SIMULATION ENVIRONMENT FOR THE INVESTIGATION OF ACTIVE ROLL CONTROL IN COMBINATION WITH VEHICLE DYNAMICS CONTROL

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KEYWORDS

Vehicle dynamics, simulation, roll control, yaw control, simulation environment

ABSTRACT

In the development of advanced chassis systems, the interaction of different active systems and their combined influence on the behaviour of the vehicle need to be closely examined. In order to optimally use the safety and comfort potentials such systems can provide, interferences must be avoided and synergetic effects need to be used. While the demand for shorter development cycles and a reduction of development costs is growing, the use of computer simulation provides a powerful means to obtain reliable results early in the development of active chassis systems.

One property of most active chassis systems is the possibility to influence the vehicle's handling characteristics. Common to all these systems is the generation of yaw moments around the vertical axis of the vehicle. Such systems are e.g., active steering, active four wheel drive, active roll controll, and individual wheel brake actuation. Depending on the driving situation, the possible influence of these systems upon driving dynamics differs greatly and simultaneous interventions of different systems can lead to both interferences and synergies.

This paper demonstrates the use of different simulation tools in the development process for active chassis systems. Using a full-vehicle simulation model, the influences of different active chassis systems upon vehicle dynamics are investigated. Co-simulation interfaces form a link to multi body simulations or fluid simulations, and rapid control prototyping is used with real time hardware for controller development.

In this work, an extension to this simulation environment is developed which allows the investigation of the yaw moment resulting from interventions of different active chassis systems. The work is focused on the simulation of interactions between an active roll control system and the actuation of individual wheel brakes generally used in vehicle dynamics control. Using the simulation environment, influences upon the dynamic behaviour of the complete vehicle equipped with both active systems can be assessed and controller specifications can be defined at very early stages of the development.

MAIN SECTION

SIMULATION ENVIRONMENT

Automotive research and development activities at the Institut für Kraftfahrwesen Aachen (ika) of RWTH Aachen University include the investigation of a broad field of issues in different types of vehicles. For this reason, a simulation environment was developed which can flexibly be used for a range of tasks in the area of chassis engineering, vehicle dynamics, and controller development.

<u>Full Vehicle Simulation Platform</u>

The basis for this simulation environment is formed by a MATLAB/Simulink full vehicle simulation model. Fundamentally, this model consists of the linear differential equations of motion of a 5 mass, 10 DoF system: translational and rotational motion of the body/vehicle along three coordinate directions as well as translational motion of the 4 wheel masses, Fig. 1. In addition to the linear time invariant (LTI) core of the model, non-linear elements such as a tyre model, steering gear, and kinematics and compliance subsystems are used in order to improve the accuracy of the model.

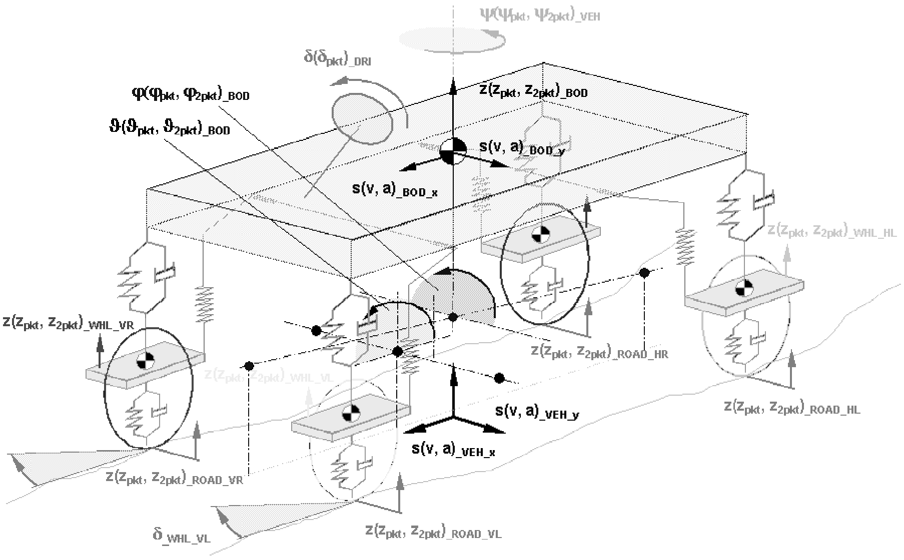

Fig. 1: Full vehicle model

The simulation model is integrated into a simulation environment including a simple driver model, a device for the visualisation of the acquired results, and a graphical user interface (GUI) for the selection and parameter definition of driving manoeuvres. A generic controller subsystem allows the integration of control algorithms for controller development. The structure of the simulation model in MATLAB/Simulink is given in Fig. 2.

The MATLAB/Simulink full vehicle model is implemented in a modular way allowing modifications and model extensions depending on the objective of different simulation projects. Some key features of the simulation platform are the following:

- All parameters of the model are defined globally in a central parameter file in the MATLAB environment, offering the possibility for simple variation of parameters and automation of simulations

- Most functional blocks can be validated individually and assembled into the model; blocks can easily be substituted by alternative versions

- Data acquired in testing (e.g., steering input, velocity, road signal) can be fed into the simulation via the GUI

- Dynamic and nonlinear behaviour of subsidiary systems can be represented in modules grouped around the LTI system of motion

- Controller structures and data interfaces are designed in order to allow export to the dSPACE rapid control prototyping platform.

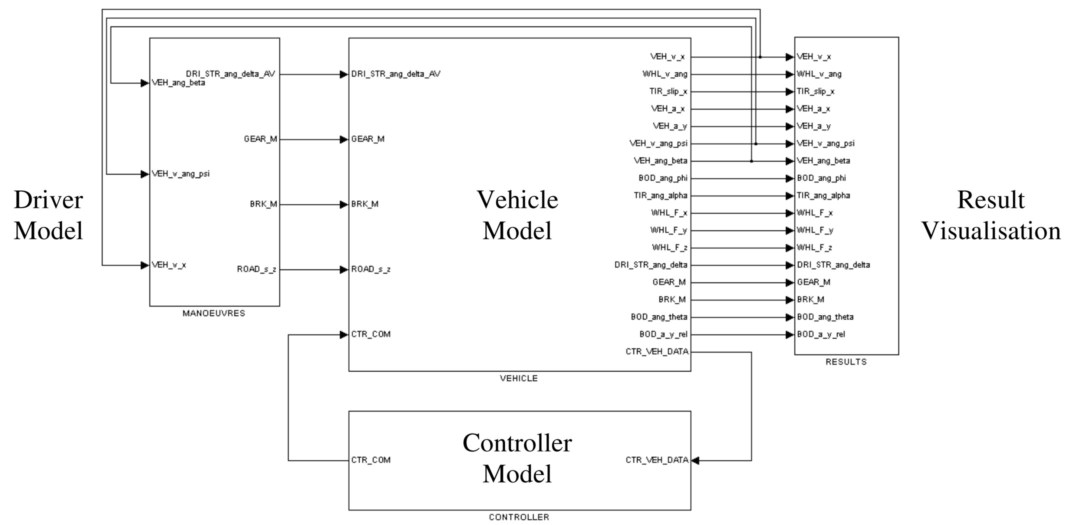

Fig. 2: MATLAB/Simulink Model Structure

The MANOEUVRES subsystem in Fig. 2 comprises the generation of driver and environment input data into the model (e.g., driver control and road input). For the simulation of closed-loop manoeuvres like steady-state cornering or double lane change, the driver is represented by a feedback controller; open-loop manoeuvres are simulated with a feedforward driver giving pre-defined input into the model. The RESULTS block facilitates the visualisation of simulation results on-line and saves the results in an export file. The CONTROLLER block is provided with the data representing sensor information in the vehicle and outputs the control commands for the chassis actuators.

Inside the VEHICLE block, the actual vehicle model is located. This model is divided into three main subsystems: longitudinal dynamics, lateral and vertical dynamics, and tyres, Fig. 3. The longitudinal dynamics part of the model contains the differential equations for longitudinal displacement of the complete vehicle considering brake and propulsion torques at each wheel as well as rotational inertia and longitudinal tyre slip from the tyre model. The lateral and vertical dynamics part of the vehicle model comprises the 5 mass system depicted in Fig. 1, representing roll, pitch, and heave of the body, vehicle yaw, and vertical displacement of the wheels. This part of the model is realised in the form of a matix system, while the system matrices are defined extenally in the vehicle parameter file. This configuration provides the possibility to adapt the model to fundamentally different suspension concepts merely by edititng the parameter file. In addition to the matrix system, the lateral and vertical dynamics subsystem contains elements for the calculation of the vehicle side slip angle, wheel travel and side force dependent toe angle, and non-linear steering gear. Both model parts are connected through the tyre model based on Pacejka's Magic Formula. Longitudinal and lateral slip are calculated for each individual tyre. The tyre parameters used are acquired in measurements at the institute's tyre test facilities.

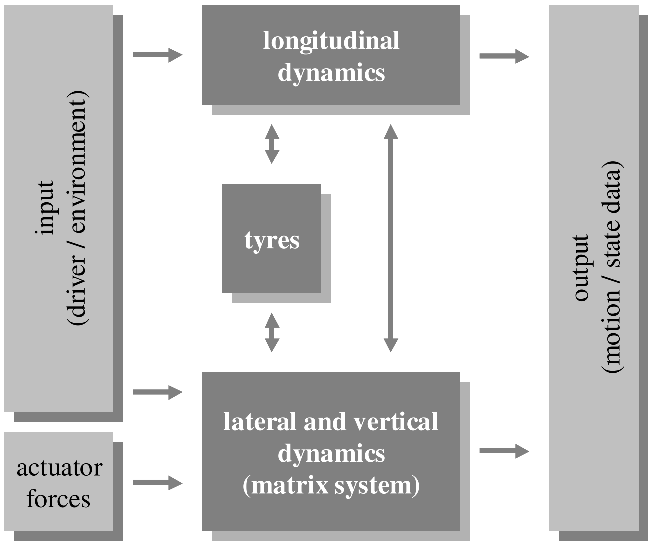

Fig. 3: Structure of the vehicle model

For the investigation of active chassis systems, simulation blocks representing the chassis actuators are integrated in the vehicle model. These actuator subsystems contain the dynamic properties of the actuator in the target vehicle and receive commands from the controller system. The actuator forces are fed into the 5 mass lateral and vertical dynamics system.

For a user-friendly operation of the simulation platform, a graphical user interface is implemented in the MATLAB environment. This GUI provides the user with a choice of open-loop and closed-loop driving manoeuvres and the possibility to load different sets of model parameters and input data into the MATLAB workspace. The major settings in the so-called 'Fahrmanöver-Generator" are

- steering angle (closed-loop, open-loop or data feed from file)
- road profile feed (time or distance based)
- longitudinal velocity / acceleration / deceleration
- simulation time and frequency.

The GUI for the simulation settings is shown in Fig. 4. The manoeuvre generator loads the parameter and data files in the workspace and runs the simulation model according to the user settings. The manoeuvre generator forms an open platform for the integration of additional data or model functions depending on the objective of different simulation investigations.

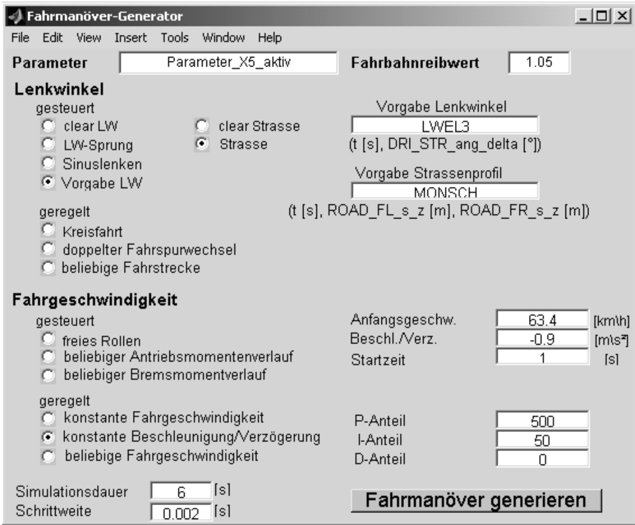

Fig. 4: Graphical user interface "Fahrmanöver-Generator"

Controller Development and In-vehicle Testing

For controller development the simulation platform described above is used in combination with rapid control prototyping (RCP) tools. Controller design and verification is carried out with the MATLAB/Simulink full vehicle model. This permits the trial of different control approaches and the assessment of influences upon vehicle dynamics at early stages of the development process.

In-vehicle controller testing and controller development with physically existing plants is carried out using an RCP environment based on a dSPACE AutoBox real-time computer. Controller models created and tested in Simulink are exported via the MATLAB Real-Time Workshop onto the dSPACE platform, where the models are calculated by a powerful real-time processor. In the controller development, production-type sensors and signals from the vehicle CAN are used.

A hardware signal conditioning unit called *i*-Box developed at ika provides signal level adjustment, hardware filtering and output of pulse width modulated signals for driving actuators. Integrated protection circuits shield the real time platform from harmful signals. In addition to the controller development functions, the AutoBox harware and software provide all functions necessary for data acquisition, visualisation and storage in testing procedures.

Co-simulation Interfaces

Since the modelling of certain components and systems in Simulink proves rather complex, specialised simulation tools are used for the investigation of complex mechanical, electric, hydraulic, or pneumatic systems. For the analysis of fluid dynamics systems in the field of chassis engineering – hydraulic actuators, hydraulic dampers, air springs, brake or steering systems – the simulation software DSH*plus* is used. Electromechanical components are simulated with SABER.

The simulation of the resulting vehicle dynamics of the complete car is investigated by means of co-simulation between specialised simulation tools and the Simulink vehicle model. In the case of DSH*plus*, co-simulation subsystems are integrated into the Simulink model replacing correspondent blocks. The implementation of individual integrators into the co-simulation blocks allows for higher simulation frequencies in numerically stiff hydraulic subsystems. At later states of the project, the complex co-simulation blocks can be replaced by simplified and validated Simulink models for improved calculation performance. In comparison to analytical modelling of hydraulic systems in Simulink, this method provides obvious advantages.

One further approach for the simulation of highly specialised systems is realised by the co-simulation between the fluid dynamics software DSH*plus* and multi body simulation (MBS) software. Such interfaces are in use at ika for the MBS programs ADAMS and SIMPACK. Thus, the functionality of complex subsystems can be assessed in detail.

COMBINED INTERVENTION OF ACTIVE CHASSIS SYSTEMS

The objective of this work is the investigation of synergies and interferences in the combined engagement of different active chassis systems for the purpose of vehicle stability control. Since stable vehicle handling is closely linked to the vehicle's yaw response upon the driver's steering input, stability control systems need to be capable of affecting yaw motions. Currently, there are a number of different systems available which offer possibilities to influence vehicle yaw. While brake-based vehicle dynamics control systems like ESP (electronic stability program) are becoming standard on high-volume models, other systems like active steering, active roll control, active suspensions, and active 4-wheel drive have only been available for for a short time on a small number of premium class vehicles.

Due to the fundamental differences between these systems, the degree of influence upon the driving behaviour of a vehicle as well as the situations in which interventions are effective differ greatly. Still, any one of the systems mentioned above is or can be used to control one common variable, the yaw rate. In order to prevent negative interactions and to fully use effects of mutual support during parallel activation, a combined contemplation of such systems in different driving situations is necessary.

Approaches with a high potential for controlling the yaw rate are brake actuation on individual wheels, steering angle interventions, and the distribution of roll moments between front and rear axle in active suspensions or anti-roll systems, Fig. 5. The highest impact on vehicle stability among these is achieved with brake systems, since any kind of brake actuation reduces vehicle speed. In this work, the strong interaction between such a brake-based system and a system for wheel load distribution is examined.

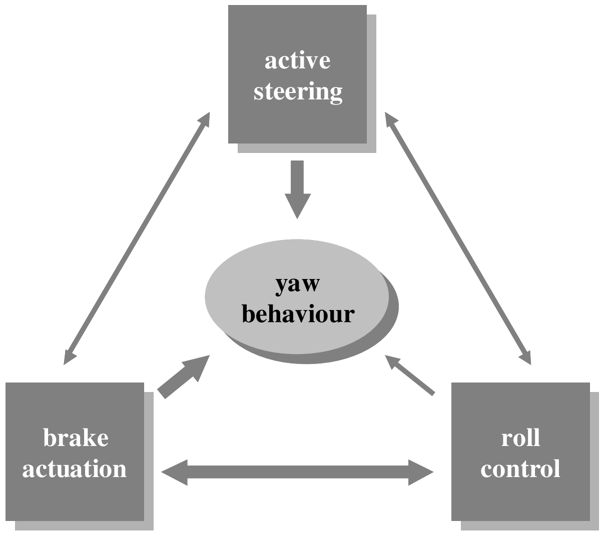

Fig. 5: Three approaches to controlling vehicle yaw

The common element for brake force systems and wheel load systems is the tyre. While brake interventions induce longitudinal tyre slip resulting in reduced lateral and increased longitudinal forces, differences in wheel load cause a decrease in side force at the concerned axle due to degressive tyre behaviour. Both the reduction of side forces and the longitudinal brake forces alter the equilibrum of moments around the vertical axis of the vehicle, leading to a yaw moment, see Fig. 6.

Fig. 6: Tyre forces affecting yaw moment

<u>Model Extension for the Assessment of Yaw Moments</u>

In this work, the simulation environment described in the first part of this paper is used to investigate combined interventions of different active chassis systems and their influence upon vehicle dynamics. This investigation hinges on the ability to assess yaw moments resulting from such interventions which act on the vehicle.

Therefore, a model extension for the quantification of yaw moments was developed and integrated into the simulation environment.

Any intervention from wheel load distribution or brake actuation influences the horizontal forces in the tyre contact patch and induces a change of the driving condition. During a wheel load intervention at constant lateral acceleration (i.e. constant side force), modified tyre slip angles occur due to the change in wheel load differences. The dependency of side force and tyre slip angle on wheel load difference for one axle of the reference vehicle is given in Fig. 7. After adapting the driving state to the changed condition, the vehicle reaches a new equilibrum of moments. Similarly, a single wheel brake actuation generates a yaw moment for a short time, hence causing a yaw motion. After the intervention is finished, an equilibrum is reached again. In both cases the yaw moment generated cannot be measured directly.

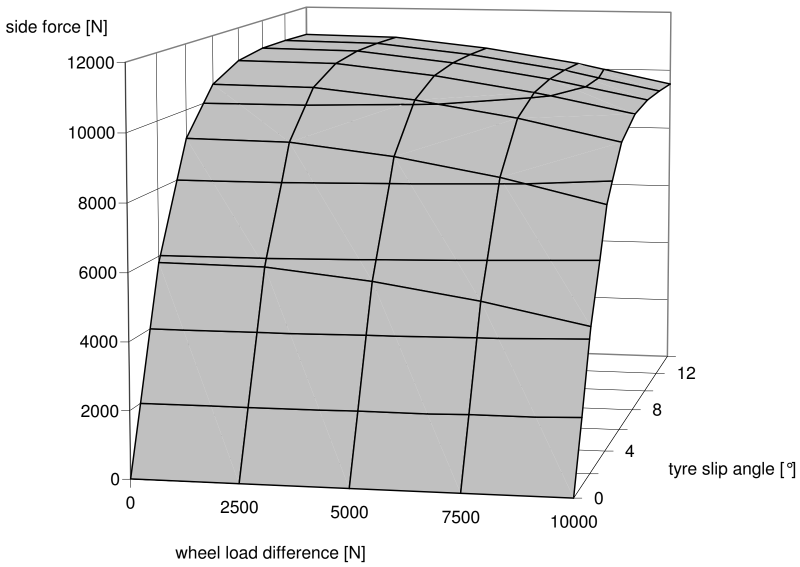

Fig. 7: Change of tyre slip angle depending on wheel load difference

In order to determine the yaw moment caused by the intervention of an active chassis system, a model extension is implemented. This extension allows for the generation of external forces and torques in the simulation acting directly upon the model vehicle. These forces and torques are controlled in order to exactly counteract the effects of the active intervention. This means that in the simulation the vehicle state is kept constant during intervention manoeuvres, not allowing the model vehicle to reach the equilibrum of moments as a real vehicle would. Thus, the yaw moment induced by the chassis interaction equals the virtual torque necessary to keep the vehicle in a constant driving condition. Fig. 8 demonstrates this principle of virtual counteracting loads using the exemple of yaw moments.

For this investigation, steady-state cornering manoeuvres are simulated at different velocities. In order to eliminate effects from tractive forces, the vehicle is simulated in a state of free coast, the velocity is kept constant by an external force. During the intervention, additional forces and torques are applied causing the vehicle to maintain its path inspite of the intervention yaw moment. Vehicle side slip angle β, yaw rate $\dot{\psi}$, longitudinal velocity v_x , and lateral acceleration a_y are kept constant during the manoeuvre. This is realised by a controller taking the values of β, $\dot{\psi}$, v_x, and a_y

before the intervention as setpoint values and using external loads on the vehicle for keeping these values constant. The torque around the vehicle's vertical axis necessary to steady the vehicle exactly represents the amount of yaw moment that can be applied by an intervention in the driving condition simulated. Using this method, different brake actuation and wheel load distribution manoeuvres are simulated in different driving conditions.

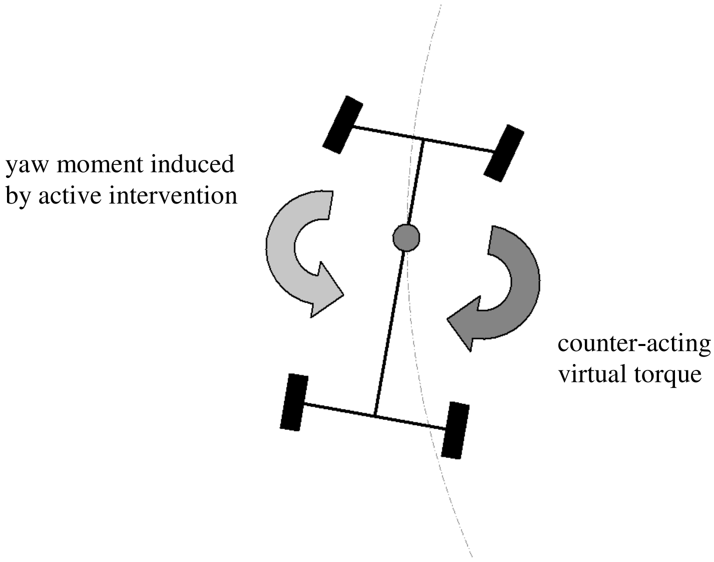

Fig. 8: Principle of virtual counteracting torque

Wheel Load Intervention

As mentioned above, the effect of wheel load distribution upon handling characteristics is based on non-linear tyre behaviour. Under the influence of wheel load differences, the increase of side force at one wheel cannot equal the decrease of side force at the other wheel of an axle. Therefore, the loss of side force needs to be compensated by increased tyre slip angles, compare Fig. 7. Consequently, wheel load differences caused by anti-roll bars or any kind of roll compensation actuator reduce the side force potential of the respective axle. In active roll control this effect can be used to continuously influence vehicle steering characteristics. By sustaining a larger portion of the overall roll moment at the front axle a tendency for understeering is provoked. Roll moment at the rear axle biases the vehicle behaviour in the direction of oversteer.

In order to obtain reliable results from the simulation, the distribution of roll moments in the model was validated by means of testing with a passive reference vehicle. The anti-roll bar configuration was varied – standard configuration, front bar only, rear bar only – and the influence of the roll stiffnes/roll moment distribution on the steering behaviour was measured, ensuring the correct representation of the effects investigated in the simulation model.

This work is based on an active suspension system incorporating one hydraulic actuator for each individual wheel. The additional wheel loads to be realised with this suspension amount to ± 4500 N per wheel at the front axle and ± 3200 N per wheel at the rear axle. As basic setup, the distribution of roll moments front/rear is set to 50/50.

Depending on the driving situation, the roll moment can dynamically be distributed in a range from 100/0 to 30/70.

This configuration is analysed with the ika simulation platform and the model extension described above. Fig. 9 displays the resulting yaw moments versus lateral acceleration for the settings 100/0 and 30/70 in relation to the neutral 50/50 distribution. The dotted lines represent the maximum yaw moment achievable when the actuators are articulated contrarily (maximum warp). A full compensation of the roll angle cannot be achieved in this mode.

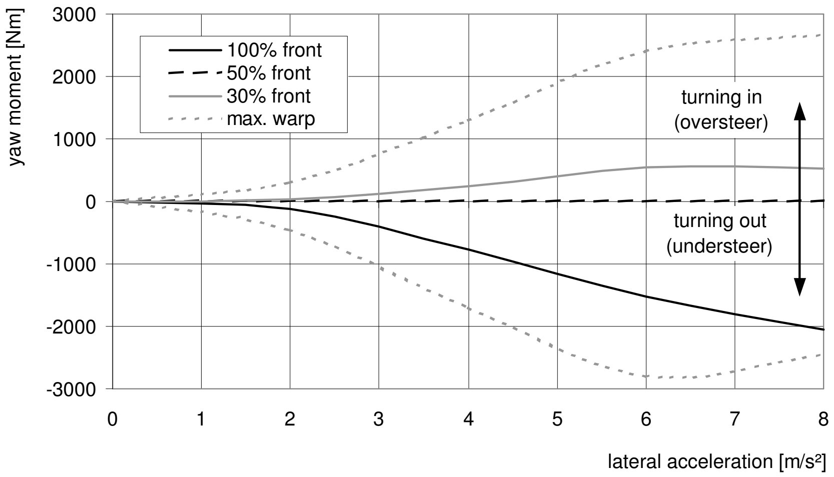

Fig. 9: Yaw moments resulting from roll moment distribution

It becomes obvious that the effect of the roll moment distribution on the yaw behaviour significantly depends on the lateral acceleration which determines the overall amount of wheel load differences to be distributed between both axles. With the settings chosen, a maximum yaw moment of 2050 Nm at 8 m/s² lateral acceleration can be applied onto the vehicle.

Active Brake Intervention

The yaw moment generated by individual wheel brake actuation originates from two mechanisms: the longitudinal force applied during the braking and the reduction of side force resulting from increased tyre slip. In Fig. 10, the combination of individual wheel brake actuations for generating oversteering and understeering yaw moments, respectively are given.
The highest potential for generating yaw moments turning the vehicle out (i.e. inducing an understeering tendency or counteracting oversteer) can be observed at the front outer wheel. The changes in both longitudinal and lateral force result in a turning-out momentum. Additionally, a brake actuation at the rear outer wheel can generate a parallel momentum, although the side force loss due to increased slip mitigates the desired effect.

The highest potential for generating a turning-in momentum (i.e. inducing oversteer or counteracting understeer) is found by braking the rear inner wheel. Again an additional momentum can be obtained from brake actuation at the front inner wheel, with equally restricted effect due to the side force loss.

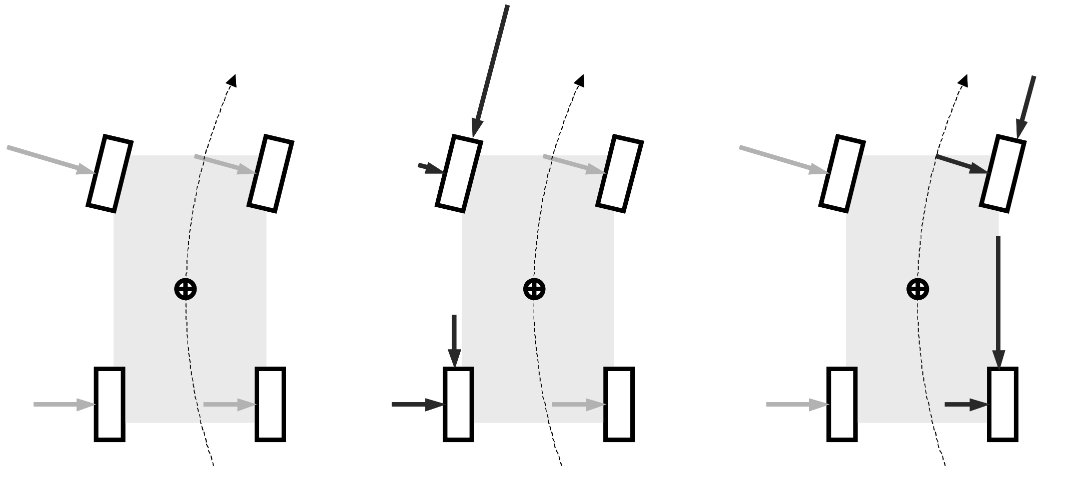

Fig. 10: Generation of yaw moments by braking individual wheels

For the simulation of brake interventions and the analysis of the yaw moments created, the simulation model is validated against test data acquired in measurements: Brake pressures were measured during ESP interventions in the reference vehicle and the transfer characteristics from brake pressure to wheel braking torque was assessed on a brake test rig. In the simulation, measured brake pressures are fed into the model and the resulting forces in the tyre contact patch are calculated by the tyre model, taking into consideration the brake system transfer characteristics.

For a direct comparison to the yaw moments determined for wheel load distribution interventions, corresponding manoeuvres are simulated. Wheel brake actuations are carried out during different steady-state cornering manoeuvres. In tests with the reference vehicle, longitudinal brake slip of around 40% was observed in severe ESP interventions. Basing the simulation parameters on these values derived from measurement, the front outer wheel is decelerated to a brake slip of 40% in order to generate turning-in yaw moments. Turning-out moments are induced by braking the rear inner wheel to the same level of longitudinal slip. Again, the model extension is used during the interventions to control additional forces and torques acting on the model vehicle for keeping the driving situation constant and measuring the yaw momentum generated.

The yaw moments induced by brake intervention generally are much higher than the moments created by shiftig wheel loads. A dependency between lateral acceleration and maximum yaw moments can clearly be seen. Increasing side force during cornering leads to increased wheel loads at the outer wheels, making the brake intervention for turning-out yaw momentum (front outer wheel) more efficient, thus providing a higher yaw potential. On the other hand, the maximum turning-in yaw momentum is largely independent of lateral acceleration.

The potential for turning the vehicle in (inducing oversteer) is significantly smaller than the potential for counteracting oversteer. While the maximum turning-in momentum remains relatively constant at 6000 Nm over the range of lateral accelerations simulated, turning-out moments exhibit a nearly linear dependency from lateral acceleration, amounting up to 17000 Nm. The results from the brake intervention manoeuvre simulations are shown in Fig. 11.

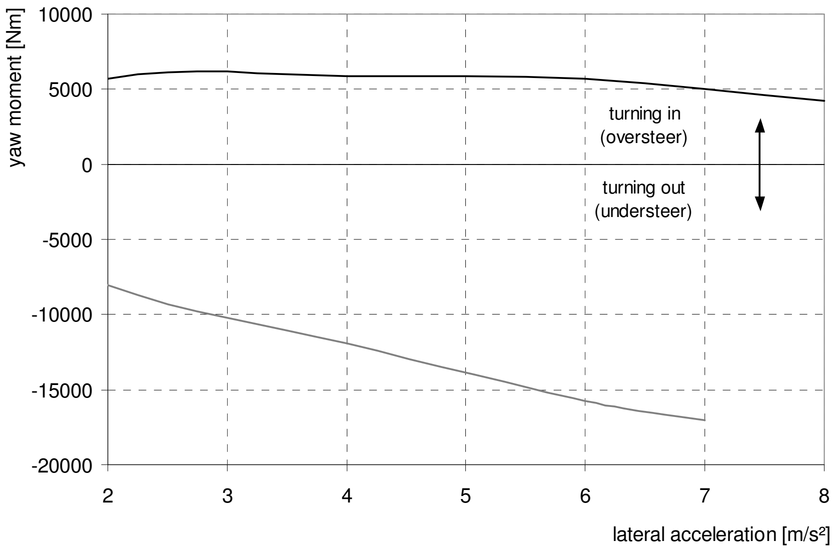

Fig. 11: Yaw moments resulting from individual brake actuation

<u>Strategies for optimised combination of interventions</u>

The properties of the two systems described above may lead to the conclusion of assigning each system to a certain range of driving state. This method is also referred to as 'friendly co-existence' of two systems. As an example, vehicle handling could be optimised in uncritical situations by way of continuously distributing roll momentum and the greater potential of brake interventions would be used in limit handling and in safety-critical states. However, the objective of this work is to asses the possibilities of improving the overall system performance by combining both systems and increasing the effect on the vehicle's stability.

A combined intervention including active wheel load distribution and individual wheel brake actuation at the same time needs to avoid negative influences of one system upon another. Furthermore, the task of controlling the vehicle's stability must be fulfilled in the most efficient way, making use of the entire potential both systems can provide. Therefore, the parallel intervention of both systems for generating turning-in and turning-out yaw moments, respectively is investigated.

In order to increase the vehicle's yaw rate (turning the vehicle in), the wheel load distribution shifts roll moment to the rear axle. While the wheel load differences at the front axle are minimised, the rear outer wheel is loaded additionally and the rear inner wheel is relieved. The increased side force at the front axle and decreased side force at the rear result in a yaw moment oriented to turn the vehicle in. For achieving yaw moments of the same direction by means of brake intervention, the rear inner wheel is braked. Even without active wheel load distribution the inner wheels are unloaded during cornering. With the braked wheel additionaly relieved by the wheel load distribution, the effect of the brake intervention is hevily reduced. In this situation the interventions of both systems conflict clearly. The roll moment distribution needs to be set to values similar to those of a conventional vehicle in order to maintain the more efficient functionality of the ESP intervention. An exchange of information between brake-based vehicle dynamics controller and the roll controll system is necessary: in the case of a brake intervention the wheel load control strategy must be altered in order to support the effect of the brake intervention.

For inducing an understeerting tendency (turning the vehicle out, reducing yaw rate), the wheel load distribution is used to shift roll moment to the front axle. In consequence the vertical tyre force is increased at the front outer wheel and decreased at the front inner wheel, while wheel load differences at the rear axle are reduced. The corresponding intervention of the active brake system is actuating the front outer wheel brake. Since the vertical force at this wheel is additionally increased by the roll moment distribution, the brake actuation results in higher longitudinal force leading to the desired yaw momentum. Due to the higher proportion of side force carried by the front outer wheel, the side force loss induced by longitudinal slip also results in increased yaw moments turning the vehicle out. Generally, both systems do not interfere whilst generating turning-out yaw moments.

Usually, the risk for the driver to lose control of the vehicle is higher in oversteering situations than during understeer. In this case both systems do not conflict and critical situations due to the interactions of both systems are not to be expected. Still, an optimised control strategy of the wheel load distribution can be used to maximise the yaw moment for stabilising the vehicle. The effect of the brake intervention can be enhanced by increasing the vertical load at the front outer wheel beyond the amount necessary for roll compensation. In the active chassis investigated in this work, the additional wheel load is generated by an actuator for the individual wheel. The same effect is achieved by producing additional wheel load differences with an active anti-roll bar system. The improvement achievable with such an optimised control strategy is shown in Fig. 12.

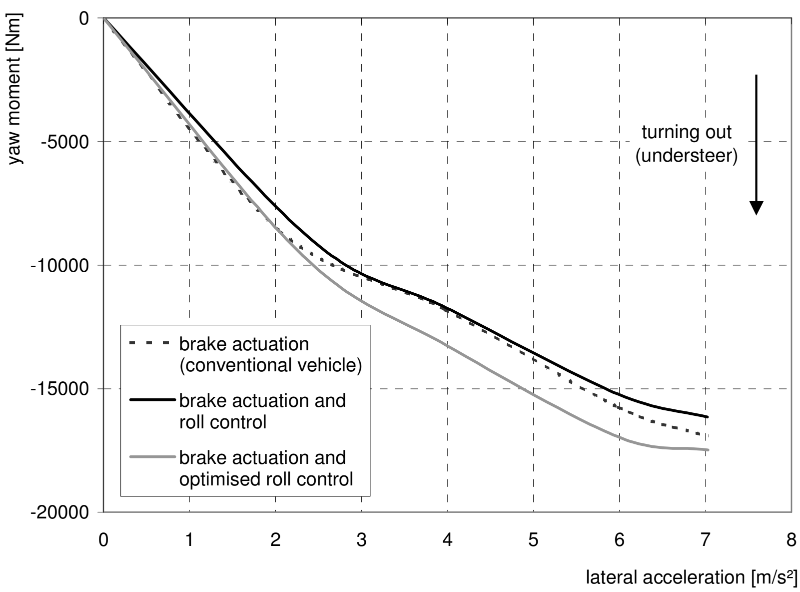

Fig. 12: Yaw moments achievable with combined interventions

If both systems act parallelly, the yaw moment generated is comparable to the yaw moment resulting from a brake intervention in a conventional vehicle (without active roll control). The minimally reduced effect for the combined actuation is due to the differences in suspension setup between the standard vehicle and the vehicle with active roll control. However, the advantage of the combined system lies in the possibility to influence the vehicle behaviour by way of continuous wheel load control before brake interventions become necessary. If the combined system is operated with an optimisted control strategy, the effects of different suspension setups are not only compensated but the overall system performance is markably increased.

CONCLUSIONS

This work establishes a process for the quantification of yaw moments generated by interventions of different active chassis systems. In vehicle testing and in standard vehicle dynamics simulation such yaw moments cannot directly be assessed. For this reason, an extension is developed on an open simulation platform for vehicle dynamics with a MATLAB/Simuklink full vehicle model. The simulation environment described offers the possibility to adapt a complex simulation model to various tasks. The integration of test track and test rig investigations is realised and interfaces for co-simulation with specialised softwae packages are provided.

This process is used for the evaluation of yaw moments resulting from roll moment distribution and individual brake interventions. For both systems the potential for generating stabilising yaw moments in the vehicle are investigated in solitary and combined operation. An optimised strategy for improved system performance is introduced and a method for preventing interference is proposed.

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