

## A HEAVY VEHICLE DYNAMICS MODEL FOR DRIVING SIMULATORS

### STEFANO SEDRAN

Obtained a master degree in 2016 in Automotive Engineering at Politecnico di Torino.



### FREDRIK BRUZELIUS

Researcher at Swedish national road and transport research institute. Adjunct Professor in Vehicle Dynamics at Chalmers.



### SOGOL KHARRAZI

Researcher at Swedish national road and transport research institute.



### BENGT JACOBSON

Professor in Vehicle Dynamics at Chalmers University of Technology – Department of Applied mechanics



### NICOLA AMATI

Associate Professor at Politecnico di Torino – Dept. Of Mechanical and Aerospace Engineering.



### Abstract

The aim of the present paper was to develop and validate a heavy vehicle dynamics model for driving simulators. The model was thought to be open, with a high level of readability and flexibility for future understanding, modification and use. A stepwise increment of the model complexity was utilized to understand the most contributing factors and to obtain a model capable of running in real-time. The model has been validated objectively using a previously validated model and through subjective evaluation in a driving simulator.

**Keywords:** Truck dynamics modeling, model complexity analysis, driving simulators

## 1. Introduction

Driving simulators provide important opportunities to study the interaction between driver and vehicle in a wide variety of situations and scenarios. The drivers' inputs (i.e. accelerator and brake pedals, steering wheel, gear clutch, cruise control selector, etc.) and the surrounding environment (i.e. road profile, road-tire friction coefficient, the wind, etc.) are fed to a vehicle dynamics model that simulates the behavior of a real vehicle. The sensation of motion is fed back to the driver through the graphical-, sound- and motion feedback- systems (Fischer, 2011). This illustrates the importance of an adequate and accurate model.

This paper presents the development of an open truck dynamics model. It is primarily intended to be used in the driving simulator Sim IV (Jansson, 2014) at the Swedish National Road and Transport Research Institute (VTI). Open models may govern for a deeper understanding, ease of maintenance of the code and a flexibility that admits modifications without limitations. Open models also fit the business of VTI, being a governmental research institute. The currently used truck dynamics model in the driving simulator is developed and owned by an OEM. This model, validated at the OEM, was further used here as a reference and a source of inspiration in the development work.

Vehicle dynamics models for heavy vehicles in driving simulators have been a research topic for decades, see for example (Bareket, 1991). The development of computer power enables a steady increase of accuracy in the simulation models used in the driving simulators. However, the computational issue remains to date and is emphasized in the simulation of longer vehicle combinations. Commercially available simulation software, such as TruckSim RT from Mechanical Simulation, can be considered as state of art models for driving simulators. However, the source code is not open to the user which limits the use in research applications and for modifications. Hence, the OEM model was used in this work as a reference due to the access to the source code.

The chosen vehicle combination for validation is an A-double, as it is one of the most promising LCVs based on the European Modular System (Nilsson, 2015 and Aurell, 2007). It consists of four units, i.e. a tractor, a semitrailer, a converter dolly and another semitrailer. In the modelled vehicle, the tractor has one steered front axle and two rear driven axles with twin tires. Both semitrailers have three axles, whereas the dolly has only two.

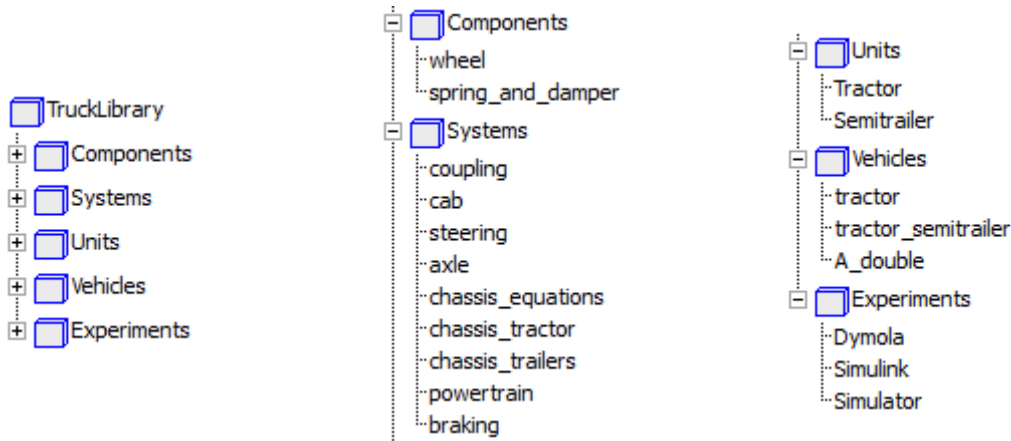
The main aim of this paper is to research the required level of detail for the modeling of each component and phenomenon for the driving of trucks in driving simulators. The required level of detail is defined in a context, i.e. a situation or a scenario, which also delimits the valid region of operation of the model. The chosen scenario was normal driving at almost constant speed on a dry road with small steering angles, i.e. typical motorway driving. The chosen scenario motivates a focus on vertical and lateral dynamics, rather than longitudinal. It also implies that extreme maneuvering, such as rollover etc., are out of the scope of the model.

The model was developed gradually by adding components and phenomena and hence increasing the complexity step-by-step. In each step, the increment was determined by focusing on what was believed to be the main contributing factors. The model and all the gradual steps to the final model have been compared to the corresponding output of the OEM model. This results in a relative validity or accuracy and relies on the validity of the OEM

model. However, it should be noticed that the required model complexity is ultimately determined by the perception of drivers of the driving simulator for this application. The final model was therefore tested in the simulator in a small scale experiment with the aim of evaluating the subjective perception of motion.

## 2. Model Description

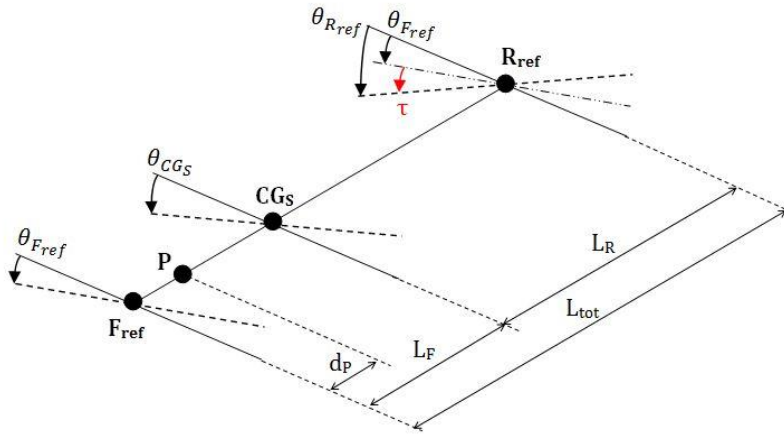
The model was developed in the commercial modeling tool Dymola (Dynamim, 2009), which provides export possibilities to Simulink. Simulink from Mathworks is one of the most frequently used software tools for real-time applications and it supports real-time computing via e.g. the XPC target environment. This is also available in the Sim IV environment. Dymola uses the programming language Modelica, which is based on an open object-oriented modeling language specification (Modelica Association, 2012). Modelica may be used with standard and commercially available libraries of components, which makes the modeling easier. However, the chosen approach was to write the entire model in plain text code and to define all involved parameters, variables and equations. The main motivation was transparency and ease of use of the code for people with no knowledge in the libraries of Modelica. Another advantage of not using proprietary libraries is the possibility to run the code in open source tools like e.g. the OpenModelica environment, see (Fritzson 2005). Finally, the structure of the passenger car model developed by Bruzelius (2013) and Obialero (2013) using the same strategy could be used as a simpler example. In fact, this strategy does not necessarily imply a single level model. A library structure can be constructed, with a number of hierarchical levels tailored for the specific application. The designed *TruckLibrary* (see Figure 1) comprises of 5 packages, called *Components*, *Systems*, *Units*, *Vehicles* and *Experiments*, each made of a number of models.



**Figure 1 – Model library structure, screenshot in Dymola environment: compact view on the left, expanded view of the 5 packages in the centre and on the right.**

As mentioned, the vehicle model was developed with a stepwise approach. The components of the final model are briefly explained here. In the chassis, the equations of motion are defined equally for all units. A Newtonian approach is used for the derivation of them. Each unit has a sprung mass, which can heave, roll and pitch with respect to the axles. Each unit is assumed to roll (and pitch) about a roll (and pitch) axis, which is found by the roll (and pitch) centers of the axles. The chassis frame of the tractor is considered to have a torsional flexibility. The other units are considered to have infinitely stiff frames. This is due to the fact

that only the inclusion of the tractor frame stiffness has a substantial effect on the resulting trajectory of the tractor itself. In this study, a non-conventional approach to the frame torsion is used. There is a roll equation for the center of gravity of the tractor sprung mass (CG<sub>S</sub>), which is complemented with a torsion flexibility equation. These two equations determine the roll angles at each longitudinal position along the tractor. A single torsion equation implies one degree of freedom. As a consequence, there is a need to define two points along the frame that will determine the length of the frame section. These points will be taken as the ideally unique points of application of torques. The two reference points are taken at the front axle and at the position of the fifth wheel.



**Figure 2 -- Coupled roll and torsion of frame, schematic representation.**

Figure 2 shows a schematic representation of the chassis frame subject to both roll and torsion. For simplicity, the frame is represented by a longitudinal line (representing the two longitudinal rails) and three transversal lines showing the frame orientation at the positions of the CG<sub>S</sub> and of the two reference points, called F<sub>ref</sub> and R<sub>ref</sub>. When torsion takes place, the roll angles at each of these three points are different, as shown in the scheme (the hatched transversal lines represent the orientation of the frame after roll and torsion). The torsion angle  $\tau$  is then defined as the difference between the roll angles ( $\theta$ ) at the two reference points.

$$\tau = \theta_{R_{ref}} - \theta_{F_{ref}} \quad (1)$$

In accordance with the assumptions of one degree of freedom and of linearity, the roll angle is linearly increasing (or decreasing, depending on the sign of  $\tau$ ) along the whole frame (Morello, 2001). As  $\theta_{CG_S}$  is the known variable from the roll equilibrium, it can be used to find all other roll angles by adding (or subtracting) a fraction of the torsion angle  $\tau$ . For instance, the roll angle at the front reference point is found by Equation 2.

$$\theta_{F_{ref}} = \theta_{CG_S} - \tau \cdot \frac{L_F}{L_{tot}} \quad (2)$$

The frame torsion angle and its rate are found by the Equation 3, where moments applied to the two reference points in the longitudinal direction of the frame, along with a linear

torsional stiffness and linear torsional damping is used. The frame stiffness is inversely proportional to the length of the frame section subject to torsion.

$$-k_t \cdot \tau - c_t \cdot \dot{\tau} + \sum M_{x,Rref} - \sum M_{x,Fref} = 0 \quad (3)$$

The torques which are not applied at the two reference points are scaled by a coefficient to partially balance the approximation of one degree of freedom. The magnitude of the moments will be reduced if applied closer to the other reference point or increased otherwise. For example, a moment applied at the point P (see Figure 2) will be assumed to be applied at F<sub>ref</sub>, but its magnitude will be multiplied by the factor in Equation 4.

$$\left(1 - \frac{d_p}{L_{tot}}\right) \quad (4)$$

The axles are modeled as rigid and each of them has one rotation (roll) and one displacement (heave) degrees of freedom relative to the sprung body. The suspension elements for each axle are two springs (with constant stiffness coefficient), two dampers (with constant linear damping coefficient with the same value in both deformation directions) and an anti-roll bar (with constant stiffness coefficient). The axle is able to heave and roll relative to the ground, as the tires are assumed to be working as springs when compressed vertically.

The rotational velocities of the wheels have a kinematic relationship with their translational velocities, which means that no relative motion (longitudinal slip) between the wheel and the road at the tire-road contact is present. In addition to this, no limits are set to the tire forces. The tires have been implemented with a simple linear model, with infinite stiffness longitudinally and finite stiffness laterally. The cornering stiffness of each tire is assumed to be proportional to its vertical load.

The cabin is a rigid body suspended on the tractor frame with the degrees of freedom of two rotations (roll and pitch) and one translation (heave) relative to the tractor frame, with a constant stiffness and a constant damping coefficient for each degree of freedom. A unique point of joint to the chassis is assumed, which transfers 3 forces and 3 moments and which is imposed to have the same velocity in the space as seen by the cabin and by the tractor chassis.

The coupling between two consecutive units is modeled in a similar way as the connection of the cabin to the tractor frame. In an A-double combination, there is a fifth wheel between the tractor and the first semitrailer, a drawbar between the first semitrailer and the dolly and a fifth wheel on a turntable between the dolly and the second semitrailer. In this model, the first and third couplings allow rotations in the yaw and the pitch directions and, instead, constrain the roll rotation; this is a usual approximation of the fifth wheel couplings. Whereas, the second coupling allows free rotations in all directions, i.e. like a spherical joint. Besides, in the first coupling, the yaw rotation is counteracted by a dry friction and linear damping torque.

The steering system is modelled as a conventional hydraulic truck steering system (Tagesson, 2014 and Rothhämel, 2013). It receives the steering wheel angle from the driver and imposes the torque on the steering wheel as a feedback. This allows for a more realistic driving

experience, although it does not affect the trajectory. Roll steer is included and the compliance of the steering torsion bar is considered. Fully Ackermann steering is used. Friction in the steering system is not considered and modelled.

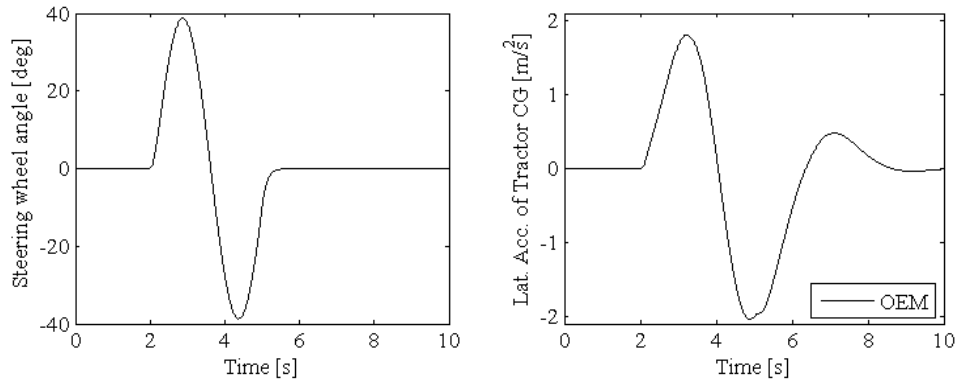
The powertrain uses a one-dimensional engine map for the torque-speed characteristics. The automatic gear shift is dependent on the engine speed without torque drops. No dynamic behavior of the engine is modelled, the retarder torque is not present and no delay from driver request to actuation is present. The braking system is modelled in a similarly simple fashion with equal brake distribution and no delays. The equations used in these two systems come from the model developed by (Bruzelius, 2013).

### 3. Simulation results

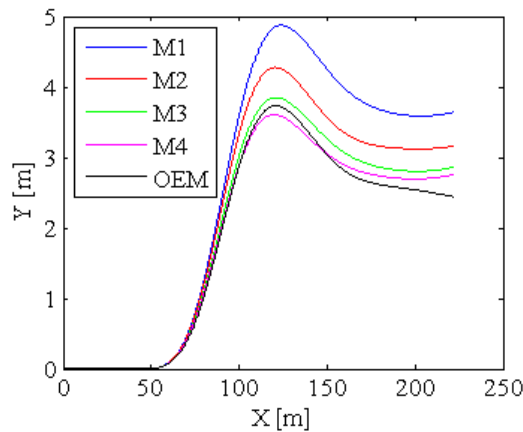
The developed Modelica model was compared to the reference OEM model with simulations in the Matlab Simulink environment. The OEM model represents an A-double combination and is implemented in the SimMechanics environment in Simulink. SimMechanics is a multibody toolbox for Simulink. For example, the frame torsion is implemented by modeling the sprung mass as two rigid bodies, linked by a spring which allows their relative motion about its own axis. The tires in the OEM model are represented by Pacejka models, in the format of PAC2002. The load case of the OEM model is very high, with yaw inertias of the two semitrailers in the order of as double as typical load cases ( $1 \cdot 10^6$  instead of  $0.5 \cdot 10^6$   $\text{kg} \cdot \text{m}^2$ ). This will excite the semitrailers in lateral maneuvers more than a typical load case, and is used as a stress test of the models.

The aim of the desktop simulations was to compare each modeling step of the Modelica model with the OEM model, to see the contribution of each specific factor. Some of these modeling steps are presented here. Furthermore, the simulations were used to show that the increase of accuracy reflects the progressive increase of complexity of the developed models. This analysis was focused on the lateral behavior of the vehicle, with the purpose of making it similar to the OEM model in the chosen driving context, without introducing unnecessary complexity. A reference maneuver was chosen to represent the selected driving context, i.e. an ISO lane change maneuver (ISO 2002), performed at a constant speed of 80 km/h. This is a common scenario used in the simulator when testing e.g. active safety features and semi-autonomous driving, see e.g. (Nilsson, 2015). When the vehicle is running at the test speed in a straight line, one full-period sinusoidal steering-wheel input is applied. The amplitude of the steering input was chosen so that the maximum lateral acceleration of the first unit is about  $2 \text{ m/s}^2$ . Figure 3 on the left shows the applied steering input, which is a single sine-wave of amplitude 40 degrees and with a period of 3 seconds. The right part of the figure shows the resulting lateral acceleration of the first unit (the tractor) of the OEM model. The longitudinal velocity was controlled to keep an almost constant value of 80 km/h during the maneuver.

The starting point was the simplest possible model, i.e. a linear single-track model with linear tires. The parameters were determined to resemble the OEM model. The dry friction and linear damping torque in the fifth wheel coupling between the tractor and the first semitrailer was included in this model. In the figures, the OEM model is denoted with the abbreviation OEM, while the single-track model with M1. Three main observations can be made from the trajectory of the tractor in Figure 4 (compare M1 with OEM).



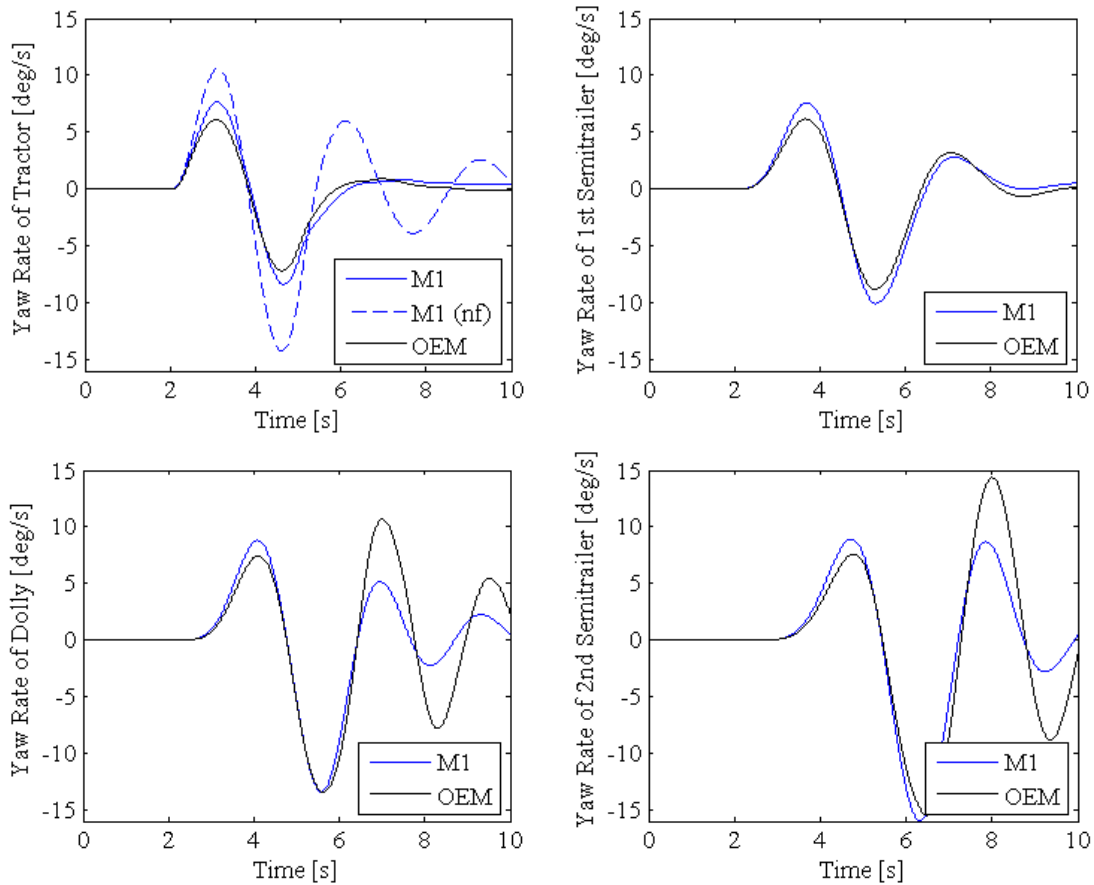
**Figure 3 – Steering wheel input and resulting lateral acceleration of the tractor centre of gravity (CG), according to the OEM model.**



**Figure 4 – Trajectory of tractor CG, according to M1, M2, M3, M4 and OEM models.**

Firstly, there is a considerable difference (about 1 meter) in lateral displacement between the M1 and the OEM model. This can mainly be explained by the absence of roll steer contribution in a M1 model where only planar motion is considered. Secondly, the tractor does not follow a path of perfect lane change in both models. This may be due to the high oscillations of the towed units, which, in turn, causes a large oscillation in the tractor trajectory (of the order of 1 meter). The oscillations are damped out after several tens of meters after the first steering input. The oscillations are largely explained by the heavy load case in the OEM model. The top-left plot in Figure 5 shows the yaw rate of the tractor as for the M1 model, the OEM model and the M1 model where the friction of the fifth wheel coupling has been removed, called “M1 (nf)”. It is clear that the difference between the OEM and the M1 model is highly increased when without friction, as the “M1 (nf)” shows oscillations with large amplitude and several peaks. We can conclude that this friction torque has a certain importance in damping down the oscillations, which, in this case, are mainly due to the excessive values of the inertias. A third observation which can be done from the trajectories is that the final heading direction is different for the two models, as, after the lane change maneuver, M1 tends towards left, whereas the OEM model towards right. This is due to a residual non-zero friction torque in the fifth wheel coupling at steady-state, which has a considerable influence in the yaw equilibrium. This is true especially for the Modelica model, which exhibits a considerable non-zero yaw rate after the maneuver. The plots of the yaw rates

in Figure 5 shows that the last two units are subjected to larger oscillations in the OEM model, due to the fact that their tires have reached the nonlinear region of their characteristics.



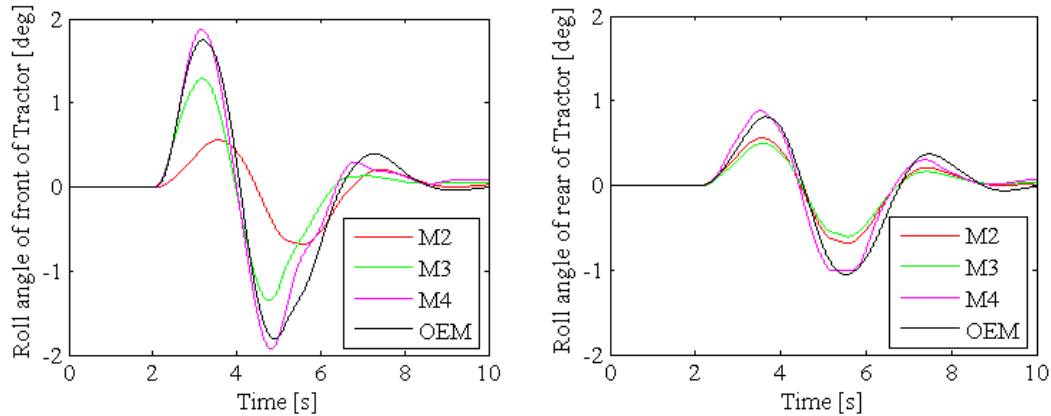
**Figure 5 – Yaw rates of the 4 units, according to M1 and OEM models.**

The following steps were taken to extend the single-track to a double-track model and to remove the assumption of small angles. An Ackermann steering was used, which means that nonlinearities are introduced. This model did not show a significant difference from M1.

The next model M2 introduces the roll, pitch and heave degrees of freedom. Each unit is here made of a sprung mass and a number of axles. In the tractor, the chassis and the cabin are considered as one rigid body. The improvement in the approximation of the tractor trajectory is clear (Figure 4) and it is explained with the roll steer contribution, which was not present in the previous models.

An important factor in the cornering dynamics of trucks is the torsional flexibility of the tractor chassis frame. Adding this factor to the model M3, the improvement in the trajectory is clear (Figure 4) and this can be explained with a more accurate roll steer contribution. Figure 6 shows how the roll angle at the front part of the tractor is different from the roll angle at the rear part (respectively the roll angles at the two reference points  $F_{ref}$  and  $R_{ref}$  defined in the previous) due to frame torsion. This difference was not modeled by the model M2. It is particularly relevant to compare the roll angle at the front part, which shows the greatest difference. Its relevance has two main reasons: it is the angle used for the roll steer term and it is the angle directly affecting the cabin and the driver of the vehicle.





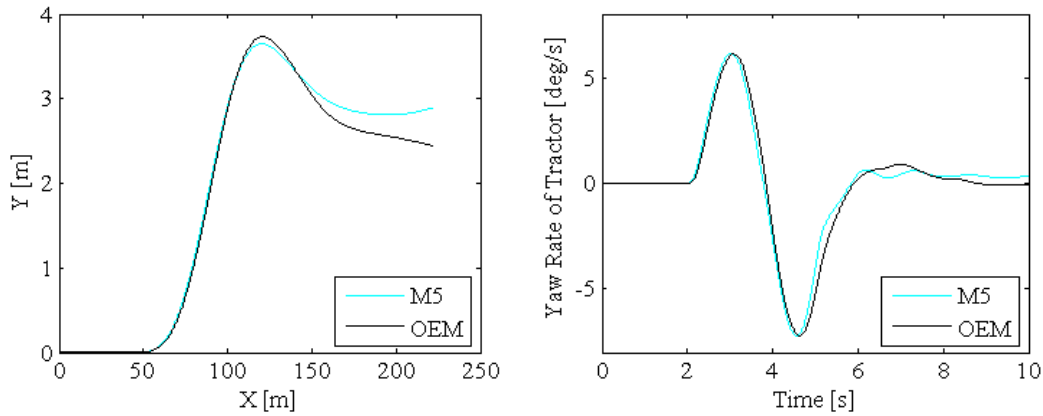
**Figure 6 – Roll angle of the front and of the rear part of the tractor, according to M2, M3, M4 and OEM models.**

The cabin as a separate body from the tractor chassis had the main motivation of the final intended application of the model, which is a driving simulator. This model considers the cabin as a body suspended on the chassis frame. This has a small effect both on the trajectory and on the roll angles.

When the roll of the axles is added in model M4, the trajectory is slightly modified (Figure 4), even though the increment in the roll angle of the front part of the tractor is important (left part of Figure 6). This might seem like a contradiction. However, the angle used in the roll steer term is the relative angle between the front part of the tractor and its first axle. This relative angle was coincident with the former in previous models, as the axles could not roll. When this degree of freedom is added, the roll angle of the tractor becomes the sum of the axle roll angle and their relative roll angle. This relative angle does not apparently change much, while the total roll angles does, and so do the cabin roll angle.

A conventional truck steering system is modeled (model M5). This allows a complete comparison with the OEM model in terms of lateral dynamics, starting from driver input. However, the steer angles at the wheels were very similar, so no particular difference exists between the models M4 and M5. Some relevant results of this final model will be presented. The trajectory of the tractor CG (Figure 7) is very close to the Modelica model and the OEM model, with a difference in the final heading direction due to the fifth wheel friction torque (as previously mentioned). Also, the yaw rates are really close (only the tractor yaw rate is shown in Figure 7), with a difference in the last oscillations of the last two units due to the nonlinearity of tires, in the same way as for the model M1. The roll angles of the first two units are very similar (Figure 6, same as M4). On the contrary, the roll angle of the last two units displays greater peaks in the OEM model, in the same way as for the yaw rates.

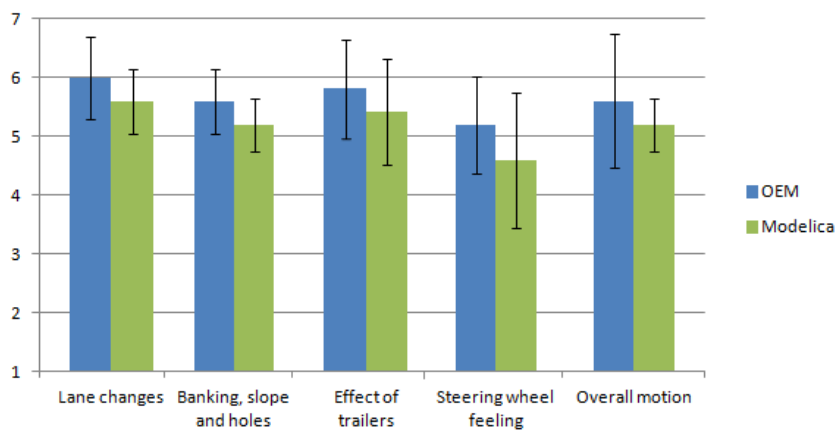
The powertrain and braking systems were the last missing parts to allow driving the model in the driving simulator, as the vehicle needs to be driven from a standstill, should be accelerated to the cruise speed and, after the maneuver, it needs to be stopped. Therefore, a powertrain and braking systems were added to the last model in order to provide a reasonable longitudinal behavior.



**Figure 7 – Trajectory of the tractor CG and yaw rate of tractor, according to M5 and OEM models.**

As the level of detail of the model was regarded as satisfactory, experiments in the driving simulator were conducted. Five volunteer test drivers holding a truck driving license drove both the final Modelica model and the OEM model without knowing which they drove. The experiment focused on the lateral and vertical dynamics rather than longitudinal. The first road section was straight and without any inclination, where the drivers could perform several lane changes at different speeds. After that, the road was still straight but included road banking, road slope and road holes. After each test, the volunteers were asked to fill in a questionnaire, where they gave a rating from 1 to 7 and give comments on the following five aspects on how the two models behave compared to reality:

- the behavior of the vehicle during lane changes
- the behavior of the vehicle during road banking, road slope and road holes
- the effect of the trailers on the vehicle motion and on the steering wheel
- the overall steering wheel feeling
- the overall motion



**Figure 8 –Average ratings (and standard deviations) of the two models.**

The average ratings of the two models are shown in Figure 8. The standard deviations are shown in the figure as lines on the top of the bars. Because of the low number of samples and

the quite similar ratings, it is not possible to draw statistical conclusions. In particular, one driver said “I think the models were behaving very similar to each other. I can’t feel any difference”. The opinions of the other drivers were that both models have a quite realistic behavior, even though the OEM model was generally regarded slightly better.

During the experiments in Sim IV, the figures of the real-time performances of the two dynamics models were noted and compared. The integration time used in the simulator experiments was  $1 \cdot 10^{-3}$  seconds. The average task execution time, i.e. the time to calculate the values of the outputs from the values of the inputs, and the percentage of time step utilization (i.e. task execution time divided by integration time, with 100% meaning that the model is not able to finish the calculations in real-time) were:

- $6.7 \cdot 10^{-4}$  seconds (i.e. 67%) for the OEM model.
- $7.4 \cdot 10^{-5}$  seconds (i.e. 7.4%) for the Modelica model, which is faster than the OEM model by a factor of 10.

#### **4. Discussion**

The comparison of the gradual steps of the Modelica model showed how the increase of model complexity is reflected in the ability to mimic the OEM model. The final results from the desktop simulations show a good match with the OEM model, at least for the first unit. This confirms how the chosen modeling process was successful. The largest source of discrepancy in the first model (M1) was the un-modeled roll steer contribution. When this contribution was added, the resulting output got much closer to the OEM model. The frame torsion was a fundamental element in this view. The cabin resulted in small differences, even if not negligible. Finally, the axle dynamics had a considerable effect, especially for the absolute values of the roll angles. As the contribution of single phenomena was studied, the validity of the conclusions is not strictly dependent on the validity of the OEM and this mitigates the importance of this assumption. One relevant outcome is that the nonlinearity of tires that is reached in the last two units of the vehicle does not considerably affect the motion of the first two units. Moreover, the fact that the tire reached their nonlinear region is mainly because of the large values of the yaw inertias of the semitrailers. A more normal loaded vehicle combination would likely result in a higher agreement. Therefore, simple linear tire models may be sufficiently accurate for applications where the risk of rollover or of loss of lateral stability is not of importance. Otherwise, the graphic system of a driving simulator may allow the driver to notice the motion of the trailers on the side-view mirrors and larger trailer oscillations may affect the perception of the lateral stability of the vehicle. A tradeoff between complexity and accuracy could be to use nonlinear tire models for the last two units. Also, the results from the simulator experiment are overall positive. The developed model was considered slightly less realistic than the OEM model, but the difference is within the range of the uncertainty.

#### **5. Conclusions and Future Work**

A truck dynamics model was created in the Modelica programming language. It is an open model and its structure gives a high level of readability and flexibility. The model was successfully implemented in the driving simulator Sim IV and run in real time with a great execution time margin. The model was subjectively compared to a reference model and it proved to represent adequately the modelled vehicle for the targeted situation of normal motorway driving. Moreover, the presented modeling process allowed obtaining a good approximation, while keeping the model complexity to a minimum. The model is a solid base

for future developments of a library of truck dynamics models and it may be used both in other driving simulators and in desktop simulations.

As future work the authors would suggest the following:

- The Modelica model can be extended to represent more vehicle combinations.
- Investigate the influence of a more detailed friction model for the fifth wheel coupling.
- Create a more detailed tire model, that can represent the nonlinear behavior both longitudinally and laterally. This may be the first step to extending the targeted driving situations and driving conditions.
- Add the tire-road friction coefficient as input to the model, possibly with one input for each wheel.
- Model a more detailed powertrain and braking systems in order to allow studying the vehicle longitudinal dynamics.
- Add some active safety functions such as ABS, ESC and Cruise Control.

## 6. References

- Fischer, M., Sehammar, H. and Palmkvist, G. (2011), “Applied Motion Cueing Strategies for Three Different Types of Motion Systems”, ASME. J. Comput. Inf. Sci. Eng.
- Jansson, J., Sandin, J., Augusto, B., and Fischer, M. (2014), “Design and performance of the VTI Sim IV”, Driving simulation conference, Paris, France.
- Bareket, Z. and Fancher, P. (1991), “Truck or bus dynamic modeling for a driving simulator”, No. UMTRI-91-26.
- Nilsson, P. (2015), “On traffic situation predictions for automated driving of long vehicle combinations”, Licentiate thesis, Chalmers University of Technology.
- Aurell, J. and Wadman, T. (2007), “Vehicle combinations based on the modular concept. Background and analysis”, Volvo Trucks, Report no. 1/2007, Committee 54: Vehicles and Transports.
- Dynasim (2009), “Dymola Multi-Engineering Modeling Laboratory. User Manual”, Vol I & II, Dynasim AB, Lund, Sweden.
- Modelica Association (2012), “Modelica Language Specification, Version 3.3”, May 2012, Available at <https://www.modelica.org/documents/ModelicaSpec33.pdf>.
- Fritzson, P., Aronsson, P., Lundvall, H., Nyström, K., Pop, A., Saldamli, L. and Broman, D. (2005), “The OpenModelica Modeling, Simulation, and Software Development Environment”, Simulation News Europe, 44(45), December 2005.
- Bruzelius, F., Gómez Fernández, J., Augusto, B. (2013), “A Basic Vehicle Dynamics Model for Driving Simulators”, International Journal of Vehicle Systems Modelling and Testing, Vol. 8, No. 4, pp 364–385.
- Obialero, E. (2013), “A refined vehicle dynamic model for driving simulators”, Master’s thesis, Chalmers University of Technology, Göteborg, Sweden.
- Morello, L., Rossini, L. R., Pia, G., Tonoli, A. (2001), “The Automotive Body”, Vol II, Springer.
- Tagesson, K. (2014), “Truck steering system and driver interaction”, Thesis for the degree of licentiate of engineering, Chalmers University of Technology, Göteborg, Sweden.
- Rothhämel, M. (2013), “Characterisation and utilisation of steering feel in heavy trucks”, Doctoral thesis in Vehicle Engineering, Royal Institute of Technology, Stockholm, Sweden.
- ISO (2002), “Road vehicles – Heavy commercial vehicle combinations and articulated buses – Lateral stability test methods”, Swedish Standard Institute SS-ISO 14791.