Test-driven full vehicle modelling for ADAS algorithm development

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Abstract

ADAS system development for safety as well as comfort is a major activity for all AOEMs and dedicated system suppliers. The development of these systems relies heavily on the availability of accurate driving dynamics models to validate the control algorithms and verify vehicle performance in realistic driving scenarios.

AOEMs usually have access to high-fidelity full vehicle models but the availability of such models poses a significant challenge to software and sub-system suppliers. In order for them to develop their ADAS solutions, an accurate test-based model identification process using prototype or benchmark vehicles would be highly desirable.

In this paper, a stepwise approach for the identification of the vehicle parameters in order to create an accurate 15-DOF vehicle dynamics model is proposed. The approach relies on an optimized use of vehicle driving test data to minimize test bench or laboratory testing. The test data is used to identify the various vehicle parameters such as inertia parameters, suspension stiffness and damping and tire stiffness. Subsequently the model is validated through simulation and test. Finally, the use of the model in context of ADAS development is illustrated by integrating it in a traffic scenario simulator allowing MiL/SiL/HiL controls validation for various driving scenarios such as a lane change or autonomous valet parking.

Introduction

In general, car OEMs and suppliers are driven to reduce the number of physical testing and shift their development towards CAE driven processes as much as possible [1]. With CAE model available at early stages of development, they can be used for benchmarking, early performance assessment and what-if games [2].

The development of ADAS algorithms in particular, and vehicle dynamics controls in general, rely heavily on the availability of accurate simulation models to that describe the vehicle dynamic driving behavior accurately. Such algorithm development based on testing alone would prove an almost impossible task as too many parameters need to be tuned in context of too many scenarios.

Car OEM’s mostly rely on dedicated laboratory measurements to derive these vehicle dynamics models using Kinematics & Compliance (K&C) test benches and Vehicle Inertial Measurement Facilities (VIMF). The data acquired from the bench measurements is used to derive the parameters of the detailed vehicle CAE model [3].

Suppliers on the other end, who often carry quite a lot of the tasks in vehicle dynamics development, don’t always have access to these extensive (and expensive) testing facilities, which leads them to use rather simplified vehicle models such as a bicycle model. As we will demonstrate in the subsequent chapters, bicycle models have significant limitations and as such limit their applicability in the vehicle dynamics development.

Similarly, it is for car OEM’s not always possible to do full detailed benchmarking on competitor vehicles as this would be prohibitively expensive.

In order to alleviate both the cost as well as the bench accessibility problems, this paper proposes a novel methodology to derive accurate 15-DOF vehicle dynamics models without resorting to dedicated test benches or even single component tests. The goal is to maximize the parameter extraction for the vehicle model in true operating conditions. This also maximizes the correlation of the model with the actual vehicle behavior as the parameters are extracted from actual driving conditions and therefor will capture the correct interaction between all the components.

The paper is built up in a number of chapters, starting with the introduction of the 15-DOF model in chapter one. The subsequent chapter will focus on the characterization of the axles which relies mostly on driving measurements but does include some laboratory work. In the third chapter, the tire model is future discussed, showing how a tire model can be derived from full-vehicle driving tests. In the fourth chapter, the focus is on the inertial properties and damping characteristics of the model and deriving those from full vehicle IMU measurements in specific driving maneuvers. In the final chapter, some applications of the model are highlighted, and it’s use in ADAS controls development illustrated.

The 15-DOF vehicle model

Vehicle dynamics models can be built in different ways, with full 3D multi-body models including flexible finite-element components being the most detailed, but also very complex and complicated to create. Less complicated but still accurate 1D simulation models are therefor used more often in the context of vehicle dynamics simulation [4]. Both due to the fact that less information regarding the vehicle is needed [5] and because they calculate much faster, which in the context of controls development and in view of HiL controls validation is not only a benefit, it also becomes a necessity for real-time applications. Nevertheless, it has been proven that 1D models are still accurate enough in the context of vehicle dynamics modeling and controls development [6].

One of the more popular 1D vehicle dynamics models is the so-called bicycle model which is proposed in literature and used in various studies [7]. In this paper however, the choice is made to use a 15-DOF model instead as it proves to be more accurate in transient behavior modeling. Figure 1 shows the different DOF utilized in such model and its implementation in Simcenter Amesim (figure 2).

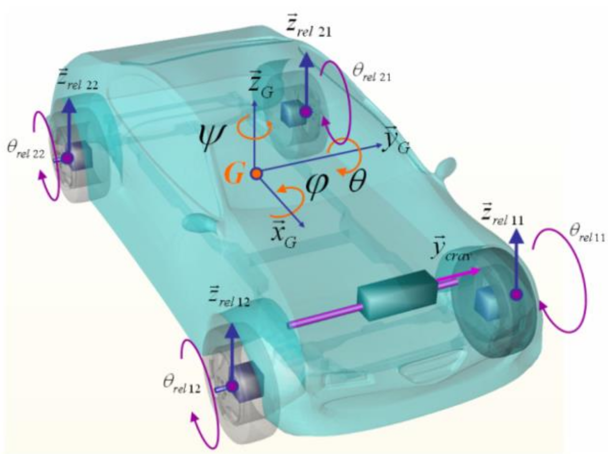

Figure 1. 15-DOF vehicle model description

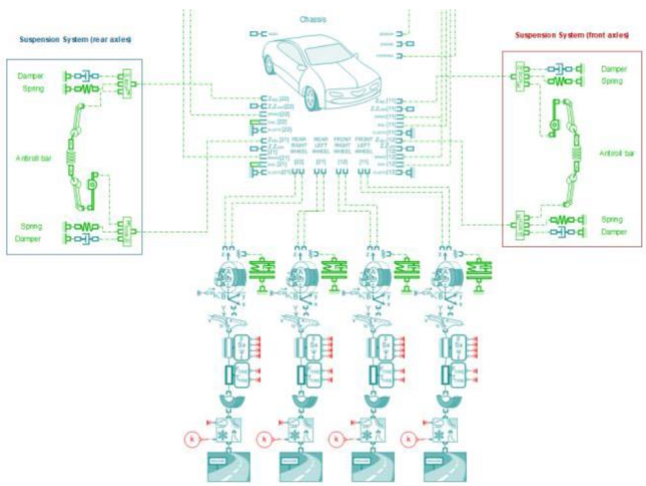

Figure 2. Simcenter Amesim 15DOF model

To show this in examples, a step steer maneuver and a constant sine steer maneuver are compared between both models.

- The first (figure 3) shows the difference between both models in a step steer maneuver and it can be clearly observed that both the overall vehicle side slip (global behavior) as well as the axle lateral force (local behavior are different)
- The second (figure 4) shows the left and right wheel tire forces. The bicycle model shows the same behavior for left and right tires, whereas the 15-DOF model shows a difference between the left and the right. This difference is explained by the different vertical loading experienced by the inner and outer wheel due to the weight transfer of the vehicle. The bicycle model does not capture this effect.

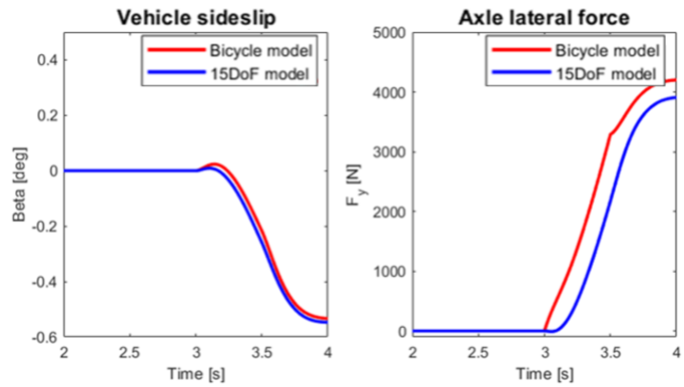

Figure 3. Difference between 15degree-of-freedom and bicycle model in the vehicle sideslip transient response in the first part of a step steer maneuver and its effect on the rear axle lateral force.

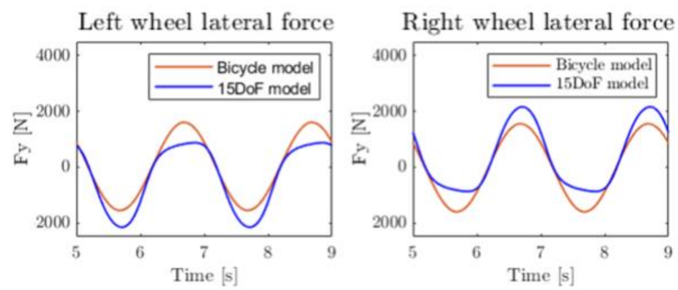

Figure 4. Comparison of the lateral force, acting on left and right rear wheels, between the 15-DOF and bicycle model in a sine steer maneuver

Chassis parameter estimation.

With the model definition chosen, the parameters of the model need to be characterized. In this chapter the focus is on the spring stiffness, anti-roll bar stiffness, tire vertical stiffness as well as the front and rear roll centers to define the roll axis and finally the center of gravity height.

Normally, these parameters would be derived from K&C test bench measurements but in this paper, the goal is to derive them from full vehicle measurements instead. Table 1 gives an overview of the maneuvers and related parameters as used for identification in this paper.

Maneuver	Specifications	Output
Ramp steer	Speed = 100km/h SWA rate = 25deg/s	Tire model parameters
Sine sweep	Speed = 100km/h Freq = 0.1 – 5 Hz SWA amplitude = 30deg	Inertia and damping
Sine*	Speed = 100km/h Freq = 0.5 Hz SWA amplitude = 30deg	Roll axis position

* This test has been validated for multiple speeds and frequencies

Table 1. Full vehicle maneuvers for parameter estimation

Spring, anti-roll bar and spring vertical stiffness

The first parameters to determine are the spring, anti-roll bar and spring vertical stiffnesses. The process to determine these starts from the observation that a car is a hyper-static system in which the suspension stiffnesses determine how the load is distributed over the 4 wheels.

This is represented by (1), where k_{ij} are the stiffness contributions and z_j and F_i are, respectively, the vertical displacements and the forces applied to the four wheels.

$$\begin{bmatrix} k_{11} & k_{12} & k_{13} & k_{14} \\ k_{21} & k_{22} & k_{23} & k_{14} \\ k_{31} & k_{32} & k_{33} & k_{14} \\ k_{41} & k_{42} & k_{43} & k_{44} \end{bmatrix} \begin{bmatrix} z_1 \\ z_2 \\ z_3 \\ z_4 \end{bmatrix} = \begin{bmatrix} F_1 \\ F_2 \\ F_3 \\ F_4 \end{bmatrix} \qquad (1)$$

As a consequence, it is possible to determine the stiffnesses by applying a set of symmetric $\begin{bmatrix} z_1 \\ z_2 \\ 0 \\ 0 \end{bmatrix} \begin{bmatrix} 0 \\ 0 \\ z_3 \\ z_4 \end{bmatrix}$ and asymmetric $\begin{bmatrix} z_1 \\ 0 \\ 0 \\ 0 \end{bmatrix} \ddots \begin{bmatrix} 0 \\ 0 \\ 0 \\ z_4 \end{bmatrix}$ vertical displacements at the level of the front and rear suspension and measuring the wheel loads and chassis displacements. The actual stiffness estimation is then an inversion process: the wheel loads are caused by the stiffnesses being compressed/extended by the displacements, hence it is possible to derive the stiffnesses from the loads if the displacements are known.

Roll centers and roll axis

A car body rolls as a result of the cornering loads acting on it. The axis around which it rolls is called the roll axis and is determined as the line that connects the roll centers at the front and rear axle planes. In a next step the vehicle roll center height will be used to determine the CoG (Center of Gravity) height.

To determine the roll center heights, 2 accelerometers at different heights measure the lateral motion of the car body. The following kinematic formula (2) would allow the extraction of the roll center height if the car body would be subject to a pure rolling motion [8].

$$H_{RC} = \frac{a_{y1} z_{s2} - a_{y2} z_{s1}}{a_{y1} - a_{y2}} \qquad (2)$$

Where H_{RC} is the roll center height, a_{y1} and a_{y2} are the accelerations of the two accelerometers in the longitudinal plane of the vehicle at different heights, and z_{s1} and z_{s2} are the vertical positions of the two accelerometers.

In laboratory conditions, and even though not easy to achieve, one could imagine applying a pure roll motion of the body on the suspension, but in real driving conditions this is definitely not possible. At any point in time, the car will be subject to motion in all 6 DOF of the body which means that formula (2) cannot be used as-is to estimate the roll center height.

As an illustration, figure 5 shows the denominator and numerator of formula (2) plotted for a laboratory measurement using shakers to excite the vehicle. Each data point on the bleu graph allows extraction of the instantaneous roll center height. The graph shows a high amount of hysteresis which is due to the effect of yaw on the data. It will also become clear that the slope of the averaged curve (in red) is completely different from the filtered curve (figure 6) and the validation data from K&C testing (figure 7)

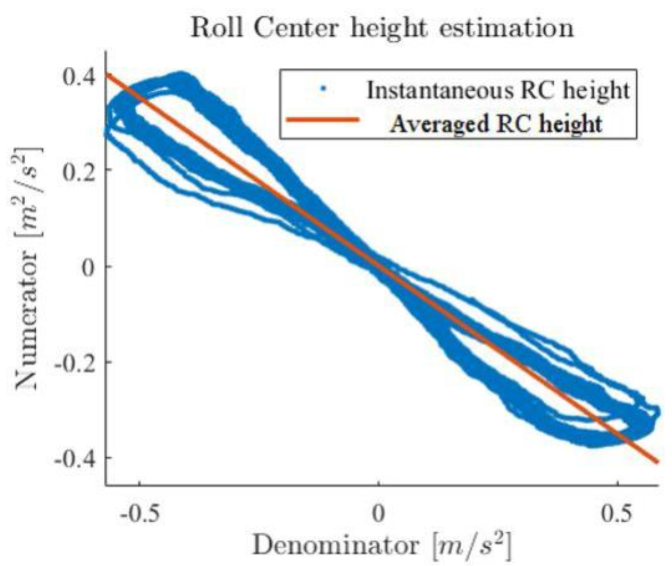

Figure 5. Roll center height estimation without vehicle 6DoF motion filtering, measured lateral acceleration used for RC height estimation

The 6-DOF data from the vehicle needs to be filtered to extract the roll components out of the full vehicle motion. This is done using a geometric reduction approach (3), where the 6 rigid body modes $(a_x, a_y, a_z, R_x, R_y, R_z)$ are decoupled starting from the accelerations measured by the accelerometers on the body (a_{ix}, a_{iy}, a_{iz}) and their positions (x_i, y_i, z_i).

$$\begin{pmatrix} a_{1x} \\ a_{1y} \\ a_{1z} \\ a_{2x} \\ \dots \\ a_{nz} \end{pmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 & z_1 & -y_1 \\ 0 & 1 & 0 & -z_1 & 0 & x_1 \\ 0 & 0 & 1 & y_1 & -x_1 & 0 \\ 1 & 0 & 0 & 0 & z_2 & -y_2 \\ \dots & \dots & \dots & \dots & \dots & \dots \\ 0 & 0 & 1 & y_n & -x_n & 0 \end{bmatrix} \begin{pmatrix} a_x \\ a_y \\ a_z \\ R_x \\ R_y \\ R_z \end{pmatrix} \qquad (3)$$

Once the data is filtered, the effects of yaw and lateral motion are removed and the roll center estimation becomes a stable process through linear regression.

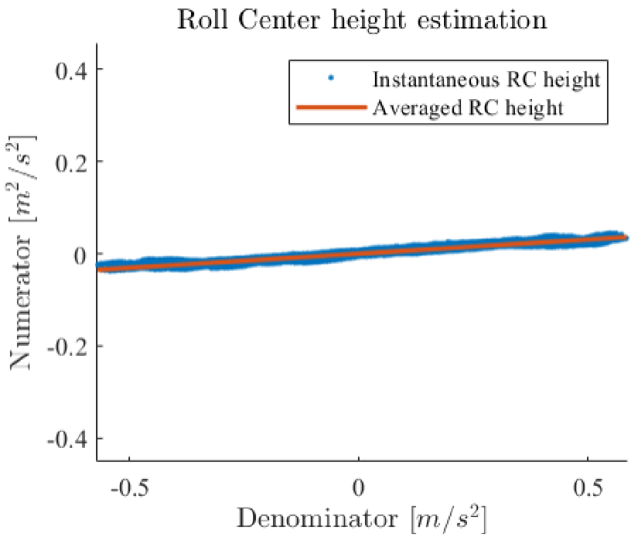

Figure 6. Roll center height estimation with vehicle 6DOF motion filtering, only pure roll motion used for RC height estimation

With the roll centers in the front and rear axle planes determined, the roll axis is defined as the line connecting front and rear roll centers.

To validate the methodology, the resulting roll centers and roll axis are compared to the results obtained from traditional K&C testing. Figure 7 shows the correlation between the roll axis as obtained from K&C testing and the same as derived from pure track driving data. The closeness of both axis shows the validity of the approach for model parameter estimation.

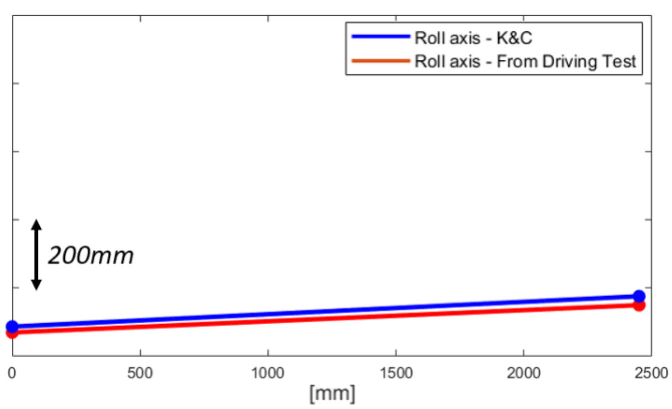

Figure 7. Comparison of the roll axis position estimated with the method proposed in the paper and the results from a K&C bench.

In addition to the advantage of deriving the roll axis from pure driving data for the purpose of extracting the parameters for the vehicle model, the method gives insight in how the roll axis might change during a steering maneuver as the estimation is done at every data time step. This provides additional insight in how the vehicle dynamics change and its correlation to the driver perception linking subjective feel to objective measurements.

Once the roll axis defined, and the roll stiffness known from the previous analyses, it is then possible to determine the COG height from the data acquired in a step steer maneuver according to formula 4:

$$M * a_y y(H_{CoG} - R_{RC}) = K_\theta \theta \quad (4)$$

Where M is the mass of the vehicle, a_y is the lateral acceleration, H_{CoG} and H_{RC} are, respectively, the height of the center of gravity and of the roll center, K_θ is the total roll stiffness of the vehicle and θ is the roll angle.

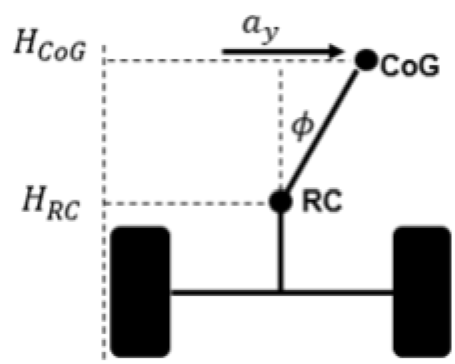

Figure 8. Schematic of the vehicle rolling behavior described by Equation 2.

Tire modeling

The last part of the vehicle model to be determined is the tire model.

Many tire models exist is literature and in software implementations. Very detailed models such as the Magic Formula tire model (MF-tire) [9] which can give an accurate representation of the tire's dynamic behavior also require dedicated test bench measurements on the tire/wheel in isolation and its parameters cannot be derived from full vehicle measurements. Very simple tire models on the other hand, such as the Dugoff tire model [10] are simpler to determine but lack the needed information to capture important details such as the influence of the tire vertical load on the self-aligning torque of the wheel.

In order to achieve the needed accuracy and at the same time only use full vehicle measurement data, a new model is developed. The model derives its parameters from global (IMU) measurements and front wheel steering angle measurements in a limited set of maneuvers. From the measurements, the cornering stiffness can be derived for a range of driving conditions. Starting from the bicycle model equations in quasi-static conditions, it is then possible to derive an adaptive cornering stiffness model.

The tire stiffness model is then created in a 2-step approach: after first estimating at the cornering stiffness at axle level, it is then divided over the left and right tires in a second step. The distribution of the axle cornering stiffness over left and right wheels is done under the assumption that the axle lateral forces distribution is proportional to the vertical load distribution over inner and outer tire (5). This assumption has been validated for moderate lateral loads using simulation models.

$$F_{y_{left}} = \frac{F_{z_{left}}}{F_{z_{axle}}} F_{y_{axle}} \quad (5)$$

To model the transient behavior, a fixed relaxation length and first order lag formulation is build in to model the lateral force build-up on each tire accurately , as presented in equation (6), where σ is the relaxation length, α is the sideslip of the wheel, v_x is tire's longitudinal speed and v_s is tire's steady state lateral slip speed. As such, the combination of dedicated transient modeling and the lateral load distribution allow the model to capture both quasi-static and transient behavior.

$$\sigma\dot{\alpha} = v_s - v_x\alpha \tag{6}$$

The last part of the tire model is to introduce the self-aligning torque. This is done by using the brush model to estimate the pneumatic trail value. The inclusion of the self-aligning moment allows for a better representation of the vehicle dynamics and can be used to update the cornering stiffness values calculated for each tires and enhance the capabilities of the presented tire model. This is represented by the comparison of the equations of the dynamics of the bicycle model (7, 8) and the updated model (9, 10), where the longitudinal forces are not included, since only the lateral dynamics is present in this dissertation.

$$F_{y_{tot}} = F_{y_1} + F_{y_2} \tag{7}$$

$$F_{y_{tot}} = F_{y_1} + F_{y_2} \tag{8}$$

$$F_{y_{tot}} = \sum_{i=1}^{4} F_{y_i} cos\delta_i \tag{9}$$

$$M_{z_{tot}} = \sum_{i=1}^{4} F_{y_i} cos\delta_i x_i + \sum_{i=1}^{4} F_{y_i} sin\delta_i y_i - \sum_{i=1}^{4} M_{z_i} \tag{10}$$

The nomenclature is explained in figure 9 and M_{z_i} is calculated in (11), where the pneumatic trail t_p is calculated from [11] under the assumption of small sideslip angles.

$$M_{z_i} = t_{p_i} F_{y_i} \tag{11}$$

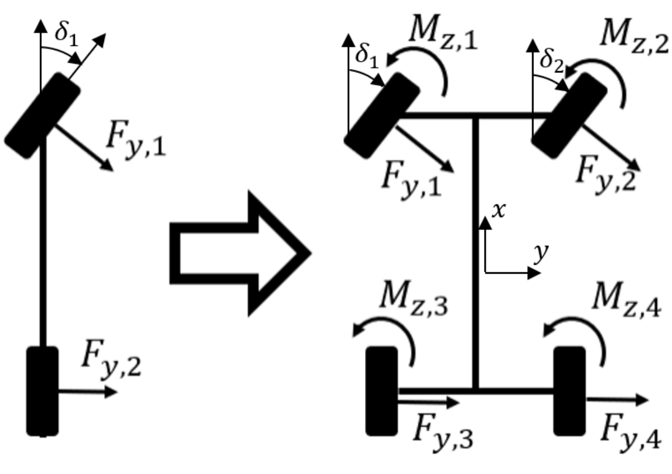

Figure 9. Wheel positions and pneumatic trails introduced in the model, in order to update the lateral wheel forces ($F_{y,i}$) and introduce the self-alignment moment ($M_{z,i}$).

Validation of the tire model through simulation

In order to validate the tire model, 3 different 15-DOF vehicle models are created:

- A reference model using a MF-Tyre/MF-Swift 6.2 [9]
- A base FV tire model with a simplified tire model using a tire representation directly from a bicycle model using a constant cornering stiffness for the tires
- A new tire based model using the tire model described in the previous sections

In this simulation-based validation step, all 3 models use the same vehicle parameters which allows to concentrate purely on the tire performance.

The first condition to compare is a ramp steer maneuver with a maximum lateral acceleration of 0.55g. A graph of the sideslip behavior shows that the base tire model with constant cornering stiffness is unable to capture the effect of the varying tire boundary condition with increasing load even though this variation is essentially quasi-static. The new tire model as proposed does capture the behavior exhibited by the reference model.

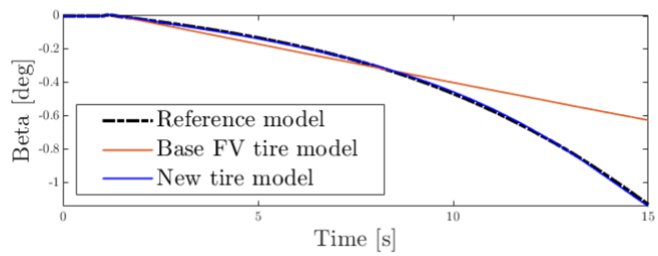

Figure 10. Comparison of the sideslip of the vehicle estimated with a constant cornering stiffness model and with the presented method in a ramp steer maneuver.

In a second validation case, the behavior at axle level is investigated. The maneuver is a step steer input with a lateral acceleration of 0.5g. Looking at the lateral force acting on the rear axle, it is shown in figure 11 that the proposed model captures bot quasi-static and transient behavior correctly.

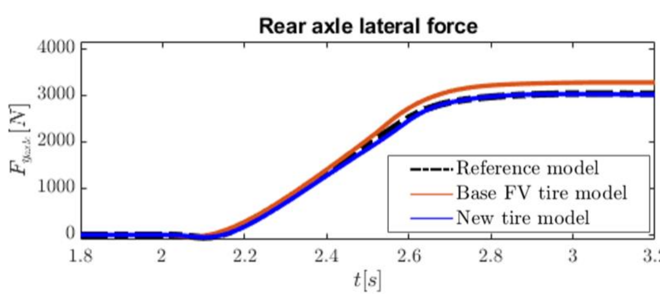

Figure 11. Comparison of the rear axle lateral force estimated with a constant cornering stiffness model and with the presented method in a step steer maneuver.

Next, the individual tire lateral forces are investigated. In figure 12 it is clear that the simplified model with constant cornering stiffness show equal forces for the inner and outer tires while the proposed tire model captures the different load behavior in a correct way. This corresponds to the fact that the force on inner and outer tire builds up differently due to the load transfer that happens while cornering. The inner tire demonstrates a flatter characteristic than the outer tire which correlates well with the reference model.

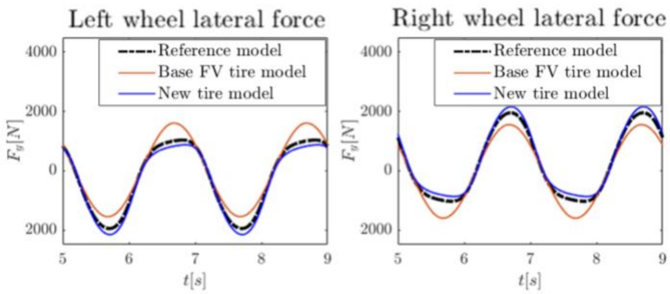

Figure 12. Comparison of the left and right wheels lateral force estimated with a constant cornering stiffness model and with the presented method in a sine steer maneuver

Finally, a validation is done for the pneumatic trail calculation in a sine steer maneuver. Again a close correlation is seen between the reference and proposed tire models in amplitude, shape and phase.

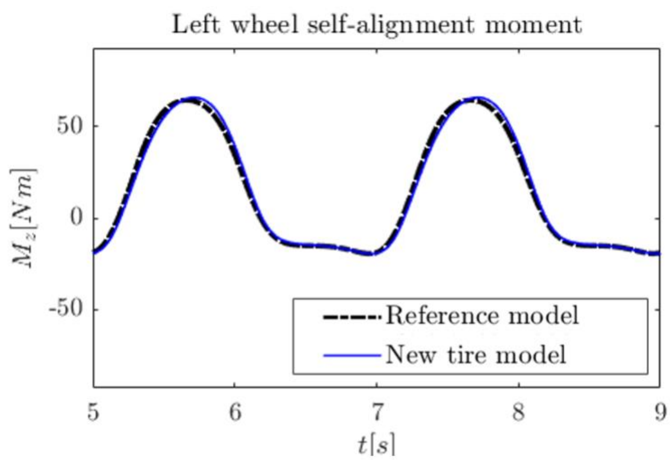

Figure 13. Self-alignment moment estimated with the presented method in a sine steer maneuver.

Validation through experimental data

The final validation of the tire model needs to be done on a real vehicle so a car has been equipped with sensors and a data acquisition system and a 15-DOF model has been created of that vehicle.

Figure 14. Instrumented vehicle for the 15DOF and tire model identification method proposed in the paper.

Again, 2 different tire models are applied. The first is a basic constant cornering stiffness tire model and the second is the new proposed model as described previously.

A first test constitutes a ramp steer maneuver which shows that the new tire model captures the real world behavior correctly whereas the constant cornering stiffness model deviates significantly.

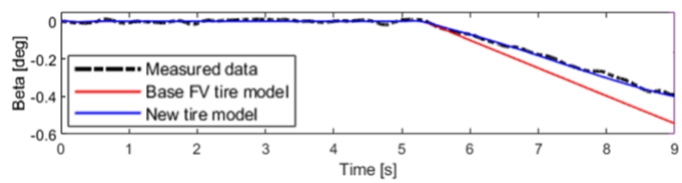

Figure 15. Comparison between the sideslip estimated with a constant cornering stiffness model and with the presented method in a ramp steer maneuver.

Second, a sine steer maneuver is performed. The new tire model correlates much better with the real world data and also captures the shape difference of the load graphs as apparent in the test data. The constant cornering stiffness model here shows a pure sine wave that is not representative of what happens due to the load transfers.

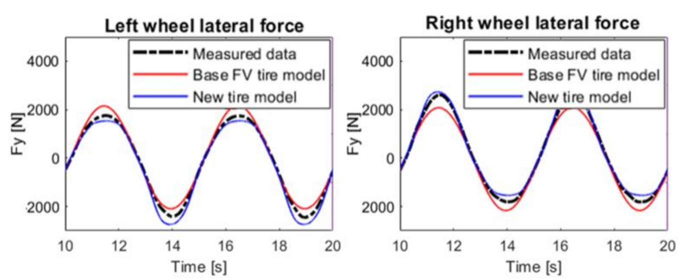

Figure 16. Comparison of the left and right wheels lateral force estimated with a constant cornering stiffness model and with the presented method in a sine steer maneuver

Finally, the self-alignment moment in the sine steer maneuver is compared between the new tire model and the test data. This comparison also shows the good correlation of the model with the measurements.

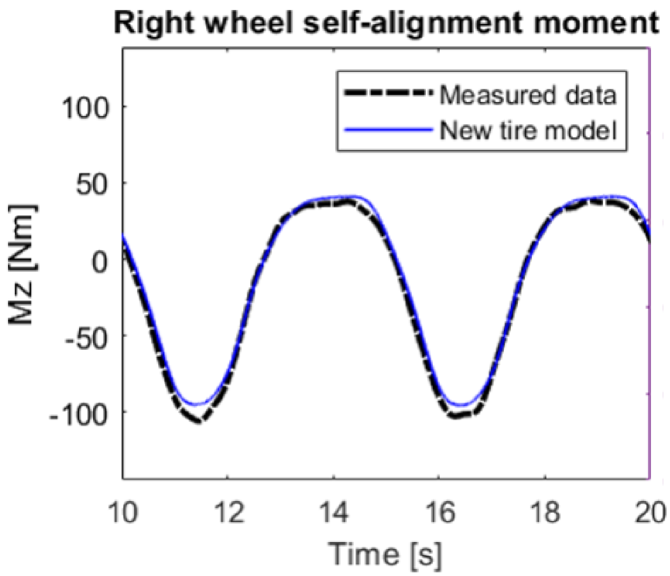

Figure 17. Self-alignment moment estimated with the presented method in a sine steer maneuver.

It is therefor concluded that the new tire model is sufficiently capable of capturing the real dynamics as seen on the test vehicle, including the left/right differences in terms of lateral loads and the self-alignment torque.

The important thing to consider here is that this tire model is not derived from expensive and extensive test bench measurements but derived from driving a vehicle through a set of dedicated maneuvers and a relatively simple sensor setup without dedicated and expensive wheel force transducers for example.

Damping and inertias

Up to this stage, all maneuvers have been estimated from static or quasi-static maneuvers, so damping and inertia have not played a significant roll. To complete the model these have to be added still.

To determine yaw, pitch and roll inertia and damping of the vehicle, a series of transient maneuvers is performed. A first approximation of the inertia matrix is obtained according to [12].

Subsequently the IMU measurements are used to decouple the pitch, yaw and roll. These are then used to update the inertia values and to extract the damping properties (figure 18)

A detailed description of this procedure is subject of a next publication on vehicle comfort assessment.

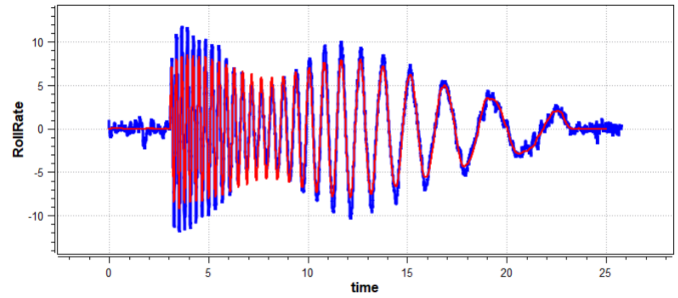

Figure 18. Estimated roll rate behavior of the vehicle (red line) and measured roll rate (blue line) in a sine sweep maneuver.

Vehicle model use cases

The 15-DOF model provides a much better model compared to a bicycle model in the context of vehicle dynamics performance evaluation but it also serves its purpose in advanced driving simulators and the development of ADAS features.

As shown in the validation case, the 15-DOF model allows accurate reproduction of track data for a given vehicle. This allows the evaluation of chassis parameters and their influence on the driving performance characteristics. It is for example possible to assess the effect of changing the anti-roll bar stiffness on the driving performance as illustrated in figure 19.

Another use-case would be in driving simulators where the 15-DOF model can capture the vehicle dynamics much better than simpler models. A bicycle model would for example not be able to capture the vehicle roll behavior, whereas in the here described model all 6 DOF of the body are present and as such captures this behavior correctly.

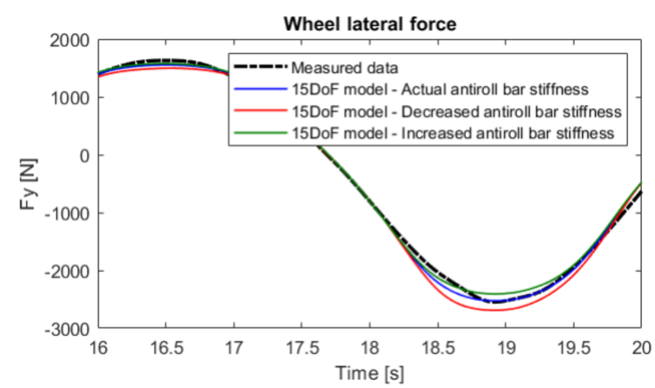

Figure 19. Comparison of the rear left wheel lateral force estimated with different values of the front antiroll bar stiffness and measured on track.

Experience has shown that using this new model in driving simulators also allows to analyze subjective feel of the vehicle and current work also includes the addition of other vehicle aspects, such as noise, to complete the driver experience and still enhance the subjective assessment.

ADAS algorithm development

It is recognized in the autonomous driving industry that simulation is an efficient method for testing and validating ADAS/AD functionalities. The traffic environment has a wide variety of parameters from road types, vehicles, pedestrians, cyclists, obstacles and weather. The number of scenarios grows exponentially with the number of parameters and can easily explode up to millions. Furthermore, all scenarios cannot be produced and reproduced easily in real life, and real road testing is valid for a specific mechanical, electrical and software configuration. If needed to adapt to a new configuration or software update, the test must be conducted again. Therefore, the major part of the ADAS functionalities will be validated through simulations.

Furthermore, MiL/HiL testing with respect to safety is particularly essential for several additional reasons:

- Physical vehicle testing for safety-critical scenarios is dangerous and expensive. An incomplete algorithm being deployed in a physical car can always cause collision.
- Development of data-driven based algorithms like machine learning and imitation learning requires datasets for training the algorithm. It is however very difficult and expensive to have datasets of scenarios relevant to safety, for example, accident or near-accident, collision avoidance and lane change at high speed.

The development of ADAS algorithms need to consider not only safety but also the passenger comfort. The vehicle model therefor needs to include roll and pitch dynamics [13]. The vehicle model as presented in this paper is particularly suited for such analyses (figure 20).

The tire model is equally important in a the ADAS algorithm development. A tire model that does not accurately represents the real tire performance will be detrimental for the performance of an ADAS algorithm [14].

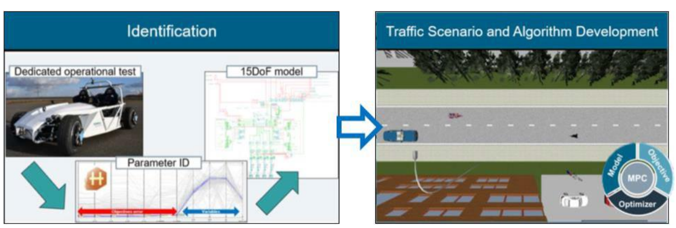

Figure 20. 15-DOF model identification approach developed for ADAS purposes. From FV acquisition a model can be obtained for the development of ADAS algorithms in a simulated environment.

Several steps need to be taken before an ADAS controller can be tested on the vehicle in a real-life environment. A first step after the offline evaluation and simulation is to evaluate the performance of the controller on a rapid control prototyping (RCP) and linked to a real-time simulation platform onto which the vehicle model and the environment simulation are deployed. The vehicle model as developed in this paper is particularly suited while the environment and sensors are simulated in Simcenter Prescan. These two platforms are linked to simulate the performance of the controller on a vehicle (figure 21).

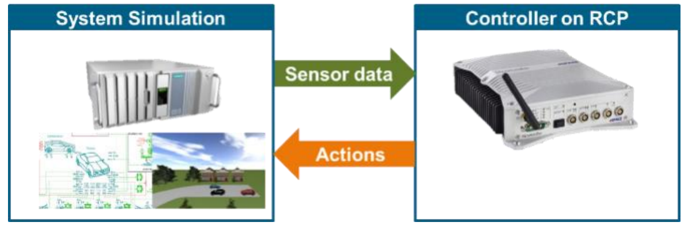

Figure 21: ADAS controller development platforms

A next step is to deploy the developed controller onto an embedded system and evaluate if the controller is still performing as designed. The rapid control prototyping system is replaced with the controller on an embedded system. The rest of the setup is kept as was for the previous step: the vehicle, environment and sensors can be simulated on a real-time platform. This approach enables a robust evaluation of the controller by a simulation-based vehicle and environment. Measurement noise and system uncertainty can be part of defined test conditions.

Conclusion

In this paper, a 15-DOF model for vehicle dynamics studies is presented that represents the vehicle in real driving conditions. The validations show that the model correlates well with the actual vehicle and tire forces and self-alignment torques are well represented.

The presented model is derived from mainly full-vehicle testing in combination with limited laboratory testing. To achieve this, the different elements of the dynamic behavior are effectively decoupled through a set of dedicated maneuvers that allow a stepwise identification of the model parameters. The required instrumentation is also limited to an IMU and some inertial sensors. No expensive measurement setup is required.

Detailed discussion is given on the topics of roll axis and roll centers as well as the tire model. The methodology described allows the determination of the roll axis at every point in time during the driving maneuver allowing a better understanding of its influence on driving dynamics. The tire as well is derived from pure vehicle measurements and therefor captures track conditions at the time of driving accurately.

Finally, some use cases for the model are described in context of driving dynamics analysis, driving simulators and the development of ADAS algorithms.

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Definitions/Abbreviations

ADAS	advanced driver-assistance system
AOEM	automotive original equipment manufacturer
CoG	center of gravity
DOF	degree(s) of freedom
IMU	inertial measurement unit
K&C	kinematics and compliance
RCP	rapid control prototyping
VIMF	vehicle inertial measurement facility