Electric power steering
Recreating steering feel

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MASTER OF SCIENCE THESIS

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Abstract

Application of electronics in the automotive industry is extensively spreading. Electric power steering is hereby an exciting improvement to steering systems since the introduction of hydraulic power steering systems, which was first introduced as early as 1970s. Drivers desire hydraulic power steering behaviour because of its natural feel while providing assistance, but it has many disadvantages such as component lifecycle, oil leaks and power consumption. The use of electronic power steering provides the opportunity to remove such problems associated to hydraulic power steering and introduce benefits in terms of adaptability, comfort and safety. Since today’s vehicles are making use of the by-wire technology for most components in the car, the additional electronic power steering will make it possible to have a full range of automated driving aids.

Even though electric power steering comes with many advantages, there are still some disadvantages which are mentioned in numerous articles. These problems are mostly described as a lack in steering feel. More precisely it’s described as if the front wheels are disconnected from the ground wheels. Many studies show different algorithms to minimise this, however it can be seen that even in the most successful products there is some complaint about this.

An important tool to analyse this problem is the use of driving simulators. The implementation of a steering model that maintains high fidelity dynamics will bring many advantages. At the moment steering systems are developed by vehicle testing which is considered expensive and time consuming. The study of steering models will give more insight for the development in steering systems and the implementation in a HIL-simulator will ease the process of steering system development.

This thesis will show the implementation of a steering model in a high fidelity steering simulator and it will propose a different method for solving the problem of steering feel loss. Finally it will conclude with driver in the loop experiments to show and evaluate the proposed method.
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T. Sezer
“Knowledge is limited. Imagination encircles the world.”

— Albert Einstein
Steering feel is important feedback for the driver during driving, it plays a big role for the feeling of safety and comfort. Apart from that it’s also something that contributes to the brand character, which separates the vehicle from other vehicles. Car manufacturers have large teams of engineers designing steering systems to achieve a desired steering feel. During this process several handling and steering experts are consulted to support the development of the steering system [1]. These specialists rely mostly on their subjective insights and experience.

Some recent studies regarding steering feel show that there is research on correlating subjective evaluations with objective assessments.[2, 3, 4, 5, 6]. This is important because it will first give a better understanding in the field of steering feel but more importantly it will help us understand the influence of the steering system design on the steering feel. By knowing the influence of steering system design on steering feel it will be possible to make realistic models of steering models for driving simulators, this will allow developing steering systems much more efficiently.

The use of driving simulators in steering system development is very useful, this is due to the adaptability of software and the repeatability of experiments. However creating high fidelity feedback is still difficult, due to both software and hardware limitations. This thesis will show a proposal to solve the steering feel problem and it will show the implementation of a steering model in a driving simulator to evaluate the proposed method.
1-1 Problem statement

As mentioned in the introduction, the use of electric power steering brought with it the steering feel problem. This can be experienced as if the steering wheel is disconnected from the wheels at the ground. But this problem manifests itself in different ways, because different car manufacturers use different hardware and different software to minimise this problem. As an example the quote below is taken from an article in a car magazine.

“There is a slight numb feeling when dead ahead. A lot of the tracking that the 997 felt on overly cambered and pot-holed roads is removed.” — *Total911 – The Porsche Magazine*

Another article from a car magazine shows a comparison between a hydraulic system and an electronic system for the same kind of car. To clarify the problem the the following paragraph from the article should help.

“For any feedback to travel from the road to the driver’s hands, EPS’s spinning electric motor must be slowed, stopped, and/or reversed. This is where the electronic controller comes into play. An array of chassis sensors—monitoring velocity, yaw rate, lateral acceleration, and other dynamic variables—keeps the controller apprised so it can issue appropriate commands to the electric-assist motor. Measuring what’s going on with the tires and the suspension indirectly amounts to intelligent guesswork, but it’s the best we have today to minimise EPS’s limitations. No wonder we found hydraulic better than electric assist in our subjective Feedback analysis. For EPS-assisted rack-and-pinion to be a genuine success before it is eventually replaced by steer-by-wire systems, it needs to strive to match what we consider the ultimate achievement in this car-design discipline: The Lotus Elise and Exige provided superb feel and feedback without pumps, motors, fluids, magnets, or artificial assists of any kind.” — *Car and Driver – Are We Losing Touch? A Comprehensive Comparison Test of Electric and Hydraulic Steering Assist*

As a result the problem statement is phrased as:

“Re-create and improve steering feel by a driving simulator study”

In order to have a better understanding of the described steering problem, Section 1-2 gives an overview of electric power steering systems. Section 1-3 shows a more detailed problem description with the problem causes. Section 1-4 describes the objectives and the approach taken.
1-2 Steering system

Assisted steering systems can be divided into three category’s, hydraulic power steering, electric power steering and steer-by-wire systems. Hydraulic power steering systems are becoming less common, while electric power steering systems are becoming more popular and steer-by-wire systems are making an entrance. Electric power steering is first introduced in the 80’s, but it’s has become more mainstream in the past ten years. This is due to the increasing demands for lower energy consumption, driving comfort and driver assistance.

Electric power steering will make it possible to have different kind of assistance systems like lane-assist and stability control but also full autonomous driving will be possible with electric power steering.

This chapter will describe the different kinds of electric power steering systems present with their advantages and disadvantages and the second subsection will describe the possible model structures.

![Figure 1-1: Four main EPS concepts](image)

1-2-1 Overview of physical steering systems

The basic functioning of electric power steering is by measuring the driver torque needed using a torsion bar. Then according to the driver intention assist torque is provided. There are four basic electronic power steering concepts that are established in the market which can be seen in Figure 1-1. The choice is based on available installation space and the required steering rack force. This section will describe these four main concepts [8]. The working principle for all systems is similar, to sum up the systems consists of:

- A torsion bar or different kind of sensing unit is used for detection of driver input torque
- A servo unit and ECU for generating the assist torque required by the driver
- A gearbox to amplify the servo torque
• An intermediate shaft with joints to make the required connections
• A rack and pinion gear for conversion of rotational motion to translational motion
• Necessary bearings to support the steering system

**EPS-Column**

The EPS-Column version is used for compact cars and mid-size cars. Here the servo unit is mounted inside the vehicle, this has advantages, like the unit doesn’t have to be water-tight and the vehicle temperature inside the vehicle is less than in the engine compartment. Having lower ambient temperature will require less from the unit casing and this will have an effect on the cost.

The motor is connected to the steering column via a worm gearbox, the worm is fixed to the motor shaft and the worm gear is connected to the steering column. Because the force is transmitted via the steering column, intermediate steering shaft and steering pinion, there are limits to the maximum achievable steering forces, that’s why it’s more suited for compact and mid-size cars.

**EPS-Pinion**

The EPS-Pinion version is used for mid-size cars and upper mid-size cars. The servo unit is mounted to steering pinion. As for the column electric power steering unit, the pinion version also makes use of a worm gearbox. Compared to the column EPS larger forces can be transmitted, because the force doesn’t have to be transmitted via the steering column and the intermediate steering shaft.

Another variant of this system is a dual pinion version, here the servo unit is fitted to a second pinion. This makes it possible to have a different drive pinion ratio than the steering ratio, which means that some performance optimization can be done. Another advantage is that the servo unit can be placed parallel to the steering rack, this makes the total unit compact and installation can be done in difficult areas.

**EPS-Parallel drive**

The EPS-parallel drive is used for the luxury class vehicles. The servo unit is placed directly parallel onto the rack, this means that again the total unit will be compact. Force transfer is done via a combination of ball screw and toothed belt, therefore this system is characterised by low friction and high efficiency. The motor is connected to the ball screw using the toothed belt, so the ball screw converts the rotational movement of the motor into a translational movement of the rack.
1-2-2 Operating principle

The operating principle of all these systems are similar and there are different modes to control the system. Sometimes switching between these modes is needed, for example after taking a turn, grip is loosened on the steering wheel and return mode can be activated to help smooth return of the steering wheel to the center position. Apart from the most common needed crucial modes which are described below, also other modes can be found in the literature to improve the steering system it’s performance, an example can be seen in Figure 1-2. These additional modes are mostly motor related to compensate for the undesired motor dynamics.

![Figure 1-2: EPS-functions [9]](image)

**Assist mode**  Boost curves based on tire friction curves are designed in order to reduce steering effort. These boost curves are then used to provide assist based on the vehicle velocity and the drivers torque, a diagram of this can be seen in Figure 1-3.

![Figure 1-3: Standard assist structure [10]](image)

The most common assist characteristics are the straight-line type, broken-line type and curve-line type[11]. The straight line is considered the most simple one, because the main parameter is the slope of the curve. This characteristic is widely used in practice. The broken-line is similar to the straight-line type, but it consists of more straight line, which has an advantage
in the low driving torque region. The curve-line assist performs also better than the straight-line assist and it doesn’t have the sharp change characteristics. So the desired assist comes from the look-up tables based on the vehicle state and drivers torque, the control structure looks similar too Figure 1-3.

He and Gu [11] show the design for a curve-line assist curve, the basis curve can be seen in Figure 1-4a. Different assist characteristics can be obtained by adjusting the five key parameters shown in this figure, where $K_1$ is the end horizontal coordinate of zone 1, and it’s also the begin coordinate of zone 2, $K_2$ is the width of zone 2, $g_3$ is the slope of zone 3, $k_4$ is the begin coordinate of zone 4 and it’s also the begin coordinate of zone 4, $k_5$ is the width of zone 5.

The 5 zones can be divided as follows:

1. Assistance dead zone, $|T_D| < K_1$: the given steering torque doesn’t demand assistance, this zone consists of just a straight line.
2. Small assistance zone, $K_1 < |T_D| < K_1 + K_2$, the steering torque requires demands small assistance, the curve type is a concave parabolic-line.
3. High assistance zone, $K_1 + K_2 < T_D < K_4$, a steering torque in this region will require more assistance, the curve type is a straight line.
4. Assistance buffer zone, $K_4 < |T_D| < K_4 + K_5$: this segment approaches the maximum assistance torque available, the curve type of this segment is convex parabolic-line.
5. Assistance saturation zone, $|T_D| > K_4 + K_5$, maximum assistance has been reached, the curve type is a straight line.

The assist curves transform simply through linear scaling depending on the vehicle velocity as can be seen in Figure 1-4b. Depending on the desired vehicle characteristics this scaling is done.
**Damping mode**  Damping mode in literature is given as a method damp undesired fast movements and vibrations. This also adds some of the viscous feeling that’s present in the hydraulic power steering systems. Mostly this is implemented by using the relation below, $T_{damp}$ is calculated by multiplying the steering wheel velocity $\dot{\phi}_{sw}$ with a damping coefficient $c_{damp}$ which is dependent on the vehicle velocity.

$$T_{damp} = c_{damp}(v_{veh})\dot{\phi}_{sw}$$

**Return mode**  Return mode is used in order to improve the returning response of the steering wheel which is caused by the reactive force from the tires. The need for this mode comes from the fact that in some cases the damping mode dampens this movement. Or another problem might be that small reaction forces might not reach the steering wheel because it gets damped out at the motor or the gears in between, this is also related to the steering feel problem described in the problem statement.

This problem is solved by using cascaded control, which can be seen in Figure 1-5. The steering wheel angle $\phi_{sw}$ is controlled by the cascaded control, where the steering wheel velocity $\dot{\phi}_{sw}$ is the inner loop. In order to reduce overshoot in case of a large position error $e_{sw}$, the position controller gain $k_{pos}$ is adjusted based on the position error $e_{sw}$.

At vehicle speeds higher steering wheel velocity’s are expected. An additional gain $x_{veh}$ is added to additional tune the return velocity, based on safety and comfort.

To handle hands off situation or loose grip situations, the steering wheel velocity $\dot{\phi}_{sw}$ is multiplied by a factor $x_\tau$ which is depended on how strong the driver holds the steering wheel.

Based on the position and and the factors described above, the position loop gives a desired velocity $\phi_{r,mod}$. This is compared to the actual steering wheel velocity $\dot{\phi}_{sw}$, the error is then multiplied with a speed gain $k_{speed}$ to obtain the intermediate return torque $T_{ar,tmp}$. The intermediate return torque $T_{ar,tmp}$ is then multiplied with a gain which is dependent on the steering torque in order to add room for some more tuning.

**Inertia and friction compensation mode**  This mode is there to compensate for inertia and friction that is caused by the assist motor assembly. This is needed because for high-way driving, when no assist is needed, the system should operate as a mechanical system.
Figure 1-5: Structure of active return module [12, 13]
1-3 Problems related to EPS systems and steering feel

Adding any assist system comes with it’s consequences. whether it’s hydraulic power steering or electric power steering. The task of the assist system is to reduce the effort, not to remove. This is done by adding assist torque while providing the driver with relevant amount of road feedback, however this isn’t always the case.

The addition of components changes steering dynamic and this might effect the steering feel in a negative way. For example the hydraulic power steering system can filter out some of the feedback coming from the tires. But still the steering wheel is connected to the wheels directly with a mechanical connection and the assist system works in parallel with the steering system. This connection is done by the torsion bar and up to a certain torque level the system works as a mechanically connected system. After a certain torsion level assist pressure is allowed to pass. This makes it possible to still pass through small inputs and also to receive feedback from the road. In order to make sure that no steering feel is sacrificed, the design of hydraulic power steering and it’s influence are studied thoroughly over the years[14]. That’s why hydraulic power steering is still preferred over electric power steering by race drivers or driving enthusiasts.

However for electric power steering these problems still remain even for the most advanced vehicles as can be read from the quote in the problem statement. This is both due to the mechanical design in the system and the operating principle which is described above. Below are the main causes of the problem.

Operating principle
From the point of operating principle it can be said that it’s very difficult to control a system like this. Because many functions that come naturally with a mechanical steering system or a hydraulically power steering system are solved by using different modes which get activated based on the state. And all these different modes require the tuning of many parameters as can be seen from the ‘Return mode’ description.

Gear connections
From the point of mechanical design it can be said that it’s difficult for small reaction forces to pass through to the steering wheel. This is because these forces are most likely damped out at the motor and it’s gear connection. Especially for small driver inputs where no assist is needed this is the case.

Wear
Another problem is the increase in wear over time which will effect the steering dynamics. The rack pinion is designed in such a way that no gear lash is present during wear[15]. This is done by pre-loading the rack spring and giving the gears a certain meshing, therefore during wear the gears always remain in contact. However friction characteristics and mechanical efficiency’s will most certainly change and this will effect the steering performance over time.

This thesis proposes a method to solve the steering feel problem. Which maintains high fidelity steering dynamics by means of a control structure.
1-4 Research objectives

To tackle with the steering feel issue as described in the previous sections, the following objectives are set:

- **Design and implementation of a high fidelity steering system**: A steering model is needed for generating steering feedback and it’s also needed for steering feel analysis. Therefore a steering model should be obtained.

- **Development of control structure to improve steering feel**: The problem can be simply described as the loss of steering feel. A control system to solve the steering feel problem should be designed.

- **Evaluation of the proposed control system based on driving simulator study**: Evaluation of both the designed steering model and the designed control system should be done. This will be done on a driving simulator because it has advantages like experiment adaptability and repeatability.
This chapter will discuss both the vehicle and steering model used in this study. The vehicle simulation software used will be described together with the method used to implement a separate steering system into the vehicle model. Steering system parametrisation and verification will be shown. Then a steering model analysis in terms of steering feel will be done. This analysis will also show the path to a control approach that will solve the steering feel problem. Finally, the control method used will be shown.

![Vehicle model diagram CarSim](image)

**Figure 2-1**: Vehicle model diagram CarSim
2-1 CarSim setup

CarSim is a commercial software package that simulates vehicle response to driver control inputs. The driver input controls in CarSim are the same as for a real vehicle, namely steering, brakes, clutch and shifting. These simulations are done in a predefined environment with road geometry, coefficient of friction, wind, etc. This gives the possibility to simulate vehicle tests like on a real test track.

The vehicle model used in CarSim is a multi-body model with a total of 27 degrees of freedom. Different vehicle classes are present in CarSim. These different vehicle types come with their own parameters, these can be adjusted to study the change in vehicle behaviour. Different control systems like traction control or ABS can be added and this can than be used for control algorithm development.

The vehicle class used for this study is the B-Class passenger car. This is a front wheel driven hatchback vehicle which can be compared to vehicles like the Audi A3. Section 1-2-1 mentioned that the system’s in mid-size cars have a lower mechanical efficiency when it’s compared to the systems’s in the luxury class vehicles. CarSim gives the possibility to change component parameters however it doesn’t allow access to the model it self. This means that in order to implement an external steering model, the CarSim steering system should be overruled. This can be achieved by setting the steering input to ‘Open loop steer control’. All steering dynamics will be disregarded and the specified steering wheel angle will be set, according to which the ground wheels will move directly with the steering wheel.

![CarSim steering system implementation](image)

Figure 2-2: CarSim steering system implementation

In order to use an external steering system, the rack displacement should be recalculated to steering wheel angle. This then should be supplied as input to CarSim and the desired rack displacement will be fed to the vehicle wheels. The steering system needs two torque inputs, one is the driver’s torque which comes from the torque sensor, the other one is the kingpin torque which comes from the vehicle model. Therefore the CarSim import/export channels should be set as in Table.2-1. The output ‘V. S1’ is the vehicle velocity, this is used for the assist characteristics.

The kingpin torque’s for each wheel is different because for a certain rack displacement, the angles for the left and right wheel are not the same. This is due to steering geometry. The CarSim software provides a look up table for both wheels, the look up table will provide the corresponding wheel angle for each rack displacement.
The ratio at which the kingpin torque’s are transformed to the steering system are also different because of the different ratio’s. Kingpin torques act on the steering system according to the rack to wheel displacement ratio, therefore these have to be recalculated accordingly.

<table>
<thead>
<tr>
<th>Import</th>
<th>Export</th>
</tr>
</thead>
<tbody>
<tr>
<td>IMP_STEER_SW</td>
<td>Mz_Kp_L1</td>
</tr>
<tr>
<td></td>
<td>Mz_Kp_R1</td>
</tr>
<tr>
<td></td>
<td>V_S1</td>
</tr>
</tbody>
</table>

Table 2-1: Carsim Import/Export channels for steering system implementation

2-2 Steering system model

In order to analyse the steering system a mathematical model should be developed. Modelling is done by developing a model for each component of the steering such as the servo motor, column, gears, rack, etc. Newton and Euler equations are used for describing each of the component dynamics and kinematics respectively as a function of time.

In literature the complexity of models is diverse, while some study’s make use of simple lumped models, others make use of complex models in which every component is modelled. However there is only a limited number of publications[16] which investigates the influence of complexity on steering dynamics.

Since steering models aren’t openly published due to intellectual property policies it’s not possible to find a verified steering model. Therefore this study parametrises a model using multi-body simulations.

2-2-1 CarSim steering model

The CarSim steering model’s description can be found in the ‘Steering system’ component section, the parameters used can be seen in Table 2-2. It can be seen that the steering system consists of two masses, with one of them being the entire upper column assembly and the other part is everything below the torsion bar. This means that everything below the torsion bar such as linkages, joints, pinion and rack are lumped into a single mass.

Note that the damping for the rack is given by 'Front steering damping’, since the rack is lumped into 'System inertia’ this damping will act on everything below the torsion bar. The terms ‘Column hysteresis’ and ‘Hysteresis angle’ are friction parameters, ‘Column hysteresis’ is the maximum friction amplitude and ‘Hysteresis angle’ decides on how fast this amplitude is reached. With the use of this description a free-body diagram is constructed as in Figure 2-3.
Using the Euler-Lagrange method the following result is obtained for the equations:

$$J_{\text{col}} \ddot{\theta}_{\text{col}} = T_{\text{driver}} - B_{\text{col}} \dot{\theta}_{\text{col}} - T_{\text{upfric}}(\beta) - T_{\text{bar}} \eta_{\text{col}}$$

$$T_{\text{bar}} = K_{tb}(\theta_{\text{col}} - \theta_{\text{sys}})$$

$$J_{\text{sys}} \ddot{\theta}_{\text{sys}} = T_{\text{bar}} \eta_{\text{rack}} - B_{\text{sys}} \dot{\theta}_{\text{sys}} - T_{\text{Kingpin}}$$

Note that additional parameters $\eta_{\text{col}}$ and $\eta_{\text{rack}}$ are added, which are respectively, the upper column mechanical efficiency and the rack mechanical efficiency. The upper column mechanical efficiency is related to the gear connection from the assist motor in the upper column, the rack mechanical efficiency is related to the rack pinion assembly. These parameters will further be used in Section 2-4.

**Figure 2-3:** Steering system diagram

<table>
<thead>
<tr>
<th>Component</th>
<th>Symbol</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Column inertia</td>
<td>$J_{\text{col}}$</td>
<td>0.1</td>
<td>kg.m$^2$</td>
</tr>
<tr>
<td>System inertia</td>
<td>$J_{\text{sys}}$</td>
<td>0.01</td>
<td>kg.m$^2$</td>
</tr>
<tr>
<td>Column damping</td>
<td>$B_{\text{col}}$</td>
<td>1.43</td>
<td>Nm.s/rad</td>
</tr>
<tr>
<td>Front steering damping</td>
<td>$B_{\text{sys}}$</td>
<td>1.10</td>
<td>Nm.s/rad</td>
</tr>
<tr>
<td>Column hysterisis</td>
<td>$T_{\text{upfric}}$</td>
<td>1.5</td>
<td>Nm</td>
</tr>
<tr>
<td>Hysteresis angle</td>
<td>$\beta$</td>
<td>2</td>
<td>deg</td>
</tr>
<tr>
<td>Torsion bar stiffness</td>
<td>$K_{tb}$</td>
<td>114.60</td>
<td>Nm/rad</td>
</tr>
<tr>
<td>Rack-pinion ratio</td>
<td>$u_{rp}$</td>
<td>0.006</td>
<td>m/rad</td>
</tr>
<tr>
<td>Upper column mechanical efficiency</td>
<td>$\eta_{\text{col}}$</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>Rack mechanical efficiency</td>
<td>$\eta_{\text{rack}}$</td>
<td>1</td>
<td></td>
</tr>
</tbody>
</table>

**Table 2-2:** Steering system parameters
2-3 Verification

2-3-1 Mechanical system

In order to verify that this steering model matches with CarSim, comparative simulations were conducted. The friction parameters for the CarSim model are set to zero, in order to verify the mechanical model first, also the assist curves are excluded for this purpose. CarSim is run with the torque input mode, this way the CarSim internal steering model will be used. The input torque to the steering system is a sine wave at 0.2 Hz that yields to a lateral acceleration of 0.2g is, this test is known is "Weave Test ISO 13674".

In parallel the constructed steering model is run with the same torque input and the the resulting reacting kingpin torque, the result of the comparison can be seen in Figure 2-4. This shows that there is a difference in amplitude of approximately 5 degrees and if looked closely also a difference in phase can be observed, which is caused by additional dynamics in the internal CarSim steering model.

![Figure 2-4: Steering system comparison](image)

In order to find the source of the difference, first the upper column dynamics are checked. This is done by only running the upper column and torsion bar in parallel with the CarSim steering model. Since CarSim gives access to both the steering wheel angle and the rack displacement all inputs are present. The rack displacement is the same as the angle of the 'System inertia' converted with the configured rack pinion ratio, this makes it possible to calculate the torsion bar torque. A diagram of this can be seen in Figure 2-5, the result can be seen in Figure 2-6.

Since the upper column gives a perfect fit, the missing dynamics is present at the part below the torsion bar that is called 'System' in the CarSim steering model. In order to find the missing dynamics an inverse model is constructed with an auxiliary extra torque '$T_x$' caused by the missing dynamics. Again '$\theta_{sys}$' is recalculated from the rack displacement and this is differentiated in order to recalculate the unknown torque as in the equation 4-1.
The calculated auxiliary torque can be seen in the upper plot of Figure 2-7. Further inspection shows that this torque profile scales linearly with the vehicle velocity. This means that this unknown torque is velocity related, checking the velocity outputs of steering related components shows that the rotational velocity of the wheels shows approximately the same period and phase with the unknown torque profile, this can be seen in Figure 2-7. The profiles aren’t exactly the same, this means that there is still something else acting on the system, however it can be seen that these two profiles contribute the most.

\[ T_x = T_{Kingpin} + B_{sys} \dot{\theta}_{sys} - J_{sys} \ddot{\theta}_{sys} - T_{bar} \]  \hspace{1cm} (2-4)

An optimization script with the 'lsqnonlin' function of Matlab\textsuperscript{®} has been implemented, such that the error between the output of the CarSim steering model and the constructed model is minimised. The unknowns are the two damping factors that are used for these velocity profiles, which are expected to be the same. The damping factors are 0.0701 and 0.060, since the optimization script finds the best fit possible the difference is accepted. For further simulations the average of the sum of both is taken as the damping factors. These are then
set in the constructed model together with the two new inputs, which are the two angular wheel velocity’s, the result can be seen in Figure 2-8.

![Wheel dynamics comparison](image)

**Figure 2-7**: Wheel dynamics comparison
Figure 2-8: Wheel dynamics comparison
2-3-2 Friction model

Two parameters are given for the steering system friction in CarSim, these are 'Column hysteresis' and 'Column damping'. The description from the CarSim manual is as follows, the parameter 'Column hysteresis' is the maximum friction torque amplitude. The parameter 'Hysteresis angle' is 1/3 of the amount of degrees needed to travel through 95% of the change in torque.

An exact model of the dynamic friction inside CarSim could not be reconstructed, because the mathematical description can not be deduced from the description. Therefore the choice has been made to use a $\tanh(x)$ approximation. The result can be seen in Figure 2-9, this is the friction during a direction change. No verification plot of the total steering dynamics with friction is included in this report.

This is because for verification the two steering systems should be run in parallel, but the resulting kingpin torque is according to the CarSim steering model output. And since the constructed steering model output friction characteristics are different, instability is caused. This can also be understood by looking at Figure 2-9 at time of 2 seconds, in the constructed model there is 0 Nm friction torque however in the CarSim model there is approximately 1.5 Nm of friction torque.

For the rest of this study the maximum friction amplitude is reduced to 0.4 Nm. This is because many studies show that for the on-centre handling a maximum amplitude of ±3-4 Nm is reached. Therefore the 0.4 Nm amplitude is set such that a torque of ±3.5 Nm amplitude is reached.

![Friction Comparison](image_url)

**Figure 2-9:** Friction comparison
2-4  Steering feel analysis

Steering feel is an important aspect during the design of the steering system it’s regarded as core element of vehicle on-centre handling. With the use of EPS systems there is great possibility to tune steering feel. The most basic requirements are low steering effort, improved return response and improved on centre feeling. These aspects are tuned by handling and steering experts by making trade-offs during the design process, this is mostly based on experience of the experts. There is no direct approach based to do this, but studies that investigate the relation between subjective and objective evaluations give some insight[4].

Since steering feel is regarded as core element of vehicle handling, there are numerous studies investigate this area[9, 4, 6]. The most common way of doing this is the on-centre handling test. At 100 km/h a steering input is applied such that the vehicle reaches a lateral acceleration of 0.2g with a frequency of 0.2 Hz. This section will explain the objective evaluation method used to quantify steering feel and these methods will be used to analyse the constructed steering system.

2-4-1  Objective test parameters

The objective test parameters are constructed by recording the following data, the steering wheel torque, the steering wheel angle and the lateral acceleration of the vehicle. Using these data the following four plots are made, the steering wheel angle versus the lateral acceleration, the steering wheel torque versus the steering wheel angle, the steering torque versus the lateral acceleration and the steering work gradient versus the lateral acceleration. From these different plots objective parameters are defined, which are described below. Several studies[17] use these parameters as baseline standards. It’s also been verified[6] that these parameters correlate with subjective observations of test drivers.

Steering wheel angle versus lateral acceleration

- **Minimum steering sensitivity**: The steepest slope before 0.1 g, the slope is then inverted and multiplied with 100 to give steering sensitivity in g/100 deg units.

- **Minimum steering sensitivity at 0.1g**: The slope at 0.1 g, the slope is then inverted and multiplied with 100 to give steering sensitivity in g/100 deg units.

- **Steering hysteresis**: Calculated from the area inside the curve between plus and minus 0.1 g, the area is divided by 0.2 g to give hysteresis in units of degrees. This parameter is related to the lag of yaw rate with respect to the steering input, the greater the phase lag is, the greater the hysteresis becomes.

Steering wheel torque versus lateral acceleration

- **Steering torque gradient at 0 g**: The slope at 0 g, this can be seen as a measure for the torque needed for a certain lateral acceleration.
• **Steering torque gradient at 0.1 g**: Similar to the slope at 0 g, however this describes the feel more during the maneuverer instead of the initial response from 0 g, also described as road feel.

• **Steering torque at 0 g**: A measure for the friction in the steering system, but this parameter is also influenced by the viscous damping and vehicle response lag.

• **Steering torque at 0.1 g**: A measure for steering effort.

• **Lateral acceleration at 0 Nm**: A measure the return-ability, this is a measure for the no torque input while having a lateral acceleration.

**Steering wheel torque versus steering wheel angle**

• **Steering stiffness at 0 deg**: The slope at 0 degrees, change in torque per angular displacement.

• **Steering wheel torque at 0 deg**: A measure for the effort and the friction.

**2-5 Objective analysis**

The problems related to steering feel is more commonly known as numb feeling in the on-centre range, which is rooted to the operation principle of the electric power steering system and the additional components that come with it. The other one is the problem of wear. Since this thesis is focused at maintaining a certain predefined steering dynamics it should be able to solve both problems. Because there is no exact data present of the numb steering feel problem, the objective analysis in this thesis will be based on the increase in wear by means of friction and decrease of mechanical efficiency. An analysis is shown for the increase in the upper column friction and the addition of rack friction. Another analysis is shown for the decrease in the mechanical efficiency of the rack and a decrease in mechanical efficiency due to the addition of the motor gear in the upper column.

**2-5-1 Efficiency variation**

The decrease in mechanical efficiency for the rack is realised by multiplying the torque acting from the torsion bar on the ‘System inertia’ with a factor ‘η’. The result can be seen in the figures below and in Table 2-3.

Decrease in efficiency has shown that the input torque has to be increased to achieve the same target lateral acceleration. This explains also the increase in torque levels in the table. It can be seen that the return-ability worsens a little bit while the sensitivity parameters remain similar.
Table 2-3: Objective steering parameters for varying rack efficiency

<table>
<thead>
<tr>
<th></th>
<th>$\eta = 1$</th>
<th>$\eta = 0.9$</th>
<th>$\eta = 0.8$</th>
<th>$\eta = 0.7$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Torque grad. at 0 deg [Nm/deg]</td>
<td>0.16</td>
<td>0.18</td>
<td>0.20</td>
<td>0.23</td>
</tr>
<tr>
<td>Torque at 0 deg [Nm]</td>
<td>1.36</td>
<td>1.39</td>
<td>1.43</td>
<td>1.48</td>
</tr>
<tr>
<td>Torque at 0 g [Nm]</td>
<td>1.65</td>
<td>1.71</td>
<td>1.80</td>
<td>1.92</td>
</tr>
<tr>
<td>Torque at 0.1 g [Nm]</td>
<td>2.88</td>
<td>3.08</td>
<td>3.38</td>
<td>3.72</td>
</tr>
<tr>
<td>Torque grad. at 0 g [Nm/g]</td>
<td>14.11</td>
<td>15.82</td>
<td>17.97</td>
<td>20.71</td>
</tr>
<tr>
<td>Torque grad. at 0.1 g [Nm/g]</td>
<td>10.43</td>
<td>11.85</td>
<td>13.70</td>
<td>15.97</td>
</tr>
<tr>
<td>Lateral acceleration at 0 N [g]</td>
<td>-0.11</td>
<td>-0.10</td>
<td>-0.09</td>
<td>-0.08</td>
</tr>
<tr>
<td>Minimum steering sensitivity [g/100deg]</td>
<td>1.09</td>
<td>1.07</td>
<td>1.06</td>
<td>1.05</td>
</tr>
<tr>
<td>Minimum steering sensitivity at 0.1 g [g/100deg]</td>
<td>1.21</td>
<td>1.20</td>
<td>1.18</td>
<td>1.17</td>
</tr>
<tr>
<td>Steering hysteresis [deg]</td>
<td>0.73</td>
<td>0.73</td>
<td>0.75</td>
<td>0.75</td>
</tr>
</tbody>
</table>

Figure 2-10: Steering torque vs. angle for rack efficiency variation
Figure 2-11: Steering torque vs. lateral acceleration for rack efficiency variation

Figure 2-12: Steering angle vs. lateral acceleration for rack efficiency variation
Similarly the torque from the torsion bar on the upper column can be multiplied with a factor \( \eta \). Physically this can be compared to the fact that some of the feedback doesn’t arrive at the driver due to some reasons as mentioned in Section 1-3. Mostly this can be caused by the addition of the motor gear in the upper column. It can be seen that the effect is basically the inverse to the mechanical efficiency variation of the rack. It is important to note here that again the steering dynamics are completely different compared to the non-deteriorated steering system.

<table>
<thead>
<tr>
<th>( \eta = 1 )</th>
<th>( \eta = 0.9 )</th>
<th>( \eta = 0.8 )</th>
<th>( \eta = 0.7 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Torque grad. at 0 deg [Nm/deg]</td>
<td>0.16</td>
<td>0.14</td>
<td>0.14</td>
</tr>
<tr>
<td>Torque at 0 deg [Nm]</td>
<td>1.36</td>
<td>1.33</td>
<td>1.31</td>
</tr>
<tr>
<td>Torque at 0.1 g [Nm]</td>
<td>1.65</td>
<td>1.59</td>
<td>1.54</td>
</tr>
<tr>
<td>Torque at 0.1 g [Nm]</td>
<td>2.88</td>
<td>2.67</td>
<td>2.48</td>
</tr>
<tr>
<td>Torque grad. at 0 g [Nm/g]</td>
<td>14.11</td>
<td>12.55</td>
<td>11.00</td>
</tr>
<tr>
<td>Torque grad. at 0.1 g [Nm/g]</td>
<td>10.43</td>
<td>9.07</td>
<td>7.77</td>
</tr>
<tr>
<td>Lateral acceleration at 0 N [g]</td>
<td>-0.11</td>
<td>-0.11</td>
<td>-0.12</td>
</tr>
<tr>
<td>Minimum steering sensitivity [g/100deg]</td>
<td>1.09</td>
<td>1.09</td>
<td>1.09</td>
</tr>
<tr>
<td>Minimum steering sensitivity at 0.1 g [g/100deg]</td>
<td>1.21</td>
<td>1.21</td>
<td>1.21</td>
</tr>
<tr>
<td>Steering hysteresis [deg]</td>
<td>0.73</td>
<td>0.73</td>
<td>0.74</td>
</tr>
</tbody>
</table>

Table 2-4: Objective steering parameters for varying upper column efficiency

![Steering wheel torque vs. Angle](image)

Figure 2-13: Steering torque vs. angle for upper column efficiency variation
Figure 2-14: Steering torque vs. lateral acceleration for upper column efficiency variation

Figure 2-15: Steering angle vs. lateral acceleration for upper column efficiency variation
2-5-2 Friction variation

Two cases are analysed here, the increase in upper column friction and the addition of rack friction, note that in the original model there is no rack friction present, everything is lumped in the rack friction in this case. Varying friction for both the upper column and for the rack give similar results. Again the effect is similar to the mechanical efficiency degradation, the overall torque levels and gradients increase while the return-ability worsens.

<table>
<thead>
<tr>
<th>Torque grad. at 0 deg [Nm/deg]</th>
<th>0.4 Nm</th>
<th>0.6 Nm</th>
<th>0.8 Nm</th>
<th>1 Nm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Torque at 0 deg [Nm]</td>
<td>1.36</td>
<td>1.59</td>
<td>1.83</td>
<td>2.04</td>
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<tr>
<td>Torque at 0 g [Nm]</td>
<td>1.65</td>
<td>1.88</td>
<td>2.13</td>
<td>2.35</td>
</tr>
<tr>
<td>Torque at 0.1 g [Nm]</td>
<td>2.88</td>
<td>3.08</td>
<td>3.30</td>
<td>3.50</td>
</tr>
<tr>
<td>Torque grad. at 0 g [Nm/g]</td>
<td>14.11</td>
<td>13.76</td>
<td>13.43</td>
<td>13.13</td>
</tr>
<tr>
<td>Torque grad. at 0.1 g [Nm/g]</td>
<td>10.43</td>
<td>10.14</td>
<td>9.93</td>
<td>9.63</td>
</tr>
<tr>
<td>Lateral acceleration at 0 N [g]</td>
<td>-0.11</td>
<td>-0.12</td>
<td>-0.14</td>
<td>-0.11</td>
</tr>
<tr>
<td>Minimum steering sensitivity [g/100deg]</td>
<td>1.09</td>
<td>1.10</td>
<td>1.11</td>
<td>1.11</td>
</tr>
<tr>
<td>Minimum steering sensitivity at 0.1 g [g/100deg]</td>
<td>1.21</td>
<td>1.22</td>
<td>1.22</td>
<td>1.23</td>
</tr>
<tr>
<td>Steering hysteresis [deg]</td>
<td>0.73</td>
<td>0.75</td>
<td>0.78</td>
<td>0.82</td>
</tr>
</tbody>
</table>

Table 2-5: Objective steering parameters for varying upper column friction

Figure 2-16: Steering torque vs. angle for upper column friction variation
2-5 Objective analysis

**Figure 2-17:** Steering torque vs. lateral acceleration for upper column friction variation

**Figure 2-18:** Steering angle vs. lateral acceleration for upper column friction variation
The results for the addition of rack friction are as follows:

<table>
<thead>
<tr>
<th>Torque grad. at 0 deg [Nm/deg]</th>
<th>0 Nm</th>
<th>0.2 Nm</th>
<th>0.4 Nm</th>
<th>0.6 Nm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Torque at 0 deg [Nm]</td>
<td>1.36</td>
<td>1.56</td>
<td>1.79</td>
<td>2.10</td>
</tr>
<tr>
<td>Torque at 0 g [Nm]</td>
<td>1.65</td>
<td>1.87</td>
<td>2.12</td>
<td>2.35</td>
</tr>
<tr>
<td>Torque at 0.1 g [Nm]</td>
<td>2.88</td>
<td>3.07</td>
<td>3.29</td>
<td>3.51</td>
</tr>
<tr>
<td>Torque grad. at 0 g [Nm/g]</td>
<td>14.11</td>
<td>13.77</td>
<td>13.44</td>
<td>13.15</td>
</tr>
<tr>
<td>Torque grad. at 0.1 g [Nm/g]</td>
<td>10.43</td>
<td>10.08</td>
<td>9.87</td>
<td>9.61</td>
</tr>
<tr>
<td>Lateral acceleration at 0 N [g]</td>
<td>-0.11</td>
<td>-0.12</td>
<td>-0.14</td>
<td>-0.15</td>
</tr>
<tr>
<td>Minimum steering sensitivity [g/100deg]</td>
<td>1.09</td>
<td>1.09</td>
<td>1.11</td>
<td>1.12</td>
</tr>
<tr>
<td>Minimum steering sensitivity at 0.1 g [g/100deg]</td>
<td>1.21</td>
<td>1.22</td>
<td>1.22</td>
<td>1.23</td>
</tr>
<tr>
<td>Steering hysteresis [deg]</td>
<td>0.73</td>
<td>0.79</td>
<td>0.86</td>
<td>0.94</td>
</tr>
</tbody>
</table>

Table 2-6: Objective steering parameters for varying rack friction

Figure 2-19: Steering torque vs. angle for rack friction variation
Figure 2-20: Steering torque vs. lateral acceleration for rack friction variation

Figure 2-21: Steering angle vs. lateral acceleration for rack friction variation
2-5-3 Discussion

The analysis have shown that both increasing the friction and decreasing the mechanical efficiency results in an overall increase in torque levels. This is as expected, because more torque has to be supplied to achieve the same target lateral acceleration. Changes in mechanical efficiency show that the torque gradients also transform, which isn’t the case for changes in friction. So depending on the sort of degradation, the steering feel will change differently.

Figure 2-22 shows the variation of mechanical efficiency for constant input, so the target of lateral acceleration is disregarded. Here it can be seen that with a decrease of mechanical efficiency, the same lateral acceleration can’t be achieved because the steering wheel angle and respectively the rack displacement decreases. Therefore this problem can be reformulated to achieve a certain reference displacement that will guarantee some predefined steering dynamics, the next section will propose a control structure to solve this problem.

![Figure 2-22: Efficiency variation with constant input](image)
2-6 Control strategy

As mentioned in the discussion of the previous section, the problem can be solved by achieving a certain rack displacement which will give the desired steering dynamics. To recap, the desired steering feel can’t be achieved due to two reasons: one of them is the electric assist motor with the gear connection and its operating principle and the other problem is wear. This section will describe the control structure and the hardware needed to solve this problem.

2-6-1 Control structure

In order to have a reference behaviour, real-time simulations have to be done to provide a reference. Since steering systems have two inputs, the driver input torque and the reactive torque from the tires, these two inputs should be available for the reference model. Torque sensors are already present in modern cars, however the reactive torque from the tire isn’t commonly available. Recently sensors like load sensing bearings[18] or tie rods with load cells[19] are being developed for vehicles.

Another piece of hardware is required to read the rack position, which can be either directly from the assist motor or with the use of a steering rack encoder. This is needed to see if the steering rack is moving as desired. This will make it possible to make sure the steering system is operating as desired from the part below the torque sensor to the steering rack.

The control strategy used is model following control, a linear PI controller is used, which has been successfully utilized for position control purposes. The control scheme in Figure 2-23 shows the control structure to handle this problem. It can be seen that the steering torque and the reactive tire torque are fed to the steering model, the difference with the actual model is calculated and the controller provides a torque to correct for this.

![Figure 2-23: Proposed control scheme](image)

In order to design the controller a simplified plant model is needed. As mentioned the available force sensors are the driver input torque sensor and the sensor that provides the tire reactive force. Therefore the model used will be the part between the torque sensor and the vehicle wheels. The model used for control design has a spring element which represents the tire friction force. Since most steering feel characteristics are determined during the sine wave test with a vehicle velocity at 100 km/h, the stiffness of the spring element is determined...
from the reactive torque and the wheel angle during a sinusoidal manoeuvre of 0.2 Hz lateral acceleration.

\[ J_{sys} \ddot{\theta}_{sys} = T_{bar} - B_{sys} \dot{\theta}_{sys} - K_{tire} \theta_{sys} \]

\[ \frac{\theta_{sys}}{T_{bar}} = \frac{1}{J_{sys}s^2 + B_{sys}s + K_{tire}} \]

After substitution of the parameter values, the transfer function is as below. It can be seen that the system has two left half plane poles, which means that the system is stable.

\[ \frac{\theta_{sys}}{T_{bar}} = \frac{100}{(s + 38.01)(s + 105.2)} \]

Figure 2-25 shows the open-loop frequency response of the system. It can be seen that there is no 0 dB crossing. Using only a feedback gain will make it possible to control the system and in order to have reference tracking integral control is added. This means that PI control will be sufficient for control. The PI controller is tuned such that the first pole of the model is cancelled by a zero and the gain is adjusted such that a phase margin of approximately 60 degrees is achieved, the resulting controller can be seen below.

\[ C_{PI} = 70 \frac{s + 38.01}{s} \]

Figure 2-26 shows the frequency response for the resulting controlled system, it can be seen that there is a bandwidth of approximately 58 rad/sec for the desired phase margin. The controller is designed using an approximate linear tire force reaction at 100 km/h. However the controller should also perform under different velocities. In order to show that the controller also performs under different conditions Figure 2-27 shows the frequency response for the same manoeuvre at 50 km/h. It can be seen that the bandwidth is approximately the same, only the phase margin has decreased, but stability still maintains. For lower velocities the tire stiffness coefficient used for plant design will not be any different, therefore that the assist characteristics will be able compensate for such variations and the controller will be valid.

Figure 2-24 shows a comparison between the controlled and the non-controlled system. The system is deteriorated by increasing friction by 5% and by decreasing mechanical efficiency by 5%. The reference is provided by the desired steering model and the controller shows successful tracking.
Figure 2-24: Comparison of the non-controlled system with the controlled system

Figure 2-25: Frequency response of the plant
Figure 2-26: Frequency response of the controlled plant

Figure 2-27: Frequency response of the controlled plant
To evaluate the performance of both the performance of the steering system and the designed control system a driving simulator will be used. Driving simulators have been used in the automotive industry first by Volkswagen (Figure 3-1) in the seventies[20]. The use varies from driver education [21] to research and development. Using simulators for research and development comes with the advantage of adaptability, this makes it a great tool for vehicle development. This means that things like experiment repeatability are possible because experiment conditions can be set accordingly[22], some factors can be excluded which might not be possible in field testing.

Steering feedback is an important aspect when it comes to driving simulators. All of the driving simulators surveys[21, 23] come to the conclusion that with an increasing number of motion cues people are able to control the virtual vehicle better. Another study[24] shows specifically that having realistic steering feedback can partially compensate for the lack of motion cues in a fixed-base simulator.

This chapter will present the driving simulator and the experiments together with the issues faced during the implementation of the steering system.

Figure 3-1: First VW driving simulator
3-1 Driving simulator

The driving simulator used is the TUDelft moving base simulator. This is a 6 DOF hydraulically controlled stewart platform. On top of the platform a cockpit from a driving is mounted. This initially was a standalone system with its own software, however now this standalone system is connected to a dSpace computer which is connected to a desktop computer on which the vehicle model runs.

The dSpace computer consists of two processing cards (DS1003) and it’s connected to the environment using an Analog to Digital (DS2002) and a digital to analog (DS2102) card.

A central modelling engine performs mathematical processing of the vehicle model which in turn provides the desired output parameters. The modelling engine makes use of a dSpace extension package which makes it possible to link the dSpace elements to the vehicle model. This package is therefore used both for vehicle dynamics analysis and real time simulation.

3-2 Steering simulator

The steering actuator used is a direct shaft connected Phase Motion Ultract II AC motor, the specifications can be seen in Table 3-1. This motor is a high performance servo motor that can be used for various things, from high precision position based uses to velocity control based uses. The motor control platform used is an AxV, this makes it possible either to use the motor for position based control or velocity based control.

<table>
<thead>
<tr>
<th>Table 3-1: Motor specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated speed [rpm]</td>
</tr>
<tr>
<td>Rated torque [Nm]</td>
</tr>
<tr>
<td>Rated current [A]</td>
</tr>
<tr>
<td>Rated power [kW]</td>
</tr>
<tr>
<td>Max. Torque [Nm]</td>
</tr>
<tr>
<td>Max Current [A]</td>
</tr>
</tbody>
</table>
For steering simulation velocity based control is used, because a reference velocity corresponds to a reference acceleration and this corresponds to a torque, so actually torque control is achieved using this method. The control loop inside the AxV controller offers a bandwidth of 16 kHz, which is sufficient for control performance.

To control the motor two sensors are used, this is the steering wheel position sensor and the torque sensor. The steering wheel position sensor is basically an encoder on the motor that gives the motor position which is logically also the steering wheel position.

In order to measure the driver’s input torque strain gauges are attached onto the shaft in a way such that a Wheatstone bridge is formed. With the deformation on the shaft, the resistance of the wires changes, this then causes a voltage difference on the constructed Wheatstone bridge. This voltage difference correlates linearly to the torque applied on the shaft.

This described the steering actuator architecture, the next sections shows how steering feedback is implemented. Also it shows some of the problems faced using this simulator and the things done to minimise these problems.

### 3-2-1 Sensors

The steering wheel angle encoder comes with the motor and this is a a SINCOS 5 channel encoder (2 absolute analogue tracks, 2 incremental analogue tracks, index track). For the encoder no calibration was needed but the torque sensor however needed calibration. This was done by attaching a hook and a weight measuring unit to the steering wheel, the result can be seen in Figure 3-4, the table below shows the final result with the conversion to 'Nm' units. Another important signal used to control the motor is the motor control voltage, depending on the voltage a reference velocity for the motor is set.
3-2-2 Steering simulator control

The steering actuator is controlled using a PID controller which works to track the reference velocity. As an example a spring damper realisation can be set in order to understand the principle, below is the equation that represents spring damper dynamics, with inertia ‘J’, stiffness ‘K’ and damping ‘B’.

\[ J\ddot{\theta} = T_{\text{input}} - B\dot{\theta} - K\theta \]

This equation shows both the torque from the driver and the reacting torque caused by the virtual stiffness and damping. The driver input torque comes from the torque sensor and the reacting torques are calculated using the encoder data. By doing this the desired angular acceleration can be calculated and integrating this will give the desired velocity for the motor input.

The only problem with this is the access to the angular velocity since the encoder signal only gives a position. Differentiating this position causes spikes because of noise in the encoder signal and this causes heavy oscillations in the actuator. So to solve this the damping torque is calculated from the desired angular velocity. A diagram of this realisation can be seen in Figure 3-5.
Using the same method the steering system can be implemented to be used on the steering simulator. The steering system used in this study consists of an upper and lower column, as mentioned in Section 2-2 these are called 'Column' and 'System'. In this model the steering wheel is lumped to the upper column and the driver torque acts onto the upper column, this upper column has its own friction and damping characteristics. The upper column is connected to the lower column by a torsion bar which is represented by a stiffness. Since the driver should experience the reaction at the steering wheel, the upper column dynamics should be simulated.

A diagram of the upper column realisation together with the steering system implemented can be seen in Figure 3-6. Again the input is the drivers torque, the torsion bar torque comes from the angle difference of the encoder and the simulated lower column angle. The upper column friction characteristics depend on the desired angular velocity signal and similarly the upper column damping characteristics come from the desired angular velocity. This desired angular velocity is then supplied to the motor controller as a reference velocity.

After implementation the driving simulator performed poor, because there where some self oscillations present and this gave an unpleasant feeling. In order to see the cause of this problem, both the sensor signals and the control signal where inspected, the next section describes the steps taken to resolve this issue.
A periodic disturbance occurs at the moment when the motor is turned on and this will affect all the signals. Figure 4-2 shows the disturbance which is present on all the sensor signals and the control signal. Another problem is that the disturbance magnitude and phase is different on all the signals, which is due to the fact that for each sensor different filters are present. This makes any kind of cancellation of the disturbance impossible.

As said the disturbance occurs when the motor is turned on, this means that the disturbance is motor related. Inverters used for motor control are known to disturb sensors and signals, but this should be a high frequency disturbance, namely in the range of 16 kHz which is the inverter switching frequency. However, the disturbance consists of multiple low frequency sinusoids, the longest period takes approximately 5.5 seconds and the shortest is approximately 0.5 seconds, so this means the lowest frequency disturbance is approximately 0.2 Hz and the highest is approximately 2 Hz. Even though the disturbance frequency is much lower than that of the motor inverter, the disturbance still occurs so somehow the source of the disturbance is still the motor inverter.

Numerous methods have been tried to find and solve the problem of the disturbance, these are described below:

**Splitting common power sources**

The installation scheme can be seen in Appendix A-2, here it can be seen that both the motor inverter and the sensor amplifiers and filters use a common power source. To see if the problem comes from the common power source, these have been separated and
the sensors and the dSpace computer are connected to a further away power source. Even with the power sources are separated a scope connected to any of the sensors show that there is still a disturbance present on the sensor lines, but the disturbance on the dSpace computer channels aren’t present in this case. Connecting either of the sensors to the dSpace computer again cause the disturbance to propagate through all the dSpace channels. This means that either there is still some connection between the sensors and the motor inverter or there is another problem. Since the problem in the installation couldn’t be found, another approach is followed to minimise the effect of the disturbance.

Reprogramming PLC
Eliminating the disturbance is difficult, but with access to the PLC of the AxV controller it’s possible to change both the control signal range and the encoder sensor range, by doing so the effect of the disturbance can be reduced. The AxV PLC can be accessed by using the RS485 port. The RS485 serial interface is a serial communication method like the more commonly used RS232 serial interface. The main difference is that the RS485 interface is less prone to noise, so higher bit rates can be achieved. Another reason for using the RS485 is the ability to use in network topology, which makes it possible to communicate with multiple motor controllers using one line.

To communicate with the AxV drive a RS232 to RS485 converter is used. An important note here is that even though the physical ports look the same, every manufacturer uses a different pinout, so a special cable has to made between the converter and the AxV drive. This interface allowed us to access and use the software from Phase Motion Control, which are Cockpit and GPLC. Cockpit is a program that makes it possible to switch between the different PLC programs. The PLC programs makes it possible to program different outputs with their signal ranges to the AxV drive.

As can be seen in Table.3-4, the encoder signal range has doubled, this means that the disturbance read in angles is twice as small. The difference in the control signal range is
almost a factor 12, which makes a huge difference. Setting lower values for the control signal range isn’t possible, because the GPLC program automatically changes it to 0.25 (rad/sec/v) when it’s set lower. The reason that the default value is set so low is that this motor has a max rpm rating of 3000 rpm, so using such low values for the control signal range would require a very high voltage for the control signal. Since a steering actuator doesn’t require this high velocity, the control signal range can be significantly reduced. This also means that the undesired movement of the motor caused by the disturbance will be much smaller.

<table>
<thead>
<tr>
<th>Sensor range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steering encoder</td>
</tr>
<tr>
<td>Torque sensor</td>
</tr>
</tbody>
</table>

**Table 3-4:** Reprogrammed signal specifications

<table>
<thead>
<tr>
<th>Control signal gain</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motor control signal</td>
</tr>
</tbody>
</table>

**Table 3-5:** Reprogrammed control signal specifications

### 3-2-4 Verification

With the change of the PLC, the setup has to be verified to be sure that the steering actuator is able to perform desired dynamics. This is done by simulating a spring damper system. The parameters chosen are as follows, the inertia ’J’ is chosen 0.1, the damping coefficient ’B’ is chosen 3.85 and the stiffness coefficient ’K’ is 3.4 Nm/rad. The transfer function for the given system is as below, the input here is the input torque and the output is the steering wheel angle.

The experiment is done by providing torque inputs that represent square waves. The actual output is then compared to the simulated output in order to check if they match, the result can be seen in Figure 3-8. This shows almost a perfect fit, however Figure 3-9 shows the a zoomed part and this shows that the disturbance is still present in the plots. From the point of feeling it can be said that all the self oscillations and the unstable behaviour are gone and the actuator can be used for experiments.
**Figure 3-8**: Comparison of simulated output versus the actual output

**Figure 3-9**: Comparison of simulated output versus the actual output (Zoomed)
3-3  Experiment

To evaluate the designed controller an experiment is designed. Evaluation is done using both subjective questionnaires and objective indicators. The experiment procedure and it’s results are presented in this section.

![Test track](image)

**Figure 3-10: Test track**

3-3-1  Experiment design

The experiment is done using a group of 14 male drivers which are either students or employees from TU-Delft. The drivers have an average age of 27.1 years (SD=4.6). The average of the driving years is 6.2 years (SD=5.0). The average skill rate given by the drivers is 4 (SD=1.47), which corresponds to the rating ’regular’.

Three cases are tested during the experiment:

- **1) Ideal model**: This model is the ideal model with standard friction and efficiency.

- **2) Deteriorated model**: In this model friction is added to the lower column, the magnitude of this friction is 0.1 Nm. Additionally the rack efficiency is decreased to 0.8. These numbers have been found by experimenting on the simulator and by recommendations from literature [25].

- **3) Deteriorated model with control**: Again the deteriorated model is used however now the controller is active to achieve the ideal model dynamics.
A track of 3 km with 5 left hand turns and 5 right hand turns is designed that will provide a lateral acceleration of approximately 0.2 g to correspond the on-center range where the steering feel is primarily described. The road is a two lane road where each lane has a width of 3.5 m accordingly to the dutch highway standards. The driver is told to keep the vehicle in the right lane until the end of the experiment. The driver didn’t know the order of the three different cases of the experiment and after each case, the driver was requested to fill in the question form.

**Subjective and objective indicators**

The questionnaire is used to evaluate the drivers subjective observations, it can be found in Appendix A-1. The questions are describe the overall feel, the torque increase, the torque level and the effort, for grading a 7-grade scale is used.

The objective indicators used to evaluate the driving performance, steering effort and driver workload are as follows:

- **Lateral deviation**: Basically it’s a measure to show how well the track is followed by the driver. It’s defined as the standard deviation of recorded distances between a fixed point of the vehicle and the lane boundary.

- **Steering entropy**: A measure for the driver workload. As the test subject drives along, he/she driver continuously assesses the situation ahead and applies smooth and predictable steering control. Smooth here means, making small corrections at a time. Steering entropy is related to the corrections, as more corrective manoeuvres are made predictability decreases. The idea is that the recorded steering angle data is resampled with a lower frequency, then a taylor expansion is used to predict the steering angle at a given time, the difference between the predicted and the actual angle is used to calculated the error. The prediction error distribution is divided into 9 bins, which is common for normal driving, the steering entropy is then calculated using the following formula:

\[
H_p = \sum_{i=1}^{9} -p_iLog(p_i)
\]

- **Steering reversal rate**: A measure for the driver workload. Again related to the corrections made by the driver, used to count the number of steering reversals within certain steering range.

- **Steering wheel steadiness**[26]: A measure for the effort. The percentage of the time that the steering wheel it’s absolute angular velocity is less then 1 deg/s.

- **Root-mean-square Torque**: A measure for the effort. The root-mean-square of the steering torque during the test.
3-3-2 Results

In order to check statistical significance, the ANOVA (analysis of variance) method is used. The significance between the three models is checked. The ANOVA method basically compares the variance between groups with the amount of variation within the groups. Since two different models are compared, the mean is expected to be different, however more important is the difference greater than expected to be caused by chance, so more simple, is there a real difference. A significant difference is considered when the 'p' value is less than 0.05.

The tables, Table.3-7 and Table.3-6 show the processed results. The objective parameters show that the first model and the last controlled model show similar results, this means that the controller is performing as desired. It can be seen that apart from the root-mean-square of the torque, none of the objective parameters are significant. This means that even though the model with increased friction requires more torque, it doesn’t influence the other objective parameters in a negative way. It can be seen that the lateral deviation decreases from the first to the third model. This can be rooted to the fact that the driver’s get more experienced after each test and they are able to improve their own performance, however it can be said that the difference is small.

<table>
<thead>
<tr>
<th>Lateral deviation</th>
<th>Model 1</th>
<th>Model 2</th>
<th>Model 3</th>
<th>Significance p</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steering entropy</td>
<td>0.30</td>
<td>0.28</td>
<td>0.24</td>
<td>0.44</td>
</tr>
<tr>
<td>Steering reversal rate</td>
<td>12.41</td>
<td>13.78</td>
<td>13.57</td>
<td>0.79</td>
</tr>
<tr>
<td>Steering wheel steadiness</td>
<td>15.62</td>
<td>15.69</td>
<td>15.14</td>
<td>0.94</td>
</tr>
<tr>
<td>RMS Torque</td>
<td>0.72</td>
<td>0.94</td>
<td>0.70</td>
<td><strong>0.001</strong></td>
</tr>
</tbody>
</table>

Table 3-6: Objective steering parameters for varying efficiency

<table>
<thead>
<tr>
<th>Overall feel</th>
<th>Model 1</th>
<th>Model 2</th>
<th>Model 3</th>
<th>ANOVA p</th>
<th>Wilcoxon p</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steering torque increase</td>
<td>4.1</td>
<td>5.1</td>
<td>4.6</td>
<td>0.33</td>
<td>0.36</td>
</tr>
<tr>
<td>Effort</td>
<td>4.4</td>
<td>4.9</td>
<td>4.3</td>
<td>0.28</td>
<td>0.72</td>
</tr>
<tr>
<td>Vehicle Response</td>
<td>3.4</td>
<td>3.9</td>
<td>4.5</td>
<td>0.25</td>
<td><strong>0.04</strong></td>
</tr>
</tbody>
</table>

Table 3-7: Subjective steering parameters for varying efficiency

The subjective ratings show that again the first and the third model are rated similarly for effort and steering torque increase. This means that the controller is performing as desired, reducing the effort caused by the friction increase and mechanical efficiency degradation. The ratings for the overall feeling and the vehicle response are rated high for the third model. The fact that is rated higher than the deteriorated model is expected, however it’s also rated higher than the first model. This is unexpected because the objective parameters show similar performance and this is because the controller is there to give the same steering dynamics as the first model.

To be sure of the effect, the statistical significance is checked. As mentioned the ANOVA method compares the three models. This shows that only the overall feel is considered statistically significant. To understand why the third model is rated higher than the first model,
the Wilcoxon rank sum test is used. This test is used when only two groups are compared. Here it can be seen that also the vehicle response can be considered significant and it’s not by chance that it’s higher rated.

The reason that it’s rated higher can be again due to the fact that the factor of driver experience plays a role here. After each test the driver get’s more acquainted with the simulator and subjectively it’s considered more pleasant. Another reason might be that the addition of the PI-controller introduces some overshoot, which is then experienced by a more responsive steering system. As mentioned, there is this problem of an disturbance in the simulator hardware. It might also be the case that the effect of the disturbance is reduced because of the introduced controller. However the exact reason remains unexplained. Apart from that it can be said that the controller is doing it’s job and it’s able to improve steering feel for a deteriorated system by maintaining desired steering dynamics, which has been verified by both the objective and subjective parameters.

![Objective ratings](image-url)

**Figure 3-11:** Objective ratings
Figure 3-12: Objective ratings

Figure 3-13: Subjective ratings
Steering feel is an important aspect of vehicle dynamics. It provides direct road feedback to a driver and therefore it plays a significant role for the feeling of safety and comfort. Apart from that it also contributes to the brand character of a vehicle. Electric power steering is becoming a mainstream of steering assist systems in passenger cars. However it brings the problem of artificial steering feel, which is mostly described as a disconnected road feel. As mentioned in Section 1-3, the assist steering system is there to reduce effort without completely removing of road feel. This section also explained the cause of the problem as follows. The main issue is the artificial steering feel due to the additional inertia of electric motor and coupling mechanisms, as well as the influence of the control mode of the electric motor. The problem statement of this thesis was therefore formulated as:

“Re-create and improve steering feel by a driving simulator study”

To achieve a thesis goal, the following objectives were formulated and respective actions were taken:

**Design and implementation of a high fidelity steering system**

- Developed a steering model providing high fidelity steering dynamics including components like steering wheel, upper column, torsion bar, rack-pinion and others with respective compliances in between;
- Verified the developed steering model by comparison to the CarSim model;
- Analysed steering components to match unknown parameters including friction amplitude;
- Analysed variation of friction and mechanical efficiency and their influence on steering feel.
Figure 4-1 shows the variation of mechanical efficiency with respect to the given steering task. It can be seen that the degradation of mechanical efficiency leads to the reduction of the steering wheel angle. This indicates that the problem can be solved by monitoring of the displacement of steering rack. Based on this fact, the second objective was formulated as:

**Development of control structure to improve steering feel**

- Developed a reference steering model regarding the “ideal” steering feel (e.g. without degradation of steering system properties like mechanical efficiency) and real-time requirements;
- Proposed feedback controller using PI strategy to guarantee the desired rack displacement according to the reference steering model (Section 2-6);
- Identified sensory requirements for the proposed control system, which requires the addition of two more sensors for measurement of rack displacement and rack force.

The advantage of the proposed control system is the elimination of use of multiply control modes to solve above mentioned problems. In this way the proposed system is able to perform according to the desired steering dynamics. However, the proposed solution needs to be investigated in details, therefore the third objective was formulated as:
Evaluation of the proposed control system based on driving simulator study

- Implemented the steering model and the proposed controller on real-time environment of driving simulator while taking steering actuator dynamics into account;

The steering actuator in this study had the issue of a certain disturbance on the system resulting in steering wheel oscillation. To minimise its influence, the signal ranges in the PLC of the steering actuator control unit were adjusted to improve signal-to-noise ratio and remove these oscillations (Figure 4-2):

![Disturbance on the encoder signal](image)

**Figure 4-2:** Disturbance on the encoder signal

- Designed a driving experiment to investigate on-centre handling performance by using three different cases: i) ideal, ii) deteriorated and iii) deteriorated with proposed control;

The hypothesis of the designed experiment is that the subjective evaluation of the deteriorated model with the proposed control will be similar to the ideal model without degradation of steering system properties.

- Carried out driver-in-the-loop experiments to collect and analyse objective and subjective criteria.

The statistical analysis of objective criteria indicates that i) and iii) cases perform similarly. The analysis of subjective indicators also demonstrates that both cases are rated similarly from the point of effort and torque. However the overall feel and vehicle response of case iii) is rated unexpectedly much higher than case i). This might be due to overshoot introduced by PI controller providing more responsive system dynamics, as a result, considering as more satisfactory drive. Another explanation might be that drivers got acquainted to the driving simulator and they compared the driving case relative to the previous one.
A-1 Questionnaire form

Age:
Driving experience in years:
How would you rate your driving skills: Incompetent - 1 2 3 4 5 6 7 - Expert

Overall steering feel: How was the overall steering feel?

| Poor | Normal | Excellent |

Steering Torque Increase: How is the increase in torque

| Low | Normal | High |

Steering effort: How is the steering effort?

| Low | Normal | High |

Vehicle response: How does the vehicle respond to steering input?

| Slow | Normal | Fast |
A-2 Steering actuator wiring scheme


Master of Science Thesis  T. Sezer


