

Durability Simulation with Chassis Control Systems: Model Depth for a Handling Maneuver

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ABSTRACT

This paper makes a contribution toward a more efficient chassis durability process for the development of passenger cars, in which the simulation of relevant load data is a time-consuming part. This is especially due to the full vehicle model complexity which is usually determined by the demands of rough road simulations. However, for the load calculation on a racetrack, time saving model approaches that are more simplified might be sufficient. Our investigation comprises two levels of vehicle model complexity: one with all chassis parts modeled in a multibody system environment and one characteristic curve based model in an internal simulation environment. Both approaches consider an original chassis control system as a Software-in-the-Loop model. By the evaluation of real-world experiments the main influence factors in terms of durability are demonstrated. With the help of those highly sensitive durability criteria the measurement and simulation results are then compared. This discussion demonstrates that both model approaches show an almost excellent performance, whereas the computation times differ significantly. This combination - a durability relevant handling maneuver with the use of an original chassis control system - has not been previously considered.

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INTRODUCTION

The method of multibody system (MBS) simulations has become a key element in the chassis development process of modern vehicles and is, among other applications, used to calculate relevant design loads and to derive durability load spectra. A virtual MBS test rig, consisting of a single front or rear axle with a fixed car body, is therefore a well-known method to derive loads of individual chassis components. However, this approach requires preprocessed measured wheel forces and moments to be applied onto the model. Hence, full vehicle simulation models are more and more used to derive loads especially in the early development stages in which no test vehicles exist.

The demands on this simulation method are very high. Due to the excitations on rough roads which are required for the durability process, the model depth needs to be appropriate: digitized road profiles with high resolution, physically based tire models with a high amount of contact points, sophisticated component models suitable for high excitation velocities and deflection, flexible bodies and chassis control systems are required. Depending on the model depth, the total computation time takes between 200 and 2000 times longer than real time. Beside those maneuvers on rough roads (mainly longitudinal and vertical loads), also maneuvers on more flat roads can be relevant for the design process. An example for this is the handling course (mainly lateral loads) in [figure 1](#).



Figure 1. Handling course at the automobile test track in Papenburg (Germany) [1].

Deriving a trajectory and velocity profile for this handling maneuver is a sophisticated and iterative process. The final computations for the development process are also characterized by variant studies. The time exposure can therefore be tremendous. Concerning this maneuver it is accommodating that the most relevant signals for the subsequent durability process are the lateral wheel forces. Hence, the individual chassis components need not necessarily be modeled. A

more simplified model approach could be sufficient and would help to speed up the whole process. Nevertheless, the quality of results must not be worse compared to the more complex model. Especially in terms of durability, minor differences in load amplitude can cause major differences in the durability assessment and estimated lifetime of chassis components.

This paper is about identifying which model depth is required to meet those high demands. Therefore, two types of full vehicle models (component-based and characteristic curve based) are regarded. An original electronic stability control (ESC) model is required for both ones. Otherwise, the vehicle model would swerve or skid, due to the fact that it moves in the critical range of driving dynamics. The validity of the results is assessed by the comparison with measurement results.

PREVIOUS WORK

A large body of literature exists for each specific component of full vehicle simulations, ranging from detailed descriptions of the MBS method itself up to the coupling of submodels through co-simulation interfaces. Therefore, only a small selection of related papers on durability analyses relevant to our approach can be discussed here.

For instance, MBS simulation in terms of durability can be found in [2], where a full vehicle model was used for the investigation of braking maneuvers. Furthermore, in [3], a full vehicle MBS model was built and stimulated on different durability roads. Also, [4] described considerable demands of full vehicle simulations on rough roads, but confirmed the potential of this method to be used in future development phases. However, none of the three above mentioned sources considered a chassis control system.

One further aspect is visible in [5], where the effect of certain active systems on the durability of attached parts was demonstrated with the help of MBS simulations. Therefore, MBS models were represented as ADAMS [6] plants in Matlab/Simulink [7], and the simulation was performed in co-simulation with the controller models. For an active spring aid (activation of an additional spring at a certain compression level; modification of the load transfer path), measured wheel forces were applied onto a (fixed to ground) rear suspension. For an active roll control, a quasi-steady-state circular test was performed by a full vehicle model. The influence of the active systems was shown; however, a durability relevant maneuver and a comparison with measurement data was not part of the investigation. Also, [8] presented a detailed description of a MBS full vehicle model, including models of the suspension, steering, drive and powertrain, engine mounts, tire, body and components, driver and a suspension control system. The accuracy of the method was verified by the comparison with measurement results for damping forces on rough roads, for the vertical tire forces and body accelerations on rough roads, as well as for the dynamic behavior of the hood on a corrugated road. The influence of the suspension control system was not directly discussed. In [9, 10], a MBS full vehicle simulation including a braking control system was demonstrated, whereas the required model depth of only the controller itself was investigated. An overview of the effort and benefit of the use of original chassis control system models was given by [11]. MBS full vehicle

simulation results and measurement results were compared for braking, damping and handling relevant maneuvers. Thereby, the issue of required vehicle model depth was not considered.

Finally, another related approach, even though it is not in the field of durability, is demonstrated by [12]. For the analysis of different vehicles and drivers on a handling course, a driver-in-the-loop simulator was built. A real-time-capable vehicle model with ten degrees of freedom was a prerequisite for this approach.

The contributions of this paper are twofold. The main contribution is considered to be the discussion about the problem-oriented required model depth for the durability relevant handling maneuver. Although several papers already exist in the field of load data generation with MBS simulations, they mostly do not focus the question of model depth. Only [12] used a “simple” and therefore real-time capable model but not in terms of durability. Furthermore, all durability approaches are mostly related to rough roads. High lateral forces, induced by high cornering speeds, are not considered. A secondary contribution concerns the use of an original active chassis control system for the complex driving dynamics control. Certain systems are indeed used in several papers, but they are limited to, e.g., suspension, braking or roll control systems.

FULL VEHICLE MODELS AND ENVIRONMENT

Vehicle Model “A”

The first model is implemented within the MBS software LMS Virtual. Lab Motion [13]. It is the more complex approach (3D-model), where all relevant chassis components are modeled. The topology of the axles as well as the mass and inertia properties of all bodies are defined according to the test vehicle. For the investigation within this paper only rigid bodies were used. At the connection points to their adjacent components appropriate joints or force elements are defined. The suspension and damping for instance are represented by translational force elements. The bushings are represented by Kelvin-Voigt based models [14]. Furthermore, the complete powertrain is modeled to ensure the correct transmission of torque from the engine to the wheels. Therefore, the engine and the gearbox are described by their mass and inertia properties and are mounted via bushings. The drive torque is then applied in terms of the gear torque onto the gear output shaft. The steering is also modeled from the steering wheel down to the steering rack with its attached tie rods. Hence, the steering torque can be applied directly onto the steering wheel by the driver model. In total, the vehicle model consists of more than 2000 parameters and 200 characteristic curves.

Several quality checks are made, for instance, the behavior of the test vehicle at quasi-static suspension movement (Kinematics and Compliance Testing, K&C) is measured at a test rig and in the simulation as well. This helps to adjust the spring and bump stop characteristics within the vehicle model. Furthermore, the dynamic behavior is checked by applying measured wheel forces and torques onto each axle (fixed car body), whereas the resulting chassis forces are compared with the measured ones. The test vehicle is also

measured at a pendulum test rig to determine the inertia properties. Moreover, all bushing and damper characteristics are measured at component test rigs.

Vehicle Model “B”

The second model is implemented within the simulation environment CASCaDE (Computer Aided Simulation of Car, Driver and Environment) [15]. It is the more simplified approach, where the chassis behavior is represented via characteristic curves. The kinematic movement of the wheels is described by polynomial functions of the wheel travel and the position of the steering rack, and is parameterized automatically from the kinematics data. This pure kinematic movement is superimposed by additional camber and toe angles resulting out of the elasto-kinematics. To this end, the results of the K&C Test are included with the help of trial functions. The spring characteristic is modeled based on the wheel travel and derived directly from the mentioned test-rig. Since no struts or other components of the chassis are modeled separately only the unsprung masses and the car body mass as well as the corresponding inertia are needed for the parametrization of the mass properties. The drivetrain with the look-up table-based engine, automatic gearbox and differential gear is modeled as 1D mechanics. The transmission path of the steering system is modeled in-depth, respecting e.g. the distributed friction. The model was validated with measurements of typical vehicle dynamics maneuvers (e.g. steady-state circular test) performed with the test vehicle.

Required Submodels for Both Approaches

A reliable and efficient description of the *road surface* is of interest no matter what simulation task is currently considered. Both demands are fulfilled by the curved regular grid (CRG) [16] format in which the road description of the handling course is available. The basic idea of this approach is the storage of the height information in a regular grid, see [figure 2](#). The reference line, which is usually in the middle of the track, describes the mapping between the global coordinates (x, y) and the road coordinates (u, v) . The height information $z(u, v)$ is then stored depending on the road coordinates in the form of a regular grid in a matrix. For every global contact point, the associated road coordinates are determined and the actual height information is obtained by a bilinear interpolation between those adjacent heights. This quite demanding task is solved by an efficient algorithm.

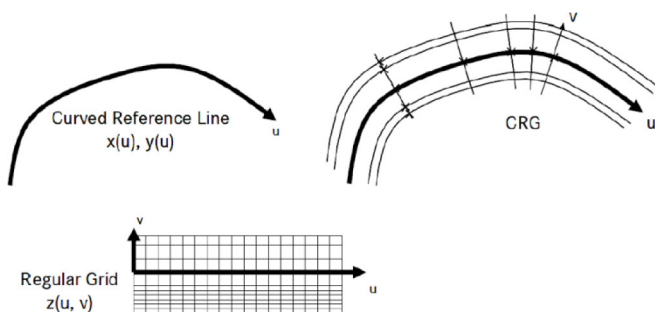


Figure 2. Basic idea of OpenCRG [16].

Tire models have a great influence on the behavior of the full vehicle models. Depending on the simulation task, there are different model depths recommended [17, 18, 19]. In this paper, the semi-empirical Pacejka model is used [20], which is sufficient for the low-frequency excitations on the handling course. The parameterization is extracted out of an existing dataset from an FTire model [21]. It was obtained by a test rig measurement of the original test vehicle tires at the Institute of Automotive Engineering of the RWTH Aachen [22].

A *driver model* is necessary in order to control the predefined velocity of the full vehicle model and to follow the desired trajectory. Therefore, two independently acting controllers are used: the pedal controller with control variables such as the accelerator and brake pedal as well as the steering controller with the steering torque as an interface to the full vehicle model. In both cases, the desired values are given depending on the current track distance which is evaluated within the OpenCRG routines. The pedal controller is a simple PID-controller with delayed D-Component and an output range of $[-1; 1]$. The control deviation is obtained from the comparison between the actual velocity and the predefined velocity profile. A value of -1 causes a full braking and a value of 1 a full accelerating action. The determination of the required steering angle is based upon the linear single-track model [23]. It is calculated depending on the current position and orientation of the vehicle model on the track, see [figure 3](#). The curvature κ of the desired trajectory is considered as an open loop variable, whereas the current track offset error Δv and the orientation error $\Psi_{veh} - \Psi_{Traj}$ are closed loop variables. All of them are provided by the online OpenCRG evaluation. The desired steering wheel angle is then adjusted by an underlying controller for the steering torque which is applied to the steering wheel. This is to avoid a kinematic constraint and discontinuous changes of the steering velocity.

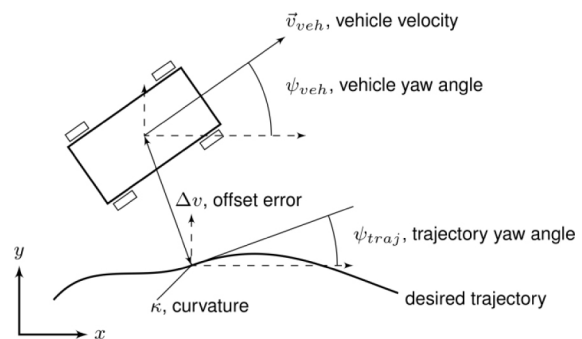


Figure 3. Basic idea of the steering controller.

The ESC control system exists as a Software-in-the-Loop (SiL) model in Matlab/Simulink. It is provided by the ESC system supplier and contains the original code applied on the electronic control unit inside the test vehicle. Hence, the whole functionality of ESC and its underlying control systems is available. In detail, this is the antilock braking system (ABS), the traction control system (TCS), and a vehicle dynamics control (VDC) with the engine drag torque control (EDC). ABS prevents the wheels from locking-up while braking through control of the braking pressures at each wheel brake. TCS ensures the traction of the driven wheels by interventions into the engine torque and/or braking of one or more wheels. EDC prevents the wheels from locking-up in case the engine drag torque locks down the wheels. This can happen if the driver quickly releases the