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DEVELOPMENT OF A TRUCK STEERING SYSTEM MODEL INCLUDING HYDRAULICS TO PREDICT THE STEERING WHEEL TORQUE

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ABSTRACT: The prediction of steering wheel torque in a truck is a challenging subject due to the number of components connecting the driver to the front wheels. In previous work, a steering-system model has been created which relates the power steering assistance torque to the input torque supplied by the driver via a look-up table. In this work the model will be extended with a more detailed hydraulic model which makes use of a Wheatstone bridge with hydraulic resistances. This makes the model valid for different flow-rates and provides more insight. To find the parameters for the model, a test setup is used to monitor the relevant pressures as well as the flow through the system. The model will be used to reproduce the measurements. It is shown that the model is suitable for the calculation of the steering wheel torque and pressures in the system with a high precision.

Keywords: Commercial vehicle, Steering-system, Hydraulics, Autonomous driving

1 INTRODUCTION

The 1 billion unit mark of vehicles in operation worldwide was surpassed in 2010. In Europe a total of 35 million heavy-duty trucks are registered in 2016. By 2030 this amount is expected to grow by 17 percent. In Asia and the Middle East this growth is expected to be even stronger (ICCT 2013). Accident statistics show that trucks and in particular tractor-semitrailers are over-represented in accident statistics. Poor rear and side visibility is a unique risk factor in truck driving leading to blind spot crashes (Schoon et al. 2008, SWOV 2009). Highway hypnosis and fatigue contributes to loss-of-control accidents and road departures on longer trips (Summala & Mikkola 1994). Another risk factor is the complex dynamics and growing sizes of trucks (Besselink et al. 2015). To improve the safety of trucks systems as anti-lock braking system and electronic stability control have become mandatory on 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 of having fully automated trucks on highways, where the truck operator is no longer a driver but a supervisor. To implement features such as automated steering, an active steering system is required. To design and implement such a system, more knowledge of a conventional truck steering system is required. This knowledge can be used, for example, to determine the position of an extra actuator to make the steering-system active. A challenging design goal in a truck steering system is the on center feel. Commercial vehicles in general make use of a hydraulic assist system which amplifies the torque supplied by the driver using a 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 al. 2008). Steering systems in truck handling simulation models, as 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).
Research on passenger cars shows that the steering-system is often modeled in a relative simple way. Most models have three degrees of freedom with a stiffness between the steering wheel and the assistance motor and a stiffness between the steering rack and the assistance motor (Parmar and Hung 2004, Song et. al 2004). (Lozia and Zardecki 2007) show that a coulomb friction model is sufficient to describe the steering wheel torque during cornering. This is also shown by (Pfeffer et. al 2008), especially around the center position. In (Rösth 2007) a concept is shown with an additional motor on the input side to enable active features like a parking pilot, lane keeping assist, emergency lane assist, active yaw control and torque reference control. He also shows that this motor can be used for friction compensation in the steering system.

Figure 1(b) shows a typical steering system layout in a commercial vehicle. From this figure it becomes apparent that the steering system contains several components connecting the steering wheel to the front wheels. The sprung cabin and limited space around the engine contribute to this complexity. 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. Literature on cars shows that modeling of the hydraulic part of the steering-system is essential to predict the steering-wheel torque and the vehicle response (Pfeffer et al. 2008 and Rösth 2007).

In (Loof et al. 2015), a steering system model with a fixed power-steering mapping of a conventional truck was created to predict the feed-back torque felt by the driver. The amount of power-steering assist was determined via lookup tables which are only valid in a limited range of operation. Literature shows that to model the hydraulic power-steering system, a Wheatstone bridge with a hydraulic cylinder is an often used approach (Amuliu 1998, Birsching 1999 and Dell’Amico 2013).

In this paper the hydraulics of the steering-system are modeled by a Wheatstone-bridge in combination with a hydraulic cylinder (see Figure 2) in order to gain insight in the functionality of the steering system and to make the model valid for different flow-rates.

The goals of this paper are:

- Extend the steering-system model with a hydraulic model.
- Determine the parameters of this hydraulic model.
- Reproduce the measurements using the improved model.

First the steering system model will be discussed in Section 2. The parametrization of this model is discussed in Section 3 and the validation of the model is made in Section 4. The conclusions are presented in Section 5.
2 STEERING SYSTEM MODEL

Figure 2 shows a reduced steering-system model which is an adapted version from (Loof et al. 2015). The driver input torque $T_{sw}$ is supplied at the steering-wheel. At the steering-wheel there is the possibility to implement a frictional torque, $T_{fric,sw}$ to model the bearings. Normally a steering column is present in between the steering house and the steering wheel. In the configuration used at the test bench, this column is replaced with a short shaft and therefore it is assumed that this stiffness is much higher than the other stiffnesses in the chain. At the input of the steering house, a second possibility is present to implement friction, $T_{fric,ha}$. The input of the steering-house is connected to the output by a series of springs. These springs represent an on-center stiffness, $k_{cent}$, to reduce the stiffness around the center position, a torsion-bar spring, $k_{tb}$ which represents the torsion-bar and a spindle spring, $k_{spindle}$ which represents the spindle. A fixed reduction, $i_{sh}$ is used as the steering-house ratio which is derived from the spindle lead and the sector-shaft radius. The mass of the spindle and hydraulic piston are lumped into one parameter $J_{sh, out}$. This inertia is also used to implement the friction, $T_{fric,ha}$ caused by the seals in the hydraulic cylinder. The output of the steering-house, (the pitman arm), is connected to an hydraulic actuator with a spring, $k_{ha}$ to represent all the stiffness of the parts in between the steering-house and the hydraulic actuator. The hydraulic actuator is connected via a kinematic ratio, $i_{pa}$ which is the result of the kinematic relations of the pitman-arm and the drag-link. The power-steering torque $T_{ps}$ is a result of the deflection of the torsion-bar. To calculate the amount of power-steering a Wheatstone bridge is used. Figure (a) shows the inside of the steering-house with the relevant pressures. We distinguish between the supply pressure $P_s$, the pressure in chamber $A$, $P_A$, the pressure in chamber $B$, $P_B$, and the return pressure $P_0$. The flow through the system is indicated with $Q_{bridge}$. Figure (b) shows the flows through the valves, the flows to the chamber are indicated by $Q_{xA}$ and $Q_{xB}$ and the the flow in and out of the system by $Q_{bridge}$ and $Q_0$.

In order to model the system a number of assumptions are made:

1. It can be seen in Figure 3(b) that four unique flow paths are present in the system. The other paths are all similar to one of these four. The assumption is made that the paths can be lumped together into four equivalent flows and flow resistances.

2. Figure 3(b) also shows that opening of the path from $Q_{bridge}$ to $Q_{xA}$ results in an equal opening of the path from $Q_{xB}$ to $Q_0$. The same holds for the flow from $Q_{bridge}$ to $Q_{xB}$ and $Q_{xA}$ to $Q_0$. Therefore symmetry is assumed and two unique flow resistances are defined: $R_1$ and $R_2$.

3. The flow through the orifices is a turbulent flow, the pressure drop and the flow through the orifice can be described by $QR = \sqrt{\Delta P}$ where $\Delta P$ is the pressure drop. The coefficient $R$ stands for the hydraulic resistance and is defined as $R_i = A_i C_d \sqrt{\frac{2}{\rho}}$ where $i$ is the index of the orifice, $A_i$ is the area of the orifice opening, $C_d$ is the discharge coefficient and $\rho$ is the density of the fluid.

4. The area of the orifice is assumed to be much smaller than the area of the channels in the steering-house. No pressure drop is assumed in between the valves and the cylinder chambers $A$ and $B$.

5. The volume in the cylinder chambers $A$ and $B$ is relatively big compared to the volume in the valves. Therefore the fluid in the valves is assumed to be incompressible and the fluid in the cylinder chambers is assumed to be compressible with bulk modulus $\beta$. 

Figure 2.: Steering-system model including Wheatstone bridge
6. Since the pump in the system is not located directly at the steering-house, a hose under high pressure with a considerable length is used. This hose can expand under pressure which is modeled by a hydraulic capacity $C_{hose}$.

7. The pressure at the output of the valve housing is used as a reference pressure, thus $P_0 = 0$.

8. The area of the piston is equal on both sides, this area is referred to as $A$.

Using mass conservation we can relate the in- and out-flow per node as shown in Figure 4. Note that in-flow is defined as negative:

$$\begin{bmatrix}
-Q_s + Q_{hose} + Q_{bridge} \\
Q_1 + Q_2 - Q_{bridge} \\
-Q_1 - Q_2 - Q_x \\
-Q_1 + Q_2 + Q_{bridge}
\end{bmatrix} = 0$$

(1)

Conservation of momentum relates the flows to the pressure drops over the orifices:

$$\begin{bmatrix}
P_s - P_A \\
P_s - P_B \\
P_B \\
P_A
\end{bmatrix} = \begin{bmatrix}
Q_1^2 R_1^2 \\
Q_2^2 R_2^2 \\
Q_1^2 R_1^2 \\
Q_2^2 R_2^2
\end{bmatrix}$$

(2)

Figure 3.: Inside of the steering-house (TRW manual)

(a) Inside of the steering-house (TRW manual)

(b) Overview of the flows through the valves

Figure 4.: Hydraulic model
Due to symmetry a number of equations are redundant. In (1) the second and fifth row, \( Q_1 + Q_2 - Q_{bridge} \), and the third and fourth row, \( Q_1 - Q_2 - Q_x \) are equal. By using the explicit definitions for \( P_A \) and \( P_B \) as given in (2), \( P_s \) can be written as \( P_s = \frac{Q_1^2 R_1^2}{Q_2 R_2} + \frac{Q_2^2 R_2^2}{Q_1 R_1} \). This results in the following set of equations:

\[
\begin{bmatrix}
2Q_1 - Q_{bridge} - Q_x \\
2Q_2 - Q_{bridge} + Q_x \\
Q_s - Q_{bridge} + Q_{hose} \\
P_A \\
P_B \\
P_s
\end{bmatrix}
= \begin{bmatrix}
0 \\
0 \\
0 \\
Q_1^2 R_1^2 \\
Q_2^2 R_2^2 \\
Q_s^2 R_s^2 + Q_2^2 R_2^2
\end{bmatrix}
\tag{3}
\]

The equivalent resistance of the whole bridge is defined as \( R_{bridge} = \frac{1}{2} \sqrt{R_1^2 + R_2^2} \). Using this definition, the explicit expressions for \( Q_1, Q_2 \) and \( Q_{bridge} \) can be written as:

\[
\begin{bmatrix}
Q_1 \\
Q_2 \\
Q_{bridge}
\end{bmatrix}
= \begin{bmatrix}
\frac{\sqrt{P_s}}{R_1} \\
\frac{\sqrt{P_s}}{R_2} \\
\frac{\sqrt{P_s}}{R_{bridge}}
\end{bmatrix}
\tag{4}
\]

The two first entries from (3) are solved for \( Q_x \) and related to the pressures by using the solutions from (4). Since compressibility in the chambers is assumed, the flow to chamber \( A \) is not necessarily equal to the flow from chamber \( B \). Therefore two definitions for \( Q_x \) are used to distinguish between the flow to chamber \( A \) and the flow from chamber \( B \):

\[
Q_x = \begin{cases}
Q_{bridge} - 2Q_2 = \frac{\sqrt{P_s}}{R_{bridge}} - \frac{2 \sqrt{P_s}}{R_s} = Q_x A \\
2Q_1 - Q_{bridge} = \frac{2 \sqrt{P_s}}{R_1} - \frac{\sqrt{P_s}}{R_{bridge}} = Q_x B
\end{cases}
\tag{5}
\]

The pressure derivatives can be calculated by making use of the continuity equation:

\[
\sum Q \frac{dV}{dt} = \frac{V}{\beta} \frac{dP}{dt}
\tag{6}
\]

where \( V \) stands for the volume. We define the volume of the chambers as:

\[
V_A = V_{A0} + Ax \quad \text{and} \quad V_B = V_{B0} - Ax
\tag{7}
\]

with \( V_{A0} \) and \( V_{B0} \) the initial volumes of the chambers, \( A \) the piston surface and \( x \) the piston position. Applying the continuity equation, (6), to \( P_s, P_A \) and \( P_B \) and substitution of (4) and (5) leads to:

\[
\frac{dP_s}{dt} = Q_s - Q_{bridge} \frac{\sqrt{P_s}}{C_{hose}} = Q_s - \frac{\sqrt{P_s}}{C_{hose}} \frac{\sqrt{P_s}}{R_{bridge}}
\tag{8}
\]

\[
\frac{dP_A}{dt} = \left( +Q_x \frac{dV_A}{dt} \right) \frac{\beta}{V_A} = \left( \frac{\sqrt{P_s}}{R_{bridge}} - \frac{2 \sqrt{P_s}}{R_s} - Ax \frac{\beta}{V_{A0} + Ax} \right)
\tag{9}
\]

\[
\frac{dP_B}{dt} = \left( -Q_x \frac{dV_B}{dt} \right) \frac{\beta}{V_B} = \left( \frac{\sqrt{P_s}}{R_{bridge}} - \frac{2 \sqrt{P_s}}{R_1} + Ax \frac{\beta}{V_{B0} - Ax} \right)
\tag{10}
\]

### 3 TEST SETUP

In order to identify the parameters for the model, a test-setup is used as shown in Figure 5. The drag-link is connected to a hydraulic actuator. The actuator force is controlled externally and the force and displacement can be measured. The pitman-arm and steering-wheel angle are measured as well as the steering-wheel torque. The supply pressure, both pressures in the chambers, the return pressure and the flow through the system are monitored. These are indicated in red and listed in Table 1.
4 PARAMETER ESTIMATION

The necessary model parameters are listed in Table 2. All parameters are measured or calculated by rules of thumb except for the valve surfaces $A_1$ and $A_2$. These will be measured in an indirect way. In order to estimate the valve areas, a staircase like force-profile is applied via the hydro-pulse actuator. The steering-wheel is fixed to the world by means of a mechanical brake. An example is shown in Figure 6. The force, $F_{hp}$, is increased stepwise over the whole operating range of the steering-house.

When the system is in steady-state ($\dot{P} = \ddot{x} = \dddot{x} = Q_{xA} = Q_{xB} = 0$) the flows in the system are given by:

$$Q_{\text{bridge}} = Q_s \quad \text{and} \quad Q_1 = Q_2 = \frac{1}{2} Q_s \quad (11)$$

This results in the following definition of the flow-resistances (due to symmetry there are two solutions for $R_1$ and $R_2$):

$$R_1 = \left\{ \frac{2\sqrt{P_s-P_n}}{2\sqrt{P_n}} \right\} \quad R_2 = \left\{ \frac{2\sqrt{P_s-P_n}}{2\sqrt{P_n}} \right\} \quad R_{\text{bridge}} = \left\{ \frac{\sqrt{P_n}}{Q_s} \right\} \quad R_1^2 + R_2^2 \quad (12)$$

The flow resistances $R_1$ and $R_2$ can be converted to valve openings via:

$$A_i = \left( R_i C_d \frac{\sqrt{\frac{2}{z}}}{\rho} \right)^{-1} \quad (13)$$

Figure 7(a) shows the estimated valve area as a function of the torsion-bar torque. For both $A_1$ and $A_2$ a good overlap between the two methods from (12) is seen for $|T_{sw}| \geq 0.5$ (outside the on-center region). In the on-center region the two methods diverge since the pressure in the chambers approaches zero. It is expected that the assumption of two unique flow-resistances does not hold in this region. Since the system cannot be parametrized without this assumption, because the division of flow within the bridge cannot be measured, it is decided to continue with this assumption and later check if the results given by the model are satisfactory. Figure 7(b) shows the resistance of the bridge calculated by (12). This figure also shows that the bridge resistance calculated via $R_1$ and $R_2$ diverges from the direct calculation via $P_n$. The assumption is made
that for zero torsion-bar torque $R_1 = R_2 = R_0$. The index 0 is given to the samples for which $|T_{tb}| < 0.1$ holds. The average of these samples is used.

$$R_{bridge0} = \frac{1}{2} \sqrt{R_0^2 + R_0^2} \rightarrow R_0 = \frac{2}{\sqrt{2}} \frac{\sqrt{P_{s0}}}{Q_{s0}} \quad (14)$$

With $R_0$ known, the area at $T_{tb} = 0$ can be calculated:

$$A_0 = \left( R_0 C_d \sqrt{\frac{2}{\rho}} \right)^{-1} \quad (15)$$

Outside the on-center region a fit is made on the data. Since it is hard to estimate the geometry of the valve openings, the choice is made to fit a function which is able to represent data well with a low amount of parameters. This function has been determined by making use of the curve-fitting toolbox in MATLAB. A function with two coefficients $a_i$ and $b_i$ is used where $i$ stands for the valve index:

$$A_i = a_i^{-1} |T_{tb}|^{-b_i} + A_0 \quad (16)$$

The points $A_{11}$ and $A_{22}$ are defined as the values of the fit at the edge of the on-center regions. From this point towards the point $A_0$ a linear increase of the valve area is assumed and this linear behavior is also used when the point $A_0$ is passed:

$$A_{ii} = a_i^{-1} |T_{tb,center}|^{-b_i} + A_0 \quad (17)$$

$$A_1 = \begin{cases} a_1^{-1} |T_{tb}|^{-b_1} + A_0 & \text{for } T_{tb} \leq -T_{tb,center} \\ A_0 + (A_0 - A_{11}) T_{tb} & \text{for } T_{tb} > -T_{tb,center} \end{cases} \quad (18)$$

$$A_2 = \begin{cases} a_2^{-1} |T_{tb}|^{-b_2} + A_0 & \text{for } T_{tb} \geq T_{tb,center} \\ A_0 - (A_0 - A_{22}) T_{tb} & \text{for } T_{tb} < T_{tb,center} \end{cases} \quad (19)$$

Due to the seals in between the piston and the cylinder friction is present in the system. It is expected that this friction depends on the pressure in the system (Suisse, 2005). To check this
the friction is plotted versus the pressure difference $\Delta P$ in Figure 8. A strong dependency on the pressure difference is seen and this is approximated with a linear function. Literature shows that the friction coefficient of the seal drops with an increasing pressure. However, since the pressure increases faster, this results in a friction force increasing with the pressure.

5 VALIDATION

The validation process is splitted into two parts, a steady-state part and a time domain simulation part. For the steady-state validation equations (8) until (10) are solved for $P_s$, $P_A$ and $P_B$ under the assumption that all time derivatives are zero. This is shown in Figure 9. Visual inspection of this plot shows that the model is able to represent the measurements very well.

To review the model response in the time domain, measurement data is used as an input for the simulation. The steering wheel angle is fixed and the hydro-pulse force and flow rate are used as an input. The simulation output is compared with the measurement in Figure 10. The simulated steering wheel torque agrees very well with the measurements and thus has a high accuracy. The supply pressure and the chamber pressures are slightly under predicted and the pressure difference shows a good match with the measurement.
6 CONCLUSIONS

A steering-system model has been extended with a model of the hydraulic power steering module and is parametrized by means of measurements with different flow rates. The measurement data has been used as an input for the model to check the validity of the model. The results show that the model is able to predict the steering wheel torque with a high accuracy. The supply pressure and chamber pressures are slightly under-predicted by the model, but the pressure difference across the piston is obtained with a high accuracy. This model provides more insight in the functionality of the steering-system and is valid for a broader range of conditions in comparison to only using a boost curve look-up table.
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