Development and validation of a modular simulation model for commercial vehicles

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Abstract In this paper a modular truck semi-trailer model is developed in the multi-body toolbox SimMechanics, which is part of MATLAB/Simulink. It can be used to analyze a wide range of active control solutions (anywhere in the model), as well as the influence of passive vehicle modifications. As the model is modular, it is relatively easy (compared to conventional multi-body software packages) to modify or replace any of the components as well as the number and type of trailer axles. The model is validated using a wide range of measurement data obtained from measurements on a real air sprung tractor semi-trailer.

Keywords: Truck model, modules, SimMechanics, measurements


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1 INTRODUCTION

Commercial vehicles are a subject of ongoing research within many companies. Besides the all important topic of durability, safety issues like roll-over (see for example Sampson and Cebon (2003)) and driver fatigueness are getting more and more attention. One of the main reasons for this trend is the large number of fatalities in traffic accidents with this type of vehicles. For example, in the Netherlands there are approximately one hundred and fifty thousand trucks versus more than seven million passenger cars. Nevertheless, when looking at traffic fatalities in the period 1999 to 2001, one in six was caused by accidents with truck involvement, Schilperoord (2007). In the same article it is also mentioned that many truck manufacturers use electronic systems to further enhance the safety of their vehicles.

Two other topics that are getting attention in this field are longitudinal vehicle control, see for example Lu and Hedrick (2005) or Zalm et al. (2007) and active suspensions, see for example Hrovat (1997), Muijderman (1997) and the market ready semi-active CDC system given in ZF Sachs (2007). All these developments require some kind of control. As such, the topic of controller design, as well as the topic of model development, is becoming more revelant.

Multi-body modeling of commercial vehicles is not a new subject. General purpose software packages like Adams, SIMPACK and LMS Virtual.Lab have already been used for many years to study the dynamic behaviour of commercial vehicles, and to investigate the effect of design changes. Furthermore, the last few years more dedicated software packages like TruckSim and veDYNA have also been developed. However, even though the interaction with control software like MATLAB/Simulink is often possible, limitations remain. Furthermore, they are typically not flexible in the vehicle configuration (e.g. obtaining results with a modified component takes a lot of effort). Moreover, reports of the experimental validation of vehicle models developed with these packages are scarce.

Nowadays, detailed truck FEM models can contain over a million degrees of freedom. A large number of these generally lie in the flexible chassis, which is the part that interconnects the main truck components. According to Jiang et al. (2001), this complexity is necessary if one is to model things like driver comfort accurately. However, these complex models can typically not be used (effectively) for real-time simulation (see for example Allen et al. (1998)), or controller design and validation. This is either because of the computational cost, or the lack of an interface with control software. Furthermore, considering the many sources of uncertainty, e.g. the number and type of trailers; loading conditions; alignment of suspension components; etc, these models may give a false sense of accuracy.

In this paper a modular truck model is presented that is suitable for controller design and evaluation. Furthermore, the model is validated using a wide range of measurements on an actual vehicle.

A modular approach is chosen to be able to deal with the wide variety of possible truck configurations, see for example Gillespie and Karamihas (2000). Furthermore, the validation is done with a real tractor semi-trailer combination. It is shown that even though the used model has some simplifications of reality (among which a
chassis with only one flexible mode) the simulation results give a good real world representation.

The paper is structured as follows. First the simulation software is discussed, followed by an in dept overview of the truck modeling. Using the developed component modules a tractor semi-trailer simulation model is constructed and validated in section 4. This paper finishes with a set of conclusions and an indication of future work.

2 SIMULATION SOFTWARE

There is a wide range of commercial multi-body software available nowadays. However, as it is desired to combine control software with the multi-body software (using a simple interface), the choice was made to use SimMechanics. It basically is the multi-body toolbox of MATLAB/Simulink, which has the great advantage that it can interact with any level of controller developed in Simulink. Furthermore, it is very easy to create sub-systems with a mix of multi-body and control elements, which can be placed in a component library as modules.

The main drawback however is the poor default visualization. This is especially bothersome when more complex systems are evaluated. Still, when using the MATLAB Virtual Reality toolbox the visualization can be greatly enhanced, as is illustrated by Besselink (2006).

3 TRUCK MODEL HIERARCHY

The MATLAB/SimMechanics model is structured in a way that resembles the physical appearance of a truck. The chassis module serves as the backbone for the model, upon which the other modules can be plugged. This way, the effect of using for example an additional trailer or different drive-line can be verified more easily. One simply has to replace the component of interest.

The model hierarchy for a tractor semi-trailer combination is depicted in Figure 1. Of course, other vehicle combinations can also be chosen. From the figure it can be seen that there are three main modules: the driver module; tractor module; and the trailer module. Herein, the last two consist of several other (sub)-modules, see Figure 2.

All the modules have their own parameters embedded in their masks and each module has its own coordinate system. As such, it is only required to specify the global position of the origin of this frame in its initial condition. All other components within the module are specified with respect to this coordinate system.

Furthermore, the modules are interconnected with a single connection (rigid constraint). This is possible due to the use of so-called ‘dummy chassis’ bodies (bodies with negligible mass) in some of the connecting modules. The components within these modules are connected to such a ‘dummy chassis’, which in turn can be connected through a weld joint (rigid connection) to the chassis module.
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For example, the cabin module uses such a ‘dummy chassis’. Without it, each of the four suspension elements would have to be connected to the chassis and variations in the location of the suspension mounts would result in changes in both modules. However, with the ‘dummy chassis’ body, these variations only influence the cabin module, which can be easily replaced to evaluate the effects. The different modules are shortly discussed in the following sections.

Figure 1  Model structure, connection of the modules.

3.1 Driver module

Modeling a driver and his or her driving behaviour is complicated as driving styles are very personal. Furthermore, as remarked in Muijderman (1997), it also depends on the type of vehicle that is being driven. The driver influences the vehicle with the throttle, brakes, steering wheel and gear-shift, depending on the desired velocity and trajectory.

The driver module has all the mentioned driver inputs embedded, be it in a simplified way. Each of the inputs, brake, throttle and steering wheel angle, is specified as a function of time. Furthermore, a continuous variable gear ratio is used to determine the torque at the wheels. Moreover, the steering input is filtered with a 20 Hz first order low-pass filter to limit the influence of measurement noise.

3.2 Tractor module

The tractor module consists of a chassis module with a cabin module, steer axle, drive axle, drive-line module, and braking module connected, see Figure 2.

3.2.1 Cabin module

A truck cabin consists of many elements and is generally optimized for driver comfort and safety. It is connected to the chassis through a relatively stiff roll bar, that limits cabin roll and longitudinal and lateral displacement (bushings) with respect
to the chassis. Furthermore, there are four air springs and dampers in vertical direction and two lateral dampers at the rear. The air springs typically have a load leveling controller embedded. The main source of uncertainty in cabin modeling lies in the loading conditions of the cabin. The load and location of the center of gravity depend on the type of driver, number of passengers and the cargo in the cabin.

The cabin is modeled as a rigid body. In lateral direction, it is fixed with two spring damper combinations at the front and dampers at the rear. In vertical direction, there are spring damper combinations at each of the cabin corners. All of the suspension elements are modeled with bump-stops and a pre-load is also included to start simulations from an equilibrium. Furthermore, a leveling controller is implemented to cope with load changes.

3.2.2 Chassis module

The tractor chassis is a ladder-like construction. It consists of two long side beams with a number of cross beams in between. As a result of its relatively low mass in comparison to the masses it interconnects, it is highly flexible with the lowest eigenfrequencies typically in the range of zero to fifteen Hertz, see Muijderman (1997) or Liebregts (2007).

The chassis module consists of two lumped masses which are connected through
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3.2.3 Steering axle module

The steering axle module consists of the rigid front axle, front axle suspension, roll stabilizer, steering system, wheel bearings with brakes and the tyres. The front axle is modeled as a lumped mass which is connected vertically to a "dummy" chassis using two linear springs and two nonlinear dampers (lookup tables). As a result, the axle has only two degrees of freedom \((z, \phi)\) with respect to the tractor chassis. Herein, the roll stabilizer is modeled as a \(z, \phi\)-joint with rotational stiffness, which is located in the roll center. The "dummy" chassis is connected to the chassis module with a weld joint.

Secondly, a body with negligible mass is connected at each end of the front axle through a revolute joint with \(\psi\) freedom. The angle at these joints is determined by the steering system. For reasons of simplicity it is chosen to prescribe the steering angle as a multiplication of the steering wheel angle and the steering ratio. Each of these bodies is connected using a revolute joint (\(\phi\) rotation) to one of the front tyres, which are modeled using the TNO Delft Tyre model, see Besselink (2006) and Pacejka (2002). The brakes generate a braking moment that is applied on these joints. These braking moments are computed in the braking module.

3.2.4 Braking module

The braking system consists of the brake and the pneumatic connections to all the braking disks on each of the wheels. In practice there will be a time-delay between the time of braking (application of brake pedal) and the time of clamping. The braking module is a simplification of the real world conditions. No load dependent brake force distribution, nor time delays are taken into account. It generates the braking moment on each of the wheels as a linear function of the normalized driver braking input (brake pedal inclination).

3.2.5 Driving axle module

The driving axle module greatly resembles the steering axle module. However, there are three differences between both modules. Firstly, the wheels are not connected to the steering system. Secondly, there are four wheels. Two at either side of the axle. Finally, the driving axle is actuated by the drive-line.

3.2.6 Drive-line module

A truck drive-line consists of an engine, a gear box and a differential gear. Furthermore, the connection of these parts to the chassis also needs to be taken into
account. The drive-line module consists of all these components. The engine and gearbox are lumped in one mass that is connected with four bushings to a dummy chassis that is rigidly connected to the chassis module. Furthermore, it is connected through a joint with four degrees of freedom ($x, \theta, \phi, \psi$) to the drive-shaft. This joint is actuated in roll ($\phi$) direction by the engine torque coming from the gearbox. This torque is computed with a lookup table using the wheel velocity and throttle as inputs. As a result, the engine and gearbox are combined as an engine with continuous variable transmission.

So, again control and multi-body components are interconnected. However, drive-line vibrations are not included in this model. Still, in steady-state the engine does generate a (realistic) torque on the chassis.

The drive-shaft is connected through a joint with two degrees of freedom ($\theta, \psi$) to the driving axle. As a result, the driving axle will not actually drive the wheels in $\theta$ direction, which simplifies the connection of the suspension elements. Instead, the driving moment is computed from the applied drive-shaft torque and the final drive ratio. This moment is directly applied to each of the wheels.

3.3 Semi-trailer module

The largest source of uncertainty comes from the trailer(s) and especially the loading conditions. The load will determine the mass and position of the center of gravity of the trailer(s). Furthermore, liquid cargo will have a very different dynamical influence on the truck-trailer combination than solid cargo, see for example Acarman and Özgüner (2006).

For this module we consider a standard semi-trailer. It consists of a flexible ladder-like chassis, three wheel axis, six tyres and some solid-state cargo. The flexible chassis is modeled as three masses which are connected with joints that allow torsion along the x-axis. At each of these joints a spring stiffness is added to mimic the chassis torsion stiffness, see Figure 3.

Rigidly connected to each of these chassis blocks (at a certain height above the frame) is a cargo block with a certain mass. Furthermore, the second chassis mass is connected to one wheel axis and the third to two. The front trailer chassis mass is connected to the chassis module with the fifth wheel: a joint that allows pitch ($\theta$) and yaw ($\psi$) rotation.

3.4 External influences

Besides the driver inputs, the vehicle is influenced by two main external influences. First, there is the road profile, which is specified in the road profile file, that is included in the tyre software. The file consists of a table that specifies the road height as a function of the traveled distance.

Secondly, there are the aerodynamic influences, which are currently not included in the model.
4 VALIDATION

In this section the designed 44 DOF model (see Figure 3) is evaluated using measurements acquired during driving tests performed with a real truck. In this paper we will focus on four different test types: longitudinal acceleration; road obstacles; double lane change; and emergency braking. Using these measurements it should be possible to evaluate all the different aspects of the model and also identify possible deficiencies.

4.1 Experimental setup

The truck under investigation is an air-sprung tractor semi-trailer. The tractor has been equipped with a number of sensors:

- 8 vertical acceleration sensors, above and below the cabin suspension;
- x,y,z acceleration sensors center of gravity chassis;
- x,y,z acceleration sensors at the front center of the cabin;
- roll, pitch and yaw velocity at the front center of the cabin;
- 8 displacement sensors, wheel and cabin suspension.

However, no additional sensors have been added to the trailer. Therefore the focus of the following validation will be on the tractor modeling. The displacement sensors

Figure 3  Schematic representation of the 44 DOF model.
have a measurement noise root-mean-square value (rms) of approximately 1 mm; the rotational velocities are distorted with measurement noise with a rms value of 0.025 deg/s; and the acceleration measurements have a measurement noise rms of 0.1 m/s². The measurements (real world and simulated) are filtered with a causal, anti-causal second order butterworth filter with a cut-off frequency of 45 Hz, which is about three times the highest frequency of interest for this model. Furthermore, the simulated accelerations are obtained using a sensor model that includes the influence of gravity.

4.2 Model parameters

The simulation model uses a large amount of parameters. All of these are physically meaningful and many can be measured directly. However, some are determined using a few basic calculations. Examples are the locations of the centers of gravity of the masses, the torsional stiffness of the various chassis elements and the orientation of the suspension elements. These measured and calculated parameter values are used in the following subsections. However, in the future a parameter optimization study may be required to further enhance the results. Such a study, will be involved due to the large amount of parameters, the complex modal vehicle behavior and sheer amount of measurement data.

4.3 Accelerating

The first test under evaluation is an acceleration test. Herein, the driver starts from standstill, quickly accelerates up to approximately 9 km/h and then presses the clutch. The test is performed in first gear. The vehicle’s longitudinal velocity is given in Figure 4. It can be seen that there is a sudden jump in the measured velocity at $t = 66$ seconds. This is a result of the fact that the CAN-bus output is zero for very low velocities.

Figure 4 Acceleration test. Vehicle longitudinal velocity measured (solid) and simulated (dashed).
The longitudinal acceleration is shown in Figure 5. Clearly, the simulated and measured accelerations are quite similar both at the chassis center of gravity and the front center of the cabin.

As a result of the longitudinal acceleration the truck will pitch backwards, see Figure 6a. However, due to the engine torque the chassis will also twist which will result in a roll velocity, see Figure 6b. Comparing the measured and simulated results in Figure 6, it can be seen that the main trend in simulated pitch and roll velocities is also visible in the measurements. However, in the measurements there are also quite a number of additional spikes and a reasonable amount of high frequent noise.
The comparison of primary suspension displacements, see Figure 7, shows a better correlation. It can be seen that the relative displacement at the front left suspension has the largest difference between measurement and simulation. This is a result of the non-smooth velocity jump at $t = 66$ seconds, see Figure 4. Even though the velocity used as reference for the simulation is filtered, the required engine torque is too high in simulation.

When looking at the relative displacements of the secondary suspension, see Figure 8, the results show a similar result.

4.4 Road obstacles

Next we consider the response to a discrete event. The event under consideration is a trapezium-shaped road profile disturbance as given in Figure 9a. The disturbance is only applied to the right side of the vehicle and is crossed with a nearly constant longitudinal velocity, see Figure 9b.

In Figures 10 and 11 the chassis and cabin accelerations, measured above and below the cabin suspension elements, are given respectively. It can be seen that the noise level on the chassis is substantially larger than that in the cabin. The main contributor to this high noise level is the engine. The measurement noise influence is negligible in comparison. Nevertheless, the acceleration spikes induced by the obstacle can be predicted very accurately both for the chassis and the cabin.
Figure 8  Acceleration test. Normalized cabin suspension displacements, measured (solid) and simulated (dashed). Positive displacements correspond to compression.

Figure 9  Discrete event. Road input (a) and longitudinal velocity (b); measured (solid) and simulated (dashed).

The angular velocities measured in the front of the cabin also show a good match, see Figure 12. Especially the relatively small pitch and yaw velocities look very similar over the depicted time-range. However, for $t > 34$ seconds a mismatch appears between the measured and simulated roll velocities. Still, as the timing of the spikes in the accelerations (Figures 10 and 11) do fit, it is unlikely that the small mismatch in longitudinal velocity, see Figure 9b, is the cause.

Finally, when looking at the suspension displacements, it can be seen that the dif-
Figure 10  Discrete event. Vertical accelerations chassis below cabin suspension elements, measured (solid) and simulated (dashed).

Figure 11  Discrete event. Vertical accelerations cabin above cabin suspension elements, measured (solid) and simulated (dashed).

...ferences between the measured and simulated response for the primary suspension (Figure 13) for \( t > 32 \) seconds are more or less of the same size as those for \( t < 32 \).
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seconds. As such, the results match reasonably well. Still, there seems to be a small error in the rear suspension modeling as there is an increase in each of the errors around $t = 34.5$ seconds. Possible causes for this effect are flexibility of the cabin and/or friction in the rear air springs.

**Figure 12** Discrete event. Pitch (top), roll (mid) and yaw (bottom) angular velocities cabin, measured (solid) and simulated (dashed).

![Pitch, Roll, and Yaw Angular Velocities](image1)

**Figure 13** Discrete event. Normalized axle suspension displacements, measured (solid) and simulated (dashed). Positive displacements correspond to compression.

![Normalized Axle Suspension Displacements](image2)
Furthermore when looking at the responses of the secondary suspension (Figure 14) it can be seen that the response of the front suspension is simulated quite accurately when the front wheel crosses the object and the same holds for the rear suspension when the rear wheel crosses the object. Though it should be noticed that the maximum values of simulation and measurement differ a little for the latter case. However, the response of the front wheels when the rear wheels cross the object and vice versa is less accurate. What causes these last mentioned differences is still unclear. It is unlikely that it is caused by the dynamics of the flexible frame though, as that should also show up in the responses of the primary suspension.

Figure 14 Discrete event. Normalized cabin suspension displacements, measured (solid) and simulated (dashed). Positive displacements correspond to compression.

4.5 Double lane change

One of the standard tests for vehicle handling is the double lane change. Herein, the driver starts with a constant velocity on a straight road. At a certain point he or she (violently) steers the vehicle to a parallel lane and subsequently back to the original lane. The measured steering wheel angle for this maneuver is used as input for the model, see Figure 15a. Furthermore, the vehicle’s longitudinal velocity is kept nearly constant around 60 km/h, see Figure 15b. The execution of this maneuver results in high lateral accelerations on the vehicle. These accelerations, measured and simulated both at the chassis center of gravity and the front center of the cabin are given in Figure 16. Clearly, the main trend of the response matches well. However, there is a small difference in maximum values. Furthermore, it can be seen that the noise level on the chassis measurement
As a result of the lateral accelerations (induced by the centrifugal force), the vehicle will roll. The roll velocity of the cabin, measured and simulated, is shown in Figure 17. Again, the main trends match nicely, while there remains a small difference in maximum values. In the same figure, the yaw-rate of the cabin is given. Clearly, the model gives an almost perfect representation of the real vehicle on this aspect. However, when comparing the responses of the primary and secondary suspension, Figures 18 and 19 respectively, a mismatch is visible at the rear. Of course, a small mismatch is expected given the deviations in roll velocity and lateral accelerations...
as mentioned before. Still, there also seems to be a torsion effect, which results in twist of the cabin.

Figure 17 Double lane change. Roll (top) and yaw (bottom) velocities cabin; measured (solid) and simulated (dashed).

Figure 18 Double lane change. Normalized axle suspension displacements, measured (solid) and simulated (dashed). Positive displacements correspond to compression.
Figure 19  Double lane change. Normalized cabin suspension displacements, measured (solid) and simulated (dashed). Positive displacements correspond to compression.

4.6 Braking

In safety studies, the braking behaviour of a truck is also an important subject. Therefore, in this section the simulated braking response of the truck is compared to that of a real vehicle. The test driver was asked to drive on a straight road at 60 km/h and brake aggressively (with a constant brake pedal inclination), in an attempt to mimic an emergency stop. The longitudinal velocity of the vehicle is shown in Figure 20b. However, as the inclination of the brake is not measured, the question arises how to come to a correct simulation of the event. This problem has been overcome by iteratively tuning the brake input of the simulation in an attempt to match the longitudinal velocities of the simulated and rear vehicle, see Figure 20b. The final brake input that is used, is given in Figure 20a.

When comparing the measured and simulated longitudinal accelerations at the chassis and the cabin, Figure 21, a profound mismatch is visible between simulation and measurement around $t = 25.5$ seconds. The measurements show relatively large acceleration spikes when the vehicle comes to a complete standstill. To prevent this effect from occurring, truck drivers normally lift the brake prior to coming to a full stop. However, this effect is clearly not included in the model.

The vertical accelerations above and below the secondary suspension, Figures 22 and 23 respectively, also show a large mismatch when the vehicle comes to a standstill. Moreover, it also shows the large influence of process noise, this despite the earlier mentioned filtering. Merely observing the noise levels at standstill ($t > 28$
Figure 20  Braking. Simulated dimensionless braking input (a) and longitudinal velocity (b); measured (solid) and simulated (dashed).

Figure 21  Braking. Longitudinal accelerations chassis center of gravity (top) and cabin front center (bottom); measured (solid) and simulated (dashed).

seconds) it is possible to get a feeling for the huge influence of engine vibrations.

From the cabin pitch velocity, Figure 24, it is again possible to identify the two spikes of forward and backward pitching. The first spike is predicted very well with the simulation model, whereas the mismatch at the second spike is substantially larger. Of course this was expected given the mismatch in longitudinal cabin accelerations (Figure 21). However, after pitching forward the measurements also show an oscillation which is damped more strongly in the simulation.
Figure 22  Braking. Vertical accelerations chassis above primary suspension elements, measured (solid) and simulated (dashed).

Figure 23  Braking. Vertical accelerations cabin above secondary suspension elements, measured (solid) and simulated (dashed).
When comparing the responses of the primary suspension displacements, see Figure 25, a small mismatch at the rear suspension is again visible. Here, the weight transfer modeling of the trailer on the truck is expected to play a role (mass and height of the center of gravity of the trailer). Furthermore, the response at the front also shows a small mismatch, which might indicate a slightly stiffer bump-stop be-
Figure 26 Braking. Normalized secondary suspension displacements, measured (solid) and simulated (dashed). Positive displacements correspond to compression.

The response of the secondary suspension, see Figure 26, shows a similar result. However, here the simulated displacements are lower than those measured. Furthermore, there is something else that strikes the attention. For $t > 26$ seconds there is a profound difference in the front displacements. Apparently, the truck cabin remains in a pitched position for quite a while after coming to a standstill. This appears to be some kind of friction effect in the front suspension, which is not included in the model.

5 CONCLUSION

In this paper a simulation model of a full truck trailer combination is presented. It can be used to evaluate the influence of any combination of different components on the dynamic behavior of a commercial vehicle. Furthermore, due to its modular design the evaluation can be performed for a wide range of vehicle combinations. Moreover, as it is constructed in SimMechanics which is the multi-body toolbox of MATLAB/Simulink, the model can also be used for the design and control of active components. Especially this last feature may prove very valuable for further developments in this field.

For the validation of the model, measurements of a variety of tests on a real tractor with semi-trailer are used. These tests were repeated in simulation, where the vehicle parameters have been obtained from measurements on the real vehicle. The
results showed overall a good resemblance between measurements and simulations. However, there are a few points of attention. First of all, the rear axle suspension deflections show a mismatch in the majority of the tests. Secondly, the deflections of the rear cabin suspension during the double lane change show a relatively large mismatch. Furthermore, a static offset was noticed in the front cabin suspension at the end of the brake test. These effects are the subject of ongoing research.

The results presented in this paper give a good picture of the dynamics of a tractor with semi-trailer. Still, despite some simplifications (especially in the drive-line and chassis modeling), the model performs quite well. In the near future this first version of the validated model (with the constructed module library) will be used for a variety of component design and control studies.

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