In vehicle truck steering-system modeling and validation

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ABSTRACT: In this paper a multi-body 44-DOF tractor semi-trailer model is coupled to a 4-DOF steering-system which includes friction and hydraulic power-steering. An extended wheel hub geometry is used to provide the correct feedback torque from the wheels. A tie-rod with stiffness has been included to connect left and right. An instrumented tractor semi-trailer is used to verify the steering-system model predictions during driving. The focus lies on the prediction of the steering-wheel torque and the vehicle velocity and steering-wheel angle are prescribed as an input for the simulation. Two tests are discussed in this paper, a J-turn at 80 km/h and sinusoidal steering-wheel input with a frequency of 0.4 Hz at 65 km/h. The comparison of the measured signals and the predicted values shows that the steering-system model is accurate. The non-linearities caused by friction and hydraulic assistance system can clearly be seen in both the measurement and the simulation.

1 INTRODUCTION

In 2016 a total 35 million heavy-duty trucks are registered. The amount of trucks is expected to grow in the future. By 2030 a growth of 17% is expected in Europe while in Asia and the Middle East this growth is expected to be more in the direction of 150% to 200% [7]. Statistics show that trucks and in particular tractor semi-trailers are over-represented in accidents in comparison to cars. Research shows that causes for this lie in poor rear and side visibility leading to blind spot crashes [10, 12]. Since truck drivers are often exposed to long working hours with a low concentration level this can result in fatigue and high-way hypnosis which can lead to loss-of-control accidents and road departures [11]. Furthermore the size of trucks is growing as well, this includes extra articulation points which result in complexer dynamics [2]. To support the driver under these difficult circumstances, a number of support systems are currently available on high-end trucks or are being developed. These systems include anti-lock braking system, electronic stability control, adaptive cruise-control, intelligent speed adaption, forward collision warning, automatic emergency braking, blind spot information, lane departure warning and curve-speed warning.

Lateral control by manipulation of the steering angle of the front wheels is a novel feature and not commonly available on the truck market. As a part of a Dutch STW project ‘Truck Merging Support’, a merging and lane changing/keeping assistant is being developed to improve safety. To investigate the possibilities for supported truck driving the current steering-system is first modeled and analyzed. Literature in passenger cars reveals that the modeling of the hydraulic power-steering system is necessary [3, 9] and that friction cannot be neglected [8] especially around the center position. Previous work [5, 6] describes the modeling and model verification of a steering-system of a commercial vehicle by means of test-bench measurements outside the vehicle. The aim in these papers is to predict input and output angles, input and output torques and pressures in the steering-system. This has resulted in the steering-system model shown in Figure 4.

In this paper the steering-system model is implemented in a multi-body truck model. A number of tests have been performed with a real instrumented vehicle on a proving ground. The model predictions are compared with the measured signals to verify the model’s validity. This paper is organized as follows: In section 2 the vehicle model and steering-system model will be discussed, section 3 describes the instrumentation of the test vehicle, section 4 discusses the tests performed and the model predictions and section 5 contains the conclusions.

2 VEHICLE AND STEERING-SYSTEM MODEL

In previous research on active cabin suspension a 44 degree of freedom (DOF) multi-body truck model
has been developed and validated [4]. As shown in Figure 1, this model contains five axles, the chassis of the tractor and trailer contain torsional flexibility and cabin and engine suspension are included. This model is constructed by making use of MATLAB SimMechanics. The tyre-road interface is taken care of by usage of the TNO-Delft tyre.

Figure 1: 44 DOF multi-body truck model as in [4]

The steering-system of a truck generally contains components as in Figure 2. The driver controls the steering-wheel which is connected to the steering-house via the steering-column and two universal joints. These universal joints are necessary to facilitate the height adjustment and for the cabin motion. The steering-house contains a hydraulic power-steering system which amplifies the input torque given by the driver. The output of the steering-house is connected to the pitman-arm which moves the drag-link. This drag-link movement is converted into a steering motion of the left wheel by means of the wheel lever. The left wheel is connected via a tie-rod to the right wheel.

Figure 2: Components in a typical truck steering-system [1]

The steering-system model (Figure 4a) is based on [5, 6]. The steering-wheel is modeled by an inertia, \( J_{sw} \), with friction, \( T_{fric,sw} \), due to the bearings. The torque felt by the driver is referred to as \( T_{sw} \) and the steering-wheel angle as \( \delta_{sw} \). The steering-wheel is connected to the input of the steering-house via two universal joints. The stiffness between the steering-wheel and the steering-house is combined in one equivalent spring \( k_{sc} \). The input to the steering-house has an angle \( \delta_{sh,in} \) and inertia \( J_{sh,in} \). Due to the bearing-friction, \( T_{fric,hu} \), is assumed here as well. The input of the steering-house is connected to the output via two springs in series which represent the torsion-bar and the spindle, the stiffness equals \( k_{tb} \) and \( k_{sp} \), respectively. A small damper \( d_{hu} \) is added to include material damping. A fixed gear ratio \( i_{sh} \) results from the spindle lead and sector shaft radius. The inertia of the components after the spindle until the pitman-arm are lumped into one equivalent parameter \( J_{sh,out} \). The output of the steering-house rotates with angle \( \delta_{sh,out} \). Friction \( T_{fric,ha} \) is present on the output as well due to the seals of the hydraulic piston. The power-steering system is modeled as a symmetric Wheatstone bridge where the deflection of the torsion-bar controls the valve openings which results in a pressure difference across the piston. The stiffness in between the steering-house and the left wheel is captured in one parameter \( k_{ha} \). The pitman-arm, drag-link and uprights are modeled in MATLAB SimMechanics. The interaction happens at the pitman-arm rotation point where a torque is applied on the SimMechanics model and the angle is fed back to the Simulink model. To generate the correct feedback torque from the wheels to the steering-system the wheel hub geometry contains the caster angle, king-pin inclination angle, camber angle, toe angle, caster offset, wheel center offset and a vertical hub offset as shown in Figure 3 for the left wheel. Both wheels are connected via the tierod with stiffness \( k_{tierod} \) and include Ackermann steering. Friction on the king-pin is accounted for by dry friction.
3 TEST VEHICLE

An instrumented tractor semi-trailer is used to verify the steering-system model in combination with the multi-body truck model. The vehicle is equipped with various sensors as in Figure 5. The cabin is equipped with a 3-axis accelerometer \( (a_xC, a_yC, a_zC) \) and a 3-axis gyroscope \( (r_xC, r_yC, r_zC) \) at the driver position. The front and rear part of the chassis are equipped with an accelerometer which only measures in the lateral direction \( (a_yF \text{ and } a_yR) \) and a gyroscope which measures the yaw-rate \( (r_zF \text{ and } r_zR) \).

The steering-system is instrumented with a measurement steering-wheel to measure the steering-wheel torque and steering-wheel angle \( (T_{sw} \text{ and } \delta_{sw}) \). The pitman-arm angle, \( \delta_{pa} \), is measured as well as the left and right king-pin angles \( (\delta_{kp,L} \text{ and } \delta_{kp,R}) \). To measure the force transferred between the wheels and the steering-house, the drag-link force, \( F_{dl} \) is measured.

4 VALIDATION OF THE MODEL

A number of tests are executed with the instrumented vehicle on a proving ground. The following tests have been done:

- Steady-state cornering from standstill to 65 km/h.
- J-turns at 80 km/h.
- A sinusoidal steering input at 65 km/h, frequency content between 0.1 and 2.7 Hz.

All the tests have been performed by a driver and not with a steering-robot. Therefore there can be some difference between similar tests and sinusoidal inputs may not always be a 100 % symmetric. The steady-state cornering tests have been used to tune the cornering and self aligning stiffness of the tyres. The J-turns are used to evaluate the model response to a step input and the sinusoidal input tests can be used to assess the model accuracy for specific frequencies. Due to the limited space only two experiments will be discussed; a J-turn since it contains information about on-center driving, a dynamic response and the steady-state behavior. Also a sinusoidal input with a frequency of 0.4 Hz will be discussed since this is a frequency which occurs a lot during high-way driving. For confidentiality reasons all graphs in this paper have been scaled and made unit-less.

In order to mimic the experiments it is important that the vehicle model has the same forward velocity as the real vehicle. To do so a cruise-controller is implemented in the model. The measured forward velocity is used as an input for the model. Furthermore the measured steering-wheel angle is used as an input to the model. As a road input for the model a flat road is assumed. In reality the road is never exactly flat and some inclination is always present. The assumption is made however that the road roughness can be neglected as well as the inclination.

Figure 6 shows the measured and predicted signals of the steering-system during a J-turn. A left-hand turn is made by applying a step-steer input after 2 seconds. The vehicle speed during this experiment is kept constant by means of the cruise control system. The final steering-wheel position is held until the end of the test. The angles are predicted accurately by the model, both the left and right king-pin angle are predicted accurately due to inclusion of the tie-rod stiffness. The pitman-arm angle seems slightly under-
Cabin 3 axis accelerometer (\( a_x^C; a_y^C; a_z^C \))

Cabin 3 axis gyroscope (\( r_x^C; r_y^C; r_z^C \))

Chassis lateral acceleration front and rear (\( a_y^F; a_y^R \))

Chassis yaw-rate front and rear (\( r_z^F; r_z^R \))

Figure 5: Side view of the motion sensors in the truck, here the following nomenclature is used: C = cabin, F = front, L = left, R = rear, a = accelerometer and r = gyroscope

predicted. The drag-link force is predicted accurately as well, during straight-line driving a small difference is seen, probably due to the road inclination which is not present in the model. The step-response and end value appear to be predicted well. The steering-wheel torque already shows a lot of vibrations even when driving straight. Since there is friction present in the system, small movements of the steering-wheel will result in torque felt by the driver. It is important to predict this behavior as well since it contributes to the on-center feeling. The peak value at 2 seconds is caused by the sudden change and is largely friction related. The steady-state value is mainly caused by the tyre feed-back and the power-steering system.

Figure 7 shows the accelerometers and gyroscopes. Interesting to see here is that the front and rear accelerometer signal differ quite a lot when the step input is applied. The model predicts similar behavior as well. The yaw-rates front and rear appear to be quite similar. The lateral acceleration in the cabin and the roll-rate in the cabin are analyzed to evaluate what the driver experiences. Both are predicted accurately as well. Interesting to see here is that the lateral acceleration in the cabin differs from both chassis measured lateral accelerations again when the step is applied.

Secondly, a sinusoidal input with a frequency of 0.4 Hz at a vehicle speed of 65 km/h is analyzed. During this experiment the vehicle speed is kept constant by the cruise-control system. Figure 8 shows the steering-system signals. Again the angles in the steering-system are predicted with a good accuracy. The drag-link force shows some mismatches around the peaks for negative values. The steering-wheel torque prediction appears to be accurate, the peak values are predicted accurately for negative values and diverge a little bit for positive values. An important thing to notice here is the sudden jump when the steering-wheel motion changes direction. Both the measurement and the simulation show this jump in steering-wheel torque, this is something that the
driver definitely notices and is caused by the friction on the input side of the steering-house. Figure 9 shows the accelerometers and gyroscopes during this experiment. Minor differences between the data and prediction exist and the measured signal of the front accelerometer seems to contain a more noise. The roll-rate prediction shows the same general trend but some differences exist.

Figure 10 shows the steering-wheel torque as a function of steering-wheel angle for this test. From this figure it becomes clear that there is friction present in the system since change of steering-wheel angle direction results in a jump in torque. Also the non-linearity of the hydraulic system can be seen in the curved shape of this plot. The top-right graph shows the lateral acceleration as a function of steering-wheel angle. From this graph it can be concluded that the phase of the signal is also correctly predicted.

Since the steering-wheel torque is important for the drivers perception, the bottom four graphs in Figure 10 show the lateral acceleration and yaw-rate as a function of steering-wheel torque. Again the non-linearity of the hydraulic power-steering system is observed since the lines are curved. Friction is visible in these graphs as well since the signal jumps horizontally upon changing direction in lateral acceleration. The bottom two figures show the yaw-rate front and rear as a function of steering-wheel torque. Again here a curved shape is seen and the horizontal jumps show the friction present in the system.

5 CONCLUSIONS

A truck steering-system model has been coupled to a multi-body tractor semi-trailer model. A steady-state cornering test with an instrumented vehicle has been done to estimate the self-aligning stiffness and the cornering stiffness of the tyres. This instrumented vehicle is also used to perform a J-turn and a 0.4 Hz sinusoidal input to verify the steering-system model predictions.

Overall the conclusion is drawn that this steering-system model in combination with the multi-body truck model and the hub geometry is able to predict all measured signals in a satisfactory manner. The angles, accelerations and yaw-rates can be predicted with a high accuracy while the steering-wheel torque, roll-rate of the cabin and the drag-link force appear to be harder to predict.
6 FUTURE WORK

Future work will consist of verification with tests using a different input frequency. When the model is completely verified it will be used to investigate the possibilities of making the steering-system active by means of an extra actuator. This active steering-system can be used to analyze different support systems such as a merging support and a lane keeping/changing assistant.

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REFERENCES


