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Validation of Longer and Heavier Vehicle Combination Simulation Models

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ABSTRACT

This paper discusses the development and subsequent validation process of generic multi-body models for commercial vehicle combinations. The model is intended for performance assessment and improving of current and future combinations for the European road network. A second goal is to employ the model for the development of driver support systems and active steering strategies for both low speed manoeuvrability and high speed stability. The model is developed in SimMechanics, which is part of the MATLAB/Simulink software. Due to its modularity, one can quickly modify the model to the desired configuration and dimensions; therefore various multi articulation vehicle models can be created. The paper further illustrates the simplified and generic modelling methods used to build particular components such as chassis, tyres or suspension in the multibody domain. A stepwise approach of the model validation is presented, followed by a sensitivity analysis illustrating the effect of the selected vehicle parameters on particular vehicle states. To conclude, the approach is applied to the validation of the simulation model using measurement data obtained from experimental testing of two different vehicle combinations.

INTRODUCTION

Increasing freight transport in the EU with the available infrastructure represents a challenging issue that cannot be underestimated. According to [2], it is predicted that freight transport in Europe will grow between 2000 and 2020 by 50%. Actual numbers are in accordance with this forecast, which underlines the link between transport and economic growth [1]. More in particular, the GDP of a country can be considered to be directly proportional to the amount of transported goods. Considering the fact that in Europe almost 75% of the inland transport is being realized through the motorways and highways, one can see a potential problem, which can arise if the current freight transport system will not be modified. The biggest restriction is coming nowadays from the EU legislation 96/53EC that prescribes the maximal length and weight of the vehicle to be 18.5 meters and 44 tones respectively. The directive, however, does not allow cross border operation of performance oriented vehicle combinations in EU, which can be longer and heavier, but if appropriately designed can also be safer, more efficient, ecological and road friendlier. This fact has been already shown by practice and by several benchmarking studies from Australia, Canada and South Africa, where these vehicles are allowed to operate [3]. In the Netherlands longer and heavier vehicles (LHV) are well accepted and can be legally used (see Figure 1.), however, as written previously, an
international cross-border operation is not allowed, which limits their potential.

Based on this point of view, a research project has started involving co-operation of academia and major commercial vehicle industry players. The project is aiming at the identification of design of future commercial vehicle concepts for the years 2020+. They will reduce CO₂ emissions, meet the future needs of logistics companies, be compatible with existing infrastructure and facilitate inter-modal cross border transport within Europe. The concepts will be based on existing modules used in road-rail-water-air. Furthermore, active wheel steering and driver support systems for maneuvering are anticipated. The development of both systems requires a fundamental understanding of the vehicle behavior, for which an appropriate vehicle model is necessary.

In general there are two ways of building a vehicle model. One can either assemble equations of motion (EOM) of the vehicle combination by hand and create a mathematical model [9] or employ software that will assemble EOM using symbolical mathematics. The second way to build the model is by a multi-body formalism that uses bodies connected by means of joints allowing/restricting particular degrees of freedom. In a second approach, the equations of motion and the constraint equations are generated automatically by the software package. In this paper we restrict ourselves to multi-body modeling only.

The multi-body modeling approach is already employed successfully in virtual prototype development for more than 20 years. One can choose nowadays between several generally applicable packages such as ADAMS, LMS Virtual.LAB, Dymola, or decide for dedicated vehicle simulation software like CarSim, TruckSim or veDYNA. These are rather easy to operate due to built-in GUI and user-friendly visualization tools [7]; however the flexibility and modularity for applied research of control systems, may sometimes prove to be a limitation. Furthermore as pointed out in [5], the papers describing the models created in some of the programmes listed above and subsequently validated against the experimental results are very rare. In this paper a generic and modular vehicle model library for multi-articulated vehicles is described, which is used to build two different vehicle combinations. The models are subsequently validated against the experimental results. The model can be directly utilized for performance assessment of various vehicle combinations as well as for the development of the controllers enhancing vehicle handling stability and low speed manoeuvrability.

The paper is structured as follows. First, the simulation environment used for model development is described. Section 2 outlines the detailed topology of the vehicle model and sub-modules, followed by a description of the instrumentation of two vehicles and test program. Section 4 proposes a stepwise methodology process for model validation, together with an optimization and a sensitivity analysis illustrating the effect of several parameters on vehicle behavior. To conclude, the approach is illustrated by validating the models against experimental data, which is being discussed further and summarized at the end of the paper.

**SIMULATION ENVIRONMENT**

The simulation environment should be able to interact with other software, necessary for future research of controllers. Besides that, the model should be sufficiently flexible and generic, thus modification of model parameters should be possible with minimal effort. Because of these reasons, and positive experience from previous projects [5], SimMechanics, which is the multi-body toolbox of MATLAB/Simulink, has been selected.

Although SimMechanics is not especially designed for simulating vehicle dynamics, one can customize it for vehicle dynamics modelling, due to its good interfacing with other MATLAB Toolboxes. An essential Toolbox for vehicle dynamics is undoubtedly the tyre model. It is represented by TNO-Delft Tyre model, which is also used by many of the listed multi-body packages. A suitable option for visualization and animation of simulated results in MATLAB is the Virtual Reality Toolbox as depicted in Figure 2., and demonstrated by [6].

**MODEL TOPOLOGY**

Models of the two vehicle combinations, LHV-B and LHV-D (see Figure 1.), have been built up by means of the Commercial Vehicle Library (CVL), which is a generic library consisting of truck, trailers and components, developed in SimMechanics by the Eindhoven University of Technology.
This library is divided in five sections:

- Vehicles (Towing vehicles)
- Trailers (Towed vehicles)
- Assemblies (Cabin, Loading Units, Axles...)
- Components (Tyre, Brake system)
- Utilities (filters, sensors...)

The structure of these models is modular in order to give the user freedom to create and develop many different combinations. At the same time, the user has the flexibility to easily customize all the components of each sub-model without operating on the entire multi-body model. The purpose of the library is to represent an average vehicle and avoid all the details that can be different for every particular vehicle as for example nonlinearities in chassis suspension, roll steer or cabin-chassis suspension layout. These can be further customized by user if necessarily needed.

The TNO Delft Tyre Model describes the vertical, lateral and longitudinal forces of the tyres, employing the Magic Formula of Pacejka [8]. The CVL includes three types of tyre files, one for steered axle, one for the driven axle and another one for the trailer axle; they report the main characteristic of the tyre (masses, inertia) and their properties, such as cornering stiffness, vertical and lateral stiffness, longitudinal slip scaling factor and many other coefficients [8, 11]. In the validation process, also the tyre files have been adapted for finding a realistic cornering stiffness.

We will demonstrate the topology of the library on building the B-Double model composed from a 2-axle Tractor, a B-dolly carriage unit and a semitrailer module; both of them equipped with 3 axles (see the Figure 3).

The tractor module is composed by a chassis divided into two parts, front and rear, connected to each other with a revolute joint and a torsional spring to reproduce the torsional stiffness of the vehicle. Cabin and engine are rigidly connected to the chassis and the steer and drive axle are linked to the chassis by means of the suspension system. The suspension model simplifies the vertical and rolling movement of the axle with respect to the chassis, placing a custom joint in the roll center of the axle. Basically, the axle module has only two degrees of freedom with respect to the chassis module: the z-axis translation and the x-axis rotation (see Figure 3). Two linear springs and two linear dampers have been implemented, one on the left and one on the right side of the axle. A torsional spring has been located in the roll center, representing the antiroll bar. On both ends of the steering axle, a body with a negligible mass is connected through a revolute joint with rotational freedom around z axis, which equals the steering angle; the angle at these joints is determined by the steering system using a prescribed steering ratio; for validation purposes, this angle has been directly measured on the wheel hubs. These bodies are then connected to the tyre module, described above, using a revolute joint. On both ends of the driven axle, these latter joints receive also the moment generated by the driveline model, which uses a continuous variable gear ratio for determining the torque at the driven wheels.

The B-Dolly unit is composed in a similar way: the chassis is divided in a front and rear part, on which the loads are applied rigidly, a torsional spring in between, suspension systems on all of the three axles. Six wheels, the kingpin and the 5th wheel are rigidly connected to the chassis, for the coupling with other towing units. The allowable degrees of freedom of this coupling are the yaw rotation and, only when the towing and the towed vehicles are aligned, also the pitch rotation. The center of gravity of the trailers depends on the towing vehicle is considered the global coordinate system with respect to which the own local systems of all the units are positioned. All the coordinate systems use a right handed system; the positive X-axis points forward, the positive Y is toward left and Z-positive is directed upward (see the Figure 3).
masses of the components and on the loading condition. Finally, the axles of the last semitrailer have been modeled as an independent suspension, in order to correctly represent the suspension of double deck trailer of the test. The roll center is placed automatically at ground level by modeling this suspension. Two transitional joints have been placed on the right and left side of the chassis, each one connected with a “suspension” body. For each side, a damper and a spring have been linked in between. Figure 3 shows a schematic overview of the degrees of freedom of the full B-Double model.

Thanks to the flexibility, offered by the CVL, also the LHV-D Combination was created with minimal modeling effort.

**EXPERIMENTS**

Full scale tests were executed to validate the simulation models. The measurements were done with two different vehicle combinations with two articulation points and length and weight exceeding the current European Union regulations (see Figure 4).

The LHV-D is composed of a rigid truck with a 20 foot container and a conventional 40 foot container semitrailer connected to the truck by means of a two axle dolly. The second configuration is a LHV-B involving a tractor linked to a B-module carrying a 20 foot container and further connected to the double-decker semitrailer with a special independent wheel suspension implying also lower roll centre than conventional. Both configurations are 25.25 meters long and are equipped with air suspension. The configuration of the vehicle combinations are selected differently on purpose, in order to validate different vehicle configurations.

Each vehicle combination is tested in two different loading conditions to cover different axle load and moments of inertia. The layout of two load distributions for LHV-D is shown in Figure 5.

![Test Vehicles a) LHV-D b) LHV-B](image)

*Figure 4. Test Vehicles a) LHV-D b) LHV-B*

The overall vehicle weight for both combinations is 60 tonnes, which is also the maximum allowed for LHVs in The Netherlands. Both vehicle combinations are instrumented in a similar way. The instrumentation involves more than one hundred measurement channels with emphasis on collecting sufficient data that enables the validation of dynamic model behaviour in longitudinal, lateral, and vertical direction. In this paper we however restrict ourselves on presenting the signals related to handling and low speed manoeuvring only. A schematic picture of the externally placed sensors on LHV-B is given in Figure 6. As can be seen, position, velocity and acceleration were measured for each vehicle body in translational and rotational direction. Additional potentiometer sensors measuring the position between chassis and axles as well as vehicle bodies with respect to each other are incorporated. The sensors are necessary due to limited accuracy of the translational and angular position measurements when evaluated from accelerometers. Such a signal might be inaccurate, because of integration error and drift [12]. Potentiometers are also used for measurement of the steering angle of towing vehicle axle for left and right wheel, as the left and the right steer angle differs when reaching bigger values. Furthermore selected signals from the CAN-bus of the towed vehicle are recorded such as the angular velocity of wheels, the engine torque, and the brake pressure. The sampling frequency is in most of cases 100 Hz only the signals coming from CAN-bus were limited to 10 Hz.
The tests were executed on a dedicated proving ground at Lelystad in Netherlands [14], which has sufficient space for all manoeuvres under consideration. The vehicles are tested in both high and low speed scenarios, but also on different frictional surfaces. Before the testing the vehicle is weighted on both loading conditions and in an empty state. Test scenarios were performed in accordance with ISO 14791:2000 E and the Australian PBS scheme [13]. In particular the test plan involved following manoeuvres:

**High Speed:**
- Sine steer input at 0.4 Hz
- Pulse steer (half sine)
- Single lane change
- Steady state cornering
- Braking on different surfaces with different intensity and curvature
- J-turn
- Maximum longitudinal acceleration

**Low speed:**
- 90 and 180 degree circle on radius of 12.5 meters
- Passing over a vertical obstacle
- Rearward parking maneuver

Each test session also involves several calibration tests when the vehicle is standing still in the straight line so the zero values for all sensors can be obtained.

### VALIDATION PROCESS

Validating the vehicle model is necessary and an important part for the model development. This step provides the confidence that the vehicle model mimics the behaviour of the real vehicle, and thus can be used for credible research. Here we propose a validation approach, which fits in the context of the model applications mentioned earlier, i.e. low speed manoeuvrability and high speed stability. The process of validation can be understood as a stepwise procedure depicted on the Figure 7.

The validation process of an articulated vehicle model should always start with the static validation. Practically it means a check of the static vertical tyre forces as they govern tyre horizontal forces caused by tyre slip. The load distribution between the axles is not only dependent on the position of gravity centre of particular vehicles but also, due to the air spring utilization, on the levelling of individual axles. The levelling is normally mastered by a fully automatic controller, however the level of precision is sometimes limited, which could result in parasitic pushing/relieving forces acting on the axles. The vertical tyre forces can be tuned in two ways, which are linked to each other. The simple movement of centre of gravity in the required direction is the first one and should always correspond to the vehicle known mass properties. The second approach compensates for suspension levelling and can be done through setting appropriate preload to vertical springs for each axle group. As the vehicle model is statically over determined, the second approach should be used only for fine tuning to ensure that vehicle body will be parallel to the ground plane. Due to large influence of the static tyre forces on the overall vehicle behaviour, correct results are crucial for consecutive validation.

Next we move to the quasi static conditions, where the vehicle handling properties will be judged. An appropriate scenario to validate the model is steady state cornering, with a constant longitudinal velocity and a constant steering angle as inputs. The key signals to be validated are:

- yaw rate
- lateral velocity
articulation angles
roll angle

To get a better understanding and quantification of the influence of particular model components such as tyres, suspension and vehicle body stiffness on above selected vehicle states a sensitivity analysis was executed. The model of the vehicle combination was set to a constant velocity and steering angle, to perform steady state cornering. Subsequently an identification of the most responsive parameters, which can be employed for optimization, was done. The following parameters have been modified in range ± 15% as being very relevant according to previous research:

• Cornering stiffness of the tyre (scaling coefficient)
• Tyre road friction coefficient
• Axle roll stiffness
• Body torsional stiffness
• Suspension spring stiffness

Based on the sensitivity analysis, listed in table of Appendix I., it can be observed that tyre cornering stiffness is the most influencing parameter for vehicle steady state handling. Its influence on yaw rate, lateral velocity and articulation angles reaches values of 20% with respect to original value, whereas in case of other parameters we obtain significantly lower values. The other tyre properties such as self aligning torque or longitudinal slip stiffness do not play a major role. For the roll angle of the 2nd trailer the suspension stiffness is the most sensitive parameter. Under the assumption of steady state cornering conditions with sufficient radius and velocity (we have chosen of R=50m and velocity above 35 km/h) that avoids the tyre scrubbing, the cornering stiffness of the tyres can be identified based on the measurement data. For this purpose we use a multivariable optimization technique. The identification only by guessing would be rather difficult because the cornering stiffnesses of all four axle groups are influencing the overall vehicle combination behaviour. As described earlier, TNO-Delft Tyre model has been used that employs the Tyre Magic Formula calculating the tyre cornering stiffness by means of a number of shaping parameters, for more details see [8]. Considering the operational range of the commercial vehicle during steady state cornering, when the side slip angles are limited, one can assume the tyres to be linear. The rollover of the vehicle would precede nonlinear tyre behaviour occurring at bigger tyre slip angles. For optimization, thus a linear single track vehicle model with linear tyres, as depicted in Figure 8., can be used. The individual tyres can be then substituted by axle group models that sum the cornering stiffnesses of the tyres, and which will be subjected to an optimization procedure.

Figure 8. Linear single track model with 2 articulations

The equations of motion defined in [9] are simplified for steady state cornering corresponding to Figure 8. The equations read:

\[
u \dot{\beta_0} (m_1 + m_2 + m_3) = -\frac{1}{u} \left( \sum_{i=1}^{4} C_i \right) v_1 + (h_1 \psi + h_2 \phi + (c_1 h_1 \psi + c_2 h_2 \phi + c_3 h_3 \psi + c_4 h_4 \phi + \alpha_i h_i) + c_i \delta_i \right)
\]

\[-h_2 u \dot{\beta} (m_2 + m_3) = \frac{1}{u} \left( [a_1 c_1 - h_1 c_2 - h_1 c_4 - h_1 c_3] v_1 + (c_3 + c_4) h_2 \psi + c_1 h_1 \psi \right) + \frac{1}{u} \left( [a_1 c_1 + b_1 c_2 + c_1 h_1 (h_1 + h_2) + c_2 h_1 (h_1 + h_2 + h_3)] \dot{\beta} + c_i \delta_i \right) \]

\[-u \dot{\beta} (m_3 a_2 + m_3 l_3) = \frac{1}{u} \left( [l_3 c_1 l_3 c_2 + l_3 c_4 l_3 c_3] v_1 + (l_3 c_1 + l_3 c_4 + l_3 c_3) \phi + l_3 c_2 \psi \right) + \frac{1}{u} \left( [c_3 l_2 h_1 + l_2 + c_4 l_2 (h_1 + l_2 + l_3)] \dot{\beta} \right) \]

\[-u \dot{\beta} (m_3 a_3) = \frac{1}{u} \left( [l_3 c_4 v_1 + l_3 c_4 \phi + l_3 c_4 \psi + c_3 l_2 h_1 + l_2 + l_3] \dot{\beta} \right) \]

Meaning of used variables can be found in the list of abbreviations at the end of this paper. Next, the equations (1, 2, 3, 4) are rearranged to the form:

\[v_{1c} = f(\dot{\beta}_M, \phi_M, \psi_M) \]

\[\dot{\beta}_c = f(\phi_M, \psi_M, v_{1M}) \]
\[ \phi_c = f(\beta_M, v_{1M}, \psi_M)[\rho] \]  
\[ \psi_c = f(\beta_M, \phi_M, v_{1M})[\rho] \]  
\[ \rho = \begin{bmatrix} C_1 \\ C_2 \\ C_3 \\ C_4 \end{bmatrix} \]  

(9)

Where \( \rho \) (9) is a vector of cornering stiffnesses to be identified and optimized. The remaining elements on the right hand side of (5, 6, 7, 8) represent experimental measurement data of the vehicle states which can be further written as:

\[ \bar{X}_M = \begin{bmatrix} v_{1M} \\ \beta_M \\ \phi_M \\ \psi_M \end{bmatrix} \]  

(10)

Gathering the left hand side of (5, 6, 7, 8) in a vector we obtain:

\[ \bar{X}_C = \begin{bmatrix} v_{1C} \\ \beta_C \\ \phi_C \\ \psi_C \end{bmatrix} \]  

(11)

Vector \( \bar{X}_C \) is basically the state vector which has been calculated from the experimental data and the vector of cornering stiffnesses. Knowledge of (10) and (11) subsequently enables to calculate the error between calculated and measured vehicle states:

\[ \varepsilon(t, \rho) = \sum_{i=1}^{M} (\bar{X}_C - \bar{X}_M)^2 \]  

(12)

The error (12) expresses the difference between measured state \( \bar{X}_M \) and the state that has been calculated \( \bar{X}_C \) based on measured states multiplied by cornering stiffnesses vector. A unique solution will only exist if the left and right side of (5, 6, 7, 8) are equal. Since the data we use for optimisation are real and contain noise the error will always occur, however can be minimized by the optimisation of the cost function:

\[ \rho^* = \arg\min_{\rho} \bar{V}(\rho) \]  

with \( \bar{V}(\rho) = \bar{E} \varepsilon^2(t, \rho) \)  

(13)

For the identification of the vector \( \rho^* \) (13) the minimization algorithm function ‘fmincon’ [10] of MATLAB has been used, which is suitable for multivariable constrained nonlinear optimization. The only constraints that have been applied are lower and upper bounds in the expected order of estimated magnitude of cornering stiffness. Adding more constraints may result in stiffness of the optimisation problem that may harm the robustness of the estimation. The results of the optimisation can be seen on Figure 9, for steady state cornering at constant velocity 45 km/h and radius of 50 meters. The red solid line represents the experimentally obtained data, dashed blue line depicts the output of the model with initial estimate (guess) of cornering stiffness and black dotted one shows the output of the model with cornering stiffnesses obtained from the optimization.
roll behaviour of the vehicle, it is recommended to adapt the model in accordance with the sensitivity table (Appendix I.) i.e. adjust the suspension roll stiffness per each individual vehicle, as the stiffness may significantly vary based on the type of vehicle. To conclude the quasi-static part it is feasible to check the validity of the model during similar operational condition, for example with a different velocity and radius.

The last level of the validation deals with the dynamic conditions when transient behaviour can be observed in manoeuvres like sine steer input or a lane change. Since majority of the vehicle parameters have been already validated during previous two steps remaining parameters to modify are:

- Damping ratio of the suspension
- Moment of inertia of vehicles

As the characteristic of the dampers are rather easy to obtain from the manufacturer, and does not vary significantly between different vehicles, one can consider the information obtained from available data sheets as reliable. Contrary, the moment of inertia differs considerably for every vehicle and depends on the bodywork and the loading layout of the cargo. A correct value involving cargo can be obtained either through calculation using parallel axis theorem or again by optimisation proposed earlier for estimation of tyre cornering stiffness.

VALIDATION RESULTS

In this section the results of all previous sections are applied and brought together. Thus a model has been validated against experimentally obtained data by means of proposed approach in three steps:

- Static validation (LHV-B)
- High speed steady state cornering (LHV-B) Low speed 90 degrees corner (LVH-B)
- High speed lane change (LHV-D)

So the results presented correspond in the first two steps to the combination B, and in last step to the combination D in order to demonstrate the validity of the generic model library if a different vehicle combination is used. As input for simulation only the steering angle of the towing vehicle and its driven wheel velocities are used. The solid black line is always used for measured data and the red dashed one for simulation output. Since there is no universal or generally applicable methodology for the judgment of the model accuracy, the validation of the model will be done by a visual inspection of the vehicle states, similarly like in case of previous work [5] or published papers from the same field [4].

Static Validation

The validation is presented for the both loading conditions when different loading pattern has been applied.
cornering stiffness, the approach described in the previous section has been employed.

Concerning the results of the high speed cornering, a very good match between all important vehicle states can be seen. A small discrepancy can be seen in case of the roll angle of tractor, which is however most likely caused by the measurement error of the sensor. In contrary the roll angle of the Trailer 2 is more consistent because this angle has been obtained by laser sensor measuring the absolute vertical distance of the vehicle body to the ground.

![Figure 11. LHV-B, 90 degree corner R12.5 @ 10km/h](image)

Concerning the results of the high speed cornering, a very good match between all important vehicle states can be seen. A small discrepancy can be seen in case of the roll angle of tractor, which is however most likely caused by the measurement error of the sensor. In contrary the roll angle of the Trailer 2 is more consistent because this angle has been obtained by laser sensor measuring the absolute vertical distance of the vehicle body to the ground.

![Figure 12. LHV-B, Steady State cornering R50m @ 45km/h](image)

The simulation output for low speed maneuvering is also in agreement with experimental data. Only a mismatch can be observed at the peak of the articulation angle between the tractor and the Trailer 1. Similar behavior has already been
observed in high speed cornering and here is magnified. It can be primarily explained by the simplified suspension model, which does not include elastic compliance of some suspension parts contributing to passive steering of the Trailer 1. Furthermore, some influence can be also accounted to the nonlinear tire behavior. The side slip angles occurring during sharp low speed cornering are certainly beyond the linear range for which the optimization was performed.

**Transient Validation**

For validation of transient (dynamic) vehicle behavior we use a lane change maneuver at 50 km/h. To discuss the results we will present the graphs for each vehicle separately. Compared to the previous measurements also the axle-chassis distance measurements were analyzed to investigate the roll behavior. The truck behavior can be seen in Figure 13, followed by the dolly in Figure 14, and the semitrailer in Figure 15.

On Figure 13, one can observe a small difference in the axle chassis distance especially in the negative region. It is caused by linearization of the dampers and the springs in the suspension model and most likely by the simplified model of the vehicle frame which allows only one degree of freedom in torsion. Omitted movement between the cabin and the chassis, which was not modeled, contributes to roll inaccuracy as well. Both effects however have minor impact on the overall combination behavior as the difference is in order of millimeters. Remaining vehicle states are in good agreement.

Besides the lagging in roll angle, no special behavior or deviation in Figure 14, are observed. The accuracy of the measured roll angle is in general very poor since the sensor calculated the angle by integrating roll rate and compensated for integration drift by other sensor signals (accelerometers and GPS), which may cause a relative big error at very small
angles. Therefore for the second measurement session with LHV-B, the roll has been obtained by the laser sensors mounted on the vehicle body and measuring vertical distance to the ground.

Figure 15. depict all effects described in the two previous paragraphs. Furthermore it shows an illustrative case of the sensor malfunction can be seen in the dolly-trailer articulation angle. The readings of the sensor between 4 and 7 seconds are apparently wrong, since they are not consistent with other measurements and the vehicle is unlikely to experience such a behavior.

CONCLUSIONS

In the paper a simplified and generic multi-body model for commercial vehicle combinations is presented and subsequently used for creating two different vehicle combinations. The models are validated by the data obtained from experimental testing. The testing was done for both combinations at different velocities and loading conditions to ensure that the captured data covers a sufficient range. Furthermore, a systematic approach is proposed to validate the vehicle combination model. The approach is composed from three steps: static, quasi-static and transient. As appears from the sensitivity analysis, the quasi-static validation part plays the most important role because the tyre cornering stiffness, which influences mostly the vehicle handling behavior, needs to be identified properly in this step. To identify the tyre cornering stiffness, we propose a multivariable constrained optimization of the cost function that sums the errors between the measured and the calculated vehicle states, employing a linear single track model. The results offer a fair estimate that is a good start for fine tuning of the model in accordance with presented sensitivity analysis. Finally the validation approach is applied to the models. Even though the multibody models are rather simplified, the graphs show a good match between the simulation output and the experimental data. As it is common in this field, the validation of the models is based solely on the visual inspection of the vehicle states. For these reasons, it is hard to objectively quantify the accuracy of the models, as there is no clear border between accurate and inaccurate models. To conclude, it might be of interest for future research to investigate a method that can be generally applicable for the assessment of any model accuracy.

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DEFINITIONS/ABBREVIATIONS

HTAS - High Tech Automotive System
EMS - European Modular System
TNO - Dutch Research Organization
GPS - Global Positioning System
TU/e - Eindhoven University of Technology
CAN-bus - Control Area Network
CVL - Commercial Vehicle Library
GUI - Graphical User Interface
CO₂ - Carbon dioxide
LHV - Longer Heavier Vehicles
LHV-B - B-Double Combination of LHV
LHV-D - D-Combination of LHV
EU - Europe
GDP - Gross Domestic Product
U - Longitudinal Velocity
v₁ - Lateral Velocity
Fy_j - Lateral Force of the tyres on the j-axle
j - Number of axle (i.e. j=3 reference axle of first trailer, the 2nd of the axle group)
C₃ - Sum of the Cornering stiffnesses of the tyres on the reference j-axle (i.e. C₃ = total cornering stiffness of the 6 tyres on the 3 axles of first semitrailer).
i - Number of the vehicle (i.e. i=3 is second semitrailer)
xᵢ yᵢ - Local coordinate system of i-vehicle
mᵢ - Mass of the i-vehicle
αᵢ - Tyre slip angle of reference tyre of i-vehicle (α₁ = front tyre slip angle of (tractor)
δ₁ - Steer angle of the front axle of the tractor.
lᵢ - Wheelbase of i-vehicle (i.e. l₂ is the distance from the kinping of first trailer to the second axle of the first trailer)
lᵢο - Distance from origin of local coordinate system of i-vehicle to the 5th wheel of i-vehicle.
e₁ - Distance from rear reference axle of i-vehicle to the 5th of i-vehicle
h₁ - Distance from the COG of i-vehicle to the 5th of i-vehicle
a₁ - Distance from the origin of local coordinate system of i-vehicle to the COG of i-vehicle.
b₁ - Distance from the COG of i-vehicle to the rear reference axle of i-vehicle.
Φ - Articulation Angle between Tractor and first Trailer
Ψ - Articulation Angle between firs Trailer and second Trailer
γ - Body Slip Angle of Tractor
β - Yaw Rate of Tractor
θ - Yaw Rate of first Trailer
ξ - Yaw Rate of second Trailer
β - Yaw Angle of Tractor
θ - Yaw Angle of first Trailer
ξ - Yaw Angle of second Trailer
Φ - Roll Rate of i-vehicle
φᵢ - Roll Angle of i-vehicle
Vᵢ - Total Lateral Acceleration of i-vehicle
dᵢL - 2nd of 3 Axle - Chassis Distance of i-vehicle - Left Side
dᵢR - 2nd of 3 Axle - Chassis Distance of i-vehicle - Right Side
LKY - Scaling Factor of Cornering Stiffness (see the Pacejka's Formula, in [2]).
Ktorsional - Body Torsional Stiffness
Kroll - Axle Roll Stiffness
Kspring - Suspension spring stiffness
μ - Tyre Frictional Coefficient
R - Corner Radius
## APPENDIX

### APPENDIX I. - SENSITIVITY TABLE

Sensitivity table simulating the steady-state cornering counter-clockwise direction at 50 km/h and R=50 m.

The Original Data reports the average values of selected vehicle states in the stationary condition and are used as reference for showing the effect of tuning a specific parameter on the behavior of the model.

<table>
<thead>
<tr>
<th>Original Data</th>
<th>Roll Angle [rad]</th>
<th>Articulation Angle [rad]</th>
<th>Yaw Rate [rad/s]</th>
<th>Lateral Velocity [m/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tractor</td>
<td>0.07</td>
<td>0.02</td>
<td>-0.08</td>
<td>0.04</td>
</tr>
<tr>
<td>Trailer1</td>
<td>0.04</td>
<td>0.04</td>
<td>-0.12</td>
<td>0.04</td>
</tr>
<tr>
<td>Trailer2</td>
<td>0.04</td>
<td>0.04</td>
<td>-0.12</td>
<td>0.04</td>
</tr>
</tbody>
</table>

### Tractor Parameters

- **K_torsional**: 15%
- **μ**: 15%
- **μ_{Front\#}\#**: 15%
- **μ_{Rear\#}\#**: 15%
- **μ_{Front\#}\#: 15%
- **μ_{Rear\#}\#: 15%
- **K_{Ly\#}\#: 15%
- **K_{Kroll\#}\#: 15%
- **K_{Kspring\#}\#: 15%

### Trailer1 Parameters

- **K_torsional**: 15%
- **μ**: 15%
- **μ_{Front\#}\#**: 15%
- **μ_{Rear\#}\#**: 15%
- **K_{Ly\#}\#: 15%
- **K_{Kroll\#}\#: 15%
- **K_{Kspring\#}\#: 15%

### Trailer2 Parameters

- **K_torsional**: 15%
- **μ**: 15%
- **μ_{Front\#}\#**: 15%
- **μ_{Rear\#}\#**: 15%
- **K_{Ly\#}\#: 15%
- **K_{Kroll\#}\#: 15%
- **K_{Kspring\#}\#: 15%