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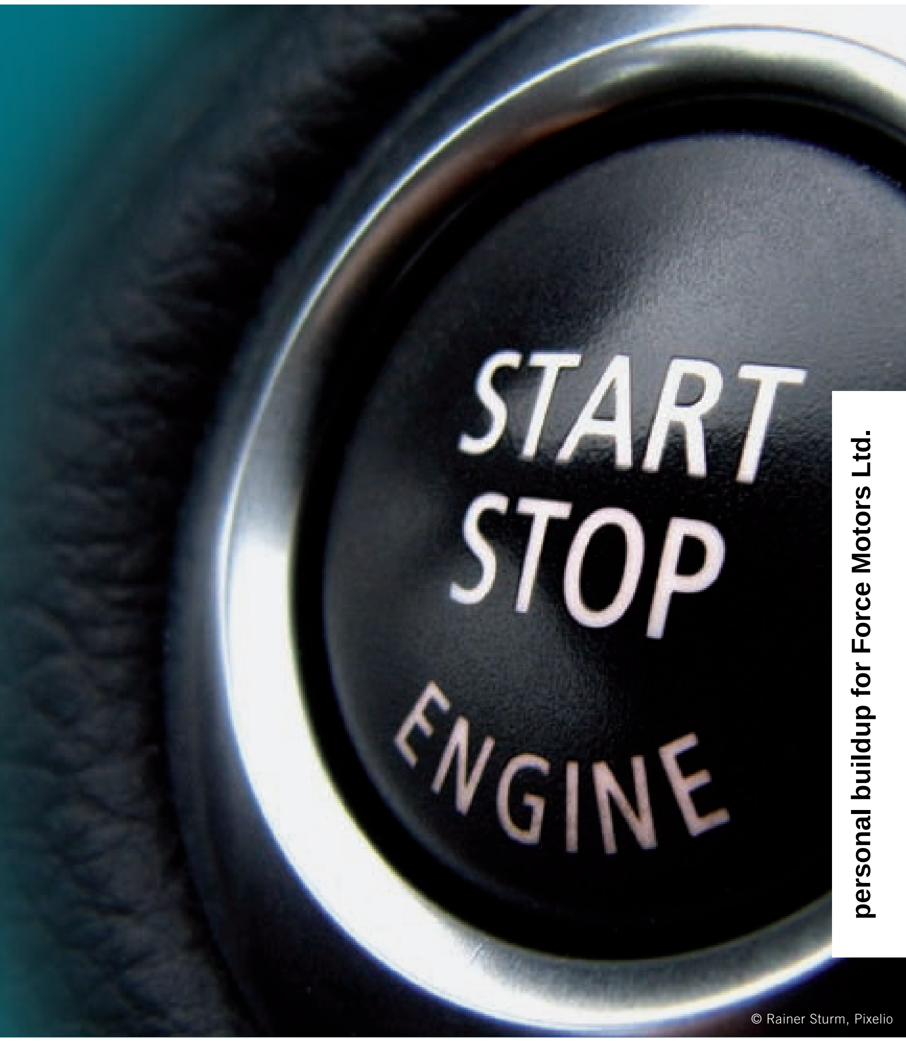
START / STOP
A CORNERSTONE TO ELECTRIFICATION

CONTENTS



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COVER STORY

START / STOP A CORNERSTONE TO ELECTRIFICATION

4 I With the introduction of the Panamera Porsche has installed the world's first start/stop system in a luxury class car with an automatic transmission, in this case the Porsche Doppelkupplung (PDK). The developers explain the operating strategy, the control mechanisms and the system integration and describe how numerous components were modified to cope with the increased loads.

COVER FIGURE Volkswagen

KEEP IT SIMPLE

Dear Reader,

Yesterday I had the opportunity to test drive a new compact car. Although I enjoyed driving the car, I had a shock when I looked at the price list. The basic model starts at 12,000 euros, while the most environmentally friendly version, which boasts a CO_2 emission of 98 grams, even costs as much as 15,000 euros. And by adding a bit of optional equipment, you can easily send the price rocketing to 20,000 euros.

There are, of course, many good reasons why cars have become so expensive and why they now cost as much in euros as they did in Deutschmarks 15 years ago. ESP and several airbags make them much safer, their engines are almost free of harmful emissions even with Euro 5, and small cars today are not really small at all. But the customers' purchasing power has not kept pace with this development.

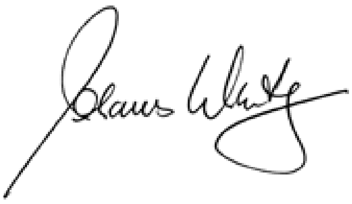
One major cost driver is often overlooked: the individualisation of equipment and body versions. It may be fun to put together various versions on a car configurator, but none of them come free of charge. All of the different powertrain and body versions need to be validated in the development process and made available in production.

Car makers can absorb these additional costs by producing in large volumes. Or, as in the premium segment, by charging a hefty mark-up. The best solution, as Volkswagen has shown, is to do both at the same time.

But what can they do if production volumes are relatively small and the brand image is not quite up to "haute couture"? Perhaps they should think back to the "Model T" Ford, the original Volkswagen Beetle or the Citroën "2CV".

My wish list today: a car that provides mobility for four adults, is safe (five stars in the Euro NCAP test) and environmentally friendly (100 grams of CO_2 or less), and has excellent design quality. The colour is optional, but the powertrain is always the same. And the whole package has a price tag of 10,000 euros.

Impossible?



JOHANNES WINTERHAGEN, Editor-in-Chief
Wiesbaden, 11 February 2010

START/STOP SYSTEM IN THE PORSCHE PANAMERA

With the introduction of the Panamera, the fourth model line from Dr. Ing. h.c. F. Porsche AG, the team in Zuffenhausen has installed the world's first start/stop system in a luxury class car with an automatic transmission, in this case the Porsche Doppelkupplung (PDK). In city driving, the sports car saves up to 1.5 l fuel per hundred kilometres in comparison to a manual transmission. The developers explain the operating strategy, the control mechanisms and the system integration and describe how numerous components were modified to cope with the increased loads.

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DEVELOPMENT OBJECTIVES

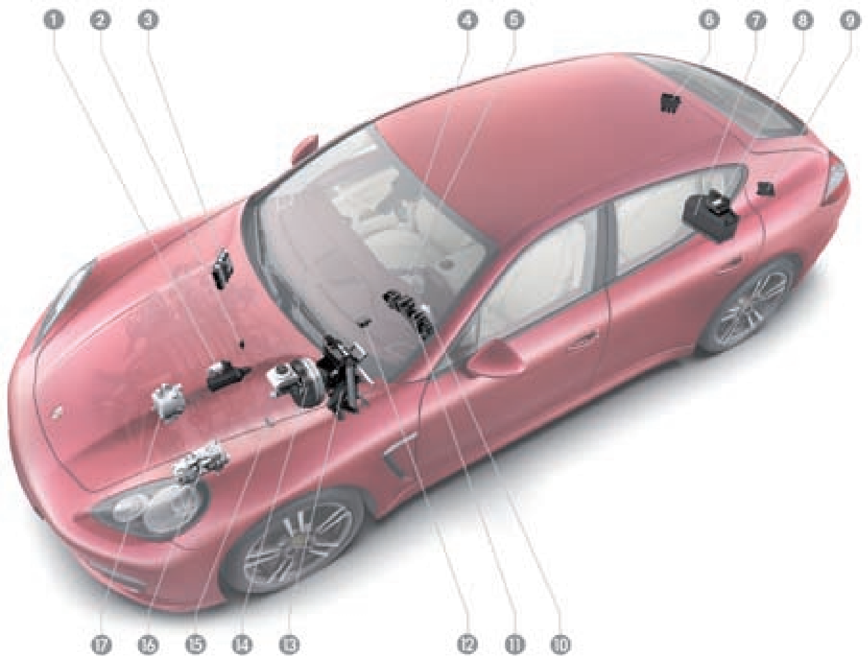
The new function supports the engineers' general development objective of achieving the lowest possible fuel consumption despite the sporty character of Porsche vehicles. Given the demand for convenient and fully-automatic operation and the best possible efficiency, this was only possible, however, by introducing new technologies into the Panamera. These include AGM batteries, engines with direct fuel injection and the comprehensive networking of the control units. All the relevant components and systems are illustrated in ❶.

In contrast to start/stop systems with manual transmissions where the driver must actively initiate the stop by engaging neutral, a new operating strategy had to be developed for vehicles with automatic transmissions. To ensure that there was no reduction in comfort, sportiness and component service life, comprehensive measures were implemented to guarantee the perfect functioning of the stop and restart operation. It was essential that customers did not experience any disruption during the standstill phase. The engineers paid particular attention to transparency and the intuitive operation of the system. Compared to the start/stop systems used with manual transmissions, it is possible to achieve significantly lower fuel consumption with the operating strategy for automatic transmissions. This is because, when the vehicle is at a standstill, the engine is switched off entirely automatically and thus more consistently and more frequently. During the stop phase, the AGM battery guarantees the electrical supply to the vehicle, as the engine-driven auxiliary systems, including the generator, are out of operation.

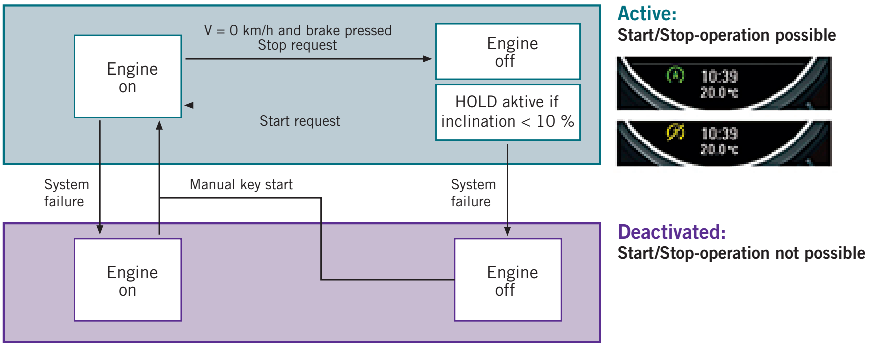
OPERATING STRATEGY

All operating conditions for the first engine stop must be fulfilled after the engine is first started using the ignition key. To ensure that the engine switch-off and restart works smoothly, the operating temperatures must first be reached. The time required for this varies depending on the climatic conditions and the warm-up behaviour.

If all the technical prerequisites for the engine stop are fulfilled, the engine is automatically switched off when a vehicle



❶ Components of the start/stop system: 1 Starter, 2 Engine temperature sensor, 3 Engine control unit, 4 Centre console switches (Level, Sport, Start/Stop, ACC etc), 5 PDK selector lever, 6 Trailer control unit, 7 AGM battery with battery sensor, 8 DC converter, 9 Gateway, 10 Steering angle sensor, 11 Instrument cluster, 12 Tilt sensor, 13 Pedals with brake switch, 14 Brake booster, 15 Engine speed sensor, 16 A/C compressor, 17 Generator



❷ Operating strategy

❸ Start/stop button

standstill is detected and the brake is engaged. An illuminated green start/stop symbol in the instrument cluster indicates this to the driver.

This switch-off is only possible within a time window of 0.85 to 2.6 seconds after the vehicle standstill is detected. Having the stop initiated after 0.85 seconds at the earliest avoids short stops, for example at stop signs or when approaching roundabouts. If a stop is not possible after 2.6 seconds from detection of the vehicle standstill, then it is no longer possible at all in this vehicle standstill phase. In this case, an illuminated yellow start/stop symbol with a line through it is displayed in the instrument cluster.

If the engine is switched off, it is automatically restarted by the system or when requested by the driver releasing the brake. The driver can drive off as usual by pressing the accelerator. After a restart, the vehicle must first be driven above a speed threshold of 2 km/h for a minimum period before another engine stop is possible. This prevents the engine switching off and on several times while the vehicle is at standstill. ❷ shows that an automatic state change between "engine on" and "engine off" is only possible when all active participants are ready for operation.

If the system is no longer ready, the start/stop function changes to the "deactivated" state and an automatic state change between "engine on" and "engine off" is no longer permitted. It is then only possible to start the engine from engine stop mode by starting it manually using the ignition key.

Various other control strategies facilitate manoeuvring, prevent unfavourable driving situations and guarantee the comfort functions. The following examples illustrate some of these control mechanisms:.

: Manoeuvring detection prevents the engine from being switched off, even if the vehicle has not moved for a brief time. As soon as the vehicle speed exceeds 8 km/h, the manoeuvring mode is switched off
: High steering angles indicate that the driver is turning and does not wish to stop
: In the case of functions requested by the driver, in particular, it is important to weigh up whether a stop is desired in these situations. Activation of the AC max or window defrost functions, for example, also prohibits stopping, as does a vehicle level adjustment requested by the driver
: When Sport mode is activated or the stability program is deactivated, the start/stop mode is likewise inactive
: To ensure drive-off quality and performance in all situations, an engine stop on steep upward or downward gradients is also prohibited. In the event of a stop on permissible inclines, the autonomous hold function is used to prevent the vehicle rolling backwards by holding the hydraulically set driver brake pressure
: If a customer does not wish to use the start/stop function, it can of course be deactivated via the start/stop button, ❸.

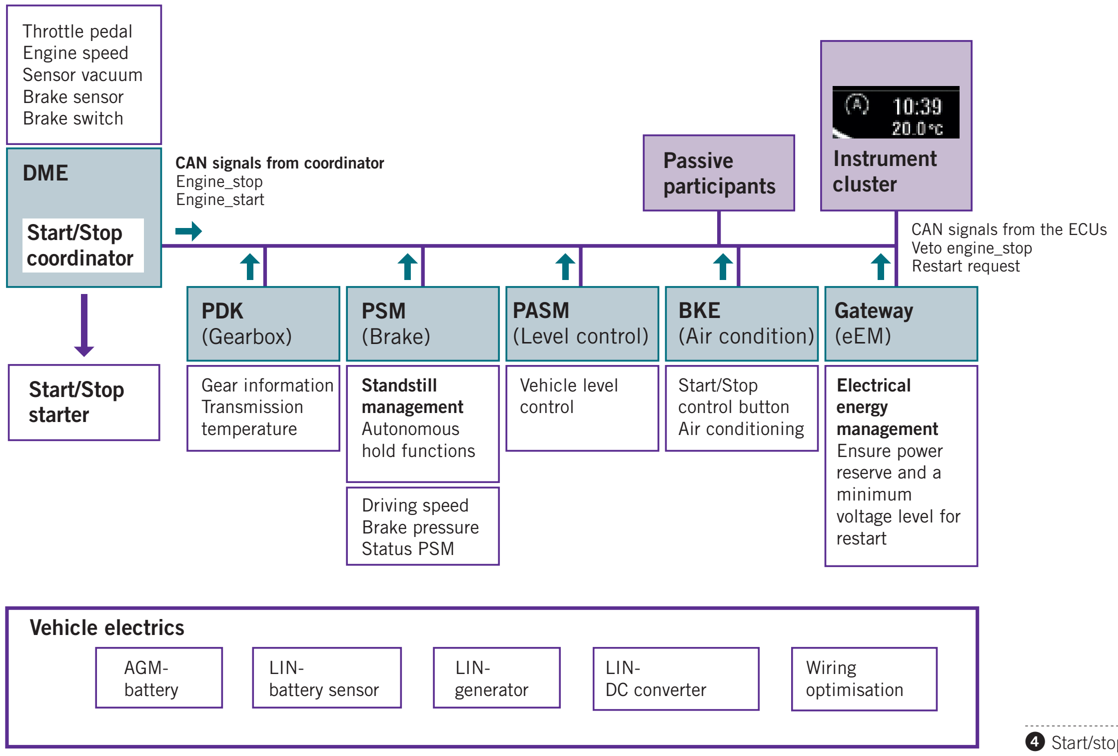
IMPLEMENTING THE FUNCTION

The requirements described for a start/stop function necessitated a number of function enhancements to existing systems and the introduction of new components. The integration and coordination of individual functions in the integrated vehicle system were amongst the most demanding tasks. An overview of the systems and components involved and the related functions is provided in ❹.

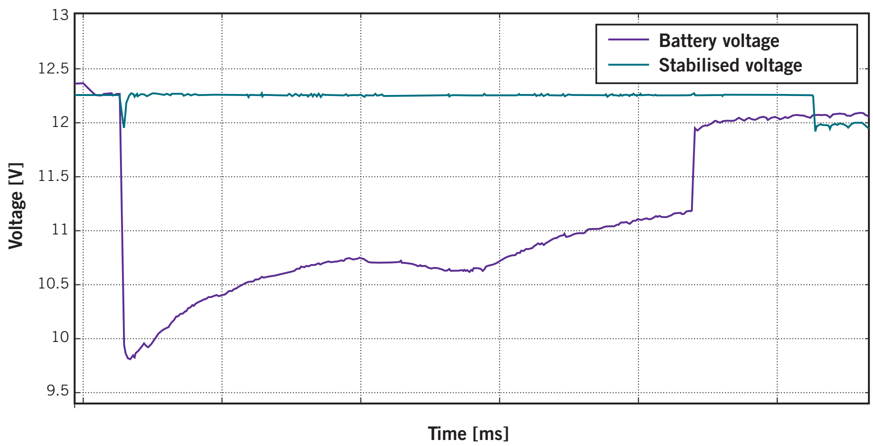
To the greatest extent possible, an automatically switched-off engine should provide the vehicle occupants with all the same normal driving functions that it does when it is running. In contrast to when the engine is started initially with the key, therefore, no restrictions to either comfort or functions are acceptable during the automatic quick-start processes. The basic operating strategy is essentially mapped by the central function coordinator in the engine control unit. The complexity of the overall system was increased considerably by the development requirements and had to be spread across several control units as a distributed function.

Numerous control units in the integrated vehicle system influence the start/stop function directly as "active participants". For example, the air-conditioning control unit uses humidity and temperature sensors to monitor the air-conditioning comfort in the vehicle and can temporarily prevent an engine stop via the CAN communication to the central coordinator. Alternatively it may, during an active engine stop operation, ask this control unit to initiate an automatic restart.

Moreover, there are scores of "passive participants" in the start/stop compound structure which exercise no direct influence on the start/stop operation, but which must detect the status of the function in order to execute special functions. For example, various automatic adjustment processes, such as the seat adjust-



❹ Start/stop system overview



❺ Stabilised voltage in the restart

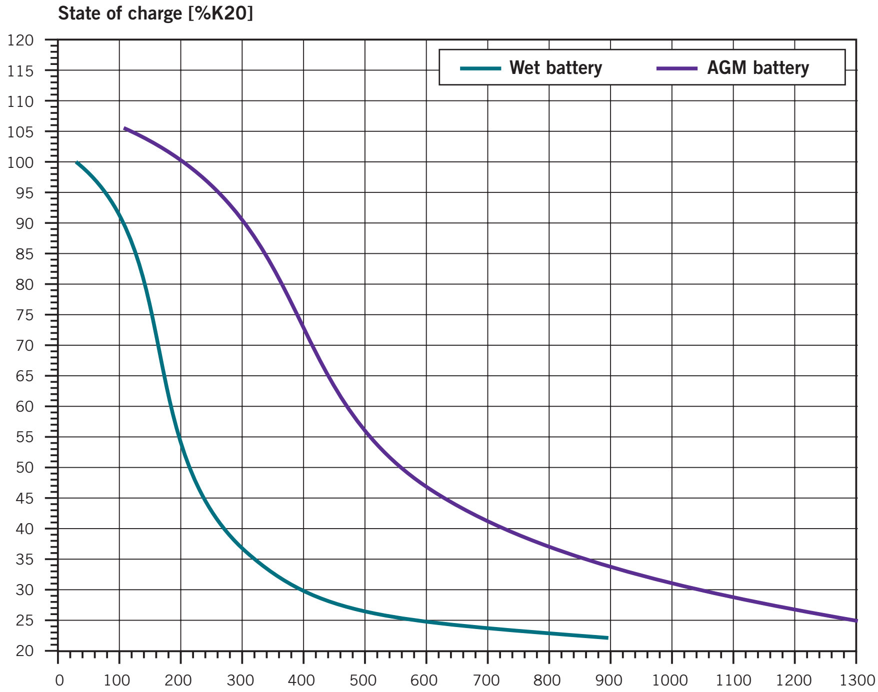
ment, electric windows, sliding roof and tailgate, must also continue during the restart process and in the event of extreme undervoltage interrupt their automatic operation in a defined manner.

All other components in the electrical vehicle system must continue to operate for the duration of the restart voltage interruption without function impairment or failure.

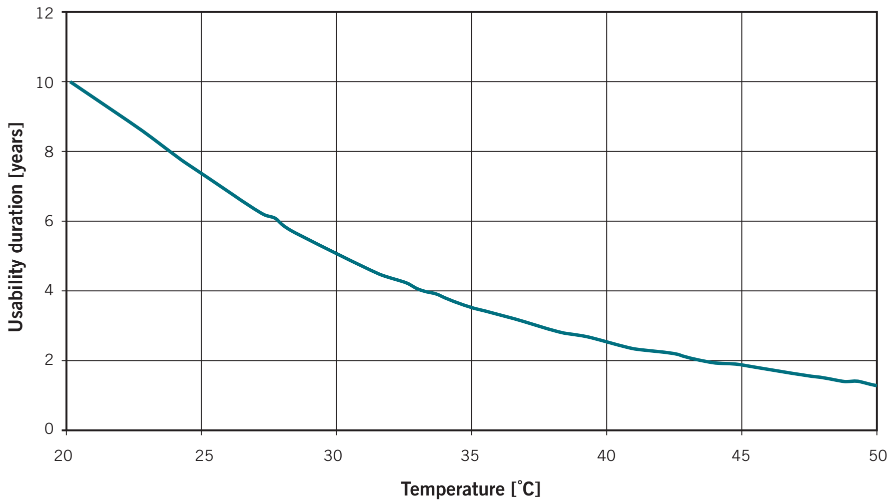
By optimising the voltage level of individual components through software and hardware changes, it was possible to implement the vehicle electrical system topology as a classic 12 V power network with only one vehicle battery. As a result, there is no need for a second battery with all the cabling and actuation controls that this entails. Additional weight and costs can therefore be avoided.

During the start-up process, the supply voltage to the consumers drops dramatically, depending on the wiring and the battery status. In order to prevent the starting voltage drop from affecting the entertainment and communications system or the interior lighting, a DC converter is used which supplies these voltage-sensitive consumers with a stabilised voltage during the restart. ❺ shows, by way of example, the voltage at the output of the DC converter in the starting process in correlation with the battery voltage.

The electrical energy management system ensures that a minimal permissible start voltage threshold is maintained at the components, that sufficient power reserves are guaranteed and that the highly-loaded battery is protected during the stop phases and during the restart. A battery sensor determines the current battery status from the voltage, current and terminal temperature variables and transmits this information to the electrical energy management system. The modelled acid temperature, the charge state



❻ Influence of depth of discharge on the cycle service life [Moll GmbH]



❼ Service life in years as a function of the average battery acid temperature [1]

and "health" (internal resistance) of the battery can thus be monitored. An engine stop can, for example, be prohibited if the battery is at risk of overheating; this is done by the electrical energy management system issuing a stop ban.

OPTIMISATION OF THE COMPONENT SERVICE LIFE

In line with the higher number of engine start processes over the vehicle service life and the new type of load, a large number of components had to be modified to cater for the new vehicle "engine stop" state. First among these was the electric starter. Porsche designed the components to accommodate eight times more starts than conventional starters. As part of the start/stop function, the starter battery has the task of supplying the vehicle electrical system during the engine stop. As a result, the battery cycling has been increased around three-fold. In order to ensure that the usual service life is still maintained, lead AGM (absorbent glass mat) batteries are now used. This technology is significantly more cycle-proof than the previous wet batteries and also makes it possible to maintain the existing battery service life under the loads typical of start/stop operation, ❻. Besides the cycling, temperature is also of critical importance to the service life of the battery, ❼. In order to protect it from extreme temperatures, the engineers located the battery in the interior, underneath the luggage compartment floor.

SWITCHING OFF AND RESTARTING

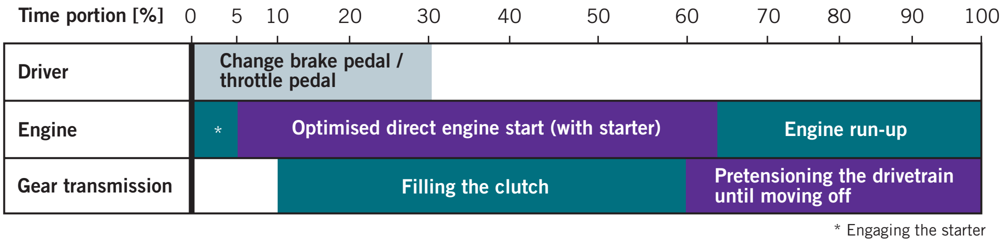
To ensure customer acceptance of the start/stop function in luxury class vehicles such as the Porsche Panamera, the switch-off and restart operations must be as unobtrusive as possible, in other words, fast and smooth. To this end, knowledge of the precise crankshaft position is required to quick-start the engine (starter-supported direct start). To determine the next cylinder in the ignition sequence, the engineering team added a rotation direction detector functionality to the existing engine speed sensors.

In addition, the moving-off process was optimised by running the initialisation processes of the engine run-up in parallel in the engine and transmission control units, ❽. As a result, the PDK clutches start to fill as early as possible, in order to accelerate the engagement operation. Likewise, the CAN communication between the engine and transmission control units during the transfer of the effective engine torque to initialise the moving-off control in the transmission was redesigned to enhance the comfort and speed of the moving-off process for start/stop.

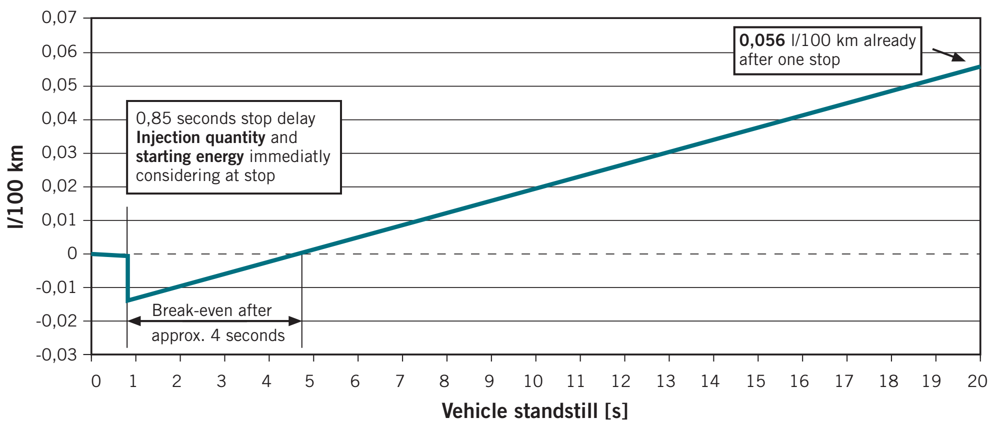
STANDSTILL MANAGEMENT

During the engine stop and, in particular, when the restart process is initiated by releasing the brake pedal, the brake system must be able to hold the vehicle until sufficient propulsion power is achieved.

This task of securing the vehicle during the engine stop and, in particular, preventing the vehicle rolling during the restart process is assumed by the autonomous hold function of the PSM brake system.



❽ Parallel moving-off process



❾ Consumption saving per engine stop relative to the NEDC, as a function of the duration of the vehicle standstill

A brake vacuum sensor on the brake booster monitors the diminishing vacuum during engine standstill. To maintain the braking effect, an engine restart is initiated well before the brake pedal becomes hard.

AVAILABILITY

To reduce consumption, the maximum stop availability is given a high priority. The consumption advantage achieved with the start/stop function is based on the fact that from a stop duration of approximately four seconds the fuel quantity required to compensate for the electrical power requirement of the starting process and to start the engine is less than that required in total when the engine is idling without a start/stop function, ❾.

Together with the PDK, the Panamera Turbo boasts a consumption saving in the NEDC (New European Driving Cycle) of 0.6 l/100 km or 14 g/km CO_2. In city driving, savings of up to 1.5 l/100 km or 35 g/km CO_2 can be achieved.

To ensure that the safety, performance and comfort criteria are met, approximately 130 vetoes from the active participants can limit the availability of the automatic engine stop when the vehicle is at a standstill. The vetoes can be divided into:

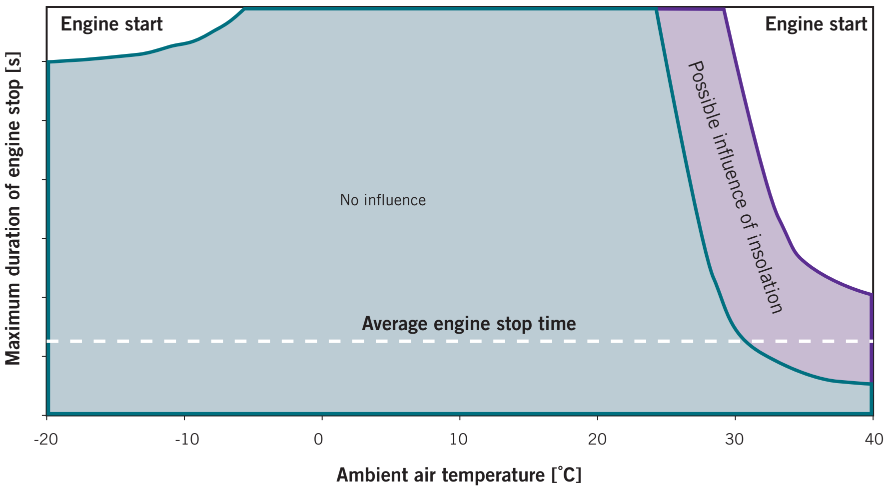
: vetoes which are transparent for the driver and which can be triggered via driver activities in the vehicle – for example, Sport mode is activated, manoeuvring mode detected or maximum interior cooling requested
: system vetoes, the cause of which may not be clearly evident to the driver under certain circumstances – for example, engine diagnostics are active during idling or the battery temperature is too high.

Furthermore, potential component errors are analysed which also leads to the deactivation of the start/stop function. This includes function monitoring on the DC converter and brake vacuum sensor, and internal errors in the active participants.

The major challenge facing the developers was to achieve maximum stop availability while avoiding comfort losses and to make the function behaviour transparent to customers through the application of the function itself.

This is explained using two examples. While the engine is stopped, it lacks the mechanical power to drive the air-conditioning compressor and the generator. The vehicle electrical system is supplied from the starter battery. The air-conditioning system takes cooling power from the storage capacity of the cooling system and heating power from the heat storage capacity of the heat exchanger. As both are finite, the combustion engine must be restarted before the driver feels any change in the air-conditioning. ❿ shows a qualitative representation of the dependence of the maximum stop duration relative to the outside temperature and sunlight. From the diagram it is clear that, on the basis of the average stop duration of around 15 seconds, which was identified in the development phase in traffic, restarts determined by the air-conditioning system occur only rarely or only in the case of extreme outside temperatures.

Because the restart request depends on internal vehicle states which are not directly evident to the driver (for example, temperature difference of the heat exchanger), restarts or stop vetoes are not always immediately transparent. One possible way of counteracting this would be to make temperature-dependent stop vetoes or restart requests dependent on the outside temperature, which is directly visible to the driver (through the display in the instrument cluster), instead of on the component temperature. When the stop availability is directly dependent on the outside temperature, any vetoes are immediately transparent but less reproducible (quickly changing air temperature and sunshine) and the system availability is additionally restricted.



❿ Qualitative representation of the maximum possible engine stop duration before the air-conditioning system issues a restart request on the basis of the ambient temperature

Another example of a temperature-dependent veto is the battery acid temperature, which must be at least -1 °C for the start/stop. With the high heating capacity of the starter battery and its housing in the vehicle interior, the start/stop function remains available even at outside temperatures of less than -1 °C. Given the large number of customers who will, it is expected, park their vehicle at night in a garage or an underground car park, start/stop will remain available for most of the time even in winter.

CONCLUSION AND OUTLOOK

In the Panamera, Porsche is introducing the start/stop function in conjunction with the Porsche Doppelkupplung, proving that it can be successfully implemented in sporty, luxury class vehicles. The high equipment rate of 90 to 95 per cent was one of the reasons for developing start/stop for the PDK. In the future, Porsche will equip most of its vehicles with a start/stop function. As this function is developed further, the system costs will be reduced, the operating strategy optimised and the function availability extended, even in conjunction with other systems such as ACC (radar-supported cruise control).

In the future, it will be possible to reduce costs as a result of a model-based software module developed by the engineers which will determine the brake vacuum level and thus render the brake vacuum sensors superfluous.

The extent to which it will be necessary to make more information available to the driver (for example, additional instrument cluster messages for transparency reasons during automatic restart situations and action recommendations in the event of system-determined stop vetoes) must be decided on the basis of feedback from the market.

REFERENCE
[1] Manual for stationary, closed gel batteries, Part 2: Assembly, commissioning and operation. Exide Technologies. Nov. 2008

THANKS

To the Co-authors at Dr. Ing. h.c. F. Porsche AG in Stuttgart: Dr. Frank-Steffen Walliser, head of Department of Hardware Integration; Martin Roth, head of Concepts & Function Energy Management, Complete Vehicle Project Manager, Start/Stop; Dr. Steffen Kehl is responsible for electrical consumption measures.

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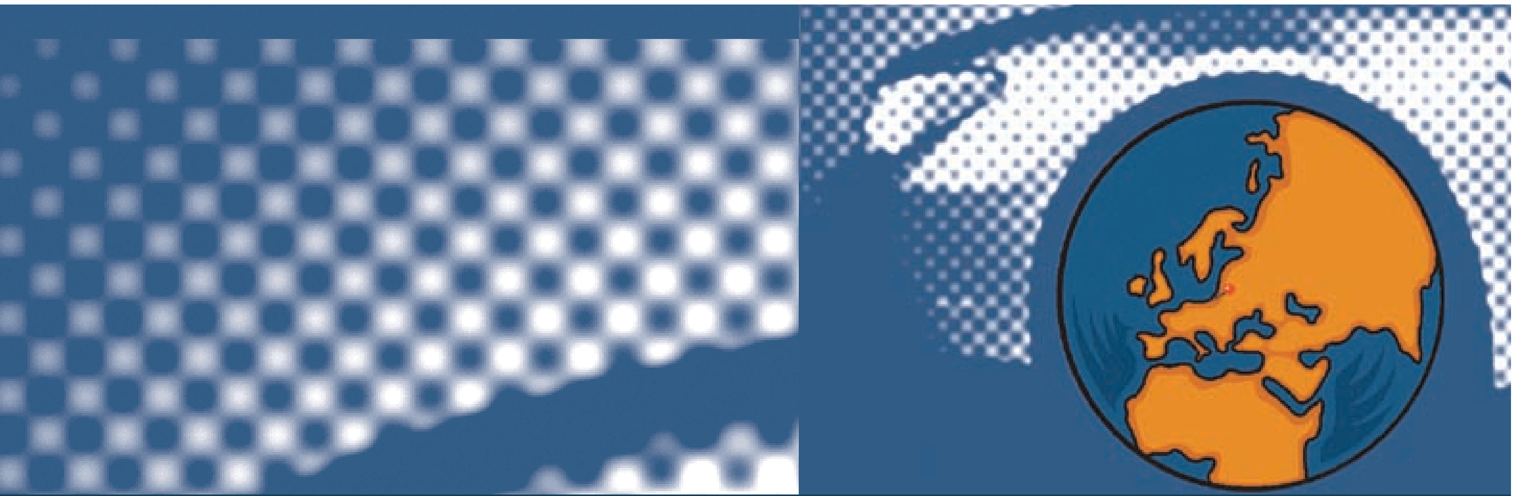

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GENERATION OF THREE-DIMENSIONAL INGREE/EGRESS PATHS INCLUDING CORRESPONDING LOAD SPECTRUMS

The ingree/egress process considerably stresses the car seats during their life time. Continuous endurance tests are to make sure that long time requirements regarding function and visual appearance of the seats are met and that any proposals regarding an improvement will be considered in the construction work. In order to reach an ingree/egress process as realistic as possible Bertrandt analyses efficiency and significance of biomechanical processes, verifying those by field observations.

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DEVELOPMENT PROCESS

From the beginning of the automotive development more and more models have been offered on the market. Reason for this fact are continuously changing individual requirements and needs of the customers. Also the dynamics of the models shows increasingly shorter life cycles as well as shorter development times making high demands towards the automotive manufacturers and sub suppliers changing the product engineering process of the product including higher quality requirements and lower development costs.

The development process is one of the most important factors in the product engineering process. During this period the course is being set to observe applicable standards and laws and to assure the function of components and systems. However, during the product engineering process the available time for tests is steadily being reduced. It is thus important to involve external test labs which handle the actual problems and develop solutions accordingly. Also the engineering partner Bertrandt focuses on developing innovative test methods and techniques which meet the requirements and conditions according to the latest state of art, ❶.

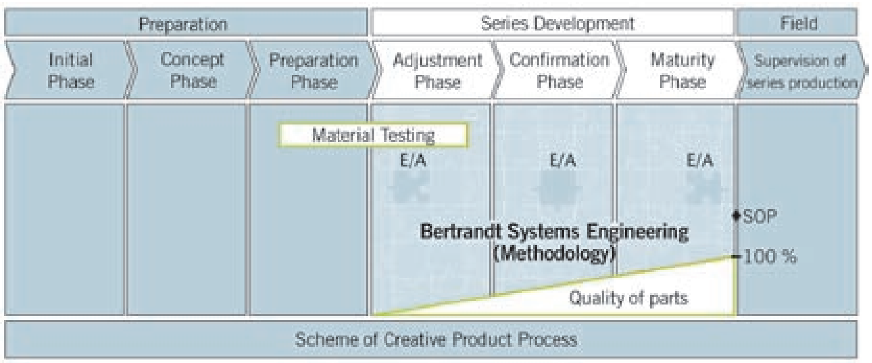
As an important part of the car development process the section „Testing“ includes life ageing and function tests of car components, modules and complete cars. Besides of the classical statical and dynamical life time tests in the interior of the car meanwhile also tests of the continuously inreasing add-ons, e.g. air conditioning and massage function, active head restraints or passenger classification systems are part of the fixed test programme due to the complex seat module. Considering the life time of the seat the ingree/egress process makes high demands regarding the mechanical properties. Although applying modern calculation and simulation proceedings still the real test at a component part, i.e. at the seat, plays an important role in the development process. Basically there is a difference between material testing during the preliminary stage, ①, and testing of components or modules during series development.

PRELIMINARY PHASE: MATERIAL TESTING

In respect to the occuring stress and the wear appearance during the preliminary phase material tests (called 2D-tests) of the cover material are most important at this stage. Among other factors rubfastness, elongation and abrasion resistance are being determined.

To determine the abrasion resistance and the pilling characteristics of textile materials the test method according to Martindale (DIN EN ISO 12947-1) is being applied. An orbital pattern is being moved towards a standard cloth simulating vertical stress. The axis of the pattern holding device, which is vertically located to the pattern area, is pivate mounted, so that a clearly translatory relative movement is given.

This method is not being applied for leather pads in cars. In this case the wear characteristics compared with the abrasion strain are usually determined by means of the Taber test device. During this test according to the friction gear



❶ Due to less available time during the development process Bertrandt offers new test methods meeting the market requirements integrating those into the product engineering process method two friction rollers push on the rotating pattern with a specified load.

All these material tests are general tests according to the regulations and requirements of DIN standards and technical delivery specifications. The actual load during the ingree/egress process is not really being considered. Much later during the creative process of the product, ①, ingree/egress test stands are applied, incorporating the interaction of all components and materials related to the complete seat system.

❷ Ingree/Egress robot

SERIES DEVELOPMENT: ROTATIONAL ALTERNATING LOAD TEST – INGREE-EGRESS-SIMULATION

The classical ingree-egress test is the rotational alternating load test. As shown in ❷ the movements of the test part as well as the stress of the seat are being generated via the dead load of the test part and pneumatic cylinders. The force is vertically being applied via the H-point of the seat shell on the seating area. The test body is simulated according to the human thigh/hip area and can either be constructed of one or three parts.

Whereas in case of the one-piece test body thigh and backside part build a consolidated strong shell the test body consisting of three pieces is devided into two thigh shells and one backside shell. This allows bending and stretching of the „thighs" around the H-point relatively to the backside shell

During the test with the three-piece seat shell the cycle is started with depositing the backside part on the seating area, whereas the „thighs" are standing parallely to the longitudinal axis of the car and afterwards also being deposited on the seating area. Afterwards the "thighs" are lifted, followed by pivoting the complete seat shell outwards around the vertical axis. To avoid any collision between test body and seat back rest the seat is simultaneously moving backwards. Now the side cheeks are being loaded by stretching respective up and down movement of the „thighs". The reverse motion sequence finalizes the test cycle.

As the „factor human being" is of great importance – besides of the car and seat specific parameters – the „classical" test stands quite quickly reach their limit. Using robot controlled seat test systems more flexible and more realisitc test processes are possible. For this reason Bertrandt applies the Occubot VI-system of KUKA Roboter GmbH, Augsburg, Germany, consisting basically of a processor with software, articulated arm robot, force/moment sensor and one-piece seat shell and thus meeting the high requirments of a flexible test process.

Owing to the mobility around all six robot axis, the possibility to determine the position of the seat shell and the force/moment measuring in the robot flange ingree/egress paths related to seat, car and persons can be realized. For more reality the SAE shell, which also simulates the human thigh/hip area, is covered with a gel mat.

Analog to a real ingree process the test starts with a slightly sloped depositing of the thigh at the shell in cheek area. The thighs are aslant to the seat. Then the robot turns the seat shell whilst moving along the cross and longitudinal axis around the vertical axis via the side cheek into the seat. Owing to the additional inclination around the longitudinal axis of the shell the actual seating position is reached. After a certain holding time the egress process is being carried out reverse.

An automatic adjustment of the ingree/egress path to the advancing damages of the foam part respective the seat is possible using a special software combined with the force/moment sensor, which is mounted at the robot flange. This guarantees a constant force level during the whole testing period.

CHALLENGE: BIOMECHANICAL COURSE OF MOVEMENT

When developing a seat the general problem arises that significant endurance tests regarding ingree/egress characteristics can only be carried out at a complete seat. The real wear appearance of the seat thus will show later on during the developing process. Any necessary considerable changes on the seat caused by the test results will certainly involve a great effort with high costs accordingly.

A special challenge for the test stands are those factors influencing the ingree/egress process. As already mentioned difficult and biomechanical individual processes like the ingree/egress process cannot satisfactory be simulated by the classical rotational alternating load test. Reason for this are the missing envelopes and also the poor programmability and controllability of the „classical" test stands.

Although test stands based on robots offer considerable advantages for these applications several challenges have to be kept in mind. The major problem is the

❸ Load situations not representable with the SAE test

transfer of the human biomechanical course of movement to the robot. Due to the physiological and geometrical deviations between human and robot dummy not all movements and load situations can be reproduced respective simulated, ❸. Special attention has to be paid to the human hip joint which enables a relative movement of both thighs towards the pelvis around each three rotatory envelopes. Whereas the real ingree/egress process goes on by separate movement of pelvis and thighs the seat shell can only be moved completely being pivoted around the projected H-point.

APPROACH: SYSTEMS ENGINEERING TO GENERATE INGREE/EGRESS PATHS

With these conclusions an important premise has been established in respect to the methodical development for generating ingree/egress paths at Bertrandt. The most important factor of the ingree/egress test was the biomechanical aspect. Placing the car seat and its wear characteristics into the center of a methodology – which is the usual procedure – became a subordinated factor. This is based on the fact that faults or wear appearance cannot be represented before all reasons are sufficiently known.

One example is parking in narrow garages or parking spaces when the small aperture angle of the door forces the driver or passenger to an unnatural course of movement during the ingree/egress process. This example shows that the ingree process starts with slipping the pelvis between door and side part. The person is standing backwards towards the car, the pelvis is located parallely to the door. When the pelvis is lowered into the seat simultaneously the torso is bent and a turning movement into the car is being made. After sitting down both legs are lifted into the car and the right knee is slided through under the steering column. The egress process is being made reverse. This causes a load situation different from that when the door is opened to a maximum extent. As this consequently leads to changed wear characteristics all factors influencing the ingree/egress process have to be considered and then integrated into a suitable methodology.

TEST INSTALLATION AND PERFORMANCE: DETERMINATION OF REAL PARAMETERS

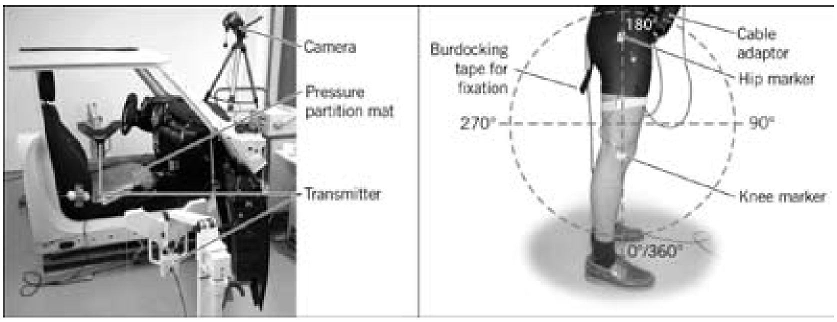
During tests with probands movements were recorded in several car and seat versions at Bertrandt. Depending on the seat versions basis, sports, multifunction and climate the main focus was put on

: parameters related to persons such as anthropometry (percentage 5 % female, 50 % male, 95 % male), biomechanics (natural/unnatural movement) and load spectrum and individual habits;

: parameters related to car and model such as high, normal or deep ingree version.

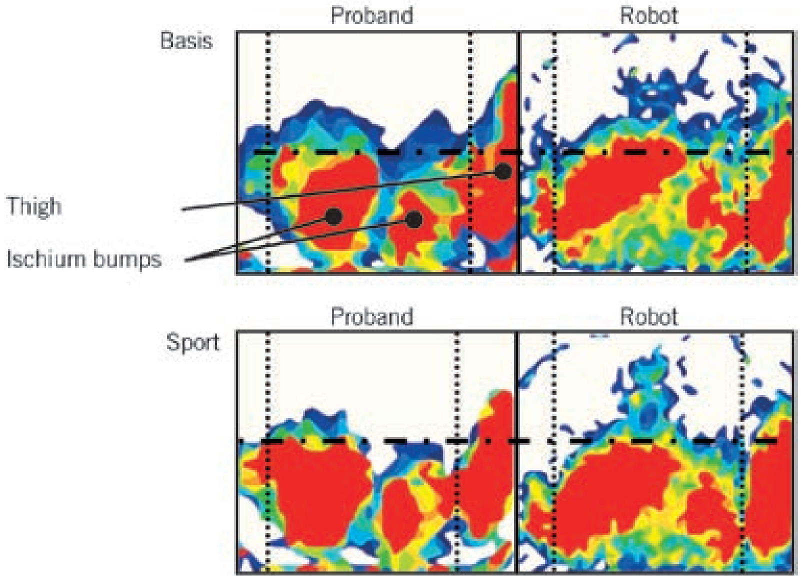
The movement was measured by means of Motion Capture with a 3D-ultrasonic measuring device of Zebris. The system is working with active markers installed at the body and sending out an ultrasonic signal, ❹. The signals are detected by two parallelly located ultrasonic microphones. Assisted by the triangulation position and movement within the room are determined. Another support also for documentation was a camera used during testing and measuring, which had been simultaneously connected with the movement and pressure data.

For movement measuring the statical and dynamical forces and moments (load spectrum) effecting the seat have simultaneously recorded. Measuring system was a pressure partition mat of Novel which was mounted on the seat, ④. In order to



❹ Preparation of test person

assure a certain statical significance at one hand and to keep the influence of fatigue as low as possible at the other hand the following test series were limited to max. 20 ingree/egress processes per test.

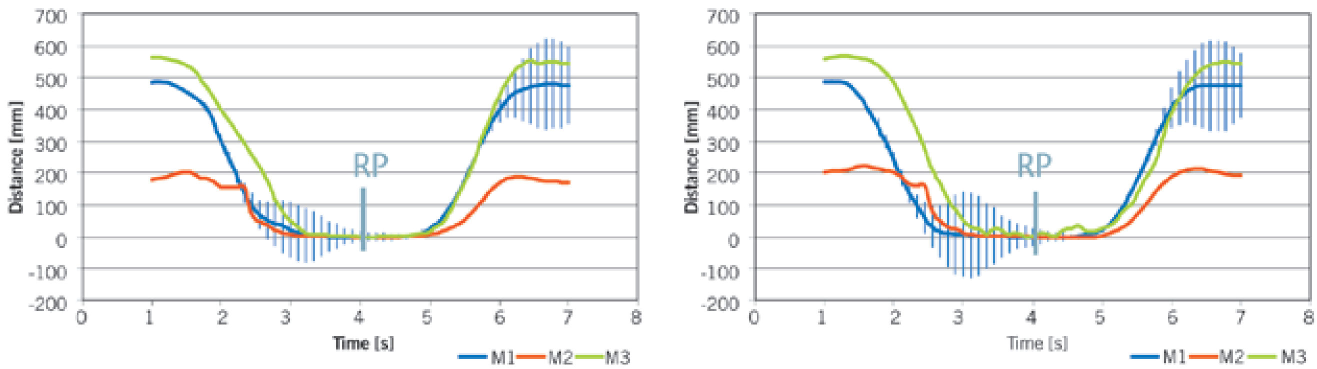


❺ Bertrandt methodology compared with test person/robot

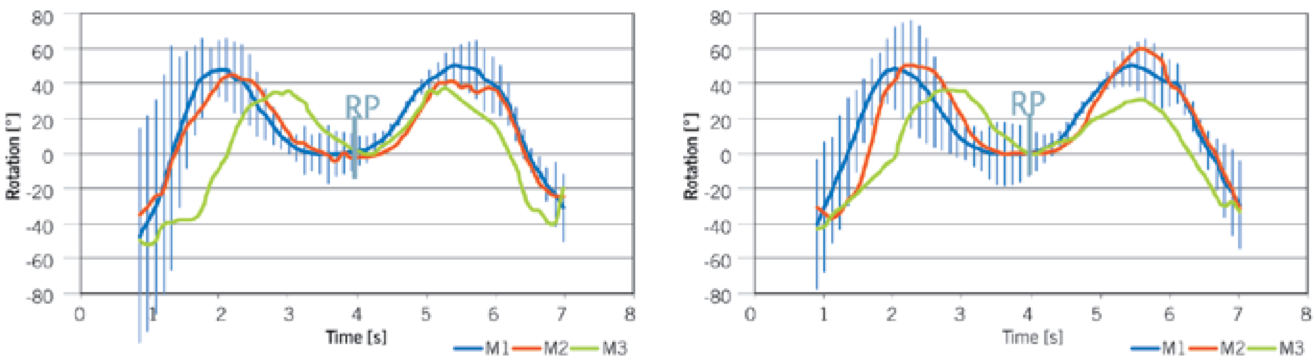
RESULTS EVALUATION: MOVEMENT PARAMETERS MEASURING OF PROBANDS

The movement to be measured was specified as course from start position (i.e. standing beside of the car) till rest position (the actual sitting) with following egress process. The main analysis focus for measuring the movement was put on to the chronologically averaged H-point depending on car and seat of the markers at the test persons (changed positions in X,Y and Z) as well as on the orientation of the pelvis (rotating around X, Y and Z axis), ❺.

❻ partly shows the results of the movement measuring path via time as a comparison between basis and sports seat. The course of the M1 line describes a normal ingree process, M2 a high ingree and M3 a deep ingree process. It shows that the changed position of the H-point is directly depending on the seat versions and car models. The phase left of the rest position in the diagram shows the ingree process, the phase in right area the egress process out of the car.



❻ Course of movement H-point into Z-direction at basis seats (left), course of movement H-point into Z-direction at sport seats (right)



❼ Orientation A (around Z-axis) at basis seats (left), orientation A (around Z-axis) at sport seats (right)

Completed are the results with those of the rotation around the individual axis, exemplary shown for Z-axis in ❼. This describes the course of the pelvis position relatively to the cross axis of the car, i.e. at an orientation A of zero degrees the pelvis is located parallely to the car cross axis. This also approves that the orientation directly depends on the seat versions and the car models.

Evaluation criteria for the load analysis were the averaged pressure courses of the individual tests over a certain time. Characteristically appearing maximum values in the curve runs have been compared related to car and seat.

Considering the time course of the resulting total load, three significant maximum values stand out, ❽. These maximal values differ depending on car and seat type in their temporally appearance as well as in their extent. In general the maximum values of the sports seats were above those of the basis seats.

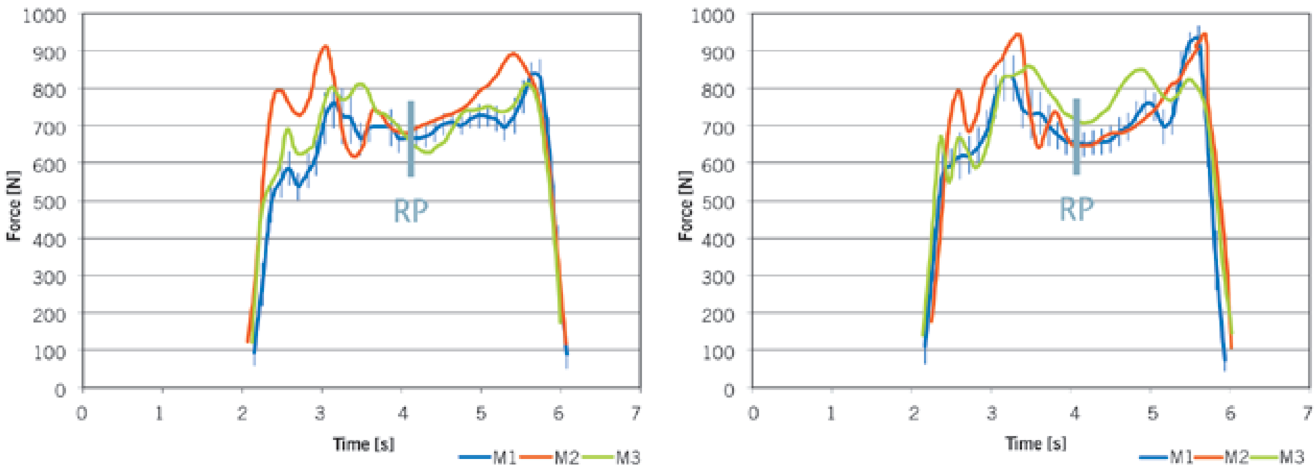
With these new determined „biomechnical data sets" a methodology for generating 3-D ingree/egress paths has been developed. This enables to generate paths nearly similar to the real load, ⑧.

It has to be pointed out that the load profiles in general approach to the test programms applied up to now. Use could be made of this fact during the temporary course and the extent of the charcteristcally determined maximum values for the component course of the seat mirror and the right side cheek as well as the resulting complete force course. Comparing the main load zones in the seating area the new generated robot curve could considerably be improved in the area of the ischium bumps and the thighs.

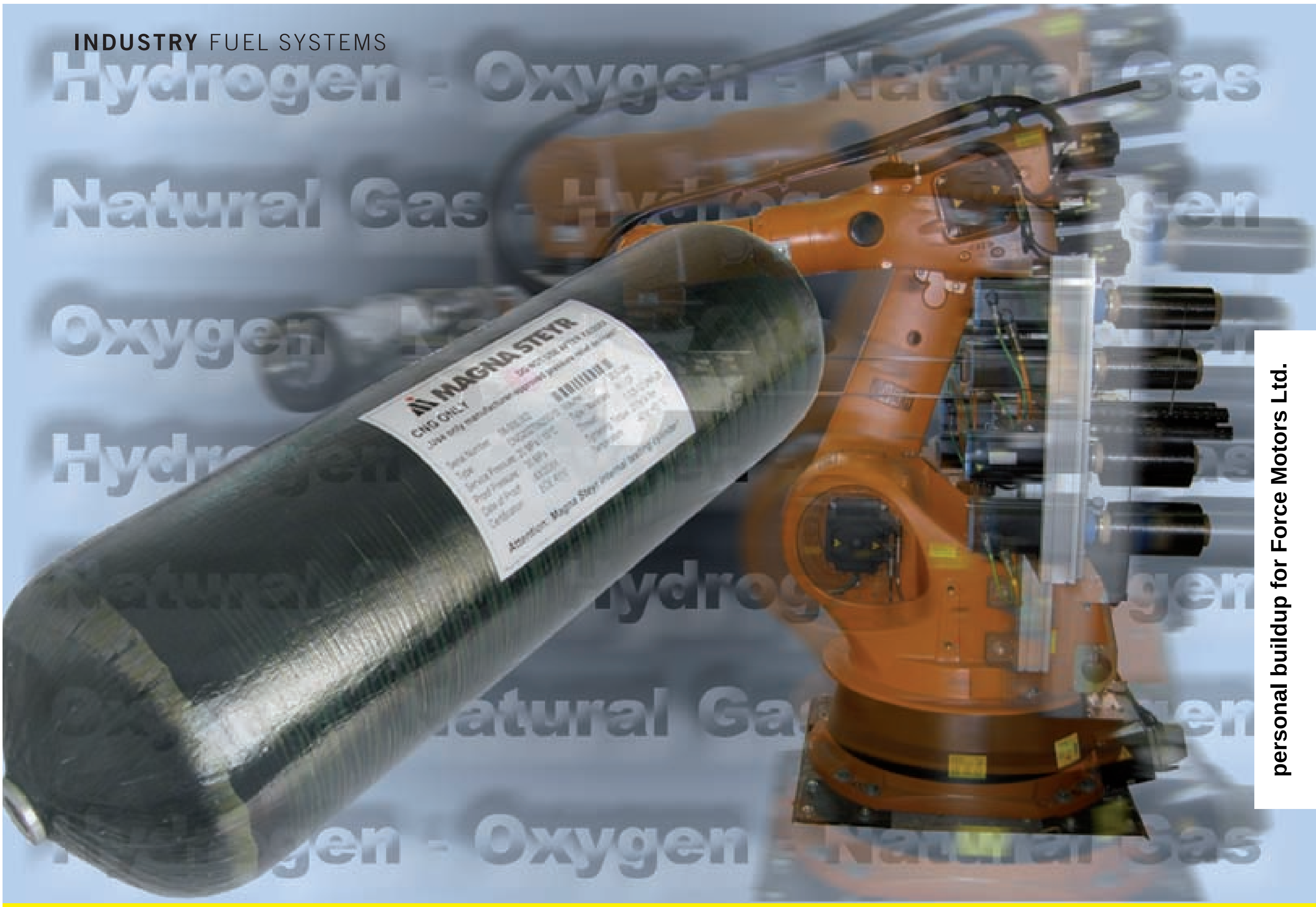
CONCLUSION AND OUTLOOK

The tests showed that the traditional test methods for simulating the ingree/egress process cannot sufficiently and realistically demonstrate the life age behaviour of car seats. Due to the combination of robotics, biomechanical basic tests and the parameters related to car and seat Betrandt was able to develop a test methodology demonstrating realistic wear appearance pictures at car seats caused by ingree/egress processes. This method considers front as well as back seats and is suitable for series development, supervision of vehicle series production and demonstration of faults, e.g. in case of the ingree wrinkling.

The tests and their results also provided interesting approaches for a methodical enhancement. The engineering staff of Bertrandt developed a methodology to be able to quickly demonstrate surface wear appearance at car seats under climate and humidity conditions. With this methodology significant results regarding material choice, appearance and workmanship of cover and the resulting extent of faults have been found. Currently Bertrandt is developing a mobile biomechanical test dummy replacing the static test body and to reach a further approach to reality.



❽ Complete force-time course at basis seats (left), complete force-time course at sport seats (right)



INNOVATIVE HIGH PRESSURE CYLINDERS FOR AUTOMOTIVE APPLICATIONS

Natural gas or hydrogen powered vehicles require special equipments for energy efficient storage of these gaseous fuels. A preferred option is the high pressure storage. Up to now heavy steel cylinders have been used in the pressure range up to 20 MPa due to their low price. Because of weight, a tendency to composite cylinders, made from carbon fiber and thermosetting resins, becomes apparent. This type of cylinder allows working pressure of 70 MPa and greater. Magna Steyr develops composite cylinders with non-load sharing plastic liners for alternative gaseous fuels in passenger cars.

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SPECIFICATIONS

In terms of an applied development, all mechanical and thermal requirements for composite cylinders, ❶, have to be harmonized with customer specifications in order to assure a safe operation at minimum weight and minimum costs. Technologies which have been subjected to aerospace industry up to now shall be transferred to the low cost segment of automotive technology without reduction in quality and safety. All cylinder components have to withstand the occurring pressure loads during operation over the lifetime up to 20 years. The leakage rate and permeation rate may not exceed the specified limit values, so that an amount of hazardous gas never will accumulate in the surrounding of the cylinder.

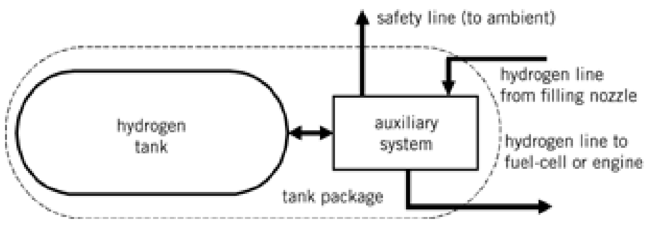
Similarly, in case of a failure, that means malfunction of safety relevant components, an accident or a rupture of the cylinder may not occur. Already in the design phase the right choice of materials has to be stated. All the fuel contacting materials have to be resistant against fuel and environmental impact. Significant for upcoming mass production is the availability of materials and its recyclability. In order to pass the benchmark with high pressure steel cylinders, energy efficient manufacturing technologies of the used materials and a high degree of automation is a crucial factor for the capability of mass production.

STATE-OF-THE-ART

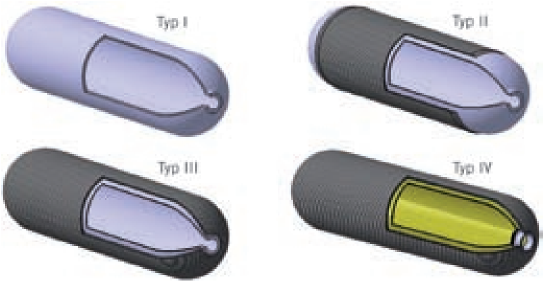
Over the last decades four typical types of high pressure cylinders have been established, ❷. The cylinders vary as well in design as in used materials. A distinction is drawn between steel or aluminum cylinder – as well without composite wrap (type I) as hoop wrapped (type II). A considerable reduction of weight can be achieved by an extensive substitution of metals by composite. Type III cylinders have a metal liner and a full composite overwrap. Type IV cylinders have also a full composite overwrap but a non-load sharing plastic liner.

For the automotive storage of compressed natural gas (CNG) the high pressure cylinders will be designed according state-of-the-art for working pressures up to 20 MPa at ambient temperature. By applying compressed gaseous hydrogen (CGH2), the working pressure is set to 70 MPa in order to reach high energy densities concerning acceptable cruising ranges.

The use of high pressure cylinders (type I) is based on long term experience in design and mass production as well as in operational lifetime and safety. This type fulfills per definition the specified values of leakage rate and permeation rate. Steel cylinders (type I) with 20 MPa working pressure have a mass-to-vol-



❶ Main components of a hydrogen storage system [2]



❷ Four typical types of high pressure cylinders as they have been established over the last decades

ume ratio between 0.8 and 1.1 kg/l. The price-to-volume ratio for adequate units is between 5 and 7 Euro/liter. Steel cylinders have cost benefits in mass production if weight is not a key aspect. Therefore, type I cylinders are mainly used for storage of natural gas which is state-of-the-art for automotive applications.

TREND OF DEVELOPMENT

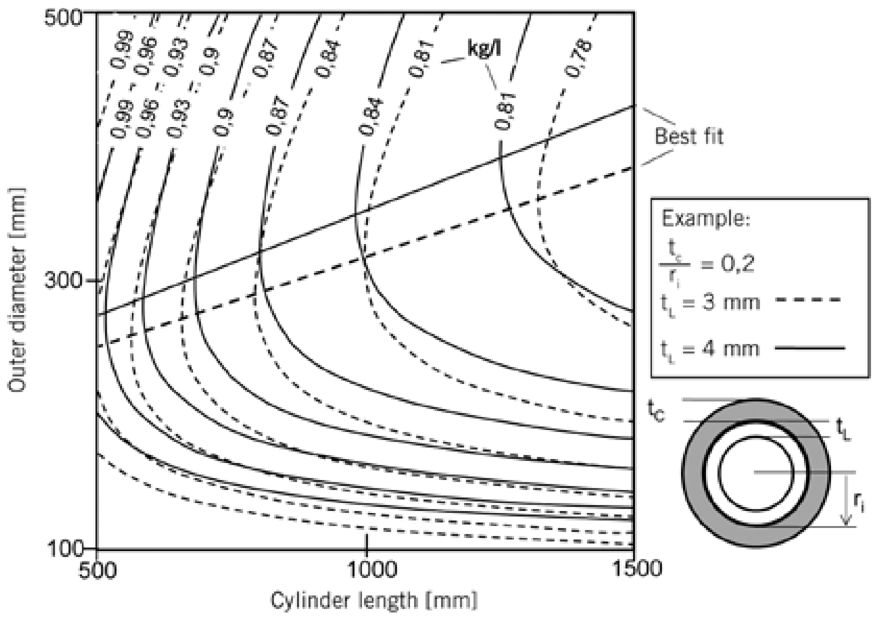
An improvement of competitiveness compared to conventional vehicles powered by gasoline or diesel and other alternative fuel storage technologies requires a significant expansion of cruising range greater than 500 km at lower costs. However, in the segment of passenger cars the effect of manufacturer specific weight restrictions will be applied increasingly, for example the maximum load per axle or restriction for compliance with inertia weight class.

This reason requires a reduction of the mass-to-volume ratio below 0.5 kg/l for high pressure cylinders with a working pressure of 20 MPa. Only type III and type IV cylinders are able to fulfill this requirement, whereas type IV cylinders have the greatest potential to reduce weight and cost. These cylinders are made of a composite wrap, mainly consisting of carbon fiber or fiber hybrid and a thermosetting matrix (for example epoxy resin). The composite shell carries the pressure load, while the thermoplastic liner adopts the function of the permeation barrier for the fuel. According to valid standards, the permeation rate must not exceed the limit value for the specified fuel over lifetime. This requires liner materials with excellent properties in permeation, impact strength, ultimate strain, low knockdown of ultimate strain as a consequence of extreme temperature and fatigue, stiffness, and heat conductivity. For state-of-the-art applications, the selection leads to available thermoplastic materials based on high density polyethylene (HDPE) as a compromise of a goal conflict, since the requirement for low permeation of plastics have strong impact on mechanical properties. Anyhow, this compromise is insufficient in order to satisfy the automotive requirements. Fundamental research and advanced product development are essential, since still new or unknown phenomena retards a mass production.

Based on valid standards for the certification of high pressure cylinders the load-carrying composite shell has to be designed for high reliability in order to withstand sustained load and cyclic load. Increasing the reliability under sustained load requires the consideration of static strength degradation of composites as a consequence of creep rupture. This property addresses mainly the definition of the safety factor. Moreover, the amount of used fibers and the corresponding expenditure is going along with the level of the safety factor. For a successful convey of this storage technology into the low-cost segment of the automobile technology, it is necessary to follow especially two aspects. First, essential adjustments of regulations regarding automotive applications are necessary, whereas safety must not be affected. Currently, a reduction of the safety factor for 70 MPa type IV cylinders from 2.35 down to 2.0 is in discussion. Secondly, inherent scatter of composites after manufacturing of high volumes has to be reduced to a minimum.

In order to realize the second aspect there is a need for new innovative production technologies which are able to guarantee a cost-effective high mass production, ❸. The wet winding process with thermosetting resins followed by the oven cure is state-of-the-art. Further developments of this technology are necessary and also their adaptations to the new requirements. This means a significant reduction of process duration and effort (increasing the degree of automation), providing a continuous high-quality impregnation of fiber rovings and increasing of the precision of fiber roving placement on the mandrel.

❸ Production of high-pressure storage systems



❹ Correlation between cylinder length and outer diameter exemplary for a 70 MPa cylinder in comparison for two different liner wall thicknesses

The impregnation quality is a significant factor for the process stability and for the burst pressure consistency. Thereby, the production tolerances play a decisive role in the design and optimization of thick-walled cylinders compared to 20 MPa composite cylinders. Additionally the influence of design specific manufacturing parameters must be balanced with the three dimensional structural mechanical aspects of the composite shell [1]. The innovative robot winding technology for winding of thick-walled composite cylinders is currently in a pilot stage.

FURTHER OPTIMIZATION POTENTIALS

An essential cost reduction potential provides the appropriate selection of cylinder dimensions under the consideration of optimal package. The mass-to-volume ratio (kg/l) of a high pressure cylinder can be improved with increasing cylinder length for constant outer diameter. Especially in case of type IV cylinders there is one particular outer diameter for each cylinder length with regard to optimal mass-to-volume ratio. Different design criteria for the total wall thickness of a type IV cylinder explain this phenomenon. The wall thickness of the load-carrying composite shell is defined via inner radius for a defined design pressure and material in contrast to the liner, where the permeation issue is a key aspect of liner wall thickness determination. These correlations are presented in ❹ exemplary for a 70 MPa cylinder (t_c/r_i = 0.2) and in comparison for two different liner wall thicknesses. The outer diameter is plotted against the cylinder length for different mass-to-volume ratios. The family of curves and the resulting best fit shifts in the direction specified by the function of the liner wall thickness.

According to experience there may be a number of possible suitable dimensions for a certain package which match the required useable gas mass. Against the background of the cost reduction an appropriate optimization of the mass-to-volume ratio must be conducted. Generally, the use of long cylinders should be preferred in vehicles. That would require potentially new vehicle platforms.

SUMMARY AND OUTLOOK

Magna Steyr develops composite cylinders with non-load sharing plastic liners for alternative gaseous fuels in passenger cars. Natural gas or hydrogen powered vehicles require special equipments for energy efficient storage of these gaseous fuels. A preferred option is the high pressure storage. Up to now heavy steel cylinders have been used in the pressure range up to 20 MPa due to their low price. Because of weight, a tendency to composite cylinders, made from carbon fiber and thermosetting resins, becomes apparent. This type of cylinder allows working pressure of 70 MPa and greater.

A pending serial production of composite cylinders poses a challenge not only for the design and production engineers. Also the purchasing and logistics have to be familiarized with the new technology and involved strongly in the design and manufacturing process. Specific requirements on components concerning material choice, production quality and tolerance of vendor parts have to be emphasized. New suppliers and manufacturers with know-how in this innovative technology have to be qualified apart from the well-known automotive supplier base.

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ESTIMATION OF THE OIL TEMPERATURE IN ADJUSTABLE VIBRATION DAMPERS

It can be presumed also for semi-active vibration dampers that a better considering of the oil viscosity can be improve control accuracy and vehicle behavior. The Baden-Wuerttemberg Cooperative State University (DHBW), the Elektronische Fahrwerksysteme GmbH, the Audi AG, and the FKA in cooperation with the Institut für Kraftfahrzeuge of RWTH Aachen University developed two simulation models. One was rather simplified for estimating the oil temperature in real-time on a control unit. The other model was built more detailed to understand the phenomena much better.

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FROM SIMPLIFIED TO DETAILED MODEL

This combined contribution between the Baden-Wuerttemberg Cooperative State University (DHBW), the Elektronische Fahrwerksysteme GmbH, the Audi AG, and the FKA in cooperation with the Institut für Kraftfahrzeuge of RWTH Aachen University, presents a model-based method for estimating the oil temperature of vibration dampers. This article is subdivided into the following sections: First of all it will address the basic principles of a vibration damper. Then two models will be described; one of them is a rather simplified version suitable for estimating the temperature in real-time on a control unit, and the other one is more detailed and applicable to a precise simulation. Finally, consolidated results will be summarized. Benefits and disadvantages will be discussed with regard to a possible implementation into a series control unit.

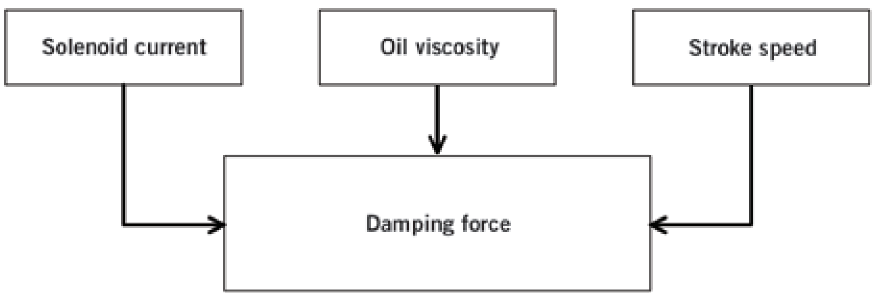
A large number of strategies for semi-active damper controls can be found in the literature, for example in [9, 10, 11]. These strategies provide an improvement of comfort and driving safety by the implementation of semi-active dampers, based on the Sky-Hook control algorithm or alternative approaches. These approaches are predominantly investigated using quarter-car up to full-scale vehicle simulations. Although the functional adjustable damper is represented by some time delay or a quantification of the damper force-velocity characteristics during these simulations, the temperature-dependent oil viscosity is hardly taken into account in these papers. The resulting damping force F_D of a semi-active vibration damper is not only determined by the damper piston speed v_D and the valve characteristic, which is adjustable because of the current feed I_S, but is also dependent on the oil viscosity ν, ❶.

Publications describing passive vibration dampers with fluid-dynamic models often take the oil viscosity into consideration. The purpose of such models is to determine the damper force by the piston speed and valve characteristics, for example, for computer-based simulations. In these publications the damper force is calculated from piston speed and valve characteristics installed in the damper, also paying attention to the thermal qualities of the damper oil (see for example [3, 4, 5]). The oil temperature strongly influences the oil viscosity and consequently the damper force to be determined. That means, the temperature is usually taken into account by a thermodynamic model within the fluid-dynamic model.

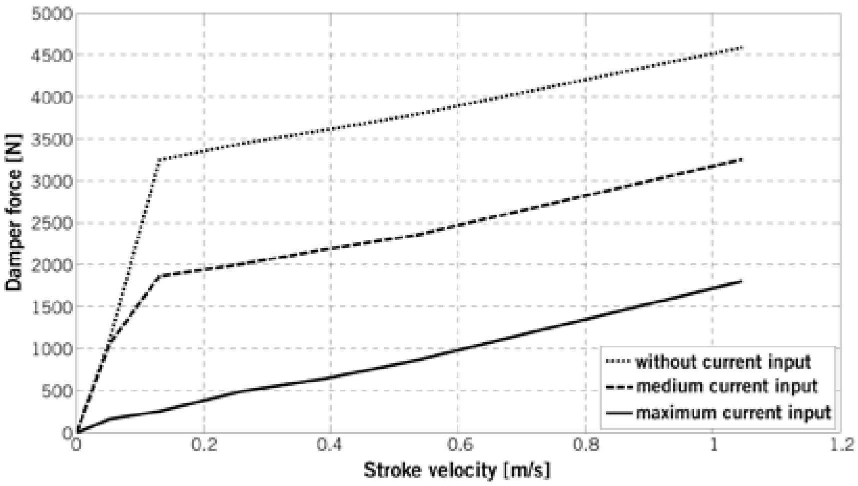
Most papers concentrate their investigation on passive vibration dampers. However, it can be presumed that considering the oil viscosity for semi-active vibration dampers, control accuracy will be improved and a more precise simulation of vehicle behavior will be possible.

The description here shows that the hydraulic and the thermodynamic properties of a semi-active vibration damper can be represented quite well by a detailed model (multi-zone model). For the establishment of this model the vibration damper has been divided into different subsystems. Afterwards the enthalpy equation was built for each subsystem. The simulation results were compared and validated with regard to measuring data recorded on a test bench. The results show that the temperature behavior of the vibration damper can be reproduced very well with this model.

Afterwards this model as one-zone model has been extremely simplified to enable an implementation on a series control unit. Even this simplified model



❶ Influencing parameters on the damper force of a hydraulic and semi-active vibration damper



❷ Example of a damper force / stroke velocity characteristic (rebound movement) of an adjustable vibration damper for different current inputs

shows an adequate agreement with the actual damper behavior, but it would not reach the quality of the detailed model.

SEMI-ACTIVE VIBRATION DAMPER

The obvious function of a vibration damper is to dampen any vehicle vibrations produced by road excitations or driver steering input. For this purpose the damper should fulfill the following tasks (see for example [1]):

: low body acceleration (for increased driving comfort)
: low dynamic wheel load (for increased driving safety)
: adequate isolation effect even for minor excitations.

When selecting a damper, its operating parameters have to be established. The damping factor adjustment, however, leads to a conflict between driving comfort and driving safety. This is due to the different natural frequencies of the vehicle body and the wheel, which will inevitably lead to a compromise between the damping requirements of body acceleration and dynamic wheel load, when applying a fixed damping factor only. A body damping rate between 0.25 and 0.35 (see for example [1]) is considered a good compromise.

In order to dampen both body and wheel loads efficiently, today semi-active vibration dampers are used in many applications. These types allow for an adjustment of the damping factor by using a valve with a variable current feed. In order to meet the above-mentioned demands for good vibration damping conditions in a vehicle, a combination of a hydraulic damper with a variable solenoid valve in the piston produces different force-velocity characteristics. ❷ shows an example of a force-velocity characteristic diagram of an adjustable vibration damper for a rebound movement.

MODELING

This Section explains the development of thermodynamic models that can be employed to estimate the oil temperature in vibration dampers. The basic idea of these models is to describe the vibration damper mathematically using differential equations based on energy and enthalpy balances. The integration of these differential equations then ensures, inter alia, the current temperature of the vibration damper.

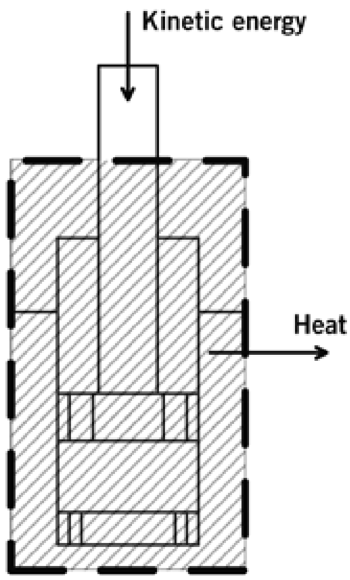
At the beginning of the modeling process, fundamental simplifications of physical reality are made before starting to design the temperature models. Within the scope of these investigations, two different models were developed. This was done with the objective to be able to evaluate the model accuracy. This active cooperation led to a one-zone model and a more complex multi-zone model.

ONE-ZONE MODEL

The one-zone model is based on the fundamental energy balance of the vibration damper. The entire vibration damper is taken as only one system, and local temperature differences inside the vibration damper are neglected. Thus, it is regarded as a “Black Box” system. This assumption is justified, since the vibration damper during operation shows a well-mixed system. The piston movement causes the oil to be blended continually and thoroughly in the interior, so that only small local temperature differences can develop. Therefore, temperature variations can be neglected, that means the vibration damper can be approximated as a specific block capacity (see for example [8]). ❸ shows the one-zone model.

Only input and output energy flows are considered. On the one hand, the kinematic piston energy means energy input flow caused by the road excitation. On the other hand, the energy output flow is the heat dissipation from the damper to the ambient air. The differences between energy input and energy output flows lead to energy changes and consequently to temperature changes of the vibration damper. The energy balance of the one-zone model can be determined by the first law of thermodynamics as shown in Eq. 1. For explanation of symbols see ❹, the index D means damper, L the air.

The power P_D absorbed by the vibration damper can be determined by the force-velocity characteristics (F_D-v_D characteristics). The F_D-v_D characteristics not only depend on the piston stroke speed v_D and



❸ System borders and energy flows using the one-zone model the current input I_S, but are also influenced by the oil temperature T_O. This relates to Eq. 2.

MULTI-ZONE MODEL

This paragraph describes the multi-zone model and the assumptions applied in this thermodynamic model, including the differential equations introduced for this purpose. The one-zone model simulates the temperature of the vibration damper with just one local temperature which represents only a rough and simplified model of reality. Various oil chambers, different materials (oil and steel), and different centers of heat generation result in obvious temperature differences between various zones of operation.

This leads to the consequence of subdividing the system into smaller sections to enhance the accuracy estimating the oil temperature, and also to determine more exactly the temperature distribution in the vibration damper. Therefore, the entire vibration damper system is divided into eight subsystems, as shown in ❺.

Eq. 3 describes the relevant parameters of system 1 (upper working chamber), see also ④. The value $\dot{m}_K$ stands for the oil flow through the piston.

The mass m_1 as well as the temperature T_1 are time dependent. This results in two terms on the left side of the Eq. 3 executing the derivation of time of the internal energy $\dot{U}_1$. On the left hand side of the Eq. 3 the variable ξ is used to indicate the damper strokes (rebound or compression mode) of the piston. If $\xi = 1$ the damper piston is carrying out rebound stroke ($v_D > 0$), and if $\xi = 0$ the damper piston is in compression stroke ($v_D < 0$).

If system 1 is in rebound stroke ($v_D > 0$), the variable ξ is equal to 1 (one) and the second term on the right hand side of Eq. 3 becomes 0 (zero). That means that the enthalpy flow leaves the control volume 1 through the piston valve into control volume 3. During the compression movement the first term on the left hand side of Eq. 3 results in 0 (zero), since the enthalpy flow is fed from the control volume boards of system 3 into system 1.

Hence, the Eq. 3 can be simplified to Eq. 4 for rebound conditions. The new introduced variables are explained in ④. The temperature changes of system 1 during rebound movement depend only on the heat dissipation to the surrounding systems.

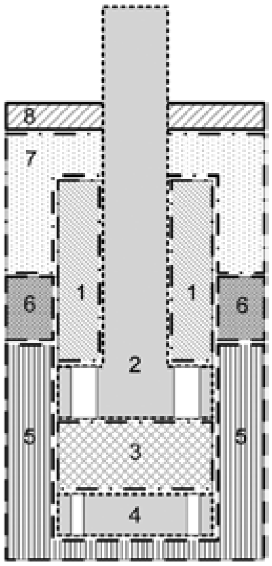
For compression movement Eq. 5 can be derived from Eq. 3. During this movement the internal energy of system 1 is influenced by the internal energy coming from system 3. Furthermore, the oil flow absorbs energy when flowing through the valve caused by friction effects. This dissipated energy results in heat and is described by the second term on the right hand side of the Eq. 5. This term is the comprehensive description of the hydraulic power loss P_h. The power loss is calculated by multiplying the pressure difference Δp of one valve and the volume flow

FORMULARY

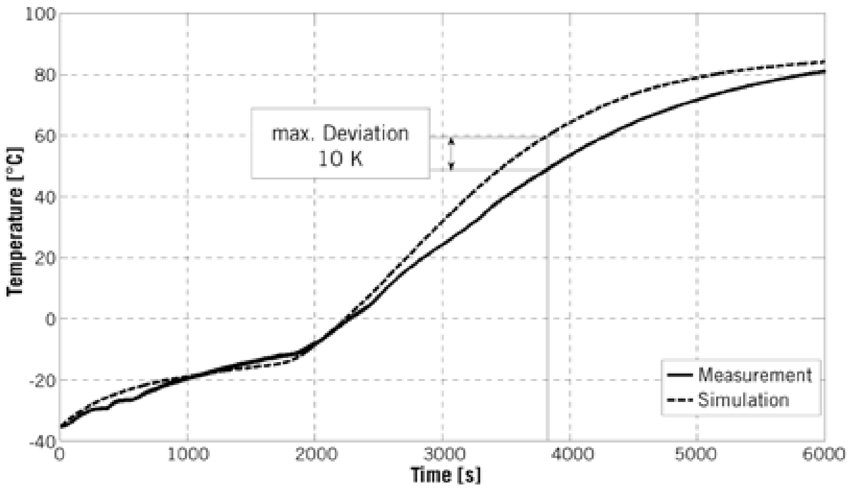
EQ. 1	$\dot{U}_D = P_D + \dot{Q}_{D-L}$
EQ. 2	$P_D(v_D, I_S, T_O) = F_D(v_D, I_S, T_O)v_D(I_S, T_O)$
EQ. 3	$c_1 m_1 \dot{T}_1 - \dot{m}_1 c_1 T_1 = -\xi \dot{m}_K h_1 - (1-\xi)\dot{m}_K h_3 + p_1 \dot{V}_1 - \dot{Q}_{1-2} - \dot{Q}_{1-6} - \dot{Q}_{1-7} - \dot{Q}_{1-8}$ with $m_1 = \rho_O l_1 (A_K - A_S)$, $\dot{m}_1 = -\rho_O v_D (A_K - A_S)$, $h_1 = c_1 T_1 + \frac{p_1}{\rho_O}$, $h_3 = c_3 T_3 + \frac{p_3}{\rho_O}$
EQ. 4	$\rho_O l_1 (A_K - A_S) c_1 \dot{T}_1 = -\dot{Q}_{1-2} - \dot{Q}_{1-6} - \dot{Q}_{1-7} - \dot{Q}_{1-8}$
EQ. 5	$\rho_O l_1 (A_K - A_S) c_1 \dot{T}_1 = \rho_O v_D (A_K - A_S) c_O (T_1 - T_3) + v_D (A_K - A_S)(p_1 - p_3) - \dot{Q}_{1-2} - \dot{Q}_{1-6} - \dot{Q}_{1-7} - \dot{Q}_{1-8}$
EQ. 6	$P_h = \Delta p \dot{V}_V$

A_K, A_S	Cross-section of piston
c_i	Specific thermal capacity of system i
c_O	Specific thermal capacity of oil
F_D	Damper force
h_i	Enthalpy flow into system i
I_S	Coil current
l_i	Length of system i
m_i, $\dot{m}_i$	Mass, mass flow of system i
$\dot{m}_K$	Mass flow through piston valve
p_i	Pressure in system i
Δp	Pressure difference
P_D	Power absorbed by shock absorber
P_h	Power loss
$\dot{Q}_{i-j}$	Rate of heat transferred energy from system i to system j
T_i, $\dot{T}_i$	Temperature and its derivation of time of system i
T_O	Oil temperature
U_i, $\dot{U}_i$	Internal Energy and its derivation of time of system i
V_i	Volume of systems i
$\dot{V}_V$	Volume flow through valve
v_D	Damper velocity
W	Work
ν	Oil viscosity
ξ	Variable for indication of rebound or compression movement
ρ_O	Oil density

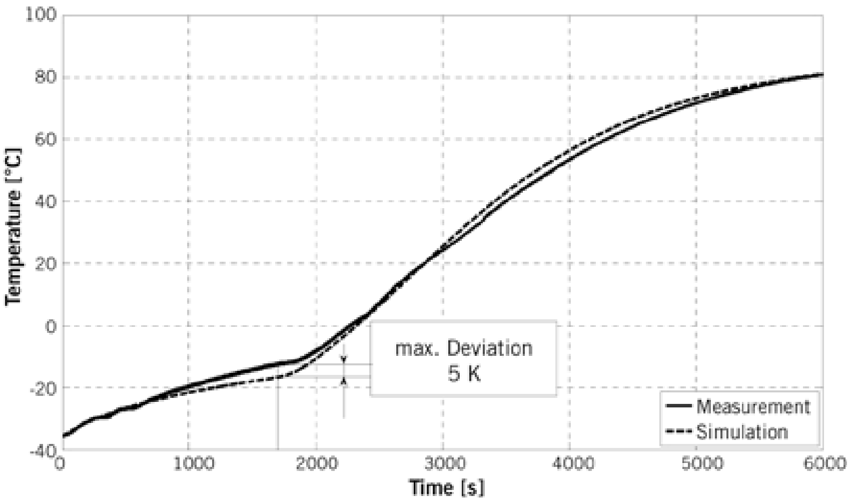
❹ Explanation of symbols



❺ Entire system vibration damper with eight subsystems: 1 upper working chamber, 2 piston and piston rod, 3 lower working chamber, 4 bottom valve, 5 lower compensating chamber, 6 upper compensating chamber, 7 air cushion, 8 sealing unit



❻ Measured and simulated oil temperature curves of a vibration damper (one-zone model); I_S = 0 mA, v_D = 0.052 m/s at ambient temperatures from -35 to +80 °C



❼ Measured and simulated oil temperature curves of a vibration damper (multi-zone model); I_S = 0 mA, v_D = 0.052 m/s, at ambient temperatures from -35 to +80 °C

$\dot{V}_V$ through this valve as described in Eq. 6. In addition to that, heat transfer to the surrounding systems takes place.

Thus, the upper working chamber in rebound and compression modes is fully described by Eqs. 4 and 5. For the other subsystems shown in ⑤, the modeling approach is similar to Eqs. 3 to 5.

COMPARISON OF THE MODELS

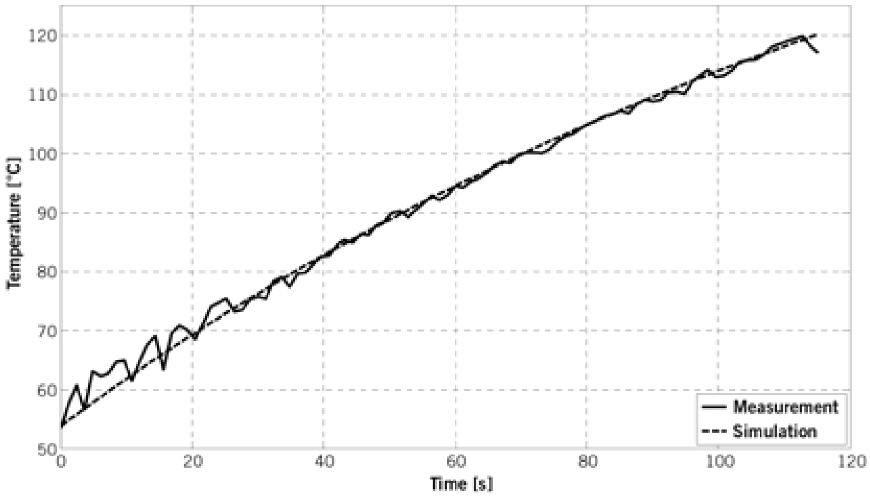
In the following the accuracy of both models is to be compared. Therefore, the one-zone model and the complex multi-zone model are shown during a heating-up cycle. The measured data are also given in the diagram. The heating-up curve as compared to the calculated temperature curve of the one-zone model shows a maximum estimation error of $\Delta T = 10\,\text{K}$, ❻. Using the more detailed multi-zone model the estimation error is not more than $\Delta T = 5\,\text{K}$. ❼ and ❽ confirm the accuracy of the multi-zone model in two different heating-up curves based on different excitations.

However, the multi-zone model needs a lot more computing capacity than the one-zone model. Hence, this model is not applicable on a control unit because of the high computing effort. Another disadvantage is the complex model parameterization requiring considerable computing time. In addition to this, several characteristic tables have to be provided which are to be adjusted to every new vibration damper by simulation on the test bench. However, each vibration damper requires individual curves for different operating temperatures, and the cooling rate of a vibration damper is very slow and time-consuming. Hence the multi-zone model is more interesting for systems requiring a high simulation accuracy, to be able to represent exactly the behavior of vibration dampers in a full-scale vehicle model.

By contrast the one-zone model is easy to parameterize. The computation effort is rather low, so that this model can easily be implemented on an automotive control unit. The model has an estimation error of $\Delta T = 10\,\text{K}$ lying within a justifiable range.

SUMMARY

The Baden-Wuerttemberg Cooperative State University (DHBW), the Elektronische Fahrwerksysteme GmbH, the



❽ Measured and simulated oil temperature curves of a vibration damper (multi-zone model); I_S = 0 mA, v_D = 1.047 m/s

Audi AG and the FKA Forschungsgesellschaft Kraftfahrwesen mbH Aachen in cooperation with the Institut für Kraftfahrzeuge der RWTH Aachen University analyzed the influence of the oil temperature on damper behavior. Furthermore, two different methods were shown to estimate the damper oil temperature. Therefore, two thermodynamic models were designed to estimate the operating oil temperature. Both models are different in their mathematical complexity, so they are recommended for different degrees of accuracy. The one-zone temperature model works along an accuracy of estimation within 10 K. The integration of this simplified model on an automotive control unit is easy, as well as its parameterization.

The multi-zone model is more complex. This model is more suited for computer-based simulations and provides an operational accuracy of estimation of less than 5 K. Apart from the estimated oil temperature to be presented; this model determines the overall temperature distribution in the vibration damper.

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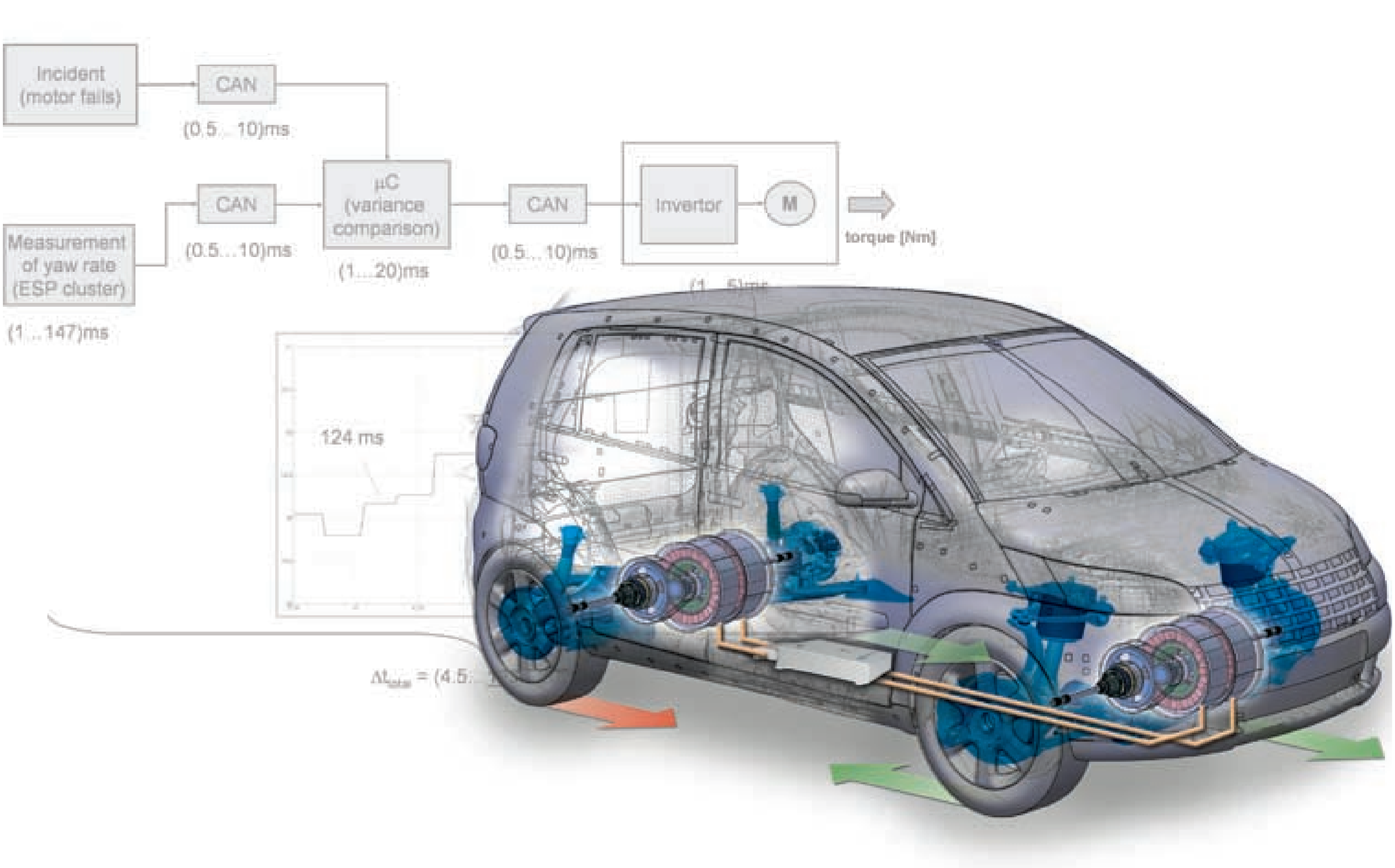

EVALUATION OF DRIVING SAFETY OF AN ELECTRIC VEHICLE DURING STEADY STATE CORNERING

The topic of this article is the assessment of the safety of an electric driven vehicle with four-wheel-drive regarding its dynamic stability. Evaluated at the FES GmbH in Zwickau (Germany) is the vehicle´s overall-dynamic behaviour in cornering, when either one of these motors fail. The results show that the vehicle reacts quite stable to the investigated failures except for situations in which the friction coefficient is very small or the lateral acceleration is very high.

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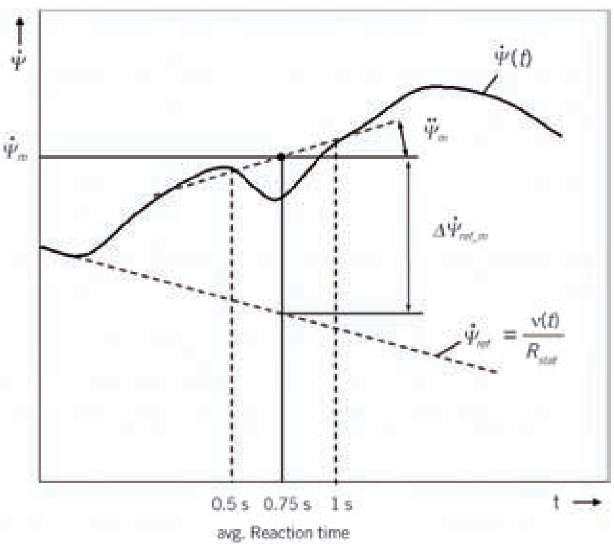


❶ Assessment-criterion for safety of a vehicles yaw reaction

STABILITY CONTROL SYSTEMS

From a chassis-engineering point of view the electric power train offers new possibilities for software operated stability control systems. By actively controlling the torque on one or more selected motors, a directed yaw moment can be applied to the vehicle. This type of control is known as torque vectoring. This principle has successfully been used in dynamic stability control systems, widely known is the Electronic Stability Program (ESP), which assists the driver in safety-critical situations to stabilise the vehicle and regain control of the situation. This system creates the correctional yaw moment by means of brake actuation on one side of the vehicle [1].

The vehicle under consideration for this study is a sports-car, driven by four separate electric motors. The motors are located inside the vehicle´s body and connected to the wheels by drive-shafts. This work explores the possibility to stabilise this vehicle via torque vectoring in safety-critical situations. In this case the critical situations are triggered by failing of one of the four electric motors.

DYNAMIC BEHAVIOUR UNDER SAFETY-CRITICAL SITUATIONS

This section describes the assessment of the vehicle´s safety under different failure-modes of the electric power-train. The possible failure modes that are investigated in this study are the three different cases:

(a) the motor increases its driving torque to its maximum value ($M = M_{max}$)
(b) the motor fails and shuts torque down to zero ($M = 0$)
(c) torque of the motor is redirected in the opposite direction, that is the motor brakes with maximum torque ($M = -M_{max}$).

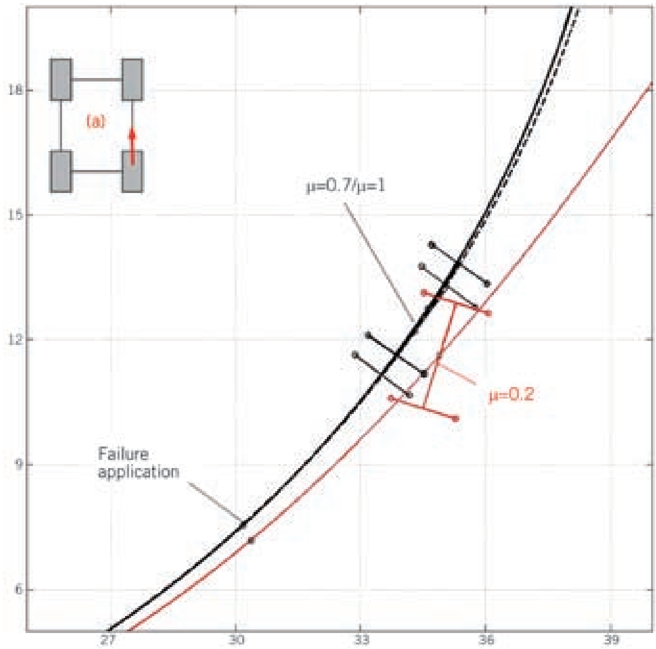
All of these failures are assumed to happen without any time-delay, that means the torque always changes in form of a step-function. The three failures (a), (b) and (c), can occur on each of the vehicle´s four wheels. In reality those failures could happen for instance due to severe malfunction of the frequency converter or due to a short in the system.

The failure-modes are simulated with a co-simulation-system of Matlab Simulink, which controls the electric power-train, and Adams Car, which simulates the vehicle´s behaviour. During steady-state-cornering manoeuvres the failures are applied separately on one wheel and in each case the reaction of the vehicle is monitored. In order to exclude any driver-related reactions, the manoeuvres are simulated open-loop, that is the steering-wheel angle is kept constant from the beginning to the end of each simulation.

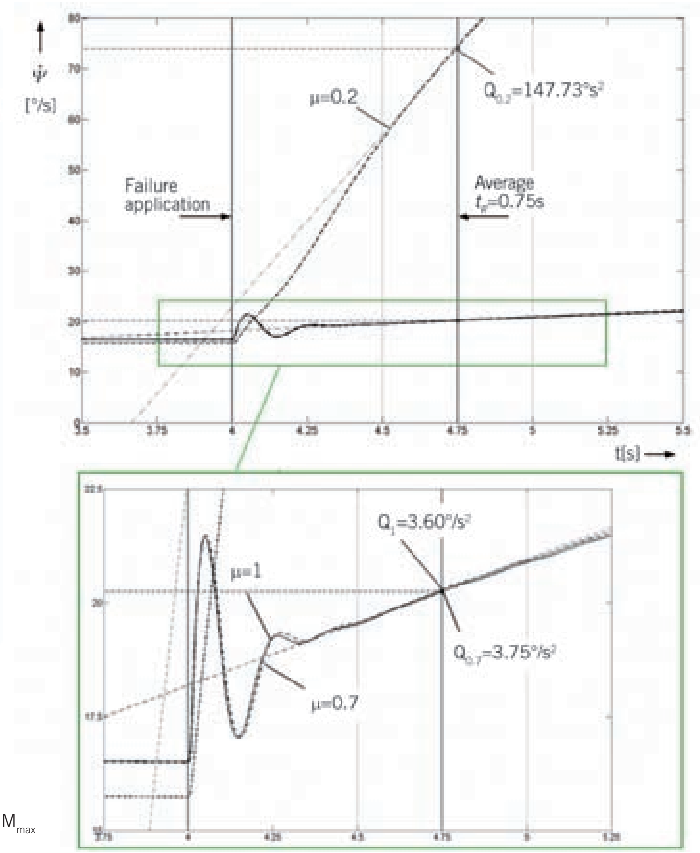
In order to decide whether the vehicle´s reaction to a failure is safety-critical, assessment-criterions for load changes during cornering developed in [2] are used. This approach is based on the evaluation of the yaw-rate difference at a reaction time of $t_R = 0.75$ s after failure-application – the 0.75 s being the average time at which a driver reacts to the situation by steering [2]. The criterion Q, ❶, contains the difference in yaw-rate, as well as the average yaw-acceleration at $t_R = 0.75$ s.

It has been shown that the vehicles yaw-reaction is to be considered critical, if the limit-value of $Q = 5°/s^2$ is exceeded [2]. If the criterion Q is less than 5°/s, that is the yaw-reaction of the vehicle can be handled by the driver; the course deviation of the vehicle from its original path, Δy, should be used as criterion to assess the safety.

The simulations are carried out with three different static lateral accelerations: $a_y = 0.23g$, $a_y = 0.5g$ and $a_y = 0.9g$, which is approximately the cornering-limit of the investigated vehicle. In addition, the simulations have been carried out for $\mu = 1$ (dry-road), $\mu = 0.7$ (wet-road) and $\mu = 0.2$ (icy-conditions), where the latter



❷ Trajectories at a_y=0.23g to failure (a) +M_{max} of rear outer wheel



❸ Yaw-reactions at a_y=0.23g to failure (a) +M_{max} of rear outer wheel

friction-coefficient could only be applied for the smallest lateral acceleration of 0.23g, as this is the cornering limit of the vehicle at $\mu = 0.2$.

❷ shows exemplary the resulting trajectories of the vehicle to failure-mode (a), that is increase to maximum torque at the outer rear wheel, at $a_y = 0.23$g and R = 30 m along with the vehicle´s positions at t = 0.75 s after failure-application. The corresponding yaw-reactions of the vehicle are displayed in ❸. It can be seen that the influence of the friction-coefficient on the yaw-response is marginal as long as long as the wheel's grip on the road is provided. There is virtually no difference in response between dry ($\mu = 1$) and wet ($\mu = 0.7$) road. However, in the case of an icy road ($\mu = 0.2$) the yaw-rate and also the yaw-acceleration become very high; the vehicle turns quickly toward the inner side of the curve and becomes unstable. The reason for this reaction is that the longitudinal slip increases to regions near 100 % (wheel spin), which leads to an almost complete loss of lateral forces at the corresponding wheel. That in turn leads to the strong over-steer reaction shown in ③.

The results of the criterions Q and $\Delta y_{0.75}$ for all considered cases are displayed in ❹. Considering the yaw-reaction, it becomes apparent that failures at the rear wheels lead to more critical situations than failures at the front-axle. Also it can be seen, that the yaw-response becomes more critical on low friction-coefficients. The exceedingly high Q-values (vehicle turns quickly inside the curve) occur in cases where the applied torque cannot be transferred to the road, this means the wheel spins or slides.

The course deviations at 0.75 s are not very high in any considered case – maximum deviation 0.84 m, refer to ④. As in the case of the yaw-response, noticeable deviations occur at very low friction coefficients or high lateral accelerations.

The results show also clearly that the failure (b) – torque is suddenly reduced to zero – is completely uncritical in all considered situations. Yaw-response and course deviation in these cases are close to zero. This is due to the fact that the changes in driving torque are quite small. A sudden collapse of these torques to zero will only have a small influence.

Therefore it can be concluded that situations where failure mode (b) can trigger a critical response are situations with high longitudinal accelerations and therefore high magnitudes of driving torque.

In summary, it can be stated that the considered vehicle reacts quite stable to the investigated failures. Safety-critical situations occur only at very high lateral accelerations and/or at very low friction coefficients. However, this behaviour depends strongly on the type of vehicle and design of the chassis. The considered vehicle is equipped with a sports-chassis and also the centre of gravity location and axle weight-distribution are those from a sports-car. The results gained here cannot be carried over to different vehicle-configurations without reservation.

CONTROLLING THE STABILITY BY TORQUE VECTORING

This section shows how the vehicle can be stabilised in case of a failure by individual distribution of driving and brake torques to the other three functional wheels (torque vectoring). A description of the detailed design of such a torque vectoring control strategy is given in [3].

❺ shows the effect of two different strategies for the failure mode given in ② at $\mu = 0.2$. It can be seen that this situation is extremely safety-critical as the vehicle spins uncontrolled around its z-axis.

One could assume that the vehicle can be stabilised by applying a brake torque to the other three functional wheels as this would reduce the longitudinal-speed and thus the yaw-rate. However, from ⑤ lower left it becomes obvious that this does not lead to a stabilisation of the vehicle at all. Using the second strategy, ⑤ lower right, is obviously much more sensible in this situation. In this case a stabilising yaw torque is created by applying a positive torque at the inner wheels and a negative braking torque at the remaining outer wheel. It can be seen that the yaw rate is reduced quickly and the vehicle stays, even though the side-slip angle is relatively high, in a stable state without spinning, ❻.

a_y=0.23 g; R=30 m; v=8.2 m/s

	Front inner			Front outer		
	μ	Q[°/s]	Δy[m]	μ	Q[°/s]	Δy[m]
M_{max} ↑	0.2	-15,28	0,56	0.2	-10,90	0,46
	0.7	-0,57	0,09	0.7	3,53	0,06
	1	0,41	0,08	1	3,38	0,06
M=0	0.2	0,05	0,23	0.2	-0,12	0,24
	0.7	-0,05	0,01	0.7	-0,11	0,01
	1	-0,05	0,00	1	-0,11	0,00
$-M_{max}$ ↓	0.2	-8,70	0,42	0.2	-14,42	0,55
	0.7	-0,83	0,06	0.7	-3,40	0,07
	1	-0,87	0,06	1	-3,35	0,06

	Rear inner			Rear outer		
	μ	Q[°/s]	Δy[m]	μ	Q[°/s]	Δy[m]
M_{max} ↑	0.2	31,76	0,37	0.2	147,73	0,42
	0.7	-0,64	0,13	0.7	3,75	0,06
	1	-0,67	0,14	1	3,6	0,07
M=0	0.2	0,19	0,23	0.2	-0,16	0,24
	0.7	-0,04	0,00	0.7	-0,17	0,01
	1	-0,04	0,00	1	-0,17	0,00
$-M_{max}$ ↓	0.2	163,09	0,35	0.2	63,05	0,35
	0.7	-0,19	0,10	0.7	-4,08	0,04
	1	-1,14	0,13	1	-7,23	0,07

not safety critical
noticable reaction tending to safety critical
safety critical

a_y=0.5 g; R=30 m; v=12.1 m/s

	Front inner			Front outer		
	μ	Q[°/s]	Δy[m]	μ	Q[°/s]	Δy[m]
M_{max} ↑	0.7	-3,39	0,23	0.7	2,90	0,06
	1	-2,48	0,15	1	2,62	0,10
M=0	0.7	0,04	0,05	0.7	-0,11	0,06
	1	0,03	0,00	1	-0,10	0,00
$-M_{max}$ ↓	0.7	1,09	0,08	0.7	-3,03	0,16
	1	0,86	0,13	1	-3,00	0,10

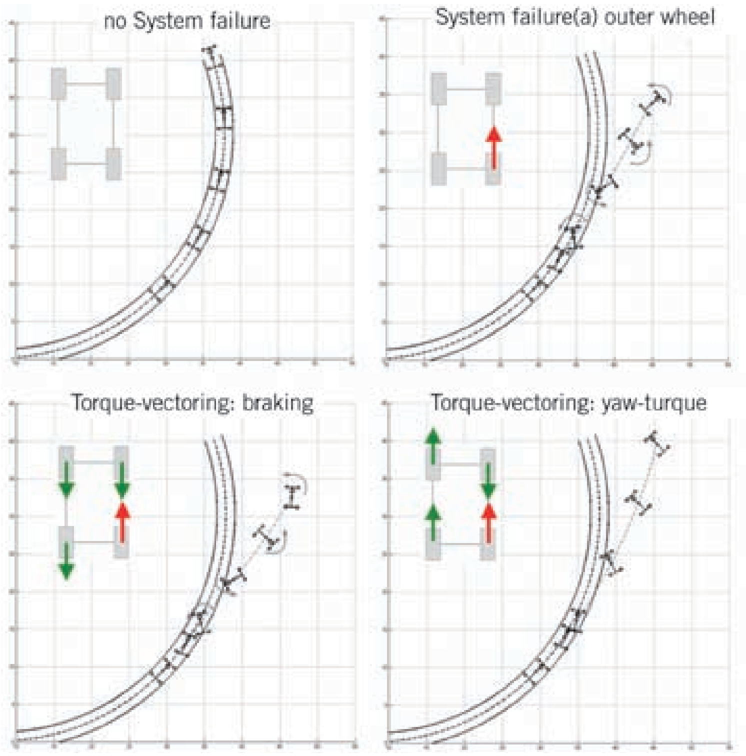
	Rear inner			Rear outer		
	μ	Q[°/s]	Δy[m]	μ	Q[°/s]	Δy[m]
M_{max} ↑	0.7	-3,59	0,25	0.7	2,35	0,02
	1	-2,76	0,18	1	2,00	0,07
M=0	0.7	0,13	0,05	0.7	-0,12	0,06
	1	0,11	0,01	1	-0,10	0,00
$-M_{max}$ ↓	0.7	423,46	0,19	0.7	-4,18	0,13
	1	2,02	0,20	1	-4,27	0,07

a_y=0.9 g; R=30 m; v=13.3 m/s

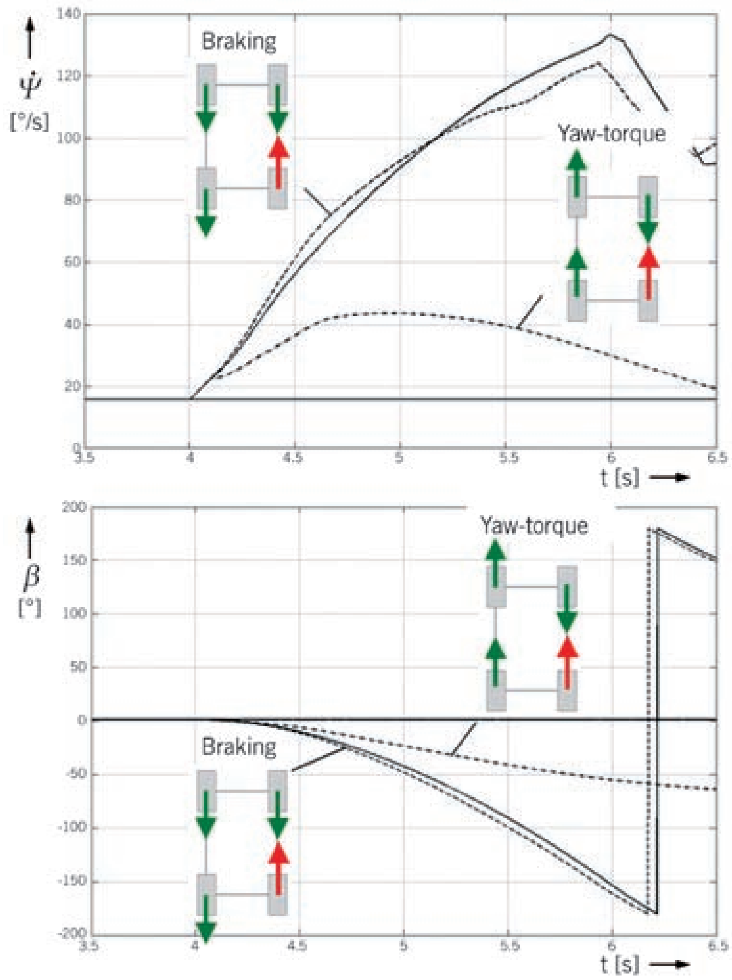
	Front inner			Front outer		
	μ	Q[°/s]	Δy[m]	μ	Q[°/s]	Δy[m]
M_{max} ↑	0.7	-6,05	0,84	0.7	1,22	0,42
	1	-4,32	0,24	1	1,57	0,13
M=0	0.7	0,28	0,51	0.7	-0,10	0,54
	1	0,22	0,001	1	-0,09	0,01
$-M_{max}$ ↓	0.7	7,42	0,34	0.7	-1,61	0,65
	1	3,3	0,21	1	-1,96	0,13

	Rear inner			Rear outer		
	μ	Q[°/s]	Δy[m]	μ	Q[°/s]	Δy[m]
M_{max} ↑	0.7	-4,94	0,77	0.7	1,56	0,45
	1	-3,63	0,19	1	1,59	0,08
M=0	0.7	0,71	0,49	0.7	0,00	0,54
	1	0,53	0,03	1	0,03	0,01
$-M_{max}$ ↓	0.7	417,24	0,40	0.7	-0,76	0,58
	1	5,24	0,30	1	-1,05	0,05

❹ Results of assessment-criterions (R=30 m)



❺ Trajectories at a_y=0.23g to failure (a) +M_{max} of rear outer wheel with and without torque vectoring



❻ Yaw-rate and slip-angle at a_y=0.23g to failure (a) +M_{max} of rear outer wheel with and without torque vectoring

This example shows the great potential of torque vectoring systems that allow for individual distribution of drive and brake torques to all wheels. In this situation a stabilisation using conventional systems that work with brake actuation alone would not have been possible.

SUMMARY

This work investigates the driving safety of an electric driven vehicle in case of different system failures of the electric motors. It has been shown that the vehicle reacts quite stable, except for situations with very high lateral accelerations or very low friction coefficients.

The example given above presents a situation that cannot be stabilised using conventional systems that work with brake actuation alone. A torque vectoring strategy that allows for individual brake and drive torque distribution at all wheels however, is capable of handling a situation like this. This shows the great potential of such control systems that can be integrated very efficiently in electric driven vehicles.

At this time FES is conducting investigations to verify and expand the results gained from this work by driving-tests.

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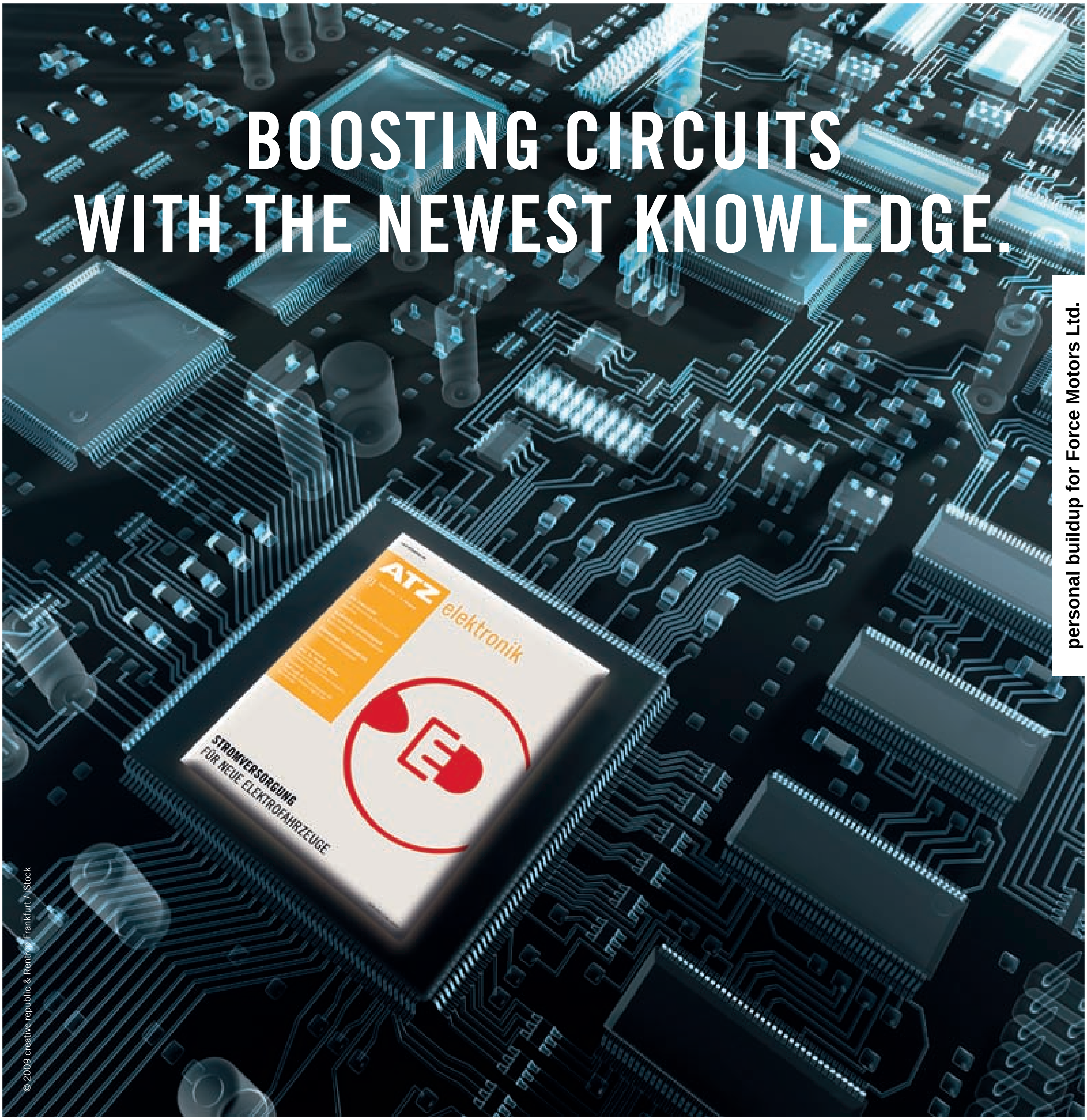
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OPTIMIZATION OF THE OPERATING COMFORT OF SHIFT LEVERS OF AUTOMATICALLY SHIFTING TRANSMISSIONS

One of the most important criterions for the customer acceptance and the quality assessment of a vehicle is the gear-shifting. This is true for both the shifting of manual transmissions and of automatic transmissions. Due to growing market shares of automatic transmissions and increasing variants an efficient method for improving the operating quality is required. For the achievement of a high shifting comfort, researchers at the Institute of Automotive Engineering (IAE), TU Braunschweig, developed a new approach which accelerates the process of shift quality adjustment with less effort. This allows a transparent, efficient and automated evaluation of the shift device as well as a computer-aided optimisation of the design.

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1 MOTIVATION AND TARGET

The customer's impression of a vehicle is determined by different components. One important component is the shift device for transmissions, which is of significant importance for the quality impression and the differentiating features of car brands. In particular the shift device, as part of the human machine interface (HMI), is a characterising element which is critically judged by the consumer during the first contact with a new vehicle.

In this context, the growing requirements of shifting quality are always a challenge for car manufacturers, who work steadily hard on optimisation of shift devices. The new generation of shift devices is a Shift-by-Wire, in which the mechanical connection between external gear shift and transmission is replaced by electrical systems. The quality-determining shift forces at the shift lever are thus freely configurable and offer more possibilities to the developer.

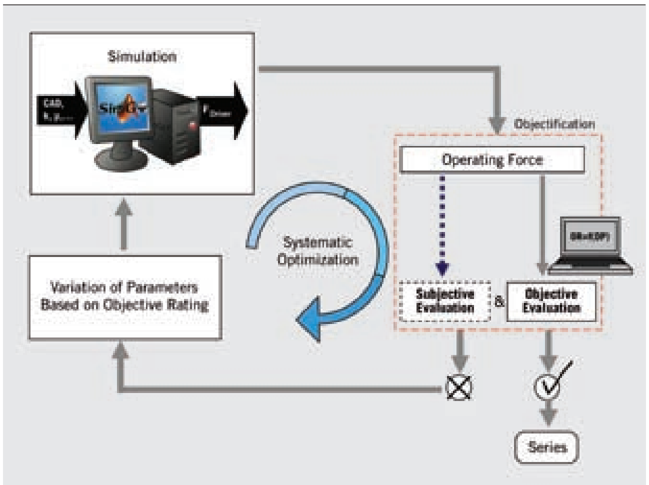
The shifting quality of shift devices is at present adjusted by experiment, at which the shift quality is evaluated subjectively. From the results of the subjective evaluation the shift device is varied based on developer experience. The quality adjustment is always done for a certain combination of vehicle, transmission and shift device.

The effort of quality adjustment with this method will increase significantly in the future. This results from the increasing number of vehicle models with automatic transmissions. Furthermore, the predicted market shares show that the market share of automatic transmission in triad markets will even increase in the future. These tendencies establish the need for a more efficient development work by means of the new methods presented here.

In terms of an efficient optimisation the aim is to implement the process of quality adjustment in the virtual work environment. For this purpose, the shift device is evaluated objectively instead of subjectively. Objectification methods were used to create the basis of objective evaluation. The methods specify the relation between parameters that can be measured and the test person's subjective perception. Furthermore, simulations are applied for the respective virtual optimisation of operating force, whereby a time and cost saving is realized, ❶.

2 MEASURING EQUIPMENT

The following measuring equipment is used for verification of simulation model and implementation of the objectification of shifting quality. It records during the shifting the parameters, which effect the shifting quality. Basically operating forces, ergonomics, haptics, optics and acoustics play an important role. The most influence criterion is however the operating forces since they identify the load for the driver and a direct haptic feedback. For this reason, the operating force is valid as the significant parameter which must be recorded by the measuring equipment. In addition, shift travel, shift time and shift angle are measured.



❶ Virtual quality adjustment of the shift lever position change by means of objective evaluation and simulation tool

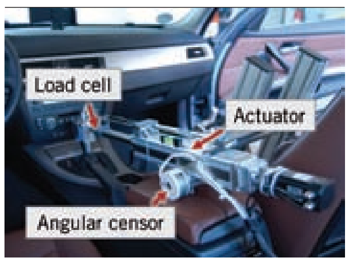
Researchers at the Institute of Automotive Engineering (IAE) of TU Braunschweig developed a shift robot, which operates the shift device automatically and records the relevant parameters at the same time, ❷.

With the shift robot the movement of a human arm is simulated. It moves the shift lever to required shift positions. The influence of the shift behaviour is eliminated and the reproducibility of the measurement is increased. Since the force sensor is directly on the shift robot, the effect of elasticity of shift knob is also taken into account. ❸ represents the shift force measured by the shift robot. For the measurement in other shift gates, such as the cross gate and in the manual mode, either the shift device or the shift robot must be turned.

The shift robot has a dimension of 50 × 50 × 600 mm. and a travel range of 200 mm. The maximum continuous force amounts to 134 N. These values comply with the results of preliminary tests with various vehicles of different brands and vehicle classes. This guarantees that the shift robot can be used in any vehicle. Furthermore, the robot arm can be adjusted in all three spatial directions, so precise positioning to the shift gate is possible.

3 OBJECTIFICATION TOOL

The objective evaluation allows an evaluation of shifting quality based on objective, measurable signals. Together with the simulation the shifting characteristics in terms of customer rating can therefore be predicted in the virtual work environment.



❷ Measurement device: shift robot

The objectification tool "Objectification of the Shift Operating Comfort" (OSOC) consists of three basic functions:

: Identification of the objective parameters
: Determination and verification of the objective rating model
: Calculation of the objective rating

An important characteristic of this tool is the ability to evaluate the shift device at any time of the development process. That means evaluation of a virtual shift device in early development stages or evaluation of a prototype.

3.1 IDENTIFICATION OF THE OBJECTIVE PARAMETERS

The tool visualises the force and automatically determines the objective parameters (OP) from the measured or simulated shift force for the individual shifting. There are approximately 72 objective parameters per shift position change to be determined as levels, differences, gradients and areas. These include maximum force level, positive force gradient and peak-to-peak value, among other things.

3.2 DETERMINATION OF THE OBJECTIVE RATING MODEL

A fundament of building the objectification model are the subjective evaluations from survey and the measured shift force of the relevant shiftings, which were provided from extensive tests performed by researchers at the IAE in the past years. Based on the subjective evaluation in form of subjective ATZ rating from 1 to 10 and the objective parameters of the relevant force, tests for outliers as well as correlation and regression analyses are performed, so the relationship of above two variables can be created. As typical regression model a multiple linear regression is used. The general form of the multiple linear regression is given by the following Equation:

EQ. 1	$OR_i = b_0 + \sum_{j=1}^{k} (b_j \cdot OP_{ij})$

Thereby, b_0, b_1, b_2, …, b_k are the regression coefficients and OR is the objective rating. After logical examination of possible objective models, suitable objective equations could be determined for each change in shift position.

The verification of objective rating model takes place through confrontation of the subjective judgments from test persons and the appropriate forecasts of objective rating model. As a quality measure, the coefficient of determination is calculated for each objective rating model.

3.3 CALCULATION OF THE OBJECTIVE RATING

With the help of objective rating model any shift force can now be evaluated. As a basis, a measured or simulated shift force is used for identification of objective parameters, which are feed into the objective rating model.

4 SIMULATION TOOL

In terms of quality adjustment, the production of the shift device variants is cost and time consuming. Therefore the "Simulation of Gearshift Operation" (SimGO) tool was developed, which allows predicting the operating force during the shifting.

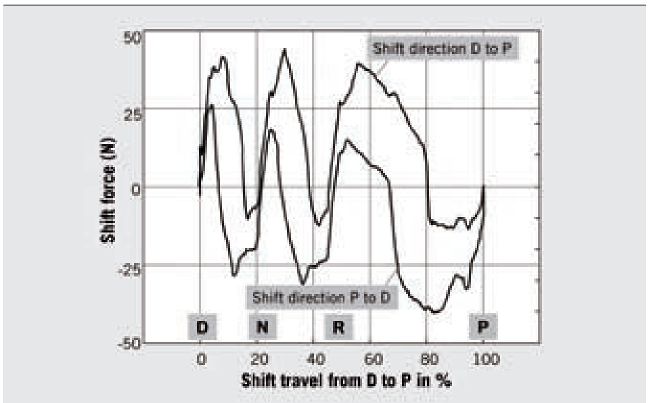
4.1 MODELING

To generate the model of shift device the influences of all components in the shift device must be taken into account. It usually consists of two main components: the external shifting and the transmission. Under the transmission the internal shifting unit and the parking mechanism are described separately as subsystems in the transmission. The driver, who operates the shift device, is described as an additional element for the control loop, ❹.

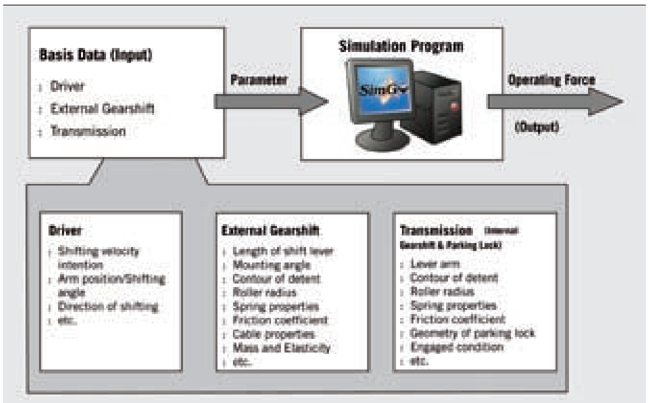
In case of shift-by-wire the transmission model is not considered, since the external shifting is connected to the transmission by electric wires and the relevant shift force is created only from the external gearshift model and the driver model.

The gear shift model was generated as a multi-body system. The parts are connected by spring and damping elements. The driver model is connected to the other models to build up a closed-loop control of the shift speed. It generates a force signal which is continuously adjusted, such that the shift lever is moved with the required speed. That means the model will increase the shift force if the required shift speed is not reached.

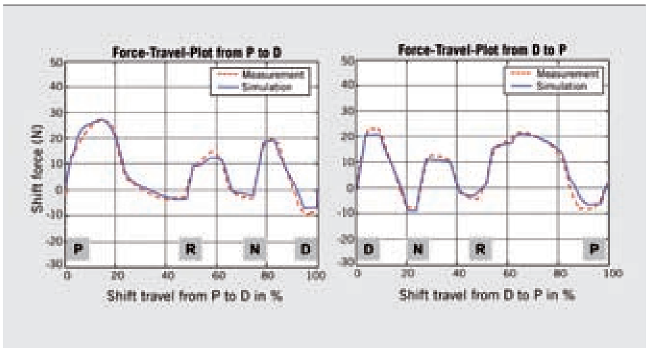
Modelling from the driver to the transmission considers dynamic effects. On the one hand, the stiffness and damping of the driver's hand, shift knob and shift lever and, on the other hand, elasticity and non-linearity of the shift cable, for example friction and



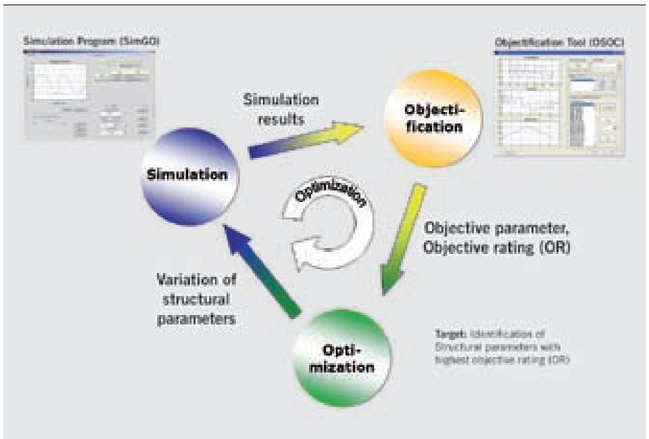
❸ Shift force of a mid-size vehicle



❹ Structure of the Simulation tool "SimGO"



❺ Comparison of measured and simulated shift force



❻ Use of the presented modules in the product development process

stroke loss, are taken into account. The parameters of shift cable are gained from analysing its connection path from the CAD model and summation of the bending angle of cable. In this context the bending radius and the length of the straight line segment can be neglected due to lower influence in comparison with the bending angle. The transmission is assumed to be a quasi-static system, in which the detent mechanism and the parking mechanism are taken into account. The detent mechanism is modelled as a characteristic diagram in the model. The parking mechanism is modelled based on the position of the connecting rod and the engagement conditions (ratchet engaged or not).

4.2 MODEL PARAMETERIZATION

To simulate the operating force the model requires several parameters, which describe both the shift device and the virtual driver behaviour. These parameters are categorized in line with geometric variables, characteristics of the components and simulation parameters. An important parameter for the precise prediction of the operating forces is the contour parameter of the detent mechanism. The detent mechanism serves to ensure an engaging of the desired shift lever position and even small changes on the contour can lead to different shifting characteristics. In the model the contour can be considered in sections three dimensionally by defining the contour in two planes: the plane in the shift direction (longitudinal plane) and the cross-sectional plane (cross-sectional plane).

4.3 MODEL VALIDATION

The simulation model was tested for many different shift devices offered by various manufacturers, which are most commonly used in today's vehicles. For example, ❺ shows the simulation results of a shift device of a compact class vehicle.

In ⑤ the measurement and simulation results for each shift direction are compared to each other. The results clearly reveal that the calculated force curves are very close to the measured ones, so the simulation leads to the desired results and the model is therefore validated.

5 CONCLUSION

Simulation and objectification in particular are of considerable importance for efficiency improvement of development process of shift devices. The tools presented here by TU Braunschweig, are used together in a process chain. Further variants can now be calculated and evaluated virtually in a simple, quick and cost effective way. The process of a typical optimisation with these tools is presented in ❻.

The design engineer changes the parameters of CAD model. Based on the design parameters from CAD model, the operating force of a new design is calculated by the presented simulation tool. With the calculated force as basis, the objective parameters are identified automatically and the objective rating is generated. This rating represents the expected shifting quality of the virtual design.

Apart from the manual optimisation this process can also be accomplished by an intelligent optimisation algorithm. The parameters of the CAD-model design are thus changed by the optimisation tool and used for the simulation. Within the design boundary, i.e. geometrical constraints of the module and the vehicle, the optimisation tool tries to find the parameter combination with the best possible objective rating. This best possible parameter combination is than implemented in form of a prototype and used in the real vehicle.

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