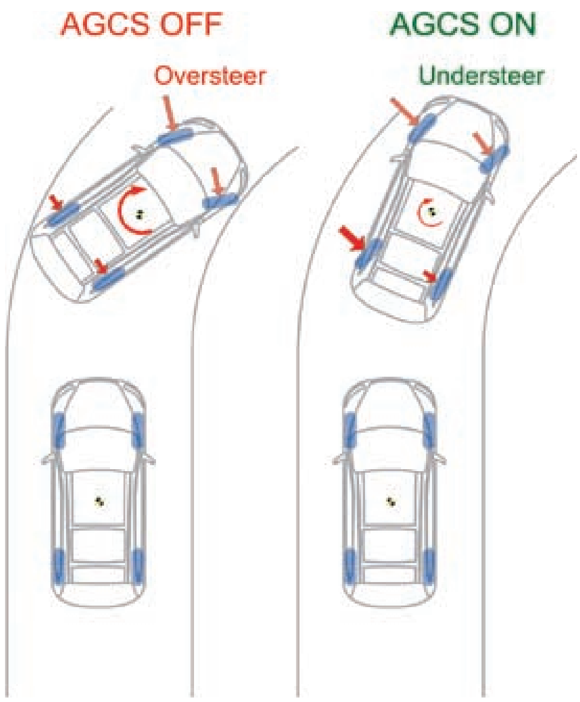

Figure 4: Vehicle trajectory in high-speed cornering

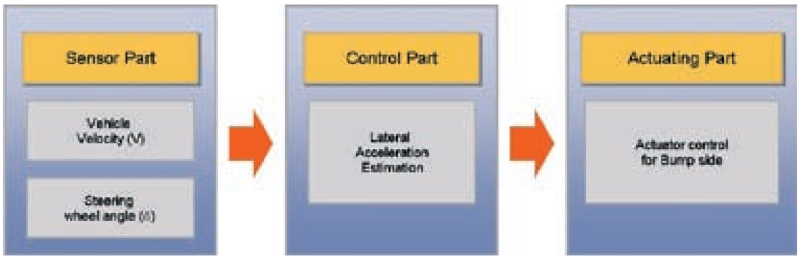

Figure 5: Control flow of AGCS system

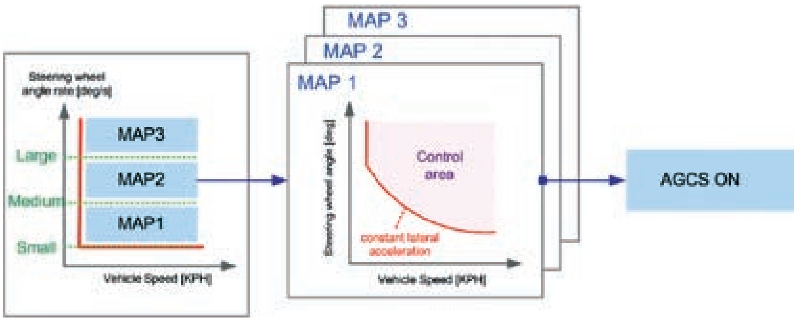

Figure 6: Control logic structure

but is optimised to control appropriately and efficiently for all driving conditions. There are three different maps based on the steering wheel angle rate. These maps provide the information about transient state of vehicle. If there is more transient state in the vehicle motion, the AGCS will engage more promptly and aggressively.

The most important parameters for the control logic are:

- control starting point: The control logic is designed to engage at constant levels of lateral acceleration for all speeds in order to guarantee a homogenous effect. The constant level is determined based on the steady-state characteristics of vehicle under different vehicle speeds. Thus, the control logic determines the actuation of the AGCS based on the steady-state lateral acceleration estimated based on the current steering wheel angle, steering wheel angle rate, and vehicle speed.
- control stroke: In order to maximise control effect and to obtain more progressive vehicle response, three different maps are defined to adjust control stroke according to vehicle lateral acceleration.

Figure 6 shows the structure of the control logic. There are three different maps (MAP1~MAP3). The AGCS controls the actuator base on these maps to generate the target toe angle. AGCS chooses MAP1 under small, MAP2 under medium, and MAP3 under the large steering wheel rate. MAP1, MAP2, and MAP3 will produce small, medium, and large actuator stroke, respectively.

4 Vehicle Performance

An extensive test programme was conducted in order to validate effects of the AGCS system. This test programme consists of subjective evaluations under various driving situations, and different handling objective tests, mainly focusing on transient response characteristics.

4.1 Subjective Evaluation

Quasi steady-state cornering and the single lane change test are conducted under AGCS on and off condition for the subjective evaluation of handling performance affected by the AGCS. Based on the evaluations from expert test drivers, the evaluation results are rated from one to ten, **Table 2**. The evaluation results and related ratings are summarised in **Table 3**. The vehicle under AGCS off conditions during lane changes shows desirable delay, damping, and response characteristics at low-mid levels of lateral acceleration. However, under high lateral acceleration, low

Table 2: Test result and driver perception ratings

Rating	~3	4	5	6	7	8	9	10
Perception	poor	objection	requires improvement	barely acceptable	fair	good	very good	excellent

Table 3: Summary of evaluation results and ratings

Test name	Evaluation item	AGCS off	AGCS on
Lane change	Response delay	7	7.2
	Response damping	6	7.5
	Rear slip generation	6.5	7.7
	Progressivity of AGCS		7.2
	Controllability at the limit	6.5	6.9
	Power-off sensitivity	6	7.3
Quasi steady state cornering	Understeering tendency	6.5	7
	Front-rear balance	6.7	7.5
	Maximum grip	7	7
	Predictability at limit	7	7.2
	Power-off sensitivity	6.4	7.8

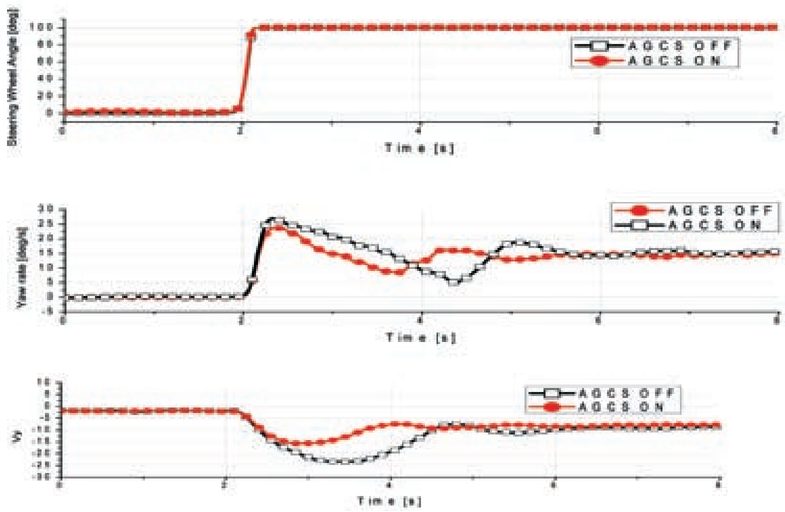

Figure 7: Step steer time results

yaw damping and excessive vehicle reaction produce too fast response, and controlling the vehicle becomes difficult. Eventually, oversteer occurs at the limit and countersteer becomes necessary. Under AGCS on conditions the vehicle shows improved transient reaction by increasing yaw damping which is one of the most important factors during cornering. Other improvements are also noticed such as faster reaction, greater progressiveness and an understeer tendency. The vehicle shows good steady-state response characteristics with understeer evolution. Predictability of the limit is sufficient, but increasing understeer tendency further improves the performance. Note that the influence of AGCS is only reached when steering wheel inputs are higher than certain degrees. Therefore, the vehicle may not be in a pure steady-state. However, this condition can be considered as a boundary of transient and steady state, thus can be used for the evaluation.

The test results show that the contributions of the AGCS are significant, and the positive contribution of AGCS on the vehicle performance. The generation of rear slip angle is more stable, and the understeer tendency at the limit of tyre cornering force is more pronounced. The results confirm better predictability of the grip limit and improved balance of front and rear slip.

Table 4: Step steer test conditions

(Parameters at 4 m/s²)	AGCS off	AGCS on
Peak rear slip angle [°]	0.22	0.22
Peak roll vel. [°/s]	0.33	0.31
Steady-st. AVz/SWA gain [°/s/ °]	0.16	0.15
90 % response time of AVz. [s]	0.60	0.55
Peak response time of ay [s]	0.32	0.29
Peak response time of AVz[s]	0.12	0.10
Peak Ay [m/s²]	9.25	9.52
Peak AVz [°/s]	-2.72	-2.84
Peak front slip angle [°]	-1.09	-1.16
Peak sides lip angle (REF) [°]	-1.41	-1.43
90 % response time of Ay [s]	7.68	8.25

4.2 Objective Tests

An extensive number of various objective tests are performed, however, the three most representative test results will be presented in this section: step steer [2], double lane change [3] and frequency response tests [4].

4.2.1 Step Steer

The step steer test is performed in order to analyse the vehicle response with high accuracy under a given steering input. Again, overshoot and delay of response are measured for different output variables, such as yaw rate, lateral acceleration, and slip angles etc. As shown in **Figure 7**, the lateral speed (related to the slip angle) of the vehicle is significantly reduced as a direct influencing factor of the AGCS. Concerning the response values of the vehicle, the effect of AGCS intervention is noticeably shown in the damping of increase yaw rate. Both the frequency and peak values of the yaw curve are lower with AGCS on test compared to off test. The numerical test results are summarised in **Table 4**.

4.2.2 Double Lane Change

Contrary to the step steer test, the double lane change is a closed-loop test. The test driver corrects the steering wheel angle to follow target path. Therefore, a comparison of the results of different test runs is much more difficult than in an open-loop test. However, the stability graph, **Figure 8**, that plots slip angle versus slip rate shows a good indication of vehicle performance under such a driving situation. Figure 8 shows that the side-slip and side-slip-rate evolution is significantly contracted with AGCS on, which indicates an improvement in vehicle stability. The numerical test results are summarised in **Table 5**.

4.2.3 Frequency Response Tests

The frequency response test proves how the rear-toe control system damps the yaw gain response throughout the range of steering input frequencies. The test results are shown in **Figure 9**. A vehicle without AGCS shows a typical yaw response with an increasing gain function until reaching resonance frequency (around 1.1 Hz). However, with AGCS on, the yaw response gain is flatter, and the increase of gain is very small. This is an indication of yaw damping increase, which proves the effect of the AGCS system in improving vehicle stability.

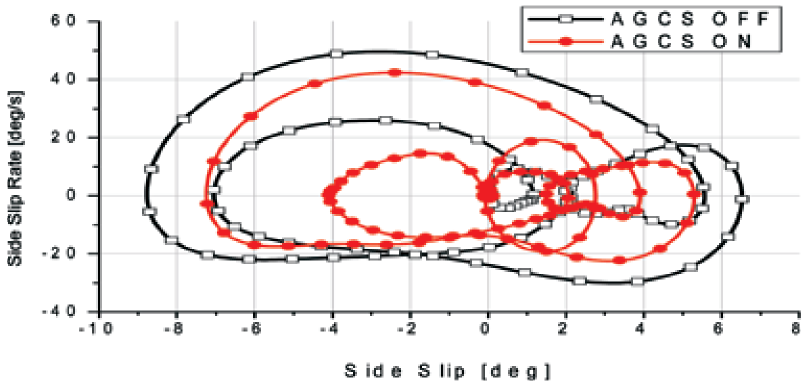

Figure 8: Stability in the double lane change test

5 Conclusion

The AGCS system is a unique active chassis control system. It is a more advanced "cause control" type system, which can operate with slower actuator and lower energy consumption compared to the conventional active control system. The variation in the suspension mounting position enables suspension to generate various, and optimised wheel trajectory for the driving conditions. In other words, AGCS can realise various suspension geometries in one suspension. More compact, efficient, and lower-price active chassis control systems such as AGCS are expected to emerge in the future.

Table 5: Double lane change test results

		AGCS off	AGCS on
Vehicle speed [km/h]		**57.4**	**60.7**
Peak averaged values	Ay [m/s2]	8.15	8.14
	AVz [°/s]	40	36
	Side slip angle [°]	7.3	5.55
	Roll [°]	4.73	4.63
	Roll rate [°/s]	20.17	19.3
	SWA [°]	235.1	223.4
	SWR [°/s]	990.5	828.2
	SWT [Nm]	7.6	5.71
Ratios	Ay/SWA	0.035	0.038
	AVz/SWA	0.174	0.167
	Roll/Ay	0.581	0.568
	SWR/SWA	0.033	0.026
Delay	SWA to Ay	0.193	0.185
	SWR to AVz	0.165	0.124
	Ay to Roll	0.039	0.293

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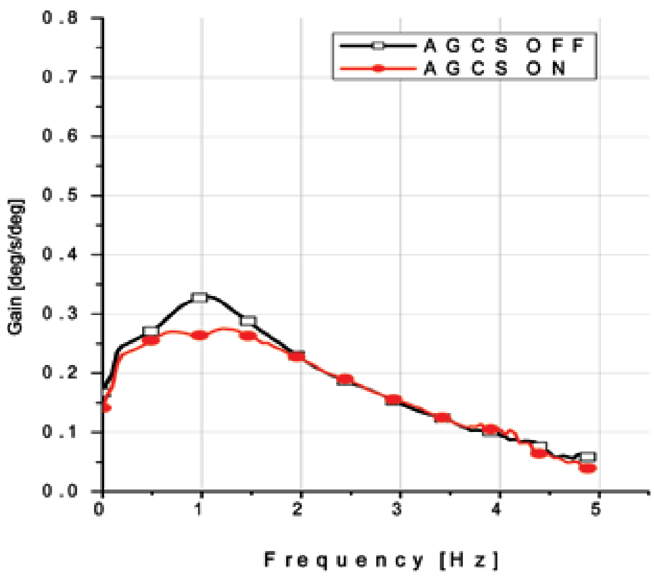

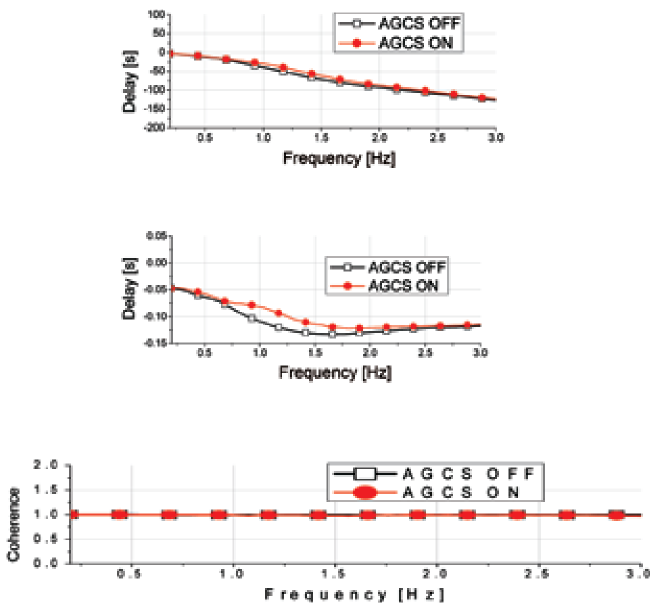

Figure 9: Frequency response test results

The Twist Beam Rear Axle
Design, Materials, Processes and Concepts

Twist beam axles by Benteler are being used not only in vehicles of the A-, B- and C-segment but also in the D-segment, in Mini Sport Utility Vehicles and four-wheel drive cars. Due to the low costs, the compact design, the reduced weight and the good axle kinematics it often represents an unrivalled solution. The trend towards improvements of the axle design as well as of the optimised and partly newly developed manufacturing technologies will continue.

1 Benchmarking

To receive a sufficient market transparency, a worldwide operating benchmarking system is needed, which systematically works out the properties respectively the advantages and disadvantages of the twist beam axles. Thus, the Benteler development satellites in Europe, America and Asia are observing the local markets and investigating all interesting new axles. The worldwide standardised procedure ensures that the determined data can be easily compared. The data are stored in a database so that all other worldwide development satellites have access to the actual information. In addition to weight, weld seam lengths and dimensions, which are data to be determined easily, material analysis, stiffness measurements and fatigue tests belong to the essential investigations to evaluate an axle concept, **Figure 1**. This requires the development of operating figures, which enables a comparison of different axles and which can be used also in the concept development phase. Axles, which lie above the expected performance, especially in terms of fatigue, are calculated by using a 3D-scan with following finite element calculation including the appearing stresses, so that the function of the axles can be evaluated. The comparison of the stresses with the results coming from the fatigue tests and the material investigations leads to the used manufacturing technologies. The experience from own developments combined with over 40 analysed twist beam axles are the basis for concept development.

2 Concept Development

The permanently reduced development periods and especially the complexity of the twist beam axles in the adjustment of the axle kinematic values arranged Benteler to strike a new path in the concept development. The aim was to be able to propose the suitable concept to the customer in shortest times, always considering the roll rate, the roll steering and the fatigue and weight requirements. For basic knowledge of the twist beam axle an analytic analogous model was developed, which delivers high accuracy in the forecast of the roll rate and the roll steering without the need of a Computer-aided Design (CAD) model respectively a finite element calculation, **Figure 2**.

2.1 Roll Rate

The roll rate k_r is defined as the roll moment divided by the roll angle, Eq. (1):

$$k_r = \frac{M_r}{\varphi_r} \quad \text{Eq. (1)}$$

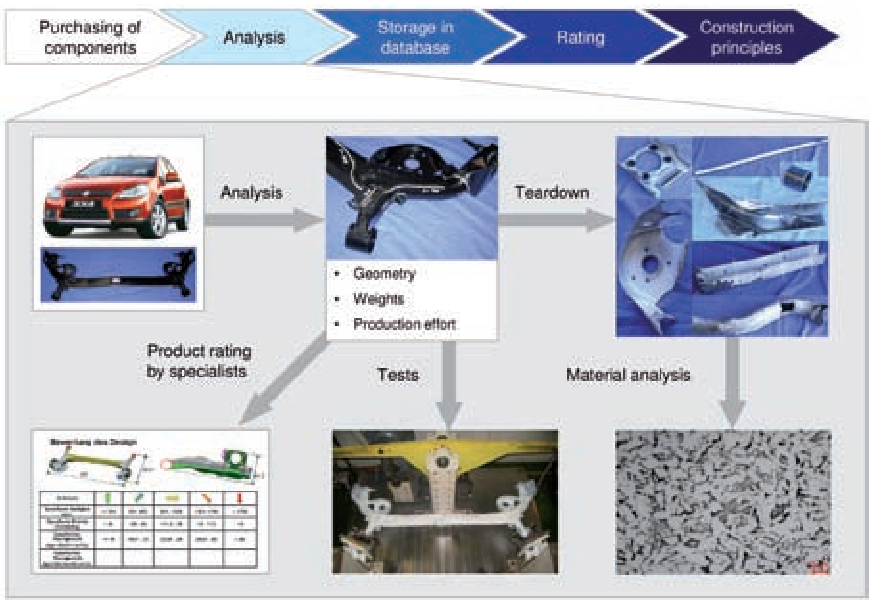

Figure 1: Procedure within the benchmark analysis of twist beam axles

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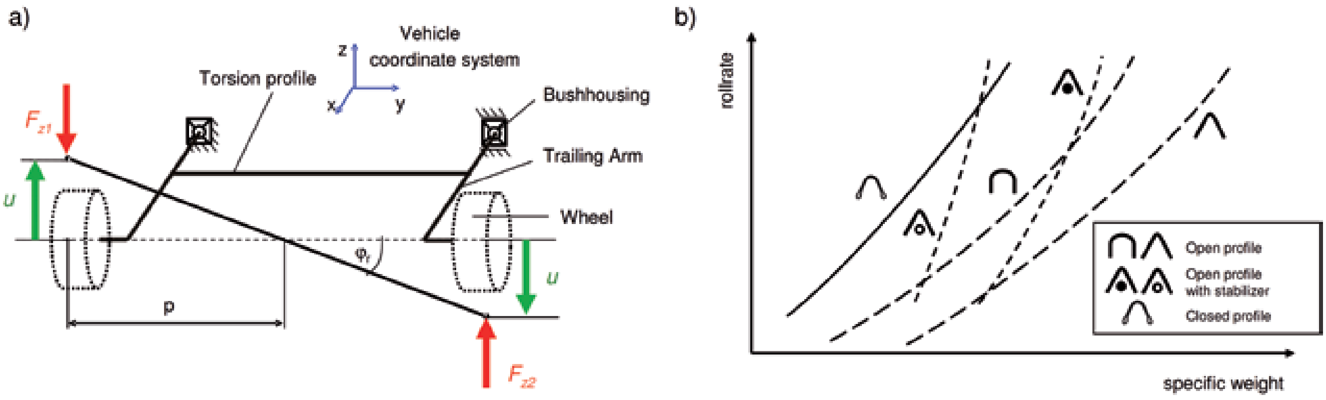

Figure 2: a) Elastokinematics in twist beam axles; b) Definition of the roll rate k_r and influence of the cross section on the roll rate and the weight

The roll moment M_r is being calculated by the forces charging at the wheel centres, multiplied with the half of the track width, Eq. (2):

$$M_r = (|F_{z1}|+|F_{z2}|) \cdot p \qquad \text{Eq. (2)}$$

The roll angle φ_r is the arcustangens of the deflection divided by the half of the track width, Eq. (3):

$$\varphi_r = \arctan\left(\frac{u}{p}\right) \qquad \text{Eq. (3)}$$

and thus represents a pure function of geometrical values.

Considering important geometrical values as axle connection points, cross beam position, wheel centre positions and corresponding diameters the roll rate can be calculated. Especially, the cross section geometry of the cross beam plays an important role for the axle adjustment. Basically, it is differed between open and closed profiles. While the torsion area moment t is defined by the circumferential and the wall thickness for an open profile, the implicit area has to be additionally considered for the closed profile. By the use of closed profiles under consideration of the roll rate, lighter cross beams and thus lighter twist beam axles can be designed. For open profiles, a parallel placed stabiliser bar is often used additionally to increase the roll rate. The comparatively simple variation of the wall thickness respectively the diameter of the stabiliser bar results in a high flexibility in adjusting the roll rate.

2.2 Semi-Analytic Stress Calculation

Due to complex geometries, especially in the transition areas between the centre of the cross beam and the connection area of the side arm, a pure analytic calculation with sufficient accuracy is not possible. Thus, in a first concept phase the maximum stress values are being calculated based on a semi-analytic approach. This means that a combination of analytic and numeric is required. For this, Computer-aided Engineering (CAE) results are being used, coming from former calculated axles, and are combined with the analytic approach out of the roll rate calculation, **Figure 3**. This is just possible because the loads, calculated as von Mises stress, are changing proportionally with the roll rate, Eq. (4):

$$k_r \sim \sigma_{vonMises} \qquad \text{Eq. (4)}$$

In the concept phase the stress calculation serves as a basis for defining the right materials and manufacturing technologies; the manufacturing costs can be influenced decisively at this stage.

2.3 Roll Steering

In the concept development phase the roll steering gradient, which is mostly defined by the customer, has to be considered beside of the roll rate. The roll steering gradient is defined as the change of the toe angle related to the roll angle, Eq. (5):

$$roll\ steering\ gradient = \frac{toe\ angle\ [°]}{roll\ angle\ [°]} \qquad \text{Eq. (5)}$$

The main influencing parameters are the track width, the length of the side arm, the position of the cross beam, especially of the shear centre, the distance between the axle connection points and the stiffness of the bushings. Similar to the roll rate calculation, a semi-analytic approach leads also here to very good results, **Figure 4**.

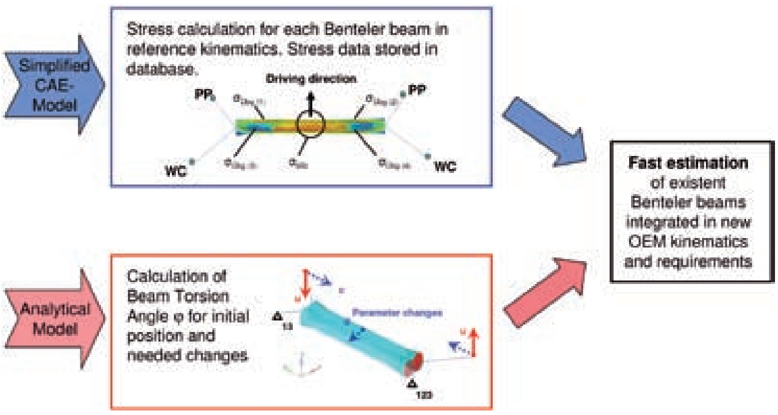

Figure 3: Stress evaluation in twist beams

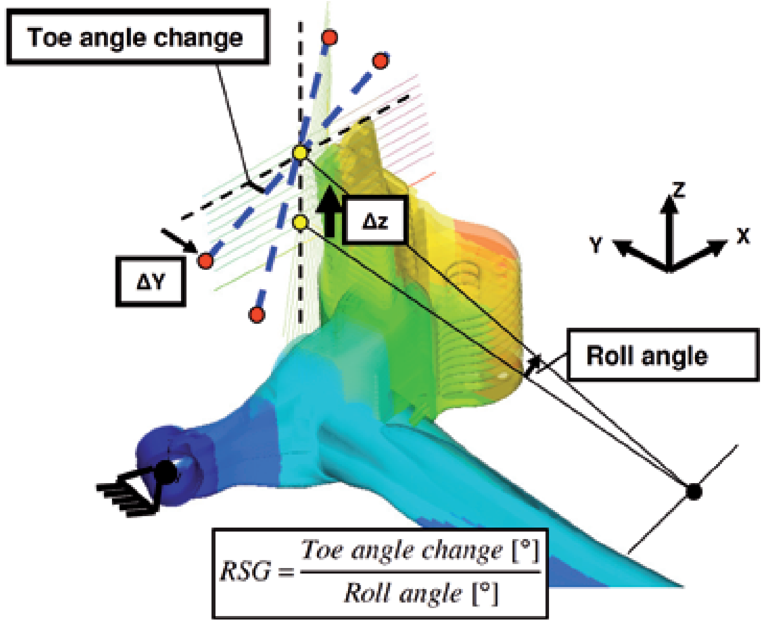

Figure 4: Definition of the roll steering gradient

Benteler has the ability to create axle concepts within one day by using this concept development and without the help of CAD and CAE which fulfil the technical requirements of the customer. At this, all profile geometries, different side arm concepts and combinations with stabiliser bars can be considered, Figure 2 right, so that the concept proposal is very exact regarding weights and costs.

3 Steel Materials and Semi-Finished Products

Depending on the vehicle segment different requirements are existent for twist beam axles which can be also achieved by the use of suitable materials and the semi-finished product. Thereby, properties like static and fatigue strength, formability, weldability, coatability, fracture toughness also at lower temperatures and the sensitivity regarding a hydrogen embrittlement play an important role, especially if ultra high-strength steels are being used. In case of open profiles, microalloyed high-strength low-alloy (HSLA)-steels in an Ultimate Tensile Strength (UTS)-range up to 550 MPa are currently prevailing. For closed profiles, mainly welded and precision rolled tubes according to DIN EN 10305-3 are being used presently, whereas the inner and outer scarfing of the tubular weld seam and the permitted surface failure depths are being exactly controlled in the tube production process. Furthermore, restricted wall thickness tolerances are required for the hot rolled strip and the tubular products to minimise the variations of the roll rate.

This can be realised by the selection of a medium hot strip mill which can guarantee highest wall thickness tolerances for the hot rolled strip (as a pre-material for the tubes). For a vehicle of the A-segment, a laser welded tube was used for the first time, which shows a reduced surface failure depth of maximum 50 µm compared with conventionally produced Resistance welding (ERW)-tubes. This was realised by a specific laser device, which scans defined blank areas before the tube forming process and directly sorts out conspicuous sheets. The laser weld seam is narrow compared to weld seams in conventionally produced ERW-tubes, so that a heat treatment step in the tube production process for homogenisation of the weld seams could be left out additionally.

As materials different steel concepts are being used currently, partly in combination with subsequent strength-increasing processes, which will be described later on. In case of medium strength requirements (yield stress; YS range around 400 MPa) usually low alloyed tubular steels, for example E355 according to DIN EN 10305-3, microalloyed HSLA-steels, for example the Benteler steel grades BTT450 (similar to S420MC) or BTR165 (similar to 22MnB5) in normalised condition are being used. These steel grades show medium high yield strength values in combination with a good formability. In case of higher strength requirements (YS-range above 500 MPa) it is advisable to use high strength HSLA- or multi phase steels. As examples,

Benteler developed tubes from the microalloyed HSLA-steel Nano-Hiten or the ferritic-bainitic steel FB590 for this application. On the Asian market, dual phase steels can be observed additionally, which are currently not available in Europe as hot rolled strip. Partly, an annealing step has to be carried out after the tube production to reduce the significant hardening in the weld seam area as well as the strain hardening from the forming process to guarantee a sufficient formability for the following twist beam forming process. This can be realised by the use of a stress relief annealing process in continuous annealing furnaces or by an inductive annealing of the weld seam directly integrated in the tube production process. In case of highest strength requirements (YS-range above 1000 MPa) only quench and tempering steels are being used, which are water hardened in a specific clamping device after the twist beam forming process, followed by a tempering process, **Figure 5**.

On the Asian market, a press hardening is carried out alternatively, that means the forming and the quenching run in the same process step by using a water-cooled stamping tool. Most of the suppliers are using the above mentioned BTR165 with variable chemical composition for this process. Advantages of this steel concept are the worldwide availability and the flexibility regarding mechanical properties, which can be reached in the quenching and tempering (QT) process and by the adjustment of the important parameters.

On the other hand an increased notch sensitivity in the QT-condition has to be mentioned, which requires a maximum inner and outer tube surface quality. Furthermore, parameters like surface

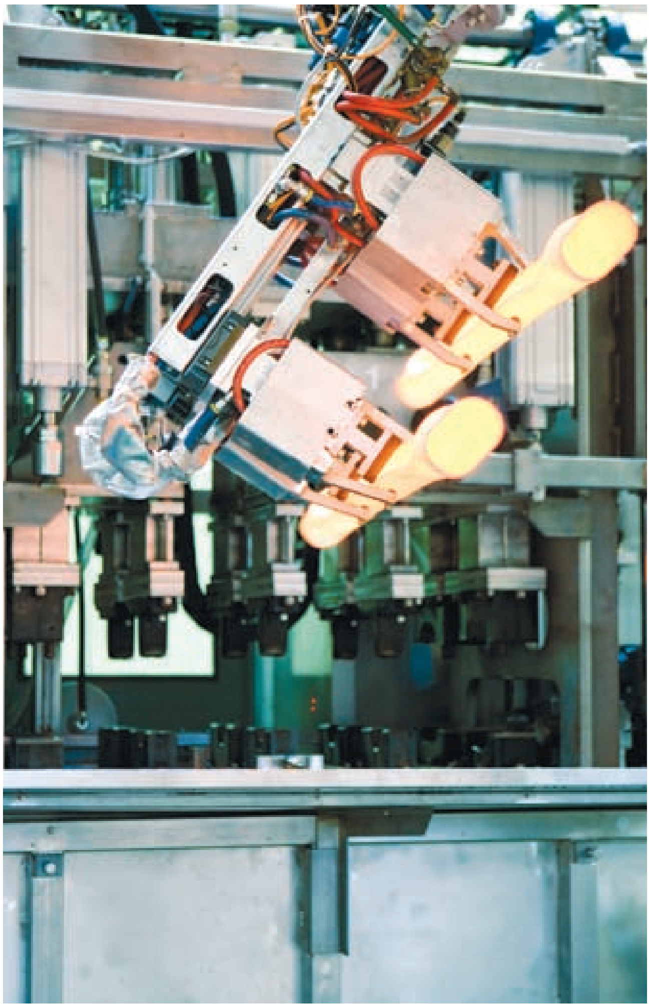

Figure 5: Twist beam made of steel BTR165 before water quenching

decarburisation, oxidation or grain coarsening from the QT-process can influence the fatigue performance of the twist beam axle. A sensitivity regarding hydrogen embrittlement is not given. **Figure 6** summarises the used steel materials for twist beams in form of the banana diagram.

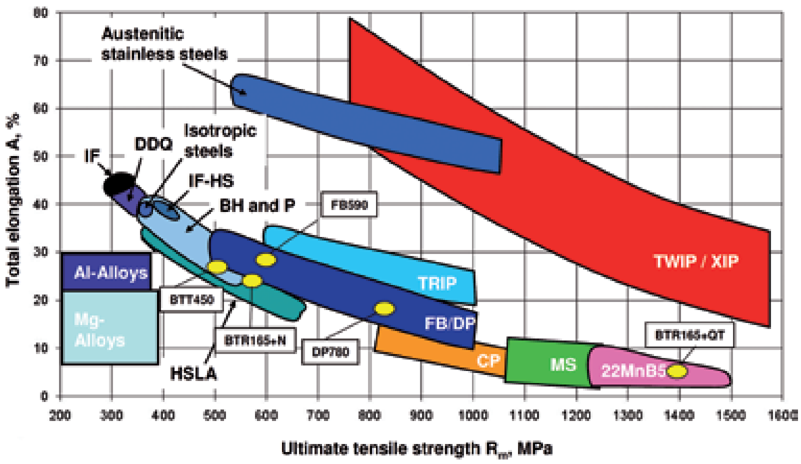

Figure 6: Used steel materials for twist beams

4 Design and Manufacturing Processes

The still existing requirement regarding lightweight construction, combined with increasing requirements concerning strength and stiffness, demand new ways and methods in the design and process development of twist beam axles. To fulfil these requirements,

Benteler is using the Reversed Approach for Process Integrated Development (RAPID) method. The classification of existing twist beam axles in requirement classes enables the selection of suitable components from a tool box in a first step for the compilation of the axle. That means that mainly standard components from existing designs are used for a first design concept in CAE. These fulfil for the most part the requirements regarding weight and elastokinematics coming from the requirement manual.

4.1 Design of the Sidearm

For the sidearms, shell or tubular design with fitted or directly welded wheel carrier connections can be selected depending on the requirements. In case of tubular design, the sidearms are produced with the Benteler Final Shape Rolled Tube (BFSRT) method to receive a final shaped component, **Figure 7**. This process is already established in the series production of sidearms for the actual Ford Fiesta or the Toyota Yaris.

The BFSRT method enables the worldwide use of the same design, independent of the local availability of tubes as semi-finished material. Variable cross sections, which are necessary to reach the stiffness requirements, can be unproblematically realised. The reduced weld seam length compared with a sidearm in shell design additionally increases the robustness of the design regarding fatigue strength. Furthermore, the placement of functional holes for the integration of attaching parts is possible without additional process steps.

4.2 Design of the Twist Beam

Similar the selection of the twist beam takes place. Dependent on the weight, cost and performance requirements a best case scenario is generated. Generally, three concepts are available:

- an open twist beam with or without a reinforcement
- an open twist beam with stabiliser bar
- a closed twist beam out of tubes.

To reach the best topology, the twist beam is being adjusted in detail according to the regularities and influencing factors on the elastokinematic and other attributes. The shifting of the kinematic point from the wheel centre and the bushing kinematic point has a direct influence on the torsional stiffness, the fatigue life of the twist beam, the roll steering as well as on the anti-dive and anti-lift behaviour of the axle, **Figure 8**. Other influencing parameters and their coupled impacts on the different manual requirements are always considered in the development process in a similar way.

At Benteler, a closed profile out of welded tubes is preferred. Depending on roll stiffness, the weight decrease lies between 2 kg and 3 kg compared with other concepts. The variation of the wall thickness in combination with the cross section leads to a variation of the torsion and bending stiffness of the twist beam. This enables a flexible production of variants without appreciable weight and cost increase. On the other hand, twist beams out of tubes are more demanding regarding the manufacturing process compared with open profiles or open profiles with stabiliser bar.

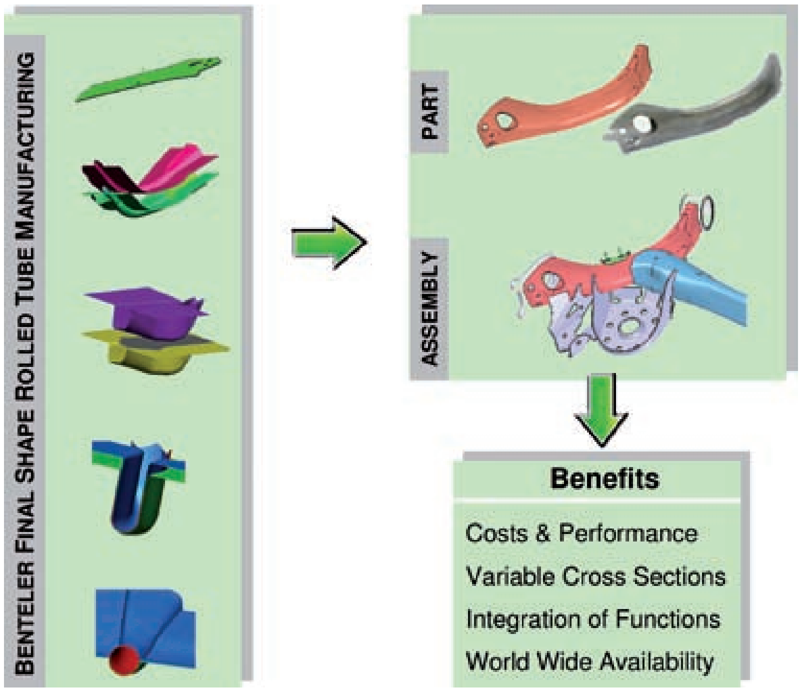

Figure 7: Function integrating rolled sidearms made of blanks

4.3 Virtual Design Validation

The created design, built from standard components, is validated with the RAPID method in the following and optimised until the complete requirements are fulfilled. The RAPID method is the result of a manufacturing orientated design adjustment process, which comprised the experiences of the last ten years at Benteler and contributes to the significant reduction of the development period. Generally, the method is suitable especially for the development of components with a high complexity factor in terms of manufacturing, that is also for the twist beam axles out of tubes.

At the beginning of the chain, there is the virtual verification of the requirements regarding elastokinematics, static

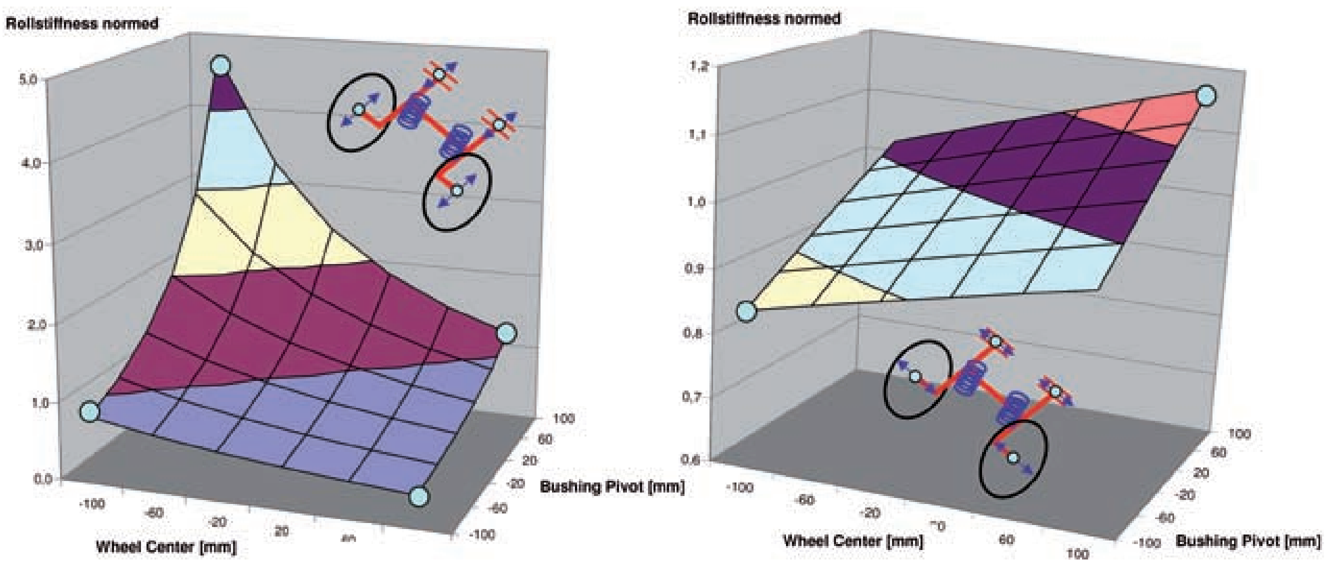

Figure 8: Topologic sensitivity investigations for twist beam axles

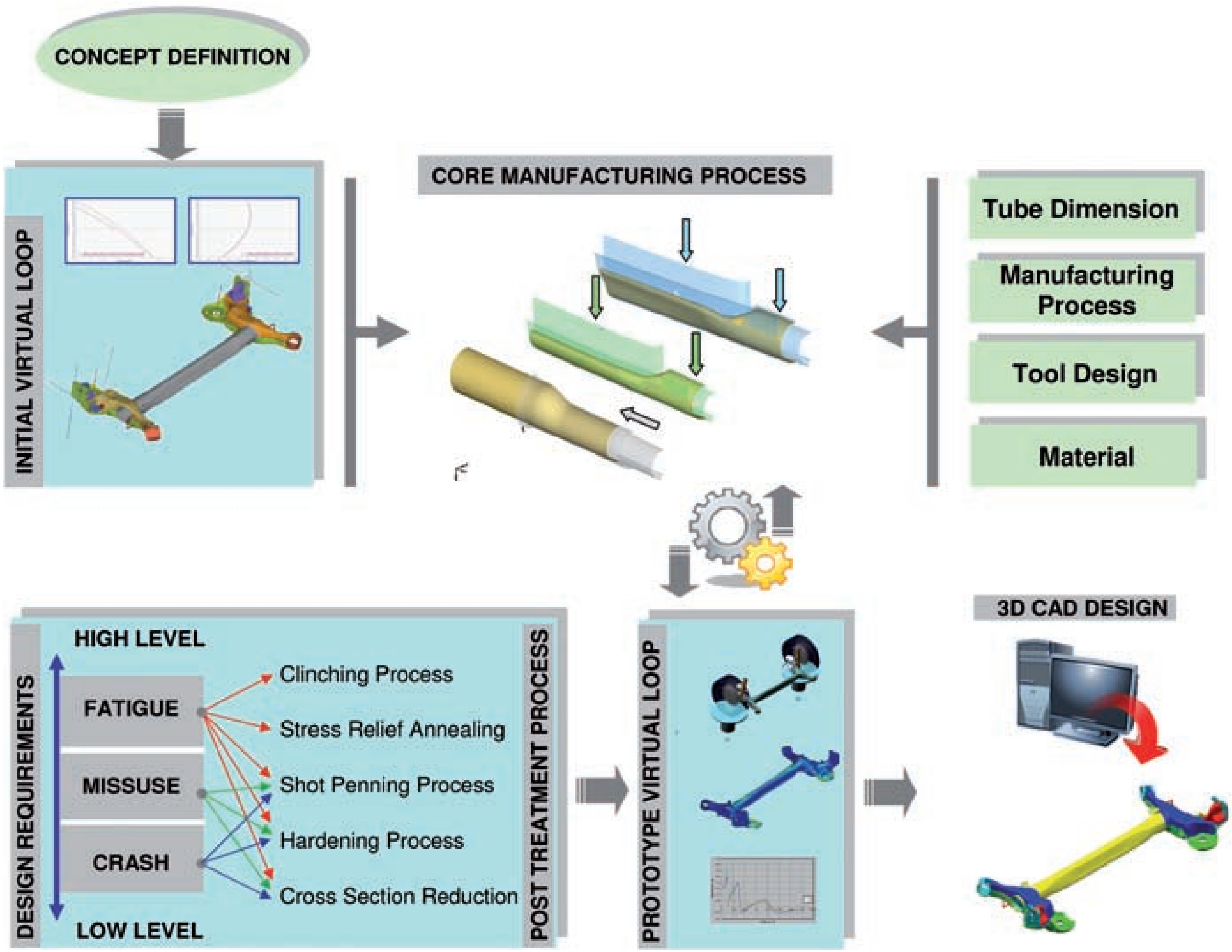

Figure 9: Schematic procedure within the RAPID method

strength, fatigue life, misuse load and eigenfrequency behaviour. Therefore, merchantable simulation tools are applied, for example Abaqus, Adams Car and Design Life, but in a linked procedure. The simulation results of the first loop serve as a basis to confirm or refine the materials for the different components proposed in the concept development phase.

The experience shows that the design of twist beam axles in a CAD system as starting geometries for the virtual validation leads to significant variances compared with real components. Due to this reason, twist beams out of tubes are not designed conventionally within the RAPID method in a CAD system, but in a multi stage well matched forming simulation process. The focus here is the process and tooling definition, **Figure 9**.

The forming process of the twist beams out of tubes is not completely predefined by the tooling, which is different to the shell design. In areas with higher loads a free material flow is preferred to reduce the forming degrees and to get a minimum material stretching. This is just possible by early using of a specific forming simulation, which considers the complete forming history. The forming simulation is additionally adjusted to the materials and the complete forming chain. Due to this connection, a forming process was developed, which guarantees sufficient material characteristics for the following component loads. With this technique a forming of high- and ultra high-strength steels is also possible to realise weight and cost saving potentials. The developed component is completely transferred for the following virtual validation of the requirements. The wall thickness distributions, the strain hardening of the materials or springback effects, resulting from the forming simulation, serve as a basis for the following calculations. This procedure guarantees a high exactness of the simulation and thus a minimised development risk.

The result of the RAPID method is a 3D CAD geometry, describing the design (inclusively tooling and process chain) and fulfilling the requirements from the manual virtually. The tools and the process definition serve as a basis for the production of prototype parts. Depending on the complexity factor, the components are ascribed to a 3D model by a pre-defined scan and used for further correlations and investigations for the finishing series design within the RAPID method.

4.4 Manufacturing Process

Due to cost reasons the general aim is the production of twist beams axles with a minimised number of process steps. The results of the virtual validation show, which process steps are necessary to fulfil the requirements defined by the manual. The selection of subsequent processes for the twist beam or the sidearms essentially depends on fatigue, misuse and crash requirements.

In the course of the platform strategy, twist beams are currently developed and produced by forming and followed annealing processes when low and medium requirements are defined. Clinching and annealing processes are mainly effective regarding fatigue strength increase up to a defined stress level without influencing crash or misuse aspects.

A further increase of fatigue life and also of the crash and misuse performance can be realised by cross section adjustments or the shot peening process. In the past, cross section adjustments were only used in case of package restrictions. Nowadays, the application of different methods at locations with high stress level can result in a local thickening of up to 35 % of the base wall thickness. By the local thickening stress peaks can be degraded. Thus, a wall thickness increase over the whole length of the twist beam is not required anymore.

For the regions with stress peaks or weld seams shot peening processes can be applied additionally. These lead to twist beams with higher fatigue life by the use of induced compression stresses. Due to the specific geometry of the twist beams the accessibility for an optimum process is not given in each case, so that the shot peening (combination of compressed air-rotational jet and turbine jet process) is just possible by the usage of the Riquochett effect. These processes are successfully applied in the aircraft industry. By the adjustment of the shot peening parameters and by using modified equipment, higher and reproducible fatigue life results can be received.

The last option to reach the highest performance level regarding fatigue life, misuse and crash requirements is – as described before – the quench and tempering process of twist beams by using suitable steel materials. Series deliveries in the frame of different projects are taking place since years. Examples are Opel Corsa and Astra, VW Polo, Mitsubishi Colt or Toyota Yaris.

Up to now, processes have been described, which are primarily applied on twist beams out of tubes. In the case of open twist beams respectively in combination with stabiliser bars, the cutting edges of the profile or the welding connection between the stabiliser bar and the sidearm/bracket are more important for the fatigue life. For the cutting edge it is necessary to realise low stresses by adjusting the geometry of the profile. If this is not possible due to package reasons or other restrictions, different cutting edge treatments, for example coining, can be used to increase the fatigue life. These processes are inducing compression stresses at the cutting edges and increase the resistance against cracks starting from these sensitive areas.

A similar process is also applied to the welding area of the stabiliser bar. Here, a wall thickness increase at the tube ends of the stabiliser bar as well as a targeted diameter increase is taking place. Both effects result in a reduction of the torsion stresses in the weld seam area between the stabiliser bar and flange respectively the sidearms and thus a fatigue life increase.

The described processes for twist beam axles have been mainly standardised. With them the requirements from the manual can be fulfilled for different platforms and by consideration of cost and weight aspects. The modularised prototype production by using spanned tools enables a high flexibility for manufacturing twist beam axles for different platform variants. Thus, different variants with variable roll stiffness can be realised in one tool. Therefore, it is also possible to supply the Original Equipment Manufacturer (OEM) with components for driving tests in an early stage, which show a defined tolerance gap regarding the driving behaviour.

5 Outlook

Besides of the advanced globalisation, the environmental protection, especially the reduction of greenhouse gases and the demand for best cost solutions are two important megatrends influencing the OEMs in the selection and the concept development of their axle systems. At this, the great potential of the twist beam axle will play an important role in comparison with the multi link axle. The twist beam axle will remain the rear axle concept of the A- and B-segment. At the same time, a trend to the twist beam axle in the C-segment can be observed. Furthermore, the application of active systems, for example active steering, will further increase the attractiveness of this concept. To fulfil all the above mentioned requirements, the following approaches are being defined by Benteler:

– securing the worldwide availability and quality of materials and semi-finished products to transpose market orientated solutions
– cost reduction by the improvement of manufacturing technologies
– development of new twist beam axle concepts with a target for weight reduction of more than 15 %
– improvement of the axle performance by application of active systems.

In common projects with several OEMs the realisation of these approaches has begun. In this connection, Benteler has the vision of being the full service supplier for innovative solutions. ■

Frontal and Side Impact Compatibility

Audi Q7 vs. Fiat 500 Crash Test

The majority of passenger cars registered in Europe offer high safety standards. To further reduce the injury risk for car occupants, we must extend the scope of safety testing and enhance the vehicle structures. Current tests show that it is no longer sufficient only to look at the occupant protection potential of a vehicle for its own. Therefore, a crash test between a Audi Q7 and a Fiat 500 was done by the ADAC. Goal of the test was to proof the features of both cars regarding their compatibility. The vehicles attested good self-protection in Euro NCAP so they were chosen for the investigations.

1 Introduction

Compatibility studies enable us to assess the interaction between two passenger cars involved in a collision. First, the vehicles' occupant protection is tested under the Euro NCAP protocol. In addition, vehicle-on-vehicle tests are performed with vehicles of different classes to gather information about a car's partner protection potential.

Consumer organisations constantly strive to refine test procedures to keep pace with the developments and today's traffic patterns. This is one of the reasons why ADAC has critically studied the compatibility issue for more than 15 years. As early as in 2005, we analysed the interaction between SUVs and compact class vehicles in a frontal collision. At the time, a VW Golf was crashed into a Kia Sorento and a Volvo XC 90 respectively. However, the results of this 2005 test were ambiguous: While the Golf's cabin stability prevented fatal injuries to the driver, the SUV's aggressive front almost invalidated partner protection.

In this year's follow up test, we crashed an Audi Q7 SUV against a Fiat 500 supermini. The test was aimed to examine the vehicles' compatibility properties. Both vehicles mastered the Euro NCAP test for occupant protection, which is why they were selected for this test.

2 Objective

2.1 Examples from ADAC Accident Research

In 1994, the results achieved by the European Enhanced Vehicle Safety Committee (EEVC) caused the Transport Research Laboratory (TRL) and the Department for Transportation (DfT) to set up the New Car Assessment Programme or NCAP in the UK. After two years, the results of the first test phase were presented to the public. In the following years, a growing number of European governments and automobile clubs joined NCAP founding the Euro NCAP consortium which established itself as a basis for consumer protection activities in the field of passive vehicle safety.

As other frontal crash procedures, Euro NCAP use an immovable block fitted with a deformable aluminium honeycomb structure. The procedure has its limits in simulating a frontal crash, since heavy vehicles "perforate" the 450 mm element and their structures engage with the steel construction behind the deformation element. On the one hand, the introduction of consumer tests has considerably improved the occupant protection of passenger cars over the past few years. On the other hand, real-life accidents show that highly stiff front structures can be critical for the crash opponent. This is the case where both frontal collisions and side impacts are concerned. Photographs of real-life side collisions show that the door panel is subject local load peaks with the longitudinal member contacting the side structure. This causes serious intrusions and tearing of the side structures, **Figure 1**.

Also in a frontal collision, heterogeneous front structures present a hazard. The occupant compartment's splash wall cannot resist the load peaks transmitted by the crash partner's longitudinal member and is caused to collapse, **Figure 2**.

Reproducing this accident scenario in a crash test impressively demonstrates

Figure 1: Incompatibility in a real-world side crash (ADAC accident research)

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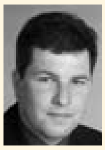

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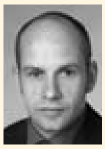

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Figure 2: Incompatibility in a real-world frontal crash (ADAC accident research)

Figure 3: Incompatibility frontal crash simulation

that the problem is repeatable under lab conditions. **Figure 3** suggests that the disparity in risk will be even greater for vehicles of different age. Even the occupants in large and heavy older family class vehicles will be exposed to very high loads with the occupant compartment collapsing and intrusions occurring in the footwell area. Like a spear, a new supermini's longitudinal member tears the splash wall in an old family car. This also translates into additional injuries for the occupants.

Preliminary investigation conducted by ADAC accident research shows that controlled energy absorption and a large interaction surface in new vehicles can minimise the additional risk of longitudinal members intruding into the crash opponent. Constructing vehicles with a focus on crashworthiness would be an important milestone in further improving the passive safety of vehicles.

2.2 Compatibility Principles

Vehicle compatibility is based on three factors:

- vehicle structure and geometry
- stiffness distribution in the deformation zone
- vehicle mass.

The vehicle size and weight are other decisive factors which cannot be materially influenced, since they vary based on the vehicle specifications. Front-end geometry, stiffness and structural configuration as well as the restraint systems used in a vehicle are variables which must be optimised so that the safety features of both vehicles involved in an accident activate efficiently. In addition, geometry and design of the deformation zone must ensure that energy is absorbed even in small overlap impacts.

Side impact performance is an additional challenge for today's vehicles. Since the vehicle sides offers much less deformation resistance than the front, ensuring crash-compatibility for optimum protection of both collision partners is a difficult task. This issue must be considered when designing a vehicle front which affords adequate partner protection.

The purpose of this crash test is to identify the difference between occupant protection and partner protection and to identify solutions. We aim at finding a possibility to test and assess the compatibility of today's vehicles in order to generate consumer information.

3 Compatibility Crash Test

3.1 Occupant Protection Structures as a Prerequisite for Partner Protection

The Euro NCAP crash test suggests that the Fiat 500 affords a very high level of occupant safety (five stars). To reduce the forces exerted on the occupants, the front structure features a centre and a lower load path forming a homogeneous deformation zone in combination with the front plate, **Figure 4**. The upper longitudinal frame rail is not attached to the

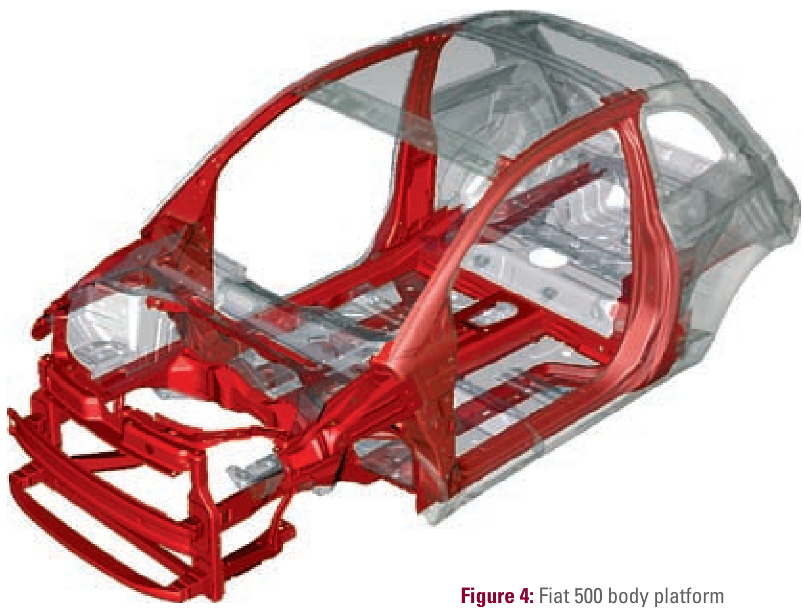

Figure 4: Fiat 500 body platform

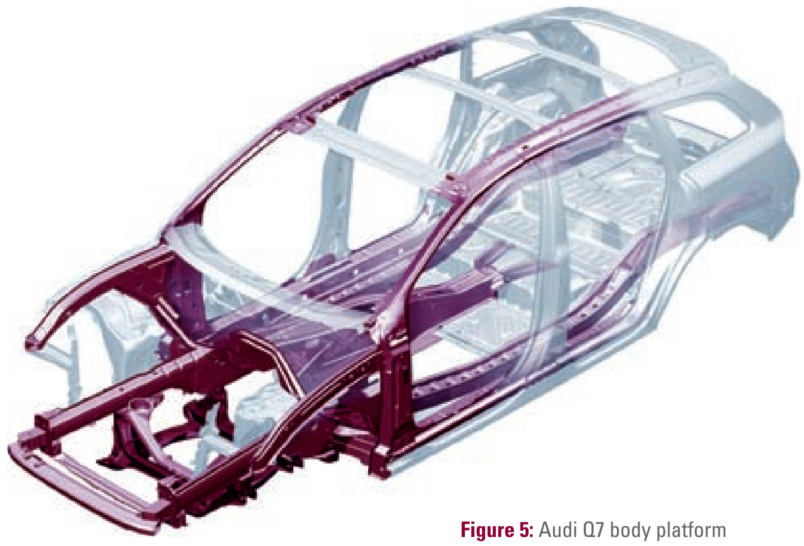

Figure 5: Audi Q7 body platform

front plate and only crumples in severer impacts or when underrunning an obstacle. Looking at the basic requirements for good occupant protection (e.g. stable passenger compartment) and partner protection (e.g. homogeneous front structure), the Fiat is well equipped for a compatibility crash.

Euro NCAP testing confirmed the Audi Q7's high self-protection level (four stars). Audi designed the front structure creating a load path over transverse members. The upper member is shorter and efficiently interacts only with larger collision partners, **Figure 5**. To prevent overriding smaller cars in a crash, a secondary energy absorbing structures are a customary solution. However, the frame beneath the Audi does not adequately engage with the structures of smaller vehicles. Since no front shield is used, the structural overlap with the crash opponent is small.

3.2 Test Procedure

Based on research investigating the compatibility of passenger cars of 1997 [1] we opted for a test setup with 50 % overlap and a test speed of approx. 56 kph for both vehicles. In this test configuration, the degree of overlap is measured at the smaller vehicle, **Figure 6**.

To analyse the injury risk, we used two 50 % male adult dummies (HIII) on the front seats and two child dummies in the second seat row representing children 1.5 and three years old. Dummy specifications, installation procedure and instrumentation were in compliance with Euro NCAP test requirements [2].

During the impact, there was little structural interaction between the Audi Q7 and the Fiat 500. With the transverse frame rails not being wide enough, the longitudinals fail to make contact. The Fiat's lower and centre longitudinals dissipate little energy, because they do not engage with the other vehicle's structures and cannot deform sufficiently. Only the Q7's front wheel offers a point where to dissipate energy. The Audi Q7's extremely stiff longitudinal engages with the cabin of the Fiat directly transfering nearly all of the crash energy. Given that the Q7's mass is about twice that of the supermini, the Fiat's occupant compartment is taken to its limits.

This compatibility crash reveals the added injury risk for the Fiat driver as the Audi's longitudinal member tears into the footwell of the Fiat like a spear threatening the driver's legs and feet, **Figure 7**.

The deformation pattern in the Audi Q7 is different. Stability of the passenger cell remains intact. Some lightweight structures around the longitudinal member deform during the impact transforming the longitudinal frame rail into a dangerous spear, **Figure 8**. The load applied by the Fiat to the area of the other car's left front wheel causes deformation of the Audi's footwell.

3.3 Occupant Protection Results

The criteria for evaluating the protection potential for vehicle passengers in the compatibility crash are in line with the Euro NCAP assessment protocol. This is based both on measurements and subjective criteria (modifiers) [3].

Frontal-crashed against the Audi Q7, the Fiat 500 demonstrates a very low

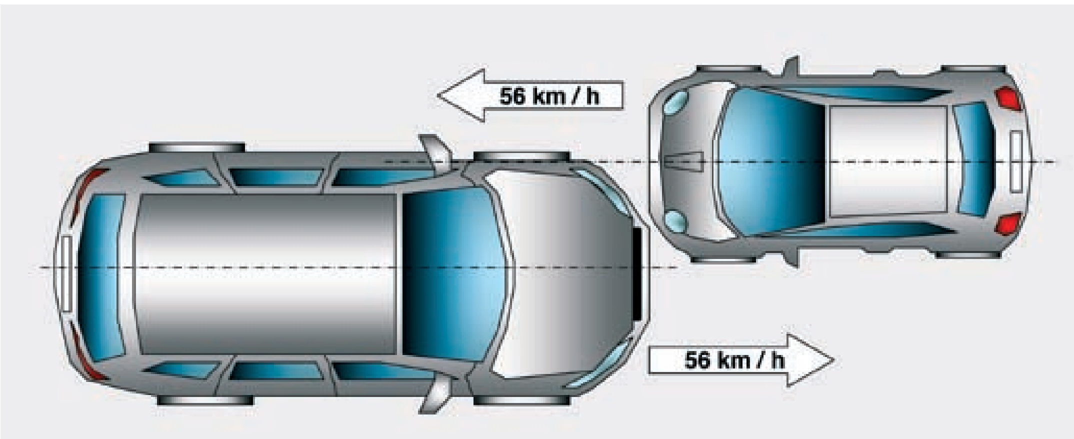

Figure 6: Test configuration

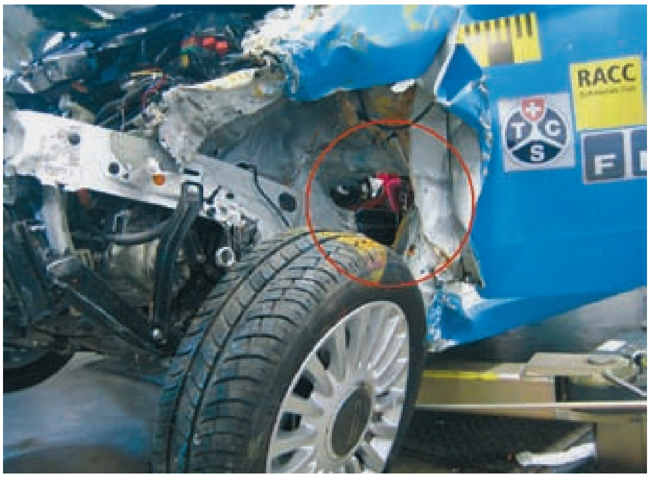

Figure 7: Hole in the Fiat's footwell

Figure 8: Audi Q7 longitudinal (visualised in red)

occupant protection potential, **Figure 9**, since the vehicles' front structures do not absorb enough energy. Because the deformation structures fail to activate and in view of the great mass disparity between the vehicles (factor 2.2), occupants in the Fiat endure a change in velocity of over 80 kph (pulse maximum at 50 g), **Figure 10**. While the stable safety cage ensures survival space for the driver, restraints such as head and knee airbags are simply overwhelmed. The Fiat's driver airbag cannot prevent the driver's head from hitting the A pillar and the driver's chest from colliding with the steering wheel. This seriously reduces the protection potential for the driver. Further, the neck, chest and leg forces measured in the driver dummy suggest that the restraint systems have reached their functional limits. The Audi's longitudinal intruding into the Fiat's splash wall poses an additional threat to the driver's footwell area of the Fiat.

The lack of partner protection in the Audi brings high forces to bear on the Fiat, which considering the high deceleration velocity severely affects especially the children on the Fiat's rear bench.

The Audi occupants still face much lower forces from the lighter collision partner as the deceleration velocity is only about 45 kph (pulse maximum at 30 g), **Figure 11**.

The results of this study demonstrate that the Fiat's good occupant protection as confirmed by the Euro NCAP results is severely reduced in a collision with an Audi Q7, while the Audi still has some potential.

3.4 Partner Protection Results

The test reveals that Fiat goes beyond current Euro NCAP requirements to make its vehicles safer. For the Fiat, the test means a speed which clearly exceeds the 64 kph required by Euro NCAP for cabin stability.

The Fiat 500 has a homogeneous front structure which is ideal for engagement with a collision partner, and a stable safety cage to ensure good occupant protection. However, the great mass disparity cancels the Fiat 500's occupant protection

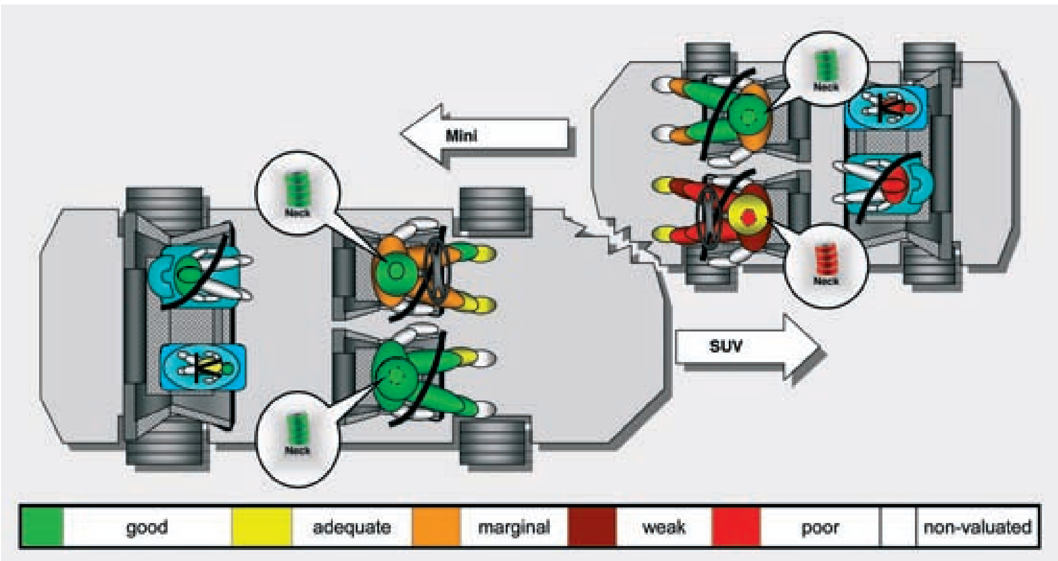

Figure 9: Protection potential in frontal impact Fiat 500 vs. Audi Q7

potential. Also, the safety margin is further reduced because the vertical mismatch of the vehicles' deformation elements prevents the dissipation of energy.

Improvements for occupants in the supermini can only be achieved by lowering force levels and designing a more homogeneous front structure for the larger vehicle. The Audi Q7 has no homogeneous front which would require several longitudinal and transverse members to ensure structural interaction with other vehicles.

4 Comparison of Car-to-car Collision and Euro NCAP Test

The chart below, **Figure 12** shows how occupant safety in a Euro NCAP crash test against a deformable barrier compares to the results of a car-to-car collision for the respective vehicle models. To be able to evaluate occupant protection data in both test configurations, the comparison is based on the points achieved by the front occupants under the Euro NCAP assessment protocol [3]. The chart contains the results of car-to-car collisions between VW Golf V vs. Kia Sorento and Volvo XC 90 (2005), and Audi Q7 vs. Fiat 500 (2008).

Based on the Euro NCAP barrier test, the Volvo XC 90 hardly loses any occupant protection points when crashed against a Golf V. Looking at the occupant values in the smaller collision partner, the level of occupant protection is already down by 60 %. The Audi Q7 vs. Fiat 500 collision is extreme with the smaller vehicle losing 94 % of its occupant protection potential compared to the Euro NCAP barrier test.

While occupant protection for larger vehicles is hardly any different in a car-to-car collision than in the Euro NCAP barrier crash, results are much worse for the smaller vehicles.

Both the Golf V and the Fiat 500 already boast very high safety levels under the Euro NCAP protocol for the frontal crash against a vehicle of the same class and weight. To establish the vehicles' level of partner protection, an additional test protocol is required, since the current frontal crash test does not generate any data about the compatibility of vehicles.

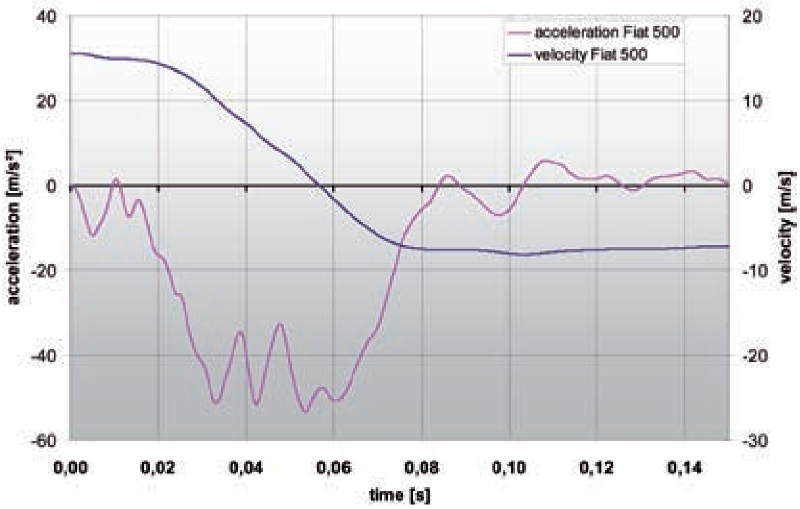

Figure 10: Deceleration velocity in the Fiat 500

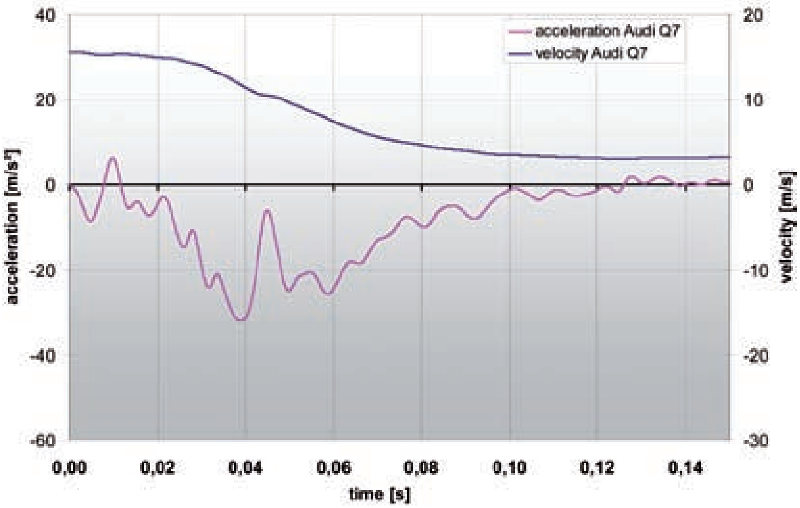

Figure 11: Deceleration velocity in the Audi Q7

5 Summary and Conclusions

Many real-world accidents demonstrate the incompatibility of vehicles in a collision. Being equipped with highly stiff longitudinal members, vehicle fronts can only absorb little deformation energy from the struck car. As a result, intrusion levels and biomechanical forces are high with the frame rails even ripping up holes in the collision partner's front and side areas.

Car buying trends, **Figure 13**, and changed accident scenarios additionally endorse the call for crash-compatible vehicles. With registrations of family cars on the decline in favour of supermini and large cars (e.g. SUV) deformation zones should be matched up in terms of geometry and structure.

ADAC's compatibility crash between a Fiat 500 for the supermini class and Audi Q7 representing SUVs confirmed the problems which also occur in real-world accidents.

Despite its good Euro NCAP crash test rating and homogeneous front structure, the supermini is pushed to its physical limits. Both children on the rear seat bench are exposed to a very high injury risk. In the Audi Q7, however, the injury risk for all occupants is relatively low.

Progressive stiffness in the impact zone of larger vehicles and structural

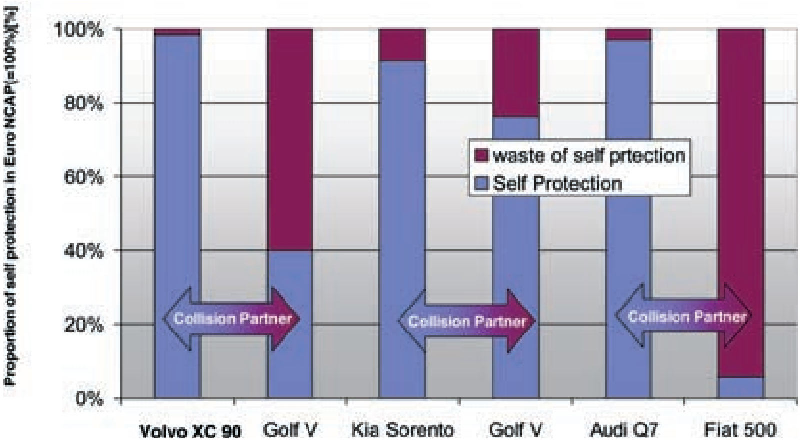

Figure 12: Occupant protection in the car-to-car collision

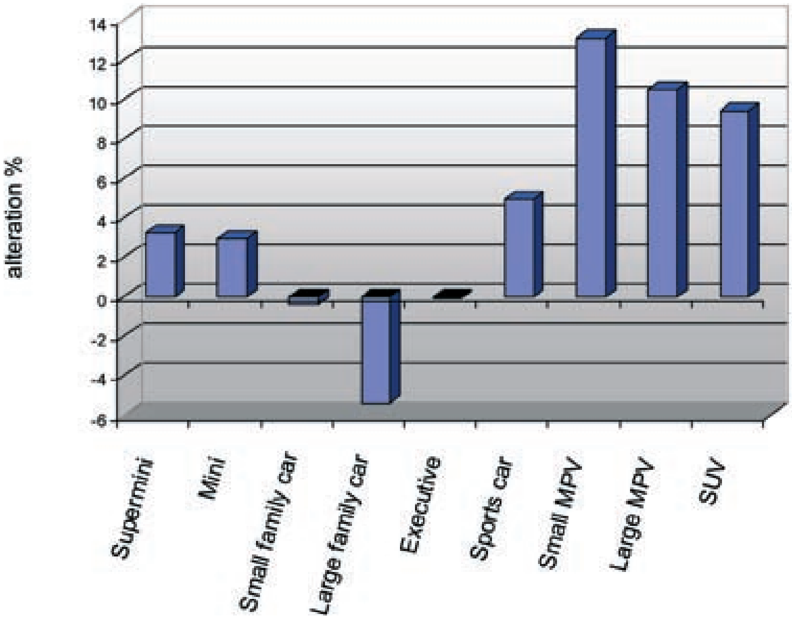

Figure 13: Alternation of motor vehicle registrations by vehicle type in Germany from 2006 to 2007

geometry are the key issues to consider when aiming to improve compatibility. This means that the geometry of all vehicles should be designed in a way that the front structures actually engage during a crash. Moreover, the structure should feature multiple load paths and transverse members for better managing the crash energy. The load path design should ensure high flexural rigidity of the transverse members to avoid local peak loads.

Front-end stiffness of heavy vehicles should be designed to preserve the occupant protection potential and not exceed the strength of the lighter vehicle's safety cage.

However, some basic considerations also apply to smaller and light vehicles. Their front-end stiffness should be sufficient to activate the front structure of the larger vehicle. The occupant department of the supermini should have sufficient integrity to force the frontal crush zone of the heavier vehicle to absorb crash energy. Moreover, the restraint systems on all seats in the smaller vehicle should be engineered to endure higher acceleration forces.

The above structural requirements for the vehicle front are significant for compatibility in a side crash as well in that they guarantee a large and even deformation zone. Structural stability in the area of the A, B and C pillars including door reinforcements will additionally enhance the integrity of the occupant department whilst activating the deformation zone of the striking vehicle.

In view of the known weaknesses in terms of compatibility it is the legislator's first and foremost task to make a partner protection test mandatory as a complement to the verification of occupant protection. Considerable research has been conducted to study crash compatibility, so that a number of solutions are available such as e.g. the results of the EEVC WG15 [4]. To improve compatibility in today's vehicle fleet, ADAC will continue its efforts in the framework of its consumer testing activities and Euro NCAP membership to promote the introduction of an additional test procedure for partner protection.

Vehicle manufacturers are challenged to commit themselves to enhancing vehicle-to-vehicle crash compatibility. This could be achieved by applying the constructive solutions found in the supermini class to large vehicles. These solutions do not require any increase in the vehicle mass, which would have added benefits in terms of CO_2.

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New Test Procedures for **Handling of Vehicles with Variable Ratio Steering Systems**

Vehicle handling tests have been standardised which makes it possible to compare vehicles for several items, such as understeer level. FEV uses some of these tests to establish scatter bands for all vehicle classes. The introduction of steering systems without a fixed ratio makes that the results of these tests are no longer valid for general vehicle comparison. Therefore a method was worked out which makes the procedures fit for future steering systems as well as the existing ones.

1 Introduction

Test procedures that describe objective tests for vehicle handling exist for more than 50 years. Many tests are developed by vehicle OEM's and several have been standardised by VDA, ISO and other organisations. The results from standardised tests generally provide no general rating or ranking but are used to quantify a certain behaviour and for comparison between vehicles. This is possible, because of the standardisation, the test conditions are the same for all tests performed according to this procedure. FEV, as a player on the market for engineering services in the field of chassis tuning, also uses a number of these standards to perform benchmark activities and for the establishment of scatter bands for vehicle handling.

With the introduction of variable steering systems there are several reasons why the test procedures and also the results obtained with them are no longer complete.

- Steering gears with mechanical variable ratio deliver different outcome for different steering angles
- Steering gears with speed dependant ratio deliver different outcome for different operation speeds.

The obtained results still give information about the perceived characteristic by the driver but no longer give information about the basic chassis characteristic of the vehicle.

In the investigation below this is further analysed for two important characteristics: understeer gradient and directness, derived from the following ISO tests:

- ISO 4138 Steady state circular behaviour [1]
- ISO 7401 Lateral transient response test methods [2]
- ISO 8726 Pseudo Random steering response test [3]

Proposals for improvement, as used by FEV, are given.

2 Steering Systems

In this chapter two steering systems will be discussed, both for front axle steering. The fact that four-wheel steering also affects the investigated characteristics is recognised but these systems are not part of the investigation.

2.1 Variable Rack Systems

Traditionally steering systems consist of a steering gear with a fixed ratio. Although variable rack solutions exist for many years, until recently the variation took place for larger steering angles, influencing the parking characteristic. Both, less direct and more direct solutions for larger steering angles are known, the first for lowering the forces that have to be applied by the driver or the assist system and the second for lowering the amount of turning, which can

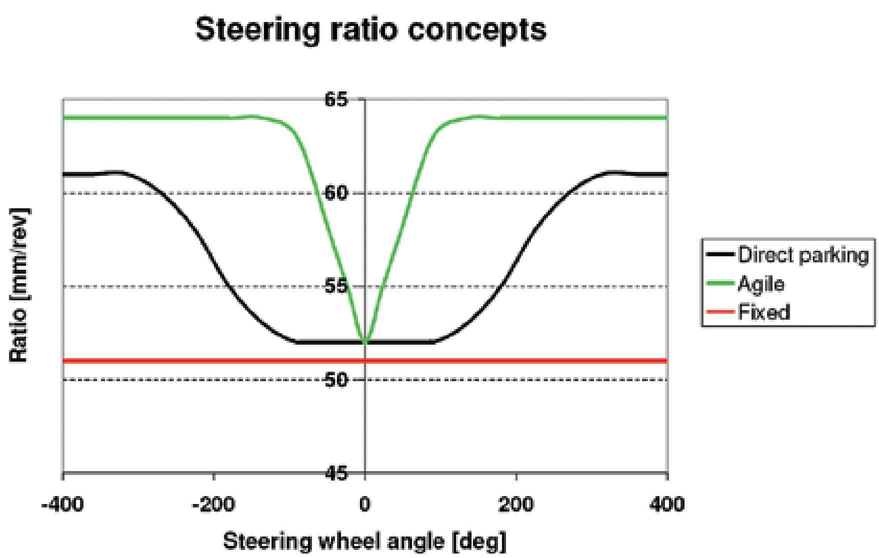

Figure 1: Variable rack concepts

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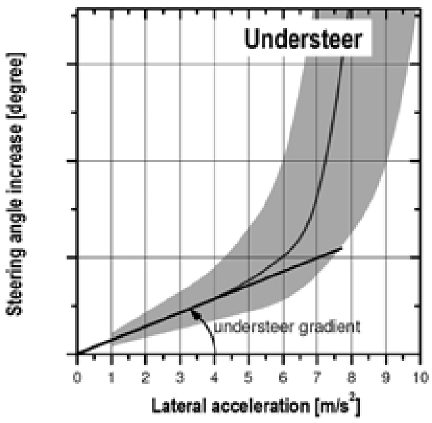

Figure 2: Understeer characteristic

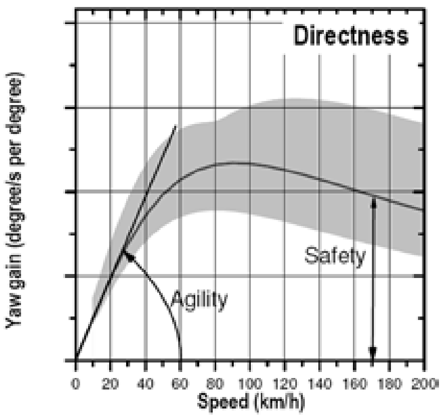

Figure 3: Typical yaw gain as function of speed

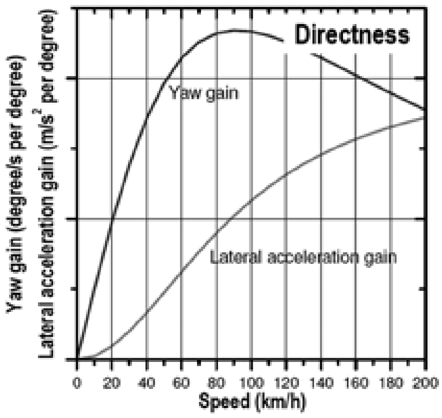

Figure 4: Yaw and lateral acceleration gain vs speed

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New are steering gears with a variable rack that provide more agility due to more direct steering ratio already after a few degrees of steering wheel rotation [Opel Corsa, Mercedes Direktlenkung], **Figure 1**.

These gears have in common that they influence the directness in the normal operation condition and therefore test results of the mentioned tests depending on the actual test conditions.

2.2 Active Front Steering (AFS)

This system basically decouples the steering wheel input of the driver from the turning of the wheels. Typically in normal driving conditions the system is used to create a speed dependant variable steering ratio where the directness is increased for low speed and decreased for high speeds [Toyota, BMW]. In this study only this functionality has been investigated. With this functionality it is clear that test results related to steering wheel angle are influenced by the vehicle speed.

3 Test Procedures and Important Characteristics

3.1 Steady State Circular Behaviour (Understeer Gradient)

The execution of this test is described in ISO 4138 and has the scope to describe handling properties during steady state cornering in the complete range of operation [1]. Basically the test is performed so that all combinations of vehicle speed, steering angle and cornering radius are covered. Since these are depending on each other, the test can be performed as constant radius test (which is commonly used), constant steering angle test or constant speed test, basically giving the same result.

From this test, several parameters and characteristics can be determined where the main focus is often on the understeer level of the vehicle. It is defined as the understeer gradient in degree/m/s² or degree/g. This gradient is positive for an understeering vehicle and negative for an oversteering vehicle. For a neutral vehicle it is zero and that means that for such a vehicle the circle can be driven with increasing speed without the need to increase the steering angle. It is also a parameter (or characteristic) that is often mentioned in publications (both scientific and popular magazines) since it gives information about the cornering power and the safety margin of a vehicle. The higher the understeer level, the lower the risk for over steer, which is considered dangerous for normal drivers.

Of course this is only an indication since the actual safety limit will be determined by the chassis tuning and tyre characteristics but it is generally considered to be a relevant parameter. At the same time a higher understeer level means a lower maximum cornering speed since the lateral tyre forces on front and rear axle become less balanced.

The driver will feel this understeer level on the steering wheel, when cornering forces increase, he has to put more steering angle. When reaching the limit he has to steer excessively and at the same time he will feel the front of car floating out of the corner.

Figure 2 shows the measured characteristic of a typical mid-size sedan it the scatter band that is obtained from measurements of a wide range of passenger cars.

The tests have been performed as constant radius test. This means the vehicle is driven at low speed on a circle with constant radius and the speed is slowly increased. In typical –understeer-vehicles this means that when speed builds up and lateral acceleration increases, the steering angle has to be increased. The amount of steering is an indicator for the understeer level. Up to 4 m/s² there is a linear behaviour (understeer gradient), followed by a progressive understeer ending at the limit of adhesion.

The test procedure for constant radius tests gives a standard for the radius of 100 m but allows the usage of smaller radii, down to 30 m. The interesting thing is that the result as plotted in Figure 2 is independent of the curve radius and therefore test results from different radii can directly be compared to each other. Unfortunately this is no longer valid when steering ratios are not fixed, as will be shown in chapter 4.

3.2 Lateral Transient Behaviour (Directness)

The execution of this test is described in ISO 7401 [2] and also in ISO/TR 8726 [3]. The pseudo random test [3] is often used since it can be performed very well without using a steering robot. The scope of this test is to describe the transient behaviour of a vehicle while steering from straight ahead driving. This test results in parameters describing the steering behaviour in the linear range of operation (up to ≈ 4 m/s²) in the time or frequency domain. Many parameters can be obtained but often the focus is on an aspect also described as directness. The definition of directness is the amount of rotation per steering angle input (yaw rate gain in degree/s per degree) for 0 Hz. This parameter can be determined for different speeds, resulting in a diagram where yaw gain is plotted as a function of speed. **Figure 3** shows an example of a typical passenger car and the scatter band from a wide range of vehicles.

In this graph the first gradient is a rating for the manoeuvrability and agility of the vehicle. If there would be no understeer in the vehicle, this gradient would remain constant, resulting in a very high yaw gain at high speed and this is not desired for normal passenger cars. The digression and decrease of the yaw rate gain over speed is a function of the understeer level and this means the steering ratio and wheel suspension need to be tuned to deliver a characteristic that fits to the vehicle both in low and high speed.

To better describe directness, a second characteristic is introduced, lateral acceleration gain. Normally this characteristic is not used or shown in publications since it is (was) completely based on the yaw gain (lateral acceleration$_{\text{steady-state}}$ = yawrate$_{\text{steady-state}}$*vehicle speed).

The two characteristics together describe what the driver feels when he is turning the steering wheel. At speeds up to 100 km/h the turning (yaw gain) will be mostly responsible for the response feeling where at high speeds the lateral acceleration gain becomes more important, **Figure 4**.

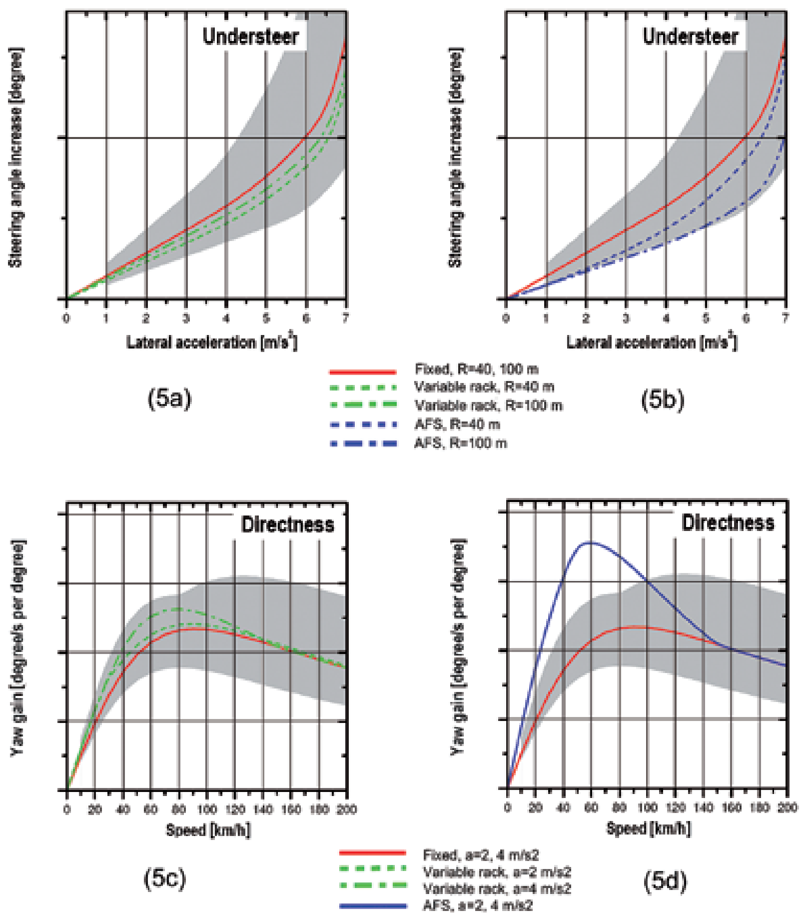

Figure 5: Effect of variable steering gears on understeer gradient

4 Effects of Variable Steering Gears

4.1 Mechanical Variable Ratio

4.1.1 Understeer Gradient

Typical steering angles for the circle test range from 20 degrees on 100 m radius to 100 degrees on 40 m radius. This means it is completely in the range where the ratio of variable rack systems is changing, Figure 1.

Figure 5 (a) shows simulation results of a vehicle with standard and variable rack on 2 different radii.

In this graph, it is clear that the variable rack results in less understeer because of the more direct steering in this condition and that the effect becomes stronger for a smaller circle radius. Also the result is depending on the test condition (circle radius). Due to this fact, comparisons between vehicles with variable ratio measured on different radii are not valid.

4.1.2 Directness

For directness, tests are performed typically at a fixed steering angle that relates to a constant lateral acceleration. From Figure 4 (lateral acceleration gain) it can be derived that with increasing speed, a smaller steering angle relates to this constant acceleration level. Since the ratio is depending on actual steering angle, the ratio will become less direct for higher speeds.

Where normally the result is independent on the chosen lateral acceleration (between 1 and 4 m/s²), with these variable steering gears the result is depending on the amplitude. Figure 5 (c) shows simulation results of the vehicle with fixed ratio and variable ratio but now for two different amplitude levels, related to 2 and 4 m/s². The first conclusion from this graph is that with the variable rack, the goal is achieved to increase the agility while maintaining the safety at high speed.

The second thing however is the fact that the result for the variable ratio in the mid-speed range is depending on the amplitude. From a driver's point of view this is a good solution since agility normally is required in a more sporty driving style with lateral accelerations above 3 m/s². From an objective measurement point of view however it means that different results can be obtained with the same test procedure.

4.2 AFS Speed Dependant Ratio

4.2.1 Understeer Gradient

In the circle test the results are influenced by the fact that on circles with larger radii the speed range is higher and therefore the ratio is less direct. This means that even for a neutral vehicle the steering angle should be increased and the larger the radius, the more increase. This increase feels like an understeer tendency and this is also what the measurement looks like. In Figure 5 (b) a plot of simulation results shows how the steering angle increase is depending on the circle radius.

4.2.2 Directness

Assuming only speed dependant ratio, no influence of amplitude is expected. The directness measured in the traditional way will give good results, both for 2 and 4 m/s². But due to the high degree of freedom of this steering system tuning, the results will no longer fit in the existing scatter band based on fixed ratio steering gears, Figure 5 (d).

5 Implication for the Discussed Test Procedures

5.1 Circle Test

The weak point in the circle test is the fact that the circle radius is not fixed. One solution could be to agree to always use the recommended 100 m but this has two disadvantages:

- Not all test tracks allow this radius.
- It only describes the behaviour for one radius.

A second solution would be to use another method, for instance the constant speed test. This however suffers from the same problem with speed dependant ratio.

The solution proposed and introduced by FEV is to perform the test for a number of radii.

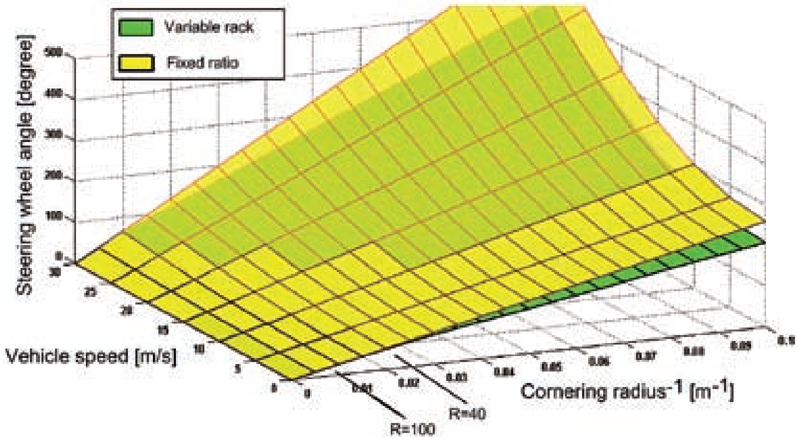

Figure 6: 3D surface for understeer behaviour

To make a complete characterisation of the steering during steady state cornering, the complete area of possible (or normally used) speeds, lateral accelerations and circle radii should be measured and shown as a 3D surface as shown in **Figure 6**. The effort for this procedure however is very high.

A good compromise between cost and benefit offers the circular-course driving for two radii (40 and 100 m). An additional proposal is to decouple the actual understeer gradient at the steering wheel from that of the vehicle handling behaviour.

The decoupling can be achieved by measuring both the steering wheel and road wheel angle. Measuring road wheel angle however is not easy and also it is taking into account the kinematic and elasto-kinematic effects that in the end determine part of the understeer level. Therefore it was chosen to measure the displacement of the steering rack as second parameter beside the steering wheel angle.

With the known relation between rack displacement and wheel rotation, we derive a real understeer characteristic of the vehicle as if it had a fixed steering ratio.

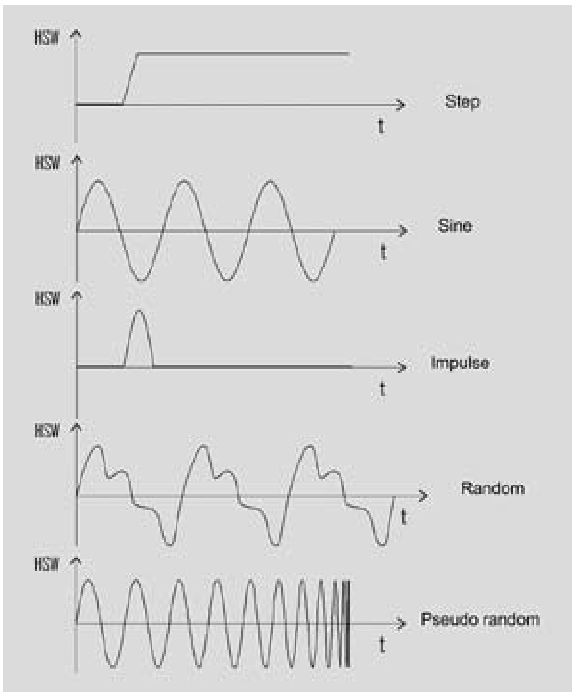

Figure 7: Input signals for determining directness

For comparison, measurements from fixed ratio steering gears can be reworked if the ratio is known. Results from other radii should be valid also for 40 and 100 m.

5.2 Lateral Transient Behaviour

The weak points in these procedures are the assumption that the behaviour is linear in the range where the tyre characteristic is linear (up to ≈ 4 m/s²). A solution would be to perform the test for the complete range of steering wheel angles but this again leads to an enormous amount of testing.

A compromise is to pick two important levels, corresponding to a steady state lateral acceleration of 2 and 4 m/s². For the determination of the directness characteristic, several steering inputs are allowed, **Figure 7**.

For each test the new approach is given:

- step steer: both 2 and 4 m/s² should be measured
- sine input: both 2 and 4 m/s² should be measured
- pulse input: less suitable for non-linear systems
- random input: less suitable for non-linear systems
- pseudo random input: both 2 and 4 m/s² should be measured.

6 Conclusions and Outlook

The examples of understeer gradient and directness show that current test methods do not allow a good comparison for vehicles with variable steering ratio. With the described changes to the test procedures these vehicles can be completely described and also be compared with other vehicles.

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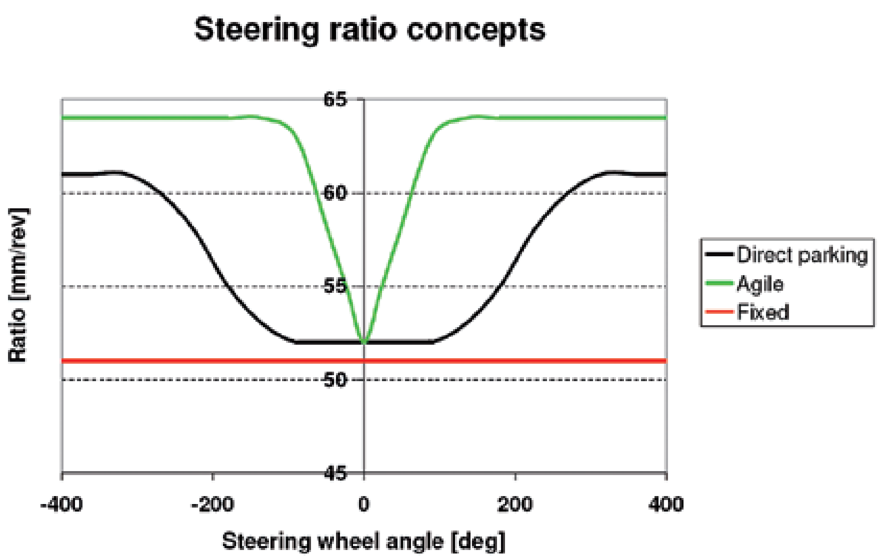

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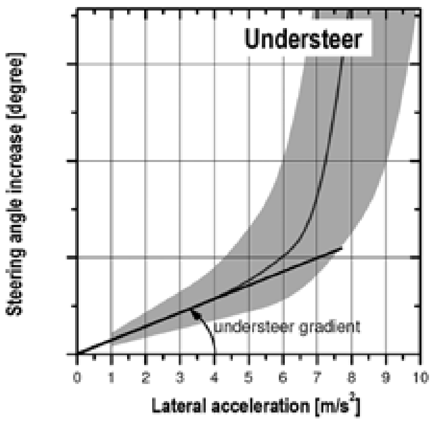

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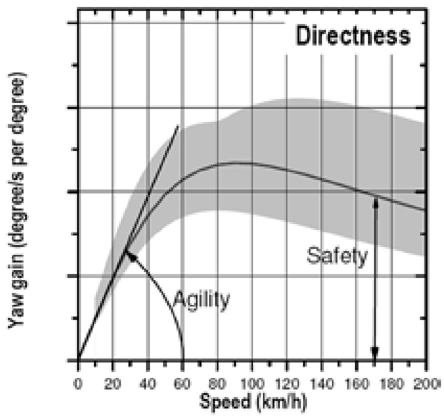

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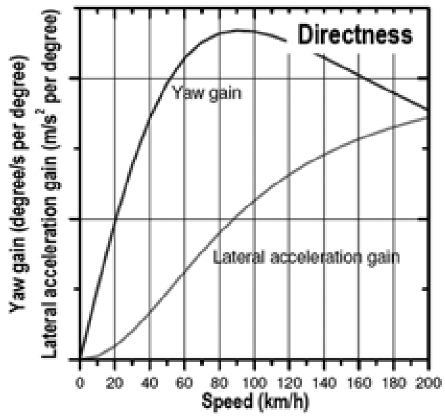

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3 Test Procedures and Important Characteristics

3.1 Steady State Circular Behaviour (Understeer Gradient)

The execution of this test is described in ISO 4138 and has the scope to describe handling properties during steady state cornering in the complete range of operation [1]. Basically the test is performed so that all combinations of vehicle speed, steering angle and cornering radius are covered. Since these are depending on each other, the test can be performed as constant radius test (which is commonly used), constant steering angle test or constant speed test, basically giving the same result.

From this test, several parameters and characteristics can be determined where the main focus is often on the understeer level of the vehicle. It is defined as the understeer gradient in degree/m/s^2 or degree/g. This gradient is positive for an understeering vehicle and negative for an oversteering vehicle. For a neutral vehicle it is zero and that means that for such a vehicle the circle can be driven with increasing speed without the need to increase the steering angle. It is also a parameter (or characteristic) that is often mentioned in publications (both scientific and popular magazines) since it gives information about the cornering power and the safety margin of a vehicle. The higher the understeer level, the lower the risk for over steer, which is considered dangerous for normal drivers.

Of course this is only an indication since the actual safety limit will be determined by the chassis tuning and tyre characteristics but it is generally considered to be a relevant parameter. At the same time a higher understeer level means a lower maximum cornering speed since the lateral tyre forces on front and rear axle become less balanced.

The driver will feel this understeer level on the steering wheel, when cornering forces increase, he has to put more steering angle. When reaching the limit he has to steer excessively and at the same time he will feel the front of car floating out of the corner.

Figure 2 shows the measured characteristic of a typical mid-size sedan it the scatter band that is obtained from measurements of a wide range of passenger cars.

The tests have been performed as constant radius test. This means the vehicle is driven at low speed on a circle with constant radius and the speed is slowly increased. In typical –understeer-vehicles this means that when speed builds up and lateral acceleration increases, the steering angle has to be increased. The amount of steering is an indicator for the understeer level. Up to 4 m/s^2 there is a linear behaviour (understeer gradient), followed by a progressive understeer ending at the limit of adhesion.

The test procedure for constant radius tests gives a standard for the radius of 100 m but allows the usage of smaller radii, down to 30 m. The interesting thing is that the result as plotted in Figure 2 is independent of the curve radius and therefore test results from different radii can directly be compared to each other. Unfortunately this is no longer valid when steering ratios are not fixed, as will be shown in chapter 4.

3.2 Lateral Transient Behaviour (Directness)

The execution of this test is described in ISO 7401 [2] and also in ISO/TR 8726 [3]. The pseudo random test [3] is often used since it can be performed very well without using a steering robot. The scope of this test is to describe the transient behaviour of a vehicle while steering from straight ahead driving. This test results in parameters describing the steering behaviour in the linear range of operation (up to ≈ 4 m/s²) in the time or frequency domain. Many parameters can be obtained but often the focus is on an aspect also described as directness. The definition of directness is the amount of rotation per steering angle input (yaw rate gain in degree/s per degree) for 0 Hz. This parameter can be determined for different speeds, resulting in a diagram where yaw gain is plotted as a function of speed. **Figure 3** shows an example of a typical passenger car and the scatter band from a wide range of vehicles.

In this graph the first gradient is a rating for the manoeuvrability and agility of the vehicle. If there would be no understeer in the vehicle, this gradient would remain constant, resulting in a very high yaw gain at high speed and this is not desired for normal passenger cars. The digression and decrease of the yaw rate gain over speed is a function of the understeer level and this means the steering ratio and wheel suspension need to be tuned to deliver a characteristic that fits to the vehicle both in low and high speed.

To better describe directness, a second characteristic is introduced, lateral acceleration gain. Normally this characteristic is not used or shown in publications since it is (was) completely based on the yaw gain (lateral acceleration$_{\text{steady-state}}$ = yawrate$_{\text{steady-state}}$ *vehicle speed).

The two characteristics together describe what the driver feels when he is turning the steering wheel. At speeds up to 100 km/h the turning (yaw gain) will be mostly responsible for the response feeling where at high speeds the lateral acceleration gain becomes more important, **Figure 4**.

4 Effects of Variable Steering Gears

4.1 Mechanical Variable Ratio

4.1.1 Understeer Gradient

Typical steering angles for the circle test range from 20 degrees on 100 m radius to 100 degrees on 40 m radius. This means it is completely in the range where the ratio of variable rack systems is changing, Figure 1.

Figure 5 (a) shows simulation results of a vehicle with standard and variable rack on 2 different radii.

In this graph, it is clear that the variable rack results in less understeer because of the more direct steering in this condition and that the effect becomes stronger for a smaller circle radius. Also the result is depending on the test condition (circle radius). Due to this fact, comparisons between vehicles with variable ratio measured on different radii are not valid.

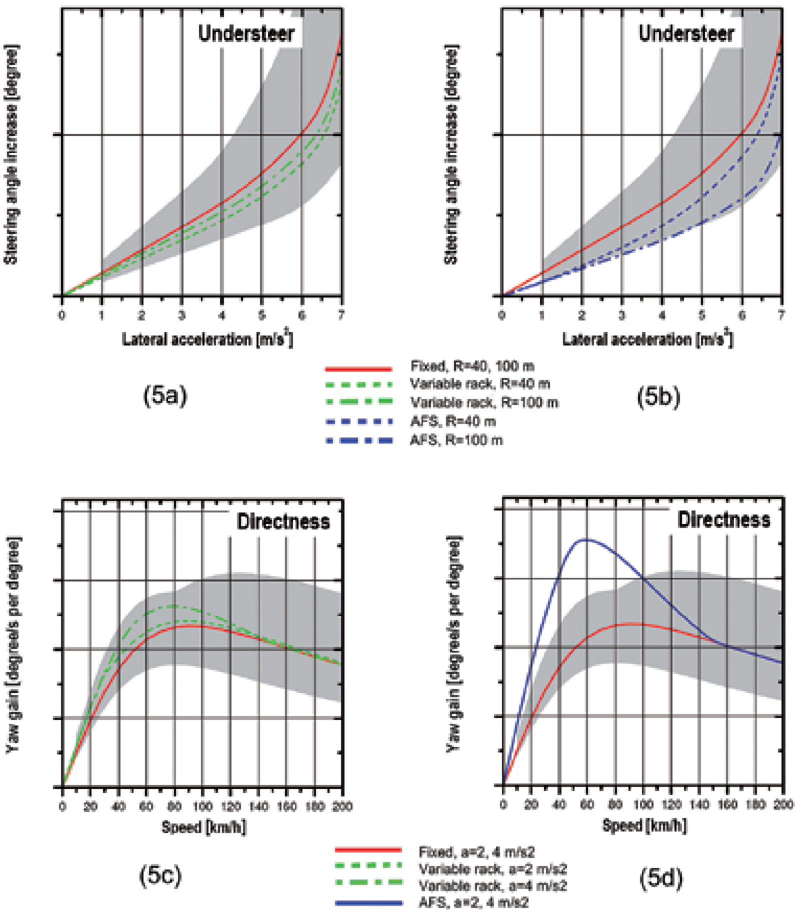

Figure 5: Effect of variable steering gears on understeer gradient

4.1.2 Directness

For directness, tests are performed typically at a fixed steering angle that relates to a constant lateral acceleration. From Figure 4 (lateral acceleration gain) it can be derived that with increasing speed, a smaller steering angle relates to this constant acceleration level. Since the ratio is depending on actual steering angle, the ratio will become less direct for higher speeds.

Where normally the result is independent on the chosen lateral acceleration (between 1 and 4 m/s²), with these variable steering gears the result is depending on the amplitude. Figure 5 (c) shows simulation results of the vehicle with fixed ratio and variable ratio but now for two different amplitude levels, related to 2 and 4 m/s². The first conclusion from this graph is that with the variable rack, the goal is achieved to increase the agility while maintaining the safety at high speed.

The second thing however is the fact that the result for the variable ratio in the mid-speed range is depending on the amplitude. From a driver's point of view this is a good solution since agility normally is required in a more sporty driving style with lateral accelerations above 3 m/s². From an objective measurement point of view however it means that different results can be obtained with the same test procedure.

4.2 AFS Speed Dependant Ratio

4.2.1 Understeer Gradient

In the circle test the results are influenced by the fact that on circles with larger radii the speed range is higher and therefore the ratio is less direct. This means that even for a neutral vehicle the steering angle should be increased and the larger the radius, the more increase. This increase feels like an understeer tendency and this is also what the measurement looks like. In Figure 5 (b) a plot of simulation results shows how the steering angle increase is depending on the circle radius.

4.2.2 Directness

Assuming only speed dependant ratio, no influence of amplitude is expected. The directness measured in the traditional way will give good results, both for 2 and 4 m/s². But due to the high degree of freedom of this steering system tuning, the results will no longer fit in the existing scatter band based on fixed ratio steering gears, Figure 5 (d).

5 Implication for the Discussed Test Procedures

5.1 Circle Test

The weak point in the circle test is the fact that the circle radius is not fixed. One solution could be to agree to always use the recommended 100 m but this has two disadvantages:

- Not all test tracks allow this radius.
- It only describes the behaviour for one radius.

A second solution would be to use another method, for instance the constant speed test. This however suffers from the same problem with speed dependant ratio.

The solution proposed and introduced by FEV is to perform the test for a number of radii.

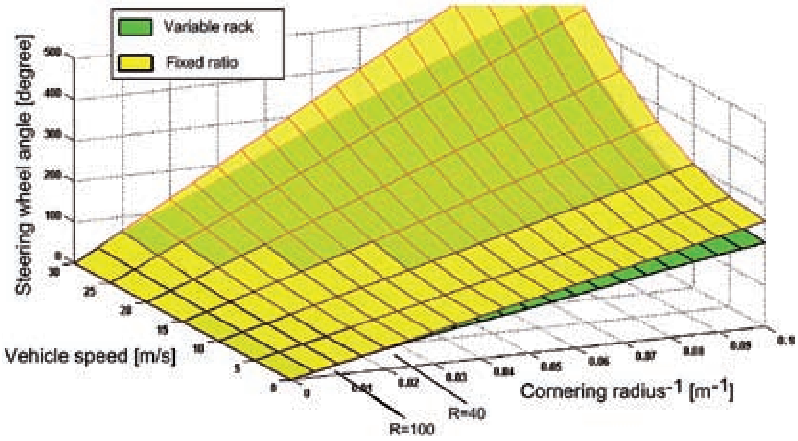

Figure 6: 3D surface for understeer behaviour

To make a complete characterisation of the steering during steady state cornering, the complete area of possible (or normally used) speeds, lateral accelerations and circle radii should be measured and shown as a 3D surface as shown in **Figure 6**. The effort for this procedure however is very high.

A good compromise between cost and benefit offers the circular-course driving for two radii (40 and 100 m). An additional proposal is to decouple the actual understeer gradient at the steering wheel from that of the vehicle handling behaviour.

The decoupling can be achieved by measuring both the steering wheel and road wheel angle. Measuring road wheel angle however is not easy and also it is taking into account the kinematic and elasto-kinematic effects that in the end determine part of the understeer level. Therefore it was chosen to measure the displacement of the steering rack as second parameter beside the steering wheel angle.

With the known relation between rack displacement and wheel rotation, we derive a real understeer characteristic of the vehicle as if it had a fixed steering ratio.

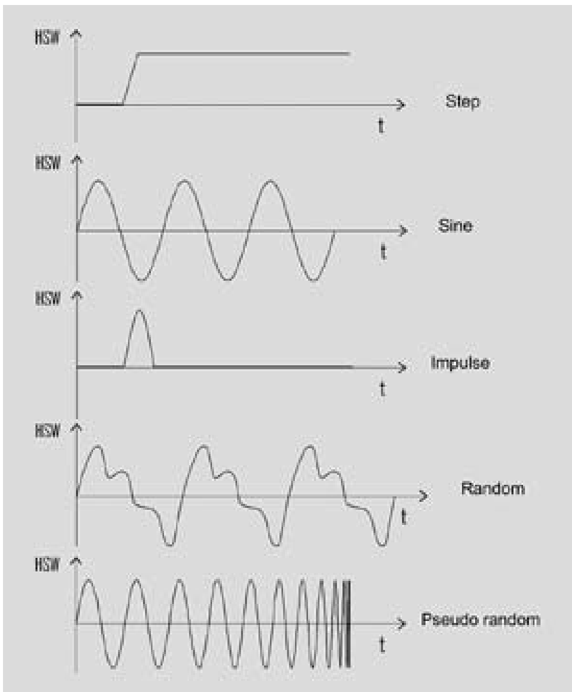

Figure 7: Input signals for determining directness

For comparison, measurements from fixed ratio steering gears can be reworked if the ratio is known. Results from other radii should be valid also for 40 and 100 m.

5.2 Lateral Transient Behaviour

The weak points in these procedures are the assumption that the behaviour is linear in the range where the tyre characteristic is linear (up to ≈ 4 m/s^2). A solution would be to perform the test for the complete range of steering wheel angles but this again leads to an enormous amount of testing.

A compromise is to pick two important levels, corresponding to a steady state lateral acceleration of 2 and 4 m/s^2. For the determination of the directness characteristic, several steering inputs are allowed, **Figure 7**.

For each test the new approach is given:

- step steer: both 2 and 4 m/s^2 should be measured
- sine input: both 2 and 4 m/s^2 should be measured
- pulse input: less suitable for non-linear systems
- random input: less suitable for non-linear systems
- pseudo random input: both 2 and 4 m/s^2 should be measured.

6 Conclusions and Outlook

The examples of understeer gradient and directness show that current test methods do not allow a good comparison for vehicles with variable steering ratio. With the described changes to the test procedures these vehicles can be completely described and also be compared with other vehicles.

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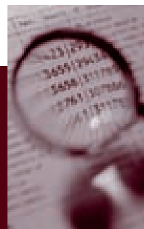

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Discomfort Glare of Tungsten Halogen and High Intensity Discharge Headlamps

For the evaluation of headlamps in everyday traffic situations, discomfort glare is an important aspect in addition to quality of illumination. The Laboratory of Lighting Technology at the Technische Universität Darmstadt has been doing research on discomfort glare evaluation for several decades. Now the impact of light source, headlamp optics and spectrum of adaptation field on discomfort glare in real traffic situations was tested for the first time within one test series for tungsten halogen and high intensity discharge (so-called HID or Xenon) headlamps. Moreover, the measured results are compared with calculated values, based on a spectral sensitivity function for discomfort glare.

1 Introduction and Background

Accident research of the past years shows a distinct pattern: only 25 % of all vehicle kilometers are driven at night, while 50 % of all traffic accidents are occurring within this time. The risk of motor vehicle accidents resulting in death increases by factor two up to three when comparing night to day ratio [1, 2, 3, 4]. Fatigue and a loss in visibility are the main aspects for it. Reaching or even exceeding the capability of the human eye causes the driver to drive with more mistakes. Therefore visibility should be optimized during nighttime traffic.

Recognizability can be improved by wearing bright clothes [5] and using retro reflective markings. Illumination is a more important factor, because visibility is increased and the number of accidents is reduced about 30 % on average [6]. 30 % of today's street lighting was built in the 1960s, showing an annual renewal of only 3 % and is not state of the art [7, 8]. Automotive headlamps are important for traffic illumination, because the most serious accidents take place out of town, where no street lighting is available. While only 25 % of all accidents occur out of town, statistics show more than 60 % of accidents with fatalities on country roads [9, 10]. The heaviness of accidents during darkness increases by factor 2.4 in towns and factor 3 on country roads, compared to daytime conditions [4]. Headlamp illumination highly contributes to traffic safety.

2 Illumination and Safety

High intensity discharge (HID) lamps have been well known in general lighting for more than six decades. First ideas to use this technology also for automotive lighting were developed in the 1930s [11]. In 1991 the first serial production car was launched with so-called Xenon headlamps [12]. The improved visibility of the driver due to the higher luminous flux available for HID lamps [13] is a milestone in automotive front lighting compared to standard tungsten halogen lamps until today [14]. Equipping all cars in Germany with HID instead of tungsten halogen headlamps, heavy nighttime traffic accidents would be reduced by 50 % on country roads and by more than 30 % on highways. In result, there would be six percent less accidents with casualties and 18 percent less accidents with fatalities [18].

It was shown in different international studies that drivers prefer HID headlamps [15, 16, 17]. HID headlamps have a higher blue part in their spectral power distribution. Peripheral visibility is improved, reaction times are shorter, obstacles are easier to detect and, in consequence, traffic safety can be improved [19, 20]. Since the beginning of automotive front lighting, there has been a tradeoff between optimizing the driver's visibility and minimizing the glare for oncoming traffic. For low beam, which is used in nearly 90 % [12] when headlamps are switched on, in the US, visibility for the driver has priority. In Europe and the rest of the world glare reduction is more important [21, 22, 23]. Today's HID headlamps produce higher luminance than tungsten halogen headlamps. Reduced reflector and lens diameters, new design specifications and the unfamiliar color temperature of HID headlamps are heating up the glare discussion from the early 1990s until today [24]. Today, 10 % of all cars in Germany are equipped with HID headlamps [25], with an upward trend for the future. Since a couple of years, committees and administrative bodies have been receiving more and more complaints about glare [26]. In the US, the National Highway Traffic Safety Administration (NHTSA) started a request for comments on glare in the year 2001. Within a short period of time, about 1800 comments have been filed from drivers [27]. While some drivers prefer HID headlamps, because the advantages are well known, other drivers would like to ban this technology in automotive headlamps in general. Research on glare of headlamps was established in the 1920s [29] and is not less important today.

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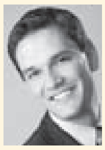
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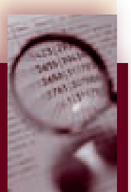

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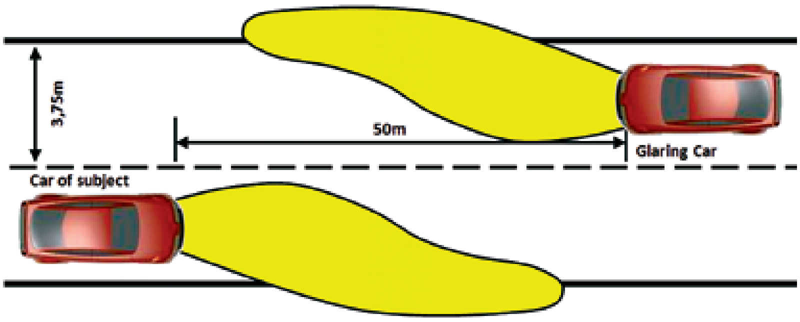

Figure 1: Test setup for discomfort glare evaluation

3 Headlamp Glare – A Short Overview

In general two different types of glare are defined. Disability glare describes a reduction of visibility, which can be measured [23], while discomfort glare causes the observer to feel uncomfortable and is distinctly different from disability glare. Disability glare is based on the physiology of the human eye and has been fairly well defined, for the first time in 1927 [28]. Light entering the ocular media is scattered and produces veiling light on the retina. Contrast sensitivity is reduced and obstacles are harder to recognize [29]. In the US, the calculation of a corresponding veiling luminance is used [28, 30]. The evaluation of the threshold increment (TI) for describing disability glare is more common in many other countries. The TI value represents how much brighter an object has to be under glare conditions compared to visibility without a glare source present. It is possible to convert from TI to veiling luminance and the other way round [31].

Discomfort glare is more complex, harder to describe and less defined. The emotional state of a person's response to discomfort glare seems to have more influence than the light source itself [32]. Moreover, discomfort glare is a subjective feeling and is different between subjects. Discomfort glare ratings of subjects can be calculated [33], based upon a common psychometrical 9-step-scale [37]. This so called DeBoer scale was also used for the research described in this paper. The impact of headlamp glare on traffic accidents is not completely known until today. Some research shows that glare can influence driving behavior [34, 35, 36] but is not causal for accidents [38]. Other results show a slight tendency that 1 % of all nighttime accidents are caused at least partly by glare [39].

Important results in glare research are:

- Illuminance at the eye is the most important factor for glare evaluation [29, 40, 41].
- The spectral power distribution has no considerable impact on disability glare, but there is an impact on discomfort glare; HID headlamps are causing more discomfort glare than tungsten halogen headlamps [16, 40, 42, 43].
- The influence of the glare source size on discomfort glare in some previous research is varying between "no impact" [40, 42] and "significant" [41].
- The S/P-ratio and sensitivity of the rods is not suitable for describing discomfort glare. The sensitivity of the blue cones is a better descriptor [44].

Discomfort glare becomes more important in today's research while disability glare is well known already. Most of recent research on discomfort glare was done under laboratory conditions. In addition, many studies used constant illuminance at the subject's eye as a parameter [42, 44]. This research is an important basis but the impact of the results for every day traffic situations is hard to define [29].

Table 1: Overview of tested headlamps (light source and optics)

Car of subject (Adaptation)	Glaring car		
Halogen H7 (Reflexion)	HID D1 (Projection)	Halogen H7 (Projection)	Halogen H7 (Reflexion)
HID D1 (Projection)	Xenon D1 (Projection)	Halogen H7 (Projection)	Halogen H7 (Reflexion)

4 Test Setup and Method

For discomfort glare evaluation in real traffic situations, cars of the same type but with different headlamps were used in field tests. All tests were performed on a separated test field with conditions comparable to German country roads. **Figure 1** shows the test setup for discomfort glare evaluation in bird's eye view. The same method was used as described in [14]. The impact of the glare source spectrum, the subject's adaptation field (headlamp spectrum of subject's car), and the headlamp optics on discomfort glare were test parameters.

A subject took place at the driver's seat and the adaptation process started while the headlamps of the subject's car were switched on. After several minutes of adaptation, the headlamps of the glaring car in a distance of 50 m on the opposite lane ("worst case" [45, 46]) were uncovered. While the subject was looking straight ahead of his own car and not into the glaring headlamps, the feeling of discomfort glare was evaluated. After a period of 30 seconds, the glaring headlamps were covered again. Glare illuminance was measured at the subject's eye at B50L point. Discomfort glare sensation of each subject was documented by using a questionnaire with a 9-step DeBoer scale [37]. The test was repeated for 15 subjects (20 up to 55 years of age), three glaring cars, and two adaptation cars (car of subject). All subjects were owners of a German driver's license. All tested cars, headlamps and headlamp optics are given in **Table 1**. The cutoff line was adjusted correctly according ECE regulations for all tested cars, all headlamps and wind screens were cleaned. The tests were performed in three different nights with comparable weather situations having some clouds and dry conditions.

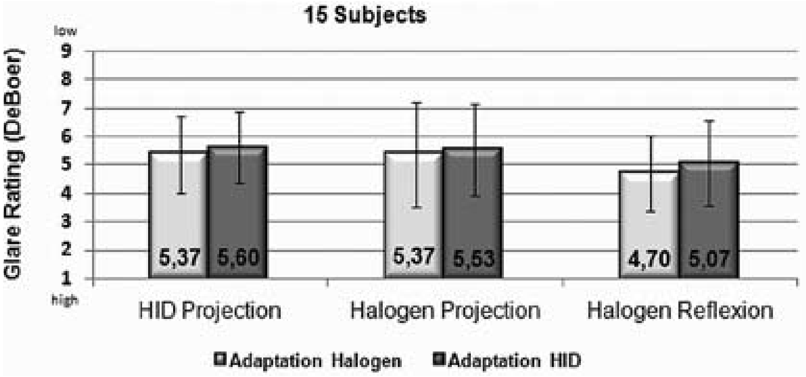

Figure 2: Discomfort glare evaluation for all tested variations

5 Results and Discussion

Figure 2 shows the mean values of discomfort glare evaluation and the standard deviations for all subjects. For each type of headlamp the DeBoer scale ratings are given for the two adaptation spectra. Lower rating values represent a higher feeling of discomfort glare.

Comparing the results for both adaptation spectra among each other, it can be seen that the feeling of discomfort glare is about 0.25 scale points lower to average for HID adaptation. But the spectrum of adaptation has no significant impact when regarding the calculated standard deviations. As a consequence, there can be shown no real effect on the feeling of discomfort glare between driving a tungsten halogen or HID headlamp car (ANOVA: $p = 0.42$). The influence of the headlamp optics on discomfort glare shows a distinct pattern. Both projection systems have nearly the same results independent of the spectrum of adaptation. But the reflexion system shows a glare rating which is about 0.5 point higher and therefore more glaring (Anova: $p = 0.08$). Both HID projection and Halogen projection headlamps show nearly identical glare ratings for comparable glare illuminance values, **Table 2**. The light source of the headlamp therefore has no significant impact on the feeling of discomfort glare under the tested conditions. Former research [16, 40, 42, 43] cannot be proved, that HID headlamps produce more glare than tungsten halogen lamps. A main reason for this finding is the higher glare illuminance coming from the tungsten halogen headlamp in the performed field tests.

Subject's age is a factor, which clearly influences the discomfort glare ratings. Higher age corresponds to a higher feeling of discomfort glare for the tested subjects. Glare illuminance is a second important aspect which has a high impact on discomfort glare sensation. The significant influence of both age and glare illuminance on discomfort glare agree with other research studies. Table 2 shows the measured values for B50L glare illuminance for the different setups. HID adaptation causes higher glare illuminance, in average about 0.14 lx higher compared to halogen adaptation. Because of the higher total luminous flux of HID lamps, more light is reflected from road surface to the subject's eye, resulting in a higher level of adaptation. As a consequence, the feeling of discomfort glare is slightly lower for HID adaptation. Table 2 shows the highest glare illuminance for the tested halogen reflexion system, independent of the spectrum of adaptation (0.36 lx and 0.52 lx). This causes a higher feeling of discomfort glare for the halogen reflexion system in Figure 2.

Table 2: Glare illuminance at subject's eyes

Car of subject (Adaptation)	Glaring car		
	HID D1 (Reflexion)	Halogen H7 (Projection)	Halogen H7 (Reflexion)
Halogen H7 (Reflexion)	0.23 lx	0.24 lx	0.36 lx
HID D1 (Projection)	0.36 lx	0.37 lx	0.52 lx

6 Calculation of Discomfort Glare

It was shown that tungsten halogen and HID headlamps can cause comparable discomfort glare ratings in real traffic situations when glare illuminance at subject's eye is similar [14]. A theoretical proof of these findings was not possible by calculation until today. An important milestone was reached in the year 2005 with establishing a spectral sensitivity function for discomfort glare in nighttime driving situations [47], which has a global maximum at 510 nm. **Figure 3** also shows the V(λ)-function and the spectral power distributions of a tungsten halogen lamp (Type H7) and a HID lamp (Type D1). Assuming additivity, which is necessary for many calculations [48], the effect of every single wavelength of a polychromatic light source can be added without influencing each other to one cumular effect. Discomfort glare sensation E(PB), photopic and scotopic illuminance E(P) and E(S) can be calculated with Eq. (1) to Eq. (3):

$$E(PB) = \int_{380\,nm}^{780\,nm} PB(\lambda) \cdot E(\lambda) d\lambda \qquad \text{Eq. (1)}$$

$$E(P) = 683\,\frac{lm}{W} \cdot \int_{380\,nm}^{780\,nm} V(\lambda) \cdot E(\lambda) d\lambda \qquad \text{Eq. (2)}$$

$$E(S) = 1699\,\frac{lm}{W} \cdot \int_{380\,nm}^{780\,nm} V'(\lambda) \cdot E(\lambda) d\lambda \qquad \text{Eq. (3)}$$

Here, PB(λ) is the spectral sensitivity function for discomfort glare [47], V(λ) is the photopic luminous efficiency function of the human eye (CIE 1924), V'(λ) is the scotopic luminous efficiency function of the human eye (CIE 1951), and E(λ) is spectral power distribution (SPD) of a light source.

By using Eq. (1) – Eq. (3), the glare factor PBF and the S/P-ratio can be calculated with Eq. (4) and Eq. (5):

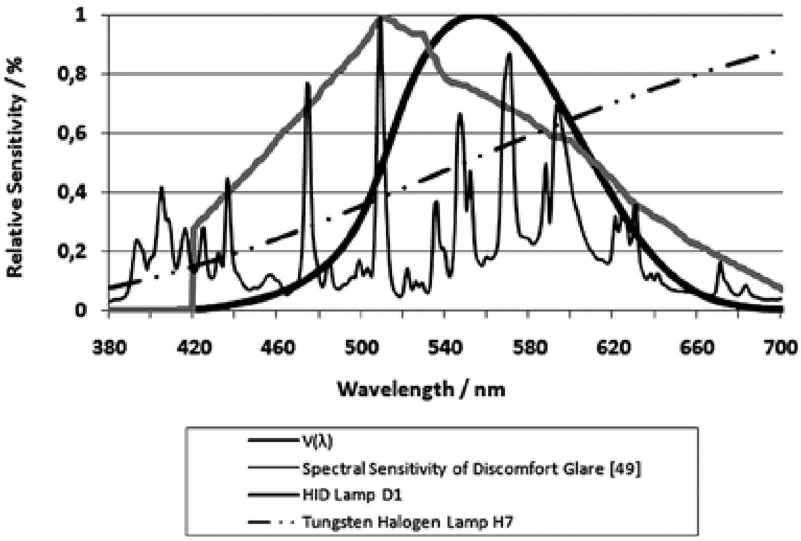

Figure 3: Spectral sensitivity function of discomfort glare and different SPD

$$PBF = 683\,\frac{lm}{W}\cdot\frac{E(PB)}{E(P)} \qquad \text{Eq. (4)}$$

$$S\,/\,P = \frac{E(S)}{E(P)} \qquad \text{Eq. (5)}$$

Table 3 shows the calculated factors for discomfort glare, the S/P-ratios and color temperatures of five automotive lamps. Calculating discomfort glare in the above mentioned way shows following aspects:

- The calculated factor for discomfort glare shows even slightly higher values for tungsten halogen lamps compared to HID lamps for identical glare illuminance. These calculations so far show the same tendency as the results from the performed field tests. Light sources with a higher blue part in their spectrum (Xenon/HID lamps) could produce comparable or even lower discomfort glare than tungsten halogen lamps.
- There is no difference in calculated discomfort glare factors PBF between normal and mercury free HID lamps.
- S/P-ratio and color temperature are highly correlated and are not appropriate for predicting discomfort sensations reliably. Both parameters cannot explain the high glare factor PBF for the tungsten halogen H7 lamp (Table 3). These findings are proved by the results from [44].

The introduced glare factor PBF seems to be one useful possibility for predicting discomfort glare for the performed field tests.

7 Summary

Cars of the same type but with different headlamps were compared by Technische Universität Darmstadt in order to test whether the correctly adjusted low beam of HID headlamps causes a higher feeling of discomfort glare in everyday traffic situations compared to tungsten halogen headlamps. Neither the field test nor the additional calculation by using the glare factor PBF showed a significant proof for higher glare rating of HID headlamps. This confirms earlier results of field tests of the authors [14].

An effect of the spectrum of the adaption field on discomfort glare could not be shown, either. Headlamp optics, subjet's age, and glare illuminance are important factors which influence discomfort glare feelings in a significant way.

Table 3: Calculated glare factors PBF, color temperatures and S/P-ratios for different types of lamps

Type of lamp	PBF	Color temperature	S/P-ratio
Tungsten Halogen H7	1.366	3200 K	1.557
HID D1 Osram	1.258	4057 K	1.64
HID D1 Philips	1.253	4099 K	1.63
HID D3 Osram	1.285	4158 K	1.71
HID D3 Philips	1.301	4290 K	1.75

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Active Hazard Braking

How Does the Driver-Vehicle System React?

One approach to increasing safety in road traffic is to assist drivers by means of an active hazard braking system. To assess drivers' behaviour, an interdisciplinary team of scientists from the Chair of Automotive Engineering (FZD) and the Institute of Ergonomics (IAD) of TU Darmstadt (Germany) has worked on behalf of six industrial partners (Audi, Bosch, BMW, Continental, MAN and Opel) in the scope of the Aktiv (Adaptive and Cooperative Technologies for Intelligent Traffic) research initiative [1], which is promoted by Germany's Federal Ministry for Economy and Technology.

1 Introduction

Safety in road traffic has greatly improved over the last 35 years. Initially, this success was achieved mainly by means of passive safety measures to mitigate the consequences of accidents. This technology is slowly reaching the limits of its safety potential. Further vehicle-safety developments are, therefore, concentrating on measures to avoid accidents and/or influence the course of accidents. The introduction of electronic stabilising systems and brake assist systems has already produced remarkable success [2]. The next generation of safety systems uses sensors to register the surroundings in order to initiate measures that will avoid a collision or reduce the impact energy before an accident takes place. As several accident analyses [3] have shown, in approximately half of the serious collision accidents, the brakes were not actuated at all. It seems expedient, therefore, to develop systems that actively initiate braking when there is a high risk of accident. This action is often embedded in an overall strategy with several previously triggered warning measures and measures to prepare the braking system and the braking assistant, plus, parallel to this, preparations for enhanced passenger protection. The Active Hazard Braking (AHB) system on which this investigation focuses is assessed as an individual measure, because, amongst other things, it cannot be assumed that timely warning alerts will be possible in very critical situations.

2 Objectives

Although it may appear plausible that an AHB would generate a higher safety impact, this has not yet been proven in real driving tests. Moreover, the impacts of faulty activation of the system were not known. In the scope of this preliminary study the impacts of two distinctive potential Active Hazard Braking system variants with different braking intensities were investigated and compared with the situation without AHB assistance (baseline):

- AHB with full deceleration (AHB full)
- AHB with partial deceleration (AHB partial)
- baseline (BL).

The obtained results may represent a substantial contribution to the development of Active Hazard Braking systems. Compared with previous driver behaviour studies [4] this test series was the first to also investigate unjustified interventions in the system that is abrupt triggering of the AHB when no danger of collision exists. Focus was placed on three variants, whose parameters are listed in the **Table**:

Table: Tested AHB and faulty activation variants

	AHB partial with subsequent brake release		AHB full with subsequent brake release		AHB full to a halt	
	Passenger Car	Truck	Passen-ger Car	Truck	Passenger Car	Truck
Deceleration specification [m/s²]	6	4	10	7	10	7
Maximum deceleration [m/s²]	7.0	3.9	9.8	8.0	9.9	8.1
Average deceleration build up [m/s³]	12	31	12	56	10	58
Average reduction in deceleration [m/s³]	-42	-25	-48	-25	-	-
Time of engagement [s]	1.3					
Time to collision at justified engagement [s]	2.0					

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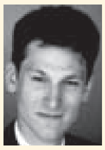

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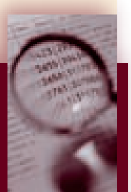

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Figure 1: The Darmstadt Test- and Evaluation Method Evita [5]

Figure 2: Test track

- AHB full to a halt
- AHB full with subsequent brake release
- AHB partial with subsequent brake release.

The AHB activation with justified intervention corresponds to faulty activation with subsequent brake release. The Time to collision (*TTC*) corresponds to the quotient of distance (*d*) and relative velocity (v_{rel}) to the vehicle ahead, Eq. (1):

$$TTC = \frac{d}{v_{rel}} \quad \text{Eq. (1)}$$

3 Methodology

The Darmstadt test and evaluation method Experimental Vehicle for Unexpected Target Approach (Evita) used to test the active hazard braking system [5, 6], differs from other driver tests of anti-collision systems as it uses an outset situation for rear-end collisions, that is a sudden abrupt braking manoeuvre following a previous car-to-car convoy as benchmark. The model is a combination of a drawing vehicle, a trailer and the collision vehicle being tested, **Figure 1**. To put the test drivers in a situation typical for rear-end collision accidents and to determine the effectiveness of the variants, they are distracted by a secondary task. During this time, the trailer (dummy target) drawn by a cable suddenly brakes. Independently of whether the test driver reacts in time or not, the trailer is pulled out of the collision area by activation of the winch in the trailer vehicle. To assess the effectiveness and the degree of disturbance, an assessment period of 2.0 s is defined beginning with the AHB engagement. As Evita ensures that a collision is automatically avoided, the procedure through to the imaginary collision with the uninterruptedly breaking dummy target is fictive. Following the action by the driver, the speed of the test vehicle is extrapolated to full deceleration as from the beginning of the Antilock Braking System (ABS) regulation, because it can be assumed that with the dummy target continuing to brake the driver will also have braked to a halt. To determine the dimensions, the velocity difference Δv is calculated between the initial measurement and the extrapolation at the end of the assessment period.

4 Test Procedures

4.1 Test Drivers

To obtain statements on the influence of the drivers' age, passenger car trials were performed with 60 test persons (male and female) from two age groups (25 to 40, on average 30 years and 50 and 65, on average 59 years). Three trials were carried out, in which 20 test persons each experienced the variants AHB full, AHB partial and baseline as their first trial. With the truck, trials were performed with 30 test persons (19 to 58, on average 37 years, most were professional drivers).

4.2 Test Track

The trials were carried out on the university's own testing grounds: the August-Euler Airfield. The course of pylons shown in **Figure 2** obliges the drivers of the AHB vehicle to follow Evita keeping to the same track without any noteworthy lateral misalignment. The foamed-plastic car on the right was placed as an alibi target for the faulty activation.

4.3 Driving Tasks

The test persons had the task of following the Evita driving in front of them at a speed of 60 km/h, keeping a distance of 20 m to 25 m, **Figure 3** left. The secondary test was to read a few lines from a route planner, Figure 3 right.

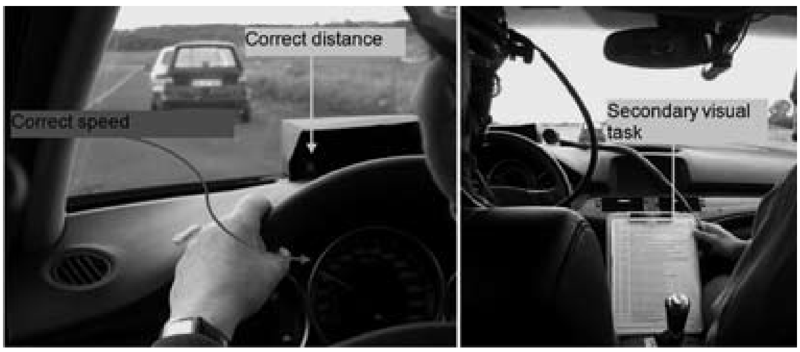

Figure 3: Left: primary task; right: secondary task

5 Results

5.1 Effectiveness

Figure 4 shows the effectiveness in the passenger car trials for the AHB variant tested, and compared to the baseline. Both AHB variants show significantly higher effectiveness compared to the baseline, but they only differ slightly from each other. In the baseline test the drivers can only perceive that the risk of collision is imminent by occasionally looking up front to check. Hence, the initial perception of danger depends mainly on the frequency and timing of the forward glances. Subsequently, the driver has to perceive that there is a danger, decide to react, move the foot to the brake pedal and actuate the brake. With AHB assistance, on the other hand, drivers are made aware of the imminent collision by a brake actuation. Moreover, drivers do not have to first mentally process the situation and then apply the brakes because the system does this automatically.

Although it could be assumed that the higher deceleration to a halt would lead to higher effectiveness, this cannot be substantiated by the results shown in Figure 4. The reason is that in the test passenger cars the build up of deceleration for both variants follows practically the same gradient; AHB full does not differ from AHB partial until approximately 800 ms at the earliest following commencement, because of the higher deceleration. As most drivers actuate the brakes very early and thereby override the braking by the AHB, full deceleration taking place later cannot achieve any detectable additional effectiveness. However, this could, for example, be achieved by building up deceleration more rapidly at the beginning of braking. In this way, with ideal, rapid full braking actuation, a speed reduction calculated at 20 ms/s = 72 km/h could be projected.

Figure 5 shows the effectiveness in the truck trials. In contrast to the car trials, an effectiveness rating is achieved at least at the level highly significant: AHB full > AHB partial > BL. The effectiveness of both AHB variants is far superior to the baseline. In trucks, however, AHB full proves to produce a higher effectiveness than AHB partial. These vehicles use a very dynamic electro pneumatic braking system with pressure storage, which allows the full deceleration variant to build up greater deceleration as early as 300 ms after commencing compared to the partial deceleration variant. The test persons cannot override the system by applying the brakes themselves until 500 ms at the earliest, which explains the advantage of the full deceleration variant.

5.2 Effectiveness Assessed by the Driver

In addition to assessing the objective effectiveness, at the end of the test drives, drivers were surveyed as to their subjective perception of the AHB variant. They assessed the effectiveness of the different braking variants without knowing which AHB variant they had experienced.

The majority of test persons assessed the effectiveness of the active hazard braking systems as effective or very effective, **Figure 6**. No significantly different assessments were made between the different variants. These results indicate on the one hand that the test persons can hardly distinguish between the degree of effectiveness of full and partial deceleration. On the other hand it is an indicator that, independent of the variant used, drivers perceive AHB as a support.

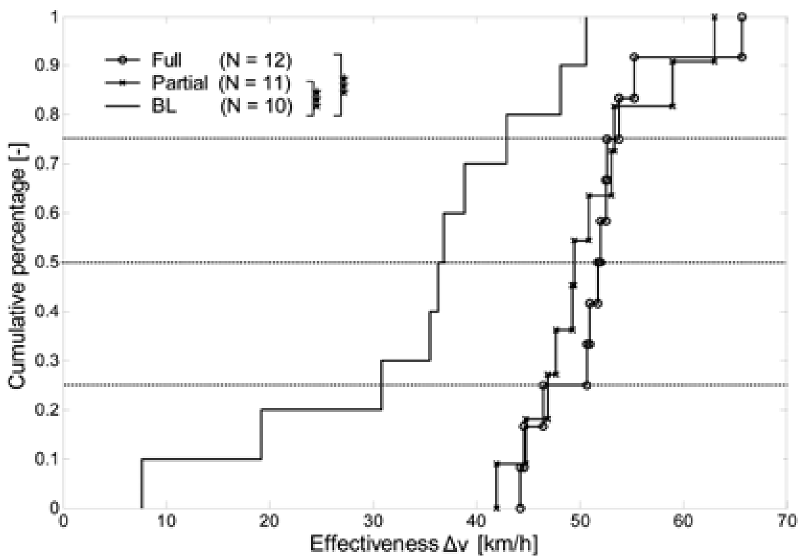

Figure 4: Cumulated percentage of effectiveness in the first trial (passenger car)

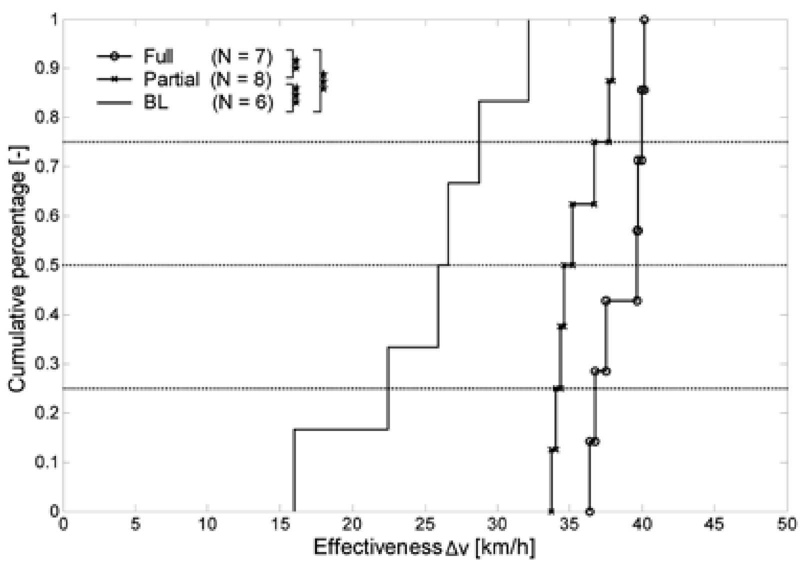

Figure 5: Cumulated percentage of the effectiveness in the first trial (truck)

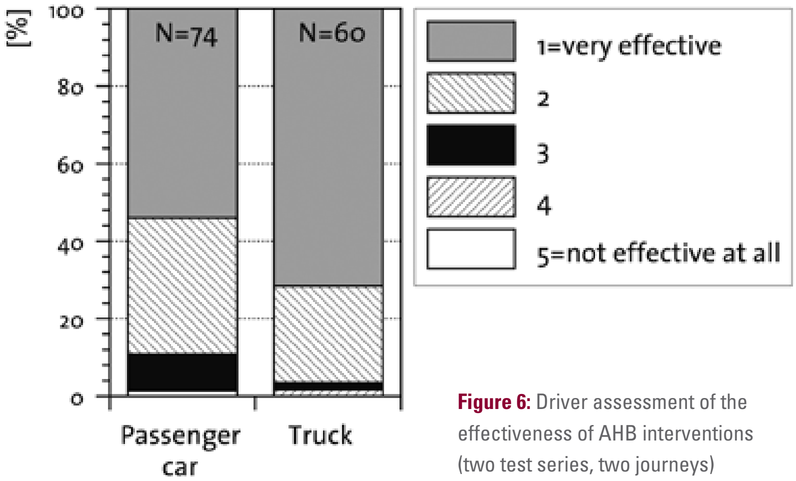

Figure 6: Driver assessment of the effectiveness of AHB interventions (two test series, two journeys)

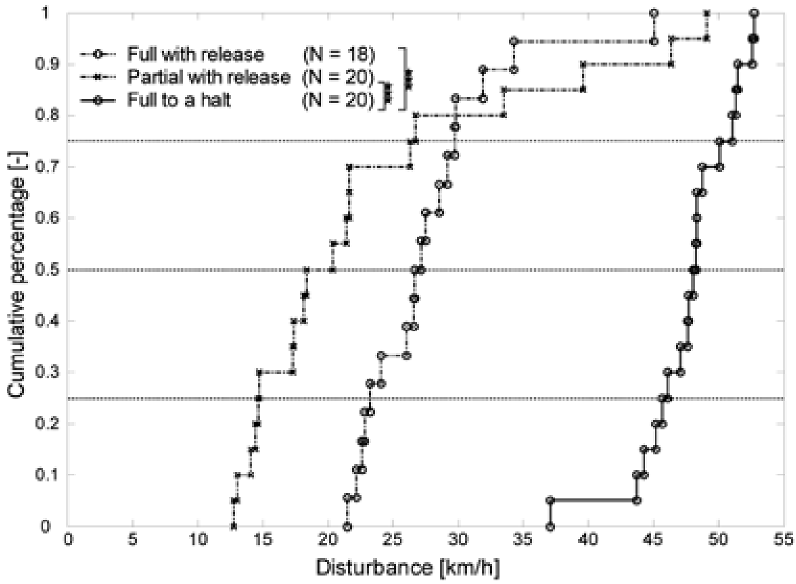

Figure 7: Cumulated percentage of disturbance, 2.0 s after faulty activation (passenger car)

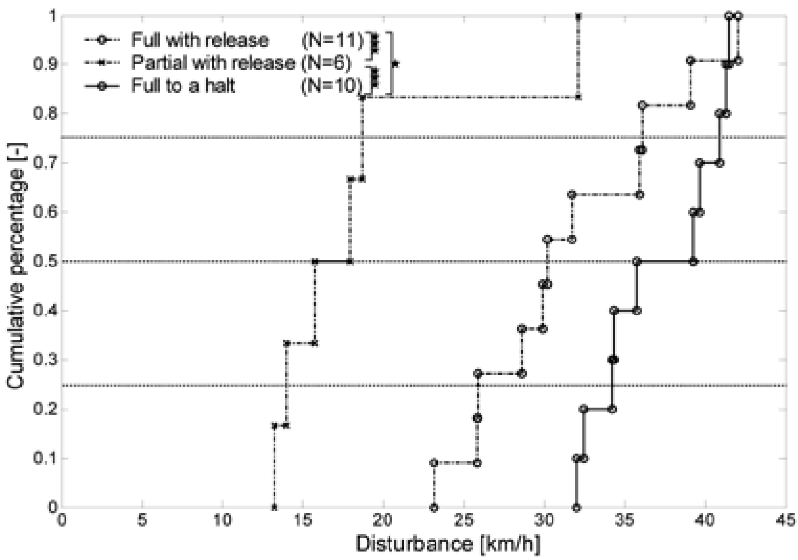

Figure 8: Cumulated percentage of disturbance, 2.0 s after faulty activation (truck)

5.3 Degree of Disturbance

The degree of disturbance Δv is defined analogue to effectiveness. **Figure 7** shows the percentage of disturbance from the passenger car trials. The vertical lines indicate for each faulty activation variant, the average measurement from those trials in which the test persons did not actuate the brakes. This illustrates, therefore, the anticipated average value of the system reaction on its own, without intervention by the driver. The square boxes placed over the individual measurements indicate a braking intervention by the driver, without giving its intensity compared to the system actuation. The spread of the degree of disturbance without driver intervention is caused by variations of the friction coefficient during the trials and the variation of the build-up of braking pressure. The latter is typical for Electronic Stability Programme (ESP) pumps with one charge piston per braking circuit.

A very significant difference exists between full deceleration to a halt and the variant with subsequent brake release. In contrast to the tests with justified brake activation only very few test persons intensify the AHB by applying the brake themselves. Most test persons recognise the faulty actuation, and consequently, the degree of disturbance of most deceleration with release tests was only higher for about a quarter of the test persons. Full deceleration to a halt considerably intensified the degree of disturbance.

Figure 8 shows the degree of disturbance from the truck trials. Because of the higher dynamic of the braking intervention, the difference between full and partial braking (both with subsequent brake release) is far clearer, so that a clear rating of the percentage of disturbance can be constructed: Faulty Activation (FA) full to a halt > FA full with release > FA partial with release.

The actuation of the brake pedal in the truck trials shows a correlation between the frequency of brake actuation and greater intensity of the faulty activation variant. Whereas only three of six drivers actuated the brakes themselves, during faulty activation with the partial deceleration nine out of ten drivers actuated the brake in the case of full deceleration to a halt and indeed eight of eleven applied the brakes for full deceleration with subsequent release.

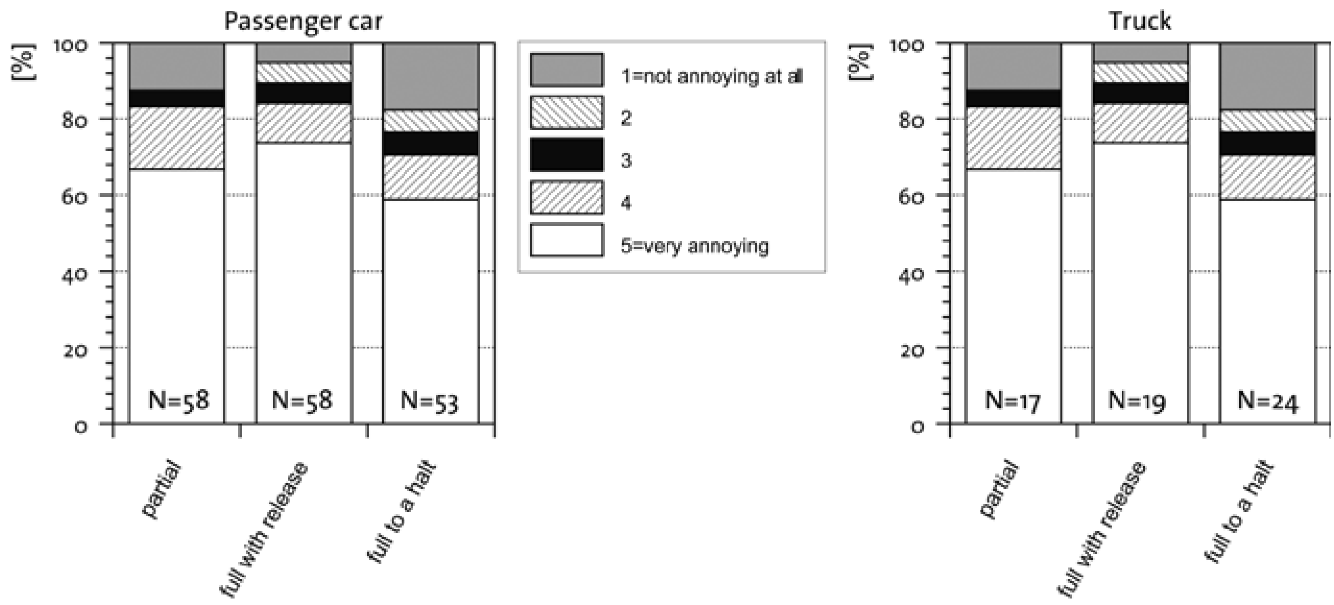

Figure 9: Subjective forgiveness of faulty activation, left for passenger cars, right for trucks

5.4 Driver-assessed Forgiveness of Faulty Activation

All test persons each experienced two accidental releases of the AHB. Subsequently, they assessed the extent to which this disturbed them, without knowing which variant they had experienced.

Figure 9 shows that the majority of drivers (of passenger cars and trucks) assessed this as disturbing and very disturbing. The drivers made a differentiation as to the degree of forgiveness, but not between the types of faulty activation.

5.5 Driver Stress

To obtain statements on the risk potential subjectively perceived by the drivers, their physically measurable reactions to their emotions when AHB was activated were calculated using their heart-beat frequency and skin conductance.

The heart-beat frequency and skin conductance indicators showed that in all critical situations all drivers had significantly higher emotional stress than when at rest. No difference was noted between the critical braking situation where a driver was assisted by AHB and barking situation where a driver was not assisted by AHB.

The analysis of drivers' emotional stress shows that stress levels are just as high with faulty activations as with critical braking situations. Only the heartbeat frequency of passenger car drivers is significantly lower for faulty activations than in critical braking situations.

To deduce statements about the visual stress of drivers, an analysis of eye tracking, **Figure 10**, examines the number of eye movements per second between defined objects. The results show that the drivers move their eyes significantly more often with faulty activations than with justified AHB interventions or in critical braking situations. From the number of changes in points of focus it can be deduced that passenger car drivers undergo far greater visual stress during faulty activations than truck drivers. These findings indicate that drivers use searching movements with their eyes try to clarify the cause of the initiated AHB. This has a detrimental effect because during the time of these eye movements the driver does not concentrate on directing his/her focus forwards, and therefore could possibly miss out on important information.

Figure 10: Eye tracking video (truck)

6 Conclusions

In the present study it was possible for the first time to substantiate the positive effects of active hazard braking in near-reality trials with test drivers. The effectiveness, expressed as the difference in speed during the danger situation, is clearly higher than when the driver is not assisted by AHB.

It is demonstrated that more intensive deceleration by the system also leads to higher disturbance and in general to higher effectiveness. Nevertheless, when deceleration build up is slow, as for example with the ESP pump in the tested passenger car, it cannot be concluded that greater deceleration automatically leads to higher effectiveness. Only a quicker deceleration build up can generate an advantage, although the disturbance is also higher.

With the variants tested a correlation will always be visible between higher impact and higher disturbance in the case of faulty activation. However, at one place indicators to solve this conflict of goals could be identified. As could be seen in the truck trials, a partial deceleration built up with higher dynamics is still suitable to attract the drivers' attention to the imminent danger of collision and leads to greater effectiveness. The accelerator pedal actions show, however, that faulty activations of the AHB are recognised as such at a very early stage, so that most drivers accepted them without applying the brakes. The drivers assessed all varieties of these faulty activations mostly as disturbing or very disturbing. The degree of disturbance was far below that of full deceleration. Because of the small amount of random samples, however, this correlation is not significant.

Drivers are always disturbed by faulty activations, although the variant AHB partial with subsequent brake release had the lowest very disturbing assessment, albeit only a small distance away from the other variants. Most test persons stated that they wanted an opportunity to override faulty activations. However, no uniform opinion could be obtained to the question of how this overriding should be structured. Applying the accelerator alone cannot be interpreted as a wish to override the system, because almost all drivers who actuate the accelerator directly before the beginning of the intervention continue to do so once the intervention has taken place.

The emotional stress is significantly higher in danger situations with AHB intervention than before starting to drive; the AHB variants examined had no influence on this. However, the findings also show that the increase is triggered primarily by the situation of danger. The AHB interventions do not create a further increase in stress, as is shown by the comparison with drives where no AHB intervention took place.

The Darmstadt Evita testing method for collision warning and collision avoiding systems allows a direct comparison of both the measures to avoid and mitigate the consequences of front-on collisions and the related disturbances with faulty activations. The results were obtained using uniform trigger thresholds and only with the forms of intervention shown. Findings to date allow initial generalised statements, but the measurements cannot be transferred to systems with different intensity and/or dynamics of braking than the examples examined. Nor have the impacts of early warning, as used in most anti-collision systems, been ascertained here.

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