Vehicle Model for Limit Handling: Implementation and Validation

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Abstract

This paper describes how a vehicle model from the VehicleDynamics Library is configured, parameterised and validated for predicting limit handling manoeuvres. Particular attention is paid to the selection of subsystem models with suitable levels of detail, as well as the selection of measurements performed and measuring equipment. A strong principle running throughout the presented work is component-based design where parameterisation is performed on subsystem levels, no tuning on the final vehicle models is done. As a final test, the vehicle model is exposed to a sinusoidal steering input. It turns out that the model is able to reproduce the vehicle's behaviour for the driving scenario selected up to the limit of adhesion.

Keywords: Vehicle Dynamics, Component-Based Modelling, Limit Handling, Validation

1 Introduction

Safety plays a prominent part in the development of vehicles. A large portion of the development is devoted to vehicle stability and the control task to maintain stability even under severe driving situations. Here, multi-body modelling becomes a powerful tool to preserve competitiveness and keep the development time within the given timescale. One reason for this strength is the ability to offer an improved understanding of the vehicle and also to support the ranking of its' design variables without any access to the physical vehicle [1]. Moreover, the development of vehicle control is facilitated if the vehicle plant pose an adequate response.

Driving conditions such as strong side motion of a vehicle are often considered to be unsafe, since the driver risks losing control of the vehicle. Such potentially dangerous situations need to be identified, and accordingly, there is a great deal of interest in reproducing this class of scenarios. However, the combination of fast transients and high accelerations triggers strong non-linear vehicle characteristics, which in turn make great demands on the model used. One example of this manoeuvre is the single lane change [2]. An example of such a *limit-handling* manoeuvre is given in Figure 1.

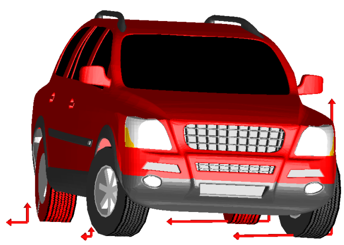

Figure 1: Simulation of the vehicle model undertaking a severe lane change maneuver. The arrows visualize the forces generated in the tyre contact patch.

A fundamental requirement when considering simulation as an alternative to real-life testing is validity. A model must not just be valid in the sense that it captures the results of already tested scenarios and pa-

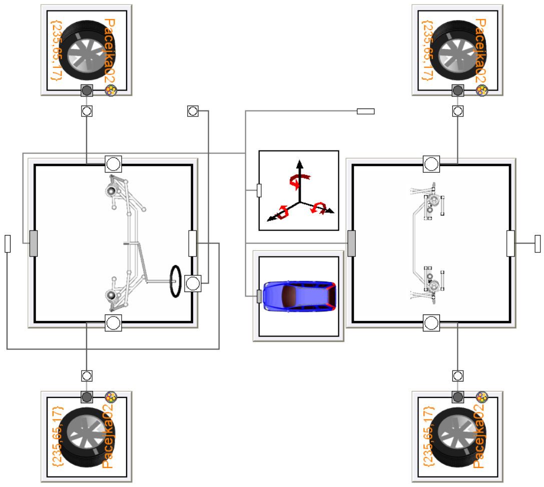

Figure 2: Diagram view of the chassis model layout with body, suspensions and wheels. The component above the body keeps track of the vehicle motion and handles initialisation.

rameterisations. It must especially be able to predict the effects of new scenarios, parameters and configurations, and therefore, the aim of this work is to show that by having valid subsystem and component models, the resulting vehicle model shall also be valid.
This paper will demonstrate how a vehicle model from the VehicleDynamics Library (VDL) [3] is put together from parameterised subsystems and verified against limit handling measurements. Special attention is given to choice and configuration of vehicle model, and parameterisation of subsystems, especially suspension characteristics.

2 Vehicle Model Configuration

The test vehicle is equipped with front McPherson and rear multi-link suspensions. The chassis model is implemented as a multi-body model in VDL with a rigid body onto which the suspensions and wheels are mounted as illustrated in Figure 2.
For the suspensions, there are two main approaches to modelling the kinematics and compliance characteristics, respectively. The kinematics can either be specified by the hard point locations of the links or as tabular characteristics depending on wheel travel (and steering for the front suspension). Compliance is either given by the individual characteristics of the elastic elements or as a lumped characteristics of the whole suspension. The required tabular characteristics and lumped elasticities can be generated from kinematics and compliance (K&C) analysis.

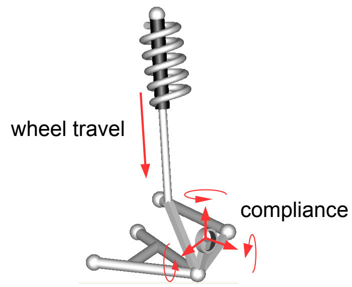

Figure 3: Animation view of the left front suspension linkage with the 7 DOF indicated.

In this application, suspension is modelled as ideal kinematic multi-body linkages with one degree-of-freedom (DOF) for wheel travel. The compliances that are caused by bushings and material deflection is lumped in one element between the wheel carrier and the hub. This approach is similar to what has been used in e.g. [4] and is illustrated in Figure 3.
The compliance adds 6 DOF for each suspension linkage, with wheel travel totally 7 DOF. Together with the front steering compliance there is a total of 7x4+1 DOF with 2 states each for position and velocity, i.e. 58 states for the suspensions.
The wheel models use the Pajeka'02 tyre force model in VDL, implemented according to [5]. This representation was chosen because it is considered to be state of the art and because there were available tyre data in this format. The tyre force model has two states for lateral and longitudinal relaxation lengths (first order dynamics). Together with the wheel's spin DOF, there are therefore 4 states per wheel and additionally, there are 6 degrees of freedom (12 states) for the vehicle's body motion, giving in total 58+4x4+12=86 states for the chassis model.

3 Suspension parameterisation and verification

As already mentioned, the model used contains both component parameters and lumped characteristics. Most parameters are taken from construction data such as geometries, masses and inertias, but some are calculated from measurements on isolated subsystems. Here, this is illustrated for the suspension compliance characteristics.
The kinematics and the compliance of the front and rear suspension have been measured in a dedicated rig where the car body is fixed and a post is mounted on

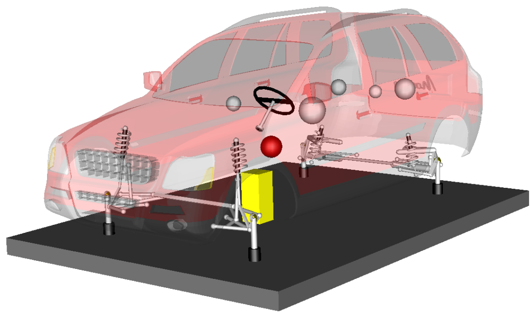

Figure 4: Virtual version of the test rig used for kinematics and compliance analysis. The chassis body is kept fixed and one actuator at each hub applies forces while the motion is registered using cameras. The spheres indicate centre of gravity and payloads.

each wheel hub (Figure 4). Using this post, forces and torques can be applied to replicate different driving scenarios. In this case, forces have been applied in the lateral ($\hat{e}_y$), longitudinal ($\hat{e}_x$), and vertical ($\hat{e}_z$) directions, both at the wheel's centre (C) and at the estimated tyre-road contact point (W). For every load case, the rotations ($\bar{p}$) and translations ($\bar{r}$) of the hub are measured.

For a force applied anywhere other than at the hub, there is both a resulting torque ($\bar{t}$) and a force ($\bar{f}$) at the hub so by comparing two equal forces applied at different locations, the torque dependency can be calculated, and thus

$$\underbrace{\begin{pmatrix} \bar{f} \\ \bar{t} \end{pmatrix}}_{\bar{F}} \rightarrow \underbrace{\begin{pmatrix} \bar{r} \\ \bar{p} \end{pmatrix}}_{\bar{\Delta}}. \tag{1}$$

Assuming that the dependency is linear, equation 1 can be rewritten as $\bar{\Delta} = \mathbf{C}\bar{F}$ where $\mathbf{C}$ is a 6x6 compliance matrix and $\mathbf{C}^{-1}$ the corresponding stiffness matrix, which is required for the compliance element.

As described in Section 2, there are 7 degrees of freedom for each suspension, 6 from lumped compliance element and 1 from wheel travel. Unfortunately, from a numerical point of view, it is hard to separate these dependencies since springs in a car are a factor >100 more compliant than the contribution from the compliance element. However, since the deflection in the compliance element is small in comparison with the total wheel travel, it is assumed that the accuracy requirement of the z-deflection from the compliance component is low. By keeping the vertical position of the measured hub fixed while forcing the opposite wheel hub to move, a force (f'_z) is implied through the stabiliser linkage. This affected the measured hub ($\bar{\Delta}'$)

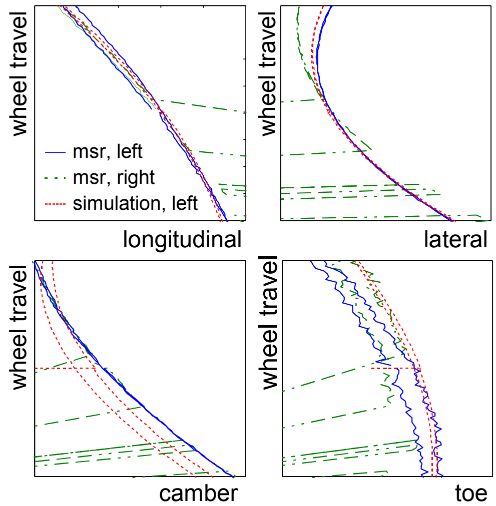

Figure 5: Suspension characteristics showing left and right side measurements (blue,green) and simulation (red). Longitudinal, lateral, camber and toe motions on the plot's x-axes and wheel travel on the y-axes.

so that a new column could be calculated from $\bar{\Delta}'/f'_z$. This column is used as a replacement in C and the original column is used to define the spring rates. Correspondingly, the linear compliance of the steering system is also extracted from these measurements.

With $\mathbf{C}$ calculated for each suspension linkage, the wheel travel tests performed in the K&C rig are carried out virtually with VDL to verify the behaviour of the suspension models. Figure 5 shows a comparison between the K&C measurement and the corresponding simulation for different wheel travel from.

Since it is a well-known fact that hydraulic dampers may deviate from the specification, all four dampers were disassembled and measured in a damper rig. The non-linear force-velocity characteristics retrieved from the damper measurements were used for purpose of modelling. This was carried out by linearising the characteristics piecewise for the compression and expansion phases respectively. One complicating aspect is the ability of the rear dampers to adapt to load changes. In brief, this can be explained as a preload in parallel with the damper that adapts slowly to the vehicle's load and the driving conditions.

4 Validation of the vehicle model

As already mentioned, limit handling involves both fast transients and highly non-linear characteristics. To create a validation in such circumstances, it is sufficient to measure the state-trajectory of the vehicle body. However, even if the response from the sim-

ulations coincides with the measured trajectory, little information can be retrieved about the correctness of the subsystem models used.
In order to identify and improve the modelling and parameterisation of these subsystems, it is advisable to extend the handling measurements to include even more mechanical phenomena. Obviously, since the tyre forces contribute substantially to the vehicle motion, added to which they are well-known to be hard to model, they become an important source to be monitored. In addition, at limit handling the vehicle executes large roll or/and pitch motions, and as a consequence, the suspension deflections become large. For large deflections, there is a significant alteration in toe and camber as illustrated in Figure 5, which in turn influences the tyre forces. Moreover, the compliance characteristics changes during the deflection. The most obvious situation is the entrance of the bump stop for large suspension compressions. For these particular reasons, the supervision of the deflection of all four corners becomes a viable option.

4.1 Instrumentation selection

Keeping the information discussed above in mind, the vehicle was equipped with a gyro-platform, four torque measuring wheels and sensors for deflection of all four corners. The gyro-platform measures the rotations (roll, pitch and yaw speeds) and accelerations (in x, y and z axes) of the car body. As illustrated in Figure 6d, the gyro-platform was mounted between the front seats. The standard wheels were replaced by the torque measuring wheels, which are able to measure tyre forces and wheel torques in and around x, y and z axes. This is possible due to strain gauges positioned at the rim. The suspension deflection instrumentation comprises levelling sensors, which measure the distance between the wheel hubs and car body.
In addition, signals from Controller Area Network (CAN) were logged in order to monitor wheel speeds and the states of the engine, brakes, gears and Haldex differential. All signals (approximately 90) were collected and sampled at 50 ms in a computer. Finally, a steering robot was mounted to support for steering input at a high level of accuracy and repeatability. Figure 6 illustrates the measurement setup of the vehicle.
One important issue was to judge and assign a span for the accuracy of these measurements. Another challenge related to the large amount of redundant data retrieved from the vehicle instrumentation. One example of this redundant data relates to vehicle speed, which can be taken from wheel speed sensors (from CAN) and also via the gyro-platform. To select the best data, information from sensors was compared, and later on, consolidated or arbitrated. In addition to this, the vehicle's corner weight and ride height were measured manually.

4.2 Driving scenarios

The driving scenarios were selected with two purposes in mind; reference and validation. To meet the first requirement, tests were carried out under conditions that allowed the measuring equipment to be tested as independently of the vehicle where possible. Typical examples are to expose the vehicle to constant conditions such as gradients in different directions.
Tests to measure the vehicle behaviour during steady-state manoeuvres were performed with both purposes in mind, and included, braking and acceleration to verify longitudinal load transfer and the resulting pitch of the body. Also steady-state cornering was executed by driving with a constant radius of 45 metres, while gradually increasing the vehicle speed up to the maximum achievable lateral acceleration. Thus body roll and tyre normal load distribution could be validated.
The handling manoeuvres used solely for validation, were selected to cover as much of the dynamics of the vehicle as possible up to its limits. This group of tests were conducted to force the vehicle into transient motions:

- Step steer using the steering robot
- Single lane change, sinusoidal steering input from the steering robot
- J-turn and simultaneous relief of the gas pedal (oversteer situation)
- Double lane change with a driver

Finally, the test procedure above was repeated under conditions where roll and yaw stability control were deactivated, in the test data presented in this report.

4.3 Validation results

Validations was finally achived on the most extreme manoeuvres which is illustrated here by a single lane change test with all active safety systems turned off. The vehicle response, both measured and simulated, is shown in Figure 7. The amplitude and the vehicle speed are set to reach the limit of available grip to trigger most non-linear characteristics. From the slight bouncing in the roll angle it can be seen that the bump

Figure 6: The tested vehicle equipped with a) wheel travel sensors, b) steering robot, c) measuring wheels and d) gyro platform

stops in the suspensions are activated which also gives the same effect on lateral acceleration. It can also bee seen that the side slip angles follow each other closely throughout just over half of the manoeuvre when they start to diverge. During the first half, the two simulations (red and green) are practically the same for all vehicle states while later on, the red and green are closer. At the time when the severe single-sine tests were carried out, the proving ground was slightly moist, as appose to the rest of the testing period when the ground was dry. A qualified guess is that μ was slightly lower when these manoeuvres were performed, which is also supported by the fact that the simulation with μ gives a behaviour closer to that measured. Another interesting aspect is the fundamentally different results for $\mu = 1$ and $\mu = 0.95$. While the first simulation shows a vehicle that slowly recovers low side slip, the latter one continues to spin out and never recovers. During this type of severe maneuvers, even very slight changes to the surrounding conditions can have a great impact on handling behaviour.

Another effect that is of particular importance during manoeuvres where a bump stop is involved is the ride height. The sudden change in load transfer can give completely different results, as illustrated in Figure 8, showing the effect of a change in ride height. The red curve shows the lateral acceleration for the setup with the measured ride height and is the same as the red curve in Figure 7. The green curve shows the same setup, but with the default ride height taken from construction parameters. The default ride height was higher in the front and lower at the rear, compared to the measured ride height. As a result, the default settings gives a later bump stop activation and thereby slower turn-in and more roll motion.

The influence on the suspension elasticity on the vehicle characteristics is well-known and provides an important means to tune a vehicle's characteristics. This effect is also seen in many of the manoeuvres investigated. However, for the limit manoeuvre presented in in Figure 7, changes in compliance have less effect on the results. As an example, a change of the rear compliance by a factor of 10 only gives very slight changes to the trajectory. This is expected to be due to the high amplitude which makes the front tyres reach their saturation limit almost immediately, thereby the load transfer has a significantly higher effect on the generated lateral force compared to changes in wheel angles [6].

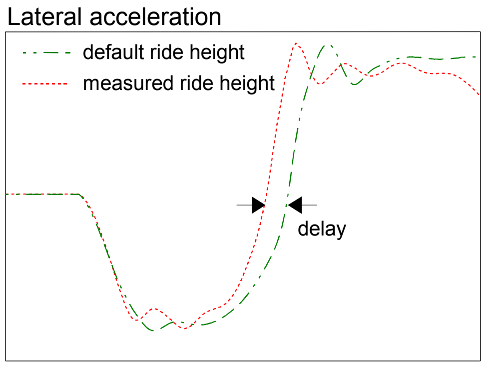

Figure 8: Lateral acceleration for two different ride height settings, the phase delay is 0.18s.

5 Conclusions

This paper has presented a methodology for validation of a vehicle model, which is to be used in a broad range of driving scenarios. A suitable approach for the selection of wheel suspension parameters has been pre-

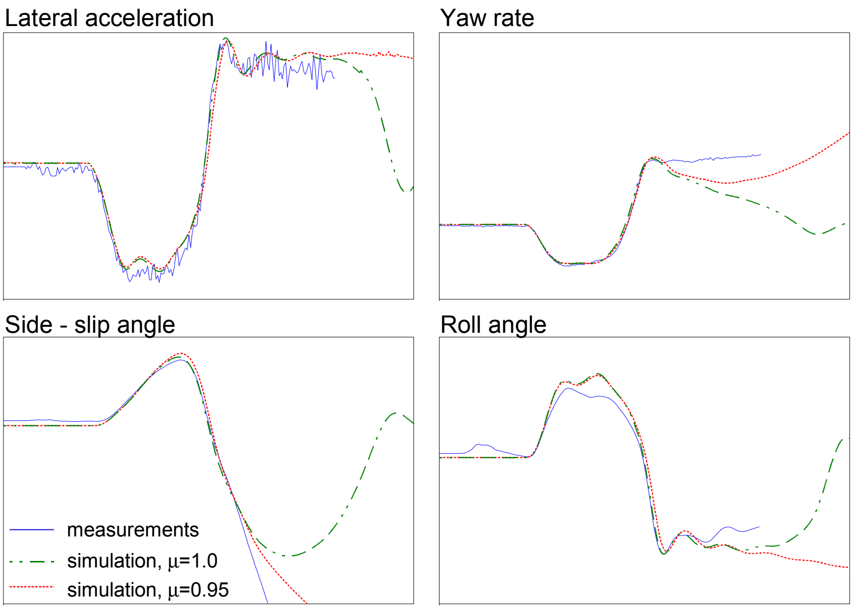

Figure 7: Response of open loop sine excitation (single lane change) at 80 km/h, measurements (blue) and simulations with $\mu = 1$ (green) and $\mu = 0.95$ (red). Lateral acceleration (upper left), yaw rate (upper right), side slip angle (lower left) and roll angle (lower right).

sented. In addition, the wheel suspension models have been validated through measurement in a chassis rig. A strong principle throughout the presented work is component-based design where parameterisations are done on sub-system levels and no tuning on the final vehicle models is performed.

The final validation involves reproduction of a real driving scenario, which has been represented here by measurement of the state-space trajectory. The results indicate that it is feasible to design a valid vehicle model, at least up to limit handling, from valid sub-systems without involving additional tuning. Finally, it is demonstrated that a minor error in the estimation of unknown environmental factors, such as road friction, risk to jeopardise the correspondence.

From these findings, it is evident that the methodology presented is a viable tool for use in the vehicle developments. In addition, it has a great potential to support for the development of safer vehicles and facilitate the development of preventative safety functions.

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