COMPONENT BASED MODELING AND VALIDATION OF A
STEERING SYSTEM FOR A COMMERCIAL VEHICLE

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ABSTRACT: The prediction of steering wheel torque in a truck is a challenging subject due to the many components connecting the driver to the front wheels. In order to accurately predict the steering wheel torque a highly detailed model is required. Steering systems in commercial vehicle handling simulation models found in literature are often modeled in a simple way while literature on passenger vehicles suggests the necessity of a higher model complexity. In this research a highly detailed model with four states, friction, stiffness, hydraulic assistance and the necessary kinematic relations is developed. This model could assist in the development of the steering system in the early stages of the design. The model is able to accurately predict the steering wheel torque, drag link force and king pin angles. The parameters for this model are estimated by means of dedicated tests and the steering system model is validated outside of the vehicle.

Keywords: Vehicle steering system, Hydraulic power-steering, Commercial vehicle, Steering system model

1 INTRODUCTION

In 2010 the 1 billion unit mark of vehicles in operation worldwide was surpassed. Only in Europe a total of 35 million heavy-duty trucks are registered, this amount is expected to grow by 17 percent in 2030. In other parts of the world this growth is expected to be the strongest in Asia and and the Middle East (ICCT 2013). Trucks and in particular tractor-semi-trailers are overrepresented in accident statistics. This is caused by the unique risk factors in truck driving, such as poor rear and side visibility, leading to blind spot crashes (Schoon et al. 2008, SWOV 2009). Secondly, on longer trips, truck drivers are vulnerable to fatigue and highway hypnosis, contributing to loss-of-control accidents and road departures (Summala & Mikkola 1994). Finally the complex dynamics and growing sizes of trucks (Besselink et. all 2015) make them prone to accidents.

First steps in improving safety have been taken, such as anti-lock braking system and electronic stability control, which are now both standard by regulations in new trucks. State of the art driver assistance systems include: adaptive cruise control, intelligent speed adaptation, forward collision warning, automatic emergency braking, blind spot information, lane departure warning and curve-speed warning. Some of these systems are applied in high-end trucks, however, most are still under active development. This development leads to the future prospect to have fully automated trucks on highways, where the truck operator is no longer a driver but a supervisor.

In order to facilitate features such as automated steering, an active steering system is required. For this, deeper knowledge of a conventional truck steering system is required such that the optimal position of an extra actuator can be determined. A particular challenge in the design of a truck steering system is on center steering feel. In general, commercial vehicles make use of a hydraulic system which amplifies the input torque of the driver with the so called boost-curve, see Figure 1(a). The application of hydraulic assistance based on torque input can result in an indirect
steering feel, especially around the center-position due to the zero gradient of the boost-curve at this position (Pfeffer et. all 2008).

Figure 1.

Steering systems in truck handling simulation models found in literature are often modeled in a simple way. In many cases they do not incorporate steering compliance or power-steering (Loth 1996) and if they do, inertias of the system components are often not considered (Govindan 2012).

Literature on steering system models in passenger cars reveals that in most cases the steering system is modeled in a relative simple manner. Most models have three degrees of freedom with a stiffness between the steering wheel and the assistance motor and a stiffness between the wheels and the assistance motor (Parmar and Hung 2004, Song et. all 2004). However, if we want to predict the steering wheel torque in an accurate way, a more detailed model is required. In (Lozia and Zardecki 2007) it is shown that a coulomb friction model is sufficient to describe the steering wheel torque during cornering. (Pfeffer et. all 2008) also shows that modeling of friction is required to predict the steering wheel torque around the center position. (Rösth 2007) shows a concept with an additional motor on the input side of the steering house to compensate for friction in the steering system and enable features like parking pilot, lane keeping assist, emergency lane assist, active yaw control and torque reference control.

Figure 1(b) shows a typical steering system layout in a commercial vehicle. From this figure it becomes apparent that the steering system contains a lot of components connecting the steering wheel to the front wheels due to a sprung cabin and limited space around the engine. This results in multiple universal joints, a hydraulic power-steering system and multiple transitions from rotation to translation and vice versa. The ratio of input torque of the driver and output torque at the front wheels is much bigger compared to a passenger vehicle and therefore low amounts of friction on the driver side can influence the on center steering feel significantly. In this research a highly detailed model is developed which takes all of these aspects into account.

The goals of this paper are defined as follows:

- Predict the steering wheel torque in the on center region as well as for larger steering wheel angles.
- Have a component based structure so it can be used to analyze the design of such a system.

First the steering system model and parameter estimation will be discussed in section 2. Section 3 shows the validation of this model outside of the vehicle. The conclusions and recommendations are presented in section 4.
2 STEERING SYSTEM MODEL AND PARAMETER IDENTIFICATION

Steering systems in commercial vehicles consist of many parts in between the driver and the wheels as can be seen in Figure 1(b). The driver actuates the steering wheel which is connected to the steering column. The steering column is connected to the steering shaft by means of a universal joint. This steering shaft is connected to the input side of the steering house via a second universal joint. Inside this steering house, see Figure 3(b), a torsion bar is present which is used to estimate the driver torque. This torsion bar operates a set of hydraulic valves which regulate the pressure to a hydraulic cylinder. The torsion bar is also connected to the hydraulic piston via a spindle which converts rotation into translation. This hydraulic piston is connected to a sector shaft where the translation is converted into a rotation again. This is the output of the steering house. The output of the steering house is connected to the pitman arm which is connected to the steering arm via the drag link. Via this steering arm the translation of the drag link is converted to a rotation around the king pin axis. The steered wheel is connected to the other wheel by means of a tie rod.

2.1 Overview

In order to understand the steering system design and to investigate a suitable position for an extra actuator, a model is required. This model should be able to accurately describe the steering wheel torque felt by the driver as well as the torque at the output side. A model with four degrees of freedom is used in order to implement friction at four locations. In between the masses, springs have been used to model the flexibility and a gear-ratio is implemented. The assistance torque is generated as a function of the input torque. The steering system model is shown in Figure 2 where the same structure as in Figure 1(b) is used. The equations of motion for the system read:

\[
\begin{align*}
J_{sw} \ddot{\delta}_{sw} &= T_{sw} - T_{fric,sw} - (\delta_{sw} \dot{u}_{uj} - \delta_{sh, in})k_{sc} \dot{u}_{uj} - T_{ecc,sw} \\
J_{sh,in} \ddot{\delta}_{sh,in} &= (\delta_{sw} \dot{u}_{uj} - \delta_{sh, in})k_{sc} - (\delta_{sh, in} - \delta_{sh, out} \dot{u}_{sh})k_{hua} - T_{fric,hua} \\
J_{sh,out} \ddot{\delta}_{sh,out} &= (\delta_{sh, in} - \delta_{sh, out} \dot{u}_{sh})k_{hua} + T_{PS} - (\delta_{sh, out} - \delta_{kp} \dot{u}_{pa})k_{ha} - T_{fric,ha} \\
J_{kp} \ddot{\delta}_{kp} &= (\delta_{sh, out} - \delta_{kp} \dot{u}_{pa})k_{ha} \dot{u}_{pa} - T_{fric,kp} - T_{kp}
\end{align*}
\]

Figure 2. Steering system model

2.2 Hydraulically Un-Actuated part

The interaction with the driver happens at the steering wheel, where a torque is applied by the driver, \( T_{sw} \). This torque is applied on the steering wheel inertia, \( J_{sw} \), and results in a steering wheel angle, \( \delta_{sw} \). The steering wheel inertia shows a coulomb friction torque \( T_{fric,sw} \) which is typical for bearings and universal joints. In order to implement the friction in the model a reset integrator model has been used (Haessign & Friedland 1991) due to its straight-forward implementation and computational performance. Since bearings in general do not only show coulomb friction but a viscous damping component as well this is added in the form of \( \dot{d}_{sw} \).
In a typical truck, height adjustment of the steering wheel is possible. In order to facilitate this height adjustment and because of packaging reasons, universal joints (U-joints) are used. The kinematic relations of a universal joint with input angle $\delta_{U\text{joint,in}}$, output angle $\delta_{U\text{joint,out}}$, torque input $T_{U\text{joint,in}}$, torque output $T_{U\text{joint,out}}$ and inclination angle $\beta$ are defined as:

\[
\delta_{U\text{joint,out}} = \arctan\left(\frac{\tan(\delta_{U\text{joint,in}})}{\cos(\beta)}\right)
\]

(5)

\[
T_{U\text{joint,out}} = T_{U\text{joint,in}} \left(1 - \sin(\beta)^2 \cos(\delta_{U\text{joint,in}})^2\right) / \cos(\beta)
\]

(6)

In this model, two U-joints are used with inclination angles $\beta_1$ and $\beta_2$. For the sake of readability the full kinematic relations for the series connection of these U-joints are not given here. We define a ratio $i_{uj}(\delta_{sw}, \beta_1, \beta_2)$ which defines the ratio caused by the two U-joints in series. This variable ratio is depending on the input angle $\delta_{sw}$ and angles $\beta_1$ and $\beta_2$.

The steering wheel center of gravity does not necessarily coincide with the rotation axis as shown in Figure 3(a). The shortest distance between the center of gravity and the rotation axis is indicated with $L_{ecc,sw}$. If the steering wheel has an angle $\theta_{sw}$ with the horizontal plane and mass $m_{sw}$ a torque will be generated as a function of the steering wheel angle:

\[
T_{ecc,sw} = m_{sw}L_{ecc,sw} \sin(\theta_{sw}) \sin(\delta_{sw})
\]

(7)

The stiffness in between the steering wheel and the input of the steering house is lumped into one parameter, $k_{sc}$. This stiffness connects to the input of the steering house which has inertia $J_{sh,in}$ and angle $\delta_{sh,in}$. A friction torque is present on the input, $T_{\text{fric},h\text{ua}}$, where $h\text{ua}$ stands for Hydraulically Un-Actuated with viscous damping $d_{h\text{ua}}$. This friction is again caused by the bearings used for the torsion bar and the spindle.

The internals of the input side of the steering house is represented by a series of springs:

1. a spring with stiffness $k_{\text{cent}}$, used to create a lower on center stiffness.
2. the torsion bar stiffness $k_{tb}$, also used to determine the amount of assistance torque.
3. the stiffness of the spindle $k_{\text{spindle}}$, connects the torsion bar to the piston.

In order to have a lower on center stiffness engaging only in the on center region, a free play $\Delta_{\text{cent}}$ is placed in parallel. In this way, the total stiffness will only be lower if the torque on this spring is within $\pm \Delta_{\text{cent}}k_{\text{cent}}$. The same strategy is used for the torsion bar by means of free play $\Delta_{tb}$. A total stiffness $k_{h\text{ua}}$ is defined as the resulting stiffness of these springs. Given the assumption that $k_{tb}\Delta_{tb} > k_{\text{cent}}\Delta_{\text{cent}}$, three possibilities for $k_{h\text{ua}}$ can be defined:
\[ k_{hua1} = k_{spindle} \tag{8} \]
\[ k_{hua2} = \left( k_{tb}^{-1} + k_{spindle}^{-1} \right)^{-1} \tag{9} \]
\[ k_{hua3} = \left( k_{cent}^{-1} + k_{tb}^{-1} + k_{spindle}^{-1} \right)^{-1} \tag{10} \]

The value of \( k_{hua} \) is based upon the torque level in the series of springs:

\[
k_{hua} = \begin{cases} 
  k_{hua1}, & \text{if } |\delta_{sh,in} - \delta_{sh,out}i_{sh}| \geq \frac{\Delta_{act} k_{cent}}{k_{hua3}} + \frac{\Delta_{act} k_{tb}}{k_{hua2}} \\
  k_{hua2}, & \text{if } |\delta_{sh,in} - \delta_{sh,out}i_{sh}| \geq \frac{\Delta_{act} k_{cent}}{k_{hua3}} \\
  k_{hua3}, & \text{otherwise}
\end{cases} \tag{11} \]

### 2.3 Hydraulically Actuated part

This stiffness is connected to the output inertia of the steering house \( J_{sh,out} \), which is the lumped inertia of the spindle, equivalent piston and sector shaft, via the ratio \( i_{sh} \):

\[
i_{sh} = \frac{R_{pa} L_{spindle}}{L_{spindle}} \tag{12}\]

where \( L_{spindle} \) is the lead of the spindle which connects the input of steering house to the hydraulic piston and \( R_{pa} \) is the radius of the sector shaft connecting the hydraulic piston to the output shaft of the steering house as shown in Figure 3(b).

The seals in the hydraulic cylinder cause a friction force which is translated to a friction torque with magnitude \( T_{fric,ha} \) where \( ha \) stands for Hydraulically Actuated. Since the hydraulic system also functions as a damper for the system (oil is squeezed through narrow channels upon movement of the piston), the assumption is made that the piston acts as a damper with coefficient \( d_{ha} \).

The output is connected to the king pin axis by means of the spring \( k_{hua} \), which represents the lumped stiffness of all components in between the steering house and the king pin axis. A ratio \( i_{pa} \) is defined, which is the kinematic ratio between the pitman arm and drag link.

The inertia of the wheels, hubs and tie-rod is lumped into one parameter, \( J_{kp} \), with angle \( \delta_{kp} \). This inertia shows a static friction torque \( T_{fric,kp} \) and a viscous damping \( d_{kp} \) caused by the needle bearings which support the king pin axis. The wheels produce a torque with magnitude \( T_{kp} \) caused by the tyres and the king pin orientation (Bakker 2012).

Power-steering torque is generated as a result of the torque acting on the torsion bar. Torsion bar torque is defined as:

\[
T_{tb} = (\delta_{sh,in} - \delta_{sh,out}i_{sh}) k_{hua} \quad \text{with} \quad |T_{tb}| \leq k_{tb} \Delta_{tb} \tag{13}\]

An empirical approach to relate the assistive torque to the torsion bar torque is used. The boost curve that models this is describe by parameters \( a, b, c \) and \( d \):

\[
T_{ps} = \text{sign}(T_{tb}) \left( a(e^{b|T_{tb}|} - 1) + c(e^{d|T_{tb}|} - 1) \right) \tag{14}\]

This definition ensures predictable behavior outside the fitting range, zero power-steering torque at zero torsion bar torque and symmetrical behavior for positive and negative torque.

### 2.4 Parameter identification

The model contains a total of 30 parameters, some of these parameters can be found by specifications of the manufacturer, others require additional testing. A total of four additional tests have been performed:
1. The piston of the steering system is blocked in order to assess the properties of the hua part, see Figure 4(a). In this way the stiffness of the spindle and torsion bar, respectively $k_{\text{spindle}}$ and $k_{tb}$, are identified by means of a least squares fit with a second order model. Also the parallel free play element $\Delta_{tb}$ can be identified. An example of the outcome of such a measurement is shown in Figure 5(a) where the change in stiffness is seen.

2. The output of the steering system is blocked by fixating the drag link to the world, see Figure 4(b). This experiment is done without power-steering in order to identify the stiffness of the $ha$ part, $k_{ha}$. An example of this test is shown in Figure 5(b). This figure also shows the reduced stiffness around the center position $k_{\text{cent}}$ as well as the parallel free play $\Delta_{\text{cent}}$.

3. The output of the steering system is free, experiments are done without power-steering to identify the friction elements $T_{\text{fric,hua}}$ and $T_{\text{fric,ha}}$. This test can also be used to check the ratio $i_{sh}$. Figure 6(a) shows an example of the friction identification on the output side of the steering house. The coulomb friction on the piston $T_{\text{fric,ha}}$ is clearly visible.

4. The output of the steering system is blocked by fixating the output shaft to a force sensor and switching the power-steering on. The input and output torques are measured as well as the input angle. This test is used to identify the coefficients $a$, $b$, $c$ and $d$ used in the power-steering model. Figure 6(b) shows the measured boost-curves together with the fits for different flow-rates. In order to estimate the torque acting on the torsion bar a compensation for the friction on the input $T_{\text{fric,hua}}$ is done before fitting.

Since all measurements have been executed at low velocities, limited information regarding the damping is present. Since the steering system manual describes a damping effect of shock forces from the steered wheels and no oscillations have been seen during the measurements, the assumption is made that the hydraulic piston is critically damped, thus $\zeta_{ha} = 1$. In this way kickback at the steering wheel is prevented. The other damping parameters have been chosen at 5\% thus $\zeta_{sw} = \zeta_{hua} = \zeta_{kp} = 0.05$ in order to prevent high frequency oscillations which are outside the scope of this research. This results in the following damping coefficients:

$$d_{sw} = 2\zeta_{sw}\sqrt{J_{sw}k_{sc}}$$

$$d_{hua} = 2\zeta_{hua}\sqrt{J_{sh,in}(k_{sc} + k_{hua})}$$

$$d_{ha} = 2\zeta_{ha}\sqrt{J_{sh,out}(k_{hua}v_{sh}^2 + k_{ha})}$$

$$d_{kp} = 2\zeta_{kp}\sqrt{J_{kp}k_{ha}v_{pa}^2}$$

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\[d_{sw} = 2\zeta_{sw}\sqrt{J_{sw}k_{sc}}\]  \hspace{1cm} (15)

\[d_{hua} = 2\zeta_{hua}\sqrt{J_{sh,in}(k_{sc} + k_{hua})}\]  \hspace{1cm} (16)

\[d_{ha} = 2\zeta_{ha}\sqrt{J_{sh,out}(k_{hua}v_{sh}^2 + k_{ha})}\]  \hspace{1cm} (17)

\[d_{kp} = 2\zeta_{kp}\sqrt{J_{kp}k_{ha}v_{pa}^2}\]  \hspace{1cm} (18)
3 VALIDATION OF THE STEERING SYSTEM MODEL

The steering system model has been validated by means of the test-setup as shown in Figure 4(b). Testing has been performed with and without the drag link attached. In this validation the power-steering will be switched on since the tests without power-steering have already been used for identification purposes. The focus in the validation process will lie on the prediction of the steering wheel torque.

3.1 Drag link fixed, power-steering on

Two situations will be considered, a situation where the system is actuated in the on center position and a situation where larger angles are applied. Figure 7 shows the steering wheel torque over time for small steering wheel angles as well as the steering wheel torque as a function of the steering wheel angle. A comparable hysteresis loop is seen here which is caused by the friction in the system.

Figure 8 shows the results for larger angles. The non linearity of the boost-curve is clearly visible as well as the friction present in the system. A good coherence with the measurements is seen. The strong non linearity when changing direction is present in the model as well which is
3.2 Drag link free, power-steering on

Again the on center situation and the large angle position will be analyzed. Figure 9 shows the results for small angles (on center behavior). The coherence with the measurements is clear but not as good as with the drag link fixed. Some a-symmetry in the measurements is seen which is not present in the model.

Figure 9 shows the response for larger angles. A position dependent friction appears to be present. This is also implemented in the model by means of a look-up table. The cause of this is in the design of the steering house, the free play is minimized in the center position which increases the friction in the system around the center position.
4 CONCLUSIONS AND RECOMMENDATIONS

A steering system model has been developed which includes the inertias, friction and flexibility of components. The eccentric properties of the steering wheel as well as the universal joints have been modeled. An empirical approach has been used to model the assistance torque delivered by the hydraulic system.

The parameters for this model have been estimated by four dedicated tests. One test consists out of blocking of the hydraulic piston to estimate the stiffness on the driver-side of the steering house. In the second test the output of the steering house is freed and the friction in the system is determined. In the third test the drag link is fixed to a force sensor and the total stiffness of the steering house is found. In the fourth test the output of the steering house is fixed to a force sensor and the power-steering is switched on. This is used to find the parameters for the boost-curve model.

The model has been validated using measurements for steady-state conditions by means of two tests, one with the drag link free and one with the drag link fixed. The power-steering is switched on and different quantities such as steering wheel torque, pitman arm angle and drag link force are measured. The model uses the steering wheel angle and the boundary conditions as an input and
the response is compared with the measurements. The steering wheel torque, pitman arm angle and drag link force can be predicted accurately.

In order to improve the steering system model, the hydraulic model can be improved. This can be done by extending the test-setup with pressure sensors inside the steering house. By measuring the pressure directly the pressure dependent friction as well as the valve characteristic can be found. The test-stand can also be improved by mounting a spring instead of a fixed drag link such that the operating condition is comparable to the situation in the vehicle. This spring should have a stiffness comparable to the combined stiffness of the two front tyres in the situation where the vehicle is running straight ahead at high-way speed.

Future work will consist out of improvement of the hydraulic model and implementation in a multi-body vehicle model. Validation will be done by a series of full vehicle tests with an instrumented steering system. Furthermore the possibility of autonomous steering via placement of an extra actuator in the steering system will be investigated with help of this model.

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