DEGREE PROJECT, IN VEHICLE ENGINEERING , FIRST LEVEL
STOCKHOLM, SWEDEN 2015

Exploration of steering feel

VLADIMIR CHEVATCO

KTH ROYAL INSTITUTE OF TECHNOLOGY

Abstract

In this thesis, the concept of steering feel, as experienced by the driver, is explored. First a literature review is conducted to highlight previous work on this topic. From this review, the Norman on-centre test and steering wheel torque are identified as important concepts, and are chosen to form the base of this thesis. Following this, steering system and tyre construction are described, and a single-track mathematical model of a car and its tyres is illustrated. Those models are then implemented in Simulink and are used to simulate the Norman on-centre test and explore the effects of vehicle mass, steering ratio and power-steering servo curves on steering wheel torque. Without power steering, vehicle mass and steering ratio are identified as having the largest effect on the steering torque. With power steering added to the model, it becomes the dominating factor in shaping the steering wheel torque, and it is concluded that future research in this area is likely to focus on power-steering and steer-by-wire effects.

Sammanfattning

I denna studie kommer begreppet styrkänsla att utvärderas. Styrkänsla är samlingsnamnet för relationerna mellan rattvinkel, rattmoment och bilens dynamiska egenskaper. Tillsammans ger de föraren information om hur bilen reagerar. Litteraturstudie görs för att belysa tidigare arbete i detta område. Utifrån litteraturstudien kan Norman ”on-centre” test och rattmoment identifieras som viktiga begrepp och fokus i denna studie kommer därför att ligga på dem. En matematisk beskrivning av styrsystemet, däckegenskaper och bilens dynamik ges. En Simulinkmodel byggs upp för att simulera effekten av bilens massa, tröghetsmoment, styrutväxling och servostyrning på rattmomentet. Det visar sig att med servostyrning frånkopplad är bilens massa och styrutväxlingen mest betydelsefulla för styrkänslan. När servostyrningen adderas har servokurvan störst effekt på bilens rattmomentsuppbyggnad. Eftersom alla bilar som säljs idag har servostyrning så kommer framtida diskussioner av styrkänsla troligtvis fokusera på servostyrning och ”steer-by-wire”-system och deras effekt på styrkänslan.

Nomenclature

T_{0g} [Nm]: Steering wheel torque at 0 g

G_{0g} [Nm/g]: Steering torque gradient at 0 g

$T_{0.1g}$ [Nm]: Steering wheel torque at 0.1 g

$G_{0.1g}$ [Nm/g]: Steering gradient at 0.1 g

$a_{y,0Nm}$ [g]: Lateral acceleration at 0 Nm

h_{st} [-]: Steering hysteresis

α : Slip angle

β : Side-slip angle

σ_x : Longitudinal slip ratio

σ_y : Lateral slip ratio

r_{eff} : Effective wheel radius

ω_w : Rotational velocity of the wheel

V_x , v : Longitudinal velocity of vehicle COG

V_y : Lateral velocity of vehicle COG

C_α : Cornering stiffness

C_σ : Tyre stiffness

δ : Steer angle

$\dot{\psi}$: Yaw rate

m : Vehicle mass

θ : Vehicle yaw moment of inertia

M_{sa} : Self-aligning torque

t_p : Pneumatic trail

F_y : Lateral tyre force

F_x : Longitudinal tyre force

F_z : Vertical tyre force

l_v, l_h : Distance from COG to front axle and rear axle, respectively

COG: Centre of gravity

Contents

1. Introduction

1.1 Background

In 2013, Alfa Romeo launched the 4C, a small, light sports car that did not have a power steering system. The stated reason for this exclusion was to save weight and increase steering feel. Steering feel is a subjective term. It can be described as the relationships between the steering wheel angle, steering wheel torque and the vehicles dynamic response such as lateral acceleration and yaw rate. These quantities provide the driver with feedback regarding how the vehicle behaves. Predictable and subjectively good steering response is therefore important for the safe operation of a vehicle and for the well-being of the driver and passengers, in addition to being an important yet often overlooked factor in the market success of a vehicle. It is a part of the overall impression a potential buyer forms of a vehicle while test driving. A vehicle that is hard to drive will not sell well. The need to characterise a vehicles steering feel in objective terms is therefore present.

1.2 Aim and limitations

The steering feel of a vehicle differs from manufacturer to manufacturer, from model to model and even between individual vehicles. This makes providing precise numerical results difficult. In addition, the vehicle dynamic model used here is simple, and, while sufficient for the purposes of this thesis, the numerical results obtained will not necessarily be applicable to any real life vehicles. Therefore, this thesis aims to provide a *qualitative* understanding of what steering feel is, why it matters, how it can be measured, and what vehicle parameters affect it the most.

1.3 Method

The method for accomplishing the stated goals is twofold. First, a literature review is conducted to obtain a theoretical understanding of steering feel, describe how it is measured, and explain the various vehicle subsystems that affect it. The results are summarised in chapters 2 and 3. Then, using the mathematical and computer models provided in chapters 4 and 5, a simple sinusoidal driving manoeuvre is simulated. The role of different vehicle properties on the steering feel is investigated by varying those parameters and observing the effect on the simulated steering wheel torque.

The thesis has been supervised by associate professor Lars Drugge at the division of vehicle dynamics at KTH.

2. Steering "feel" measurement and earlier work

While driving a vehicle, the driver has two main sources of feedback: the first is visual feedback which includes what the driver sees, like the position of the vehicle on the road, distance to other vehicles and other visual stimuli, like signal and brake lights. The other source is haptic feedback. This is the feel of the vehicle through forces, torques and vibrations transferred to the driver via the chassis and the steering wheel. The focus of this thesis is steering characteristics, so the need to define steering feedback arises.

The concept of steering itself is well defined; for ground vehicles, which normally cannot perform pure lateral motion, it is the ability of the driver to point the vehicle in the right direction correctly, that is, control vehicle heading. Steering "feel" on the other hand, is a more elusive concept. In casual contexts, words like "precise", "slow", "heavy" or "loose" are used to describe how the vehicle feels to operate. The desire is to allow the driver to position the vehicle on the road with the greatest amount of precision for the lowest amount of effort, which is why a need to relate subjective opinions to objective metrics arose. Physical quantities that can be measured on the steering wheel include steering torque, steering wheel angle and steering wheel vibrations. As the movement of the steering wheel is limited to rotation, steering torque is the quantity of interest in this study. A way to characterize steering feel is therefore to relate steering torque, defined as the amount of torque the driver exerts on the wheel, to various other parameters, such as steering angle, vehicle yaw rate and lateral acceleration. Those relations, in turn, can shed light on what defines good steering.

First, the tests and parameters vehicle manufacturers investigate when designing the steering system are described. These can be broadly split into two categories. "On-centre" steering denotes the steering regime where the steering wheel angle is small and the inputs slow. This is characteristic of a typical highway situation, where lane-keeping and lane-changing are the primary manoeuvres. "Off-centre" steering denotes the regime where steering input is larger. Here steering response is assumed to be linear, and the driver gets more feedback from the vehicle dynamic feel than the steering wheel. As "off-centre" tests include much of the vehicle dynamics, it is hard to isolate torque feedback mechanisms to the driver. Much of the research therefore is focused on either "on-centre" tests, or steady state cornering [2]. The most referred article in the context of "on-centre" steering is a 1984 technical paper by K.D. Norman, Objective Evaluation Method for On-centre Steering [5]. Norman proposes a standard test to evaluate steering feel.

The test procedure is as follows: a vehicle is accelerated up to highway speeds, typically 100 km/h. A low frequency sinusoidal steering input is then applied, with a typical frequency of 0.2 Hz and an amplitude sufficient to cause 0.2 g of lateral acceleration. The steering angle itself varies with respect to steering sensitivity, but is typically around 20 degrees. Note that the author chose to define lateral acceleration as the vehicles' yaw rate multiplied by its speed. This is not fully equivalent to the "true" lateral acceleration of the vehicle, due to the fact that true lateral acceleration calculation needs to take into account the roll behaviour. This practice however, is followed by other tests [3], so the results are consistent. Norman then plots three relations to illustrate the behaviour and extract useful quantities from:

- Steering wheel angle versus lateral acceleration (Figure 1).
- Steering wheel torque versus lateral acceleration (Figure 2).

- Steering wheel torque versus steering wheel angle (Figure 3).

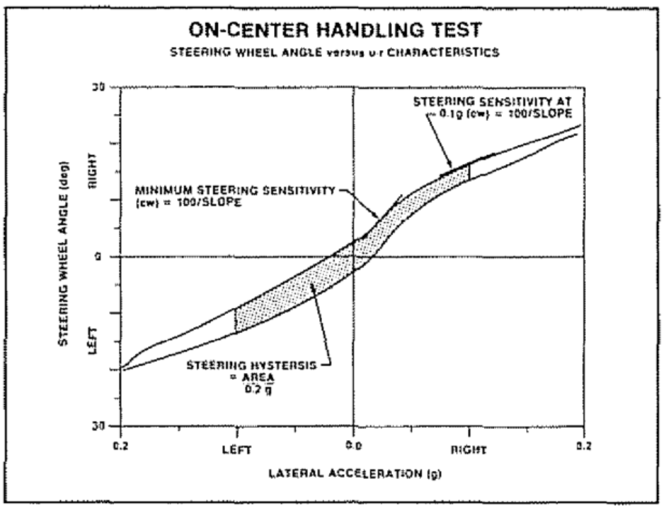


Figure 1: Steering angle vs lateral acceleration [5].

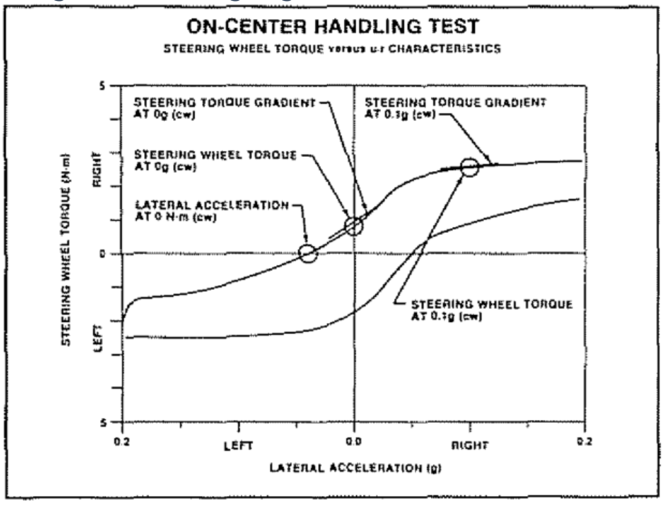


Figure 2: Steering wheel torque vs lateral acceleration [5].

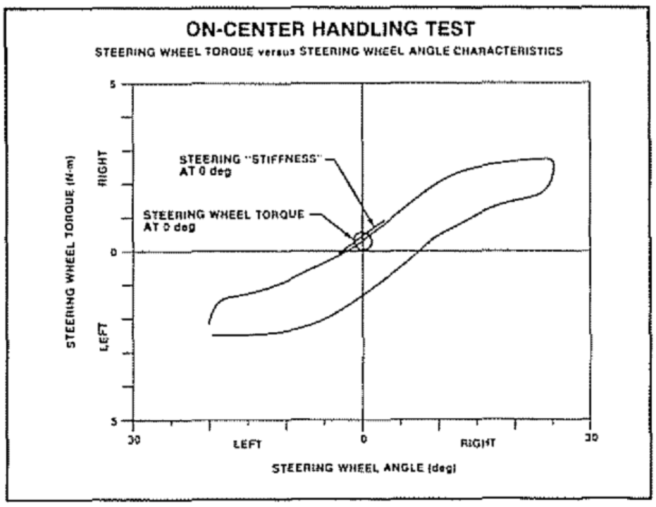


Figure 3: Steering wheel torque vs steering wheel angle [5].

From these plots, several key parameters are identified:

T_{0g} [Nm]: Steering wheel torque at 0g – an indication of friction in the steering system.

G_{0g} [Nm/g]: Steering torque gradient at 0g – change in torque vs change in lateral acceleration, related to “road feel” and directional sense.

$T_{0.1g}$ [Nm]: Steering wheel torque at 0.1g – a measure of steering effort.

$G_{0.1g}$ [Nm/g]: Steering gradient at 0.1g – related to road feel just off the straight ahead direction.

$a_{y,0Nm}$ [g]: Lateral acceleration at 0 Nm – an indication of returnability.

h_{st} [-]: Steering hysteresis – related to the time delay between steer input and yaw rate.

As steering “feel” is subjective in nature, there is much variation between vehicle manufacturers when it comes to properties like steering torque. A concise numerical presentation of the results is therefore difficult. Norman presents typical values by vehicle class. The values for a mid-sized, front-wheel drive, power-steered foreign car (Domestic in Norman’s case means US-made cars, so “foreign” in this context means European or Asian) are presented in Table 1 [5]. Higuchi and Sakai [3] performed the same test on 51 vehicles, a mix of European and Japanese cars. As their work concerned itself with deriving mathematical descriptions of the results, they do not present clear numerical values for these quantities. However, these can be read manually from the graphs they present, and are extracted and presented in Table 1. These values differ slightly from Normans, but the differences can easily be explained by time (Norman did his test in the 80s, Higuchi and Sakai in the 2000s) and choice of cars, due to the large variation between car manufacturers.

Table 1: On-center steering results

	T_{0g}	G_{0g}	$T_{0.1g}$	$G_{0.1g}$
Norman	0.9	20	2.43	7.9
Higuchi and Sakai	1.6	20.8	2.84	10.2

In addition to on-centre tests, results for steady state cornering tests relating steering wheel torque and torque gradient to vehicle velocity have been obtained from a related study. Data for steady state cornering as shown in Figures 4 and 5 shows a wide spread in results [2].

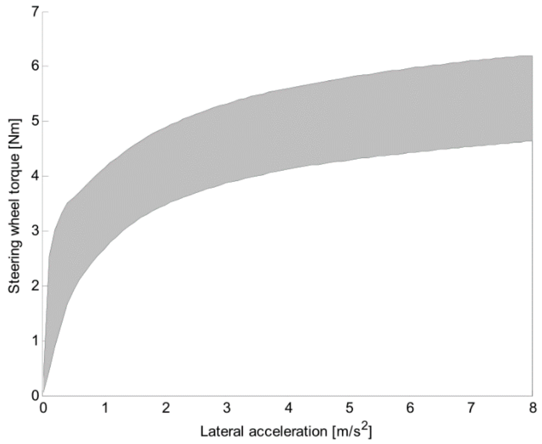


Figure 4: Measured sports car steering-torque range during steady-state cornering [2].

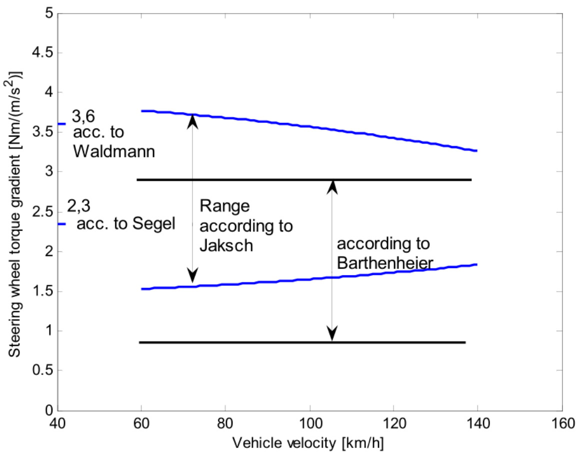


Figure 5: Range of steering wheel torque gradient vs velocity [2].

Given the wide spread of torque and torque gradients, it can be expected that simulated on-centre tests will also show similar results, at least with respect to shape and order of magnitude.

3. Steering system components and construction

With the relevant relationships between torque and steering angle identified in the previous chapter, the need arises to relate them to vehicle component systems, such as suspension, steering system geometry and tyre selection. A simple steering system model is used to explore the effects of different components on steering torque. The simplest model is a rigid (no suspension) Ackermann steering geometry, such as the one shown in Figure 6 [1]:

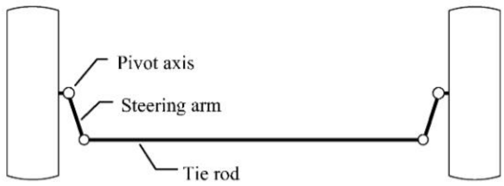


Figure 6: A basic Ackermann steering setup [1].

Here, wheel torque is transferred directly to the steering wheel via the tie rod geometry and a basic rack-and-pinion system (not shown). The torque around the pivot axis is what the driver needs to control in order to steer the wheels. A more realistic model that may be used in a modern car includes more subsystems and components, and an example of this can be seen in Figure 7 [6].

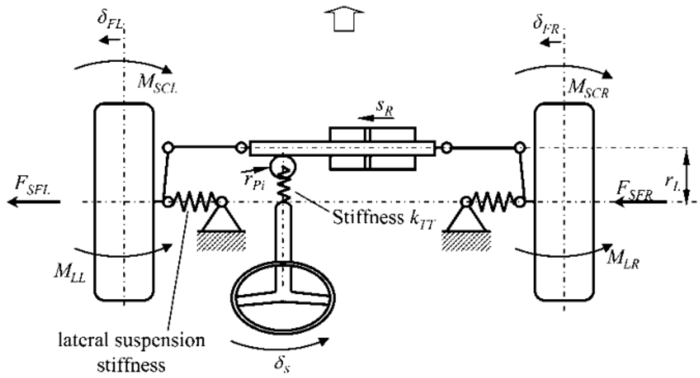


Figure 7: A more realistic steering system model [6].

This model includes properties such as power steering, steering ratio, steering column stiffness and lateral suspension stiffness, and is a much better example of real-life steering systems. Ultimately, however, the driver still needs to control the torque around the steering axis, just like in the simple Ackermann system. To understand how forces and torques from the road transfer to the steering wheel, the wheel geometry needs to be considered. The wheel itself may be attached to the suspension in various ways that affect its orientation in 3-D space. The angles the wheel creates with each plane are called caster, camber and toe angles, and are shown in Figures 8, 9 and 10, respectively. These angles create an offset between the forces acting on the tyre and the steering (or pivot) axis. These offsets are what causes the torque around the steering axis. Another lever arm is created due to the tyre deforming at the road surface. These offsets are called "Caster trail", "Camber trail" and "Pneumatic trail", respectively. A non-zero toe-angle causes the tyre to experience lateral forces even when the car is moving straight ahead and affects the steering angle at which the tyre provides maximum lateral force. A non-zero Camber angle will affect tyre grip, as it can be used to maintain vertical tyre orientation and thus proper tyre-road contact though a corner where the vehicle tends to roll. The simulation model used here is a single track model (see chapter 4.2.1) which cannot account for camber or toe angles, and therefore are not included in the analysis.

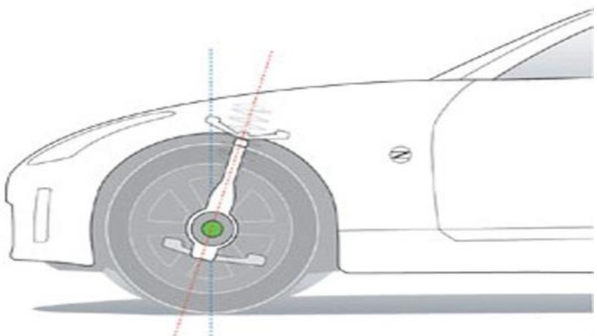


Figure 8: Caster angle is defined as the angle the steering axis creates with the vertical axis. The associated lever-arm is called the Caster trail. Positive caster (the steering axis leads the contact patch) is shown [10].

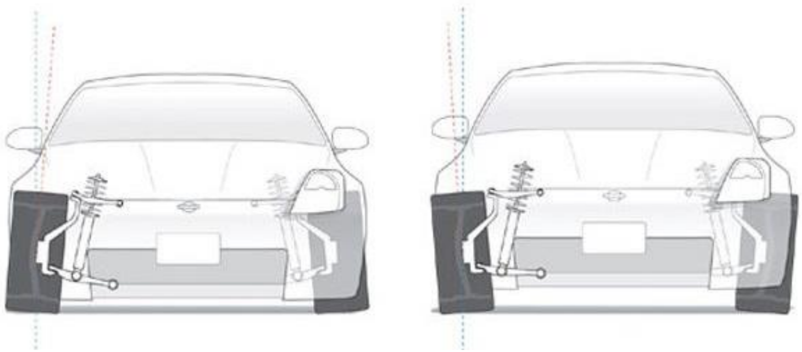


Figure 9: Camber angle is defined as the angle the wheel axis creates with the vertical axis. The associated lever-arm is called the Camber trail, or Scrub radius. Positive Camber is defined with the top of the wheel leaning outwards [10].

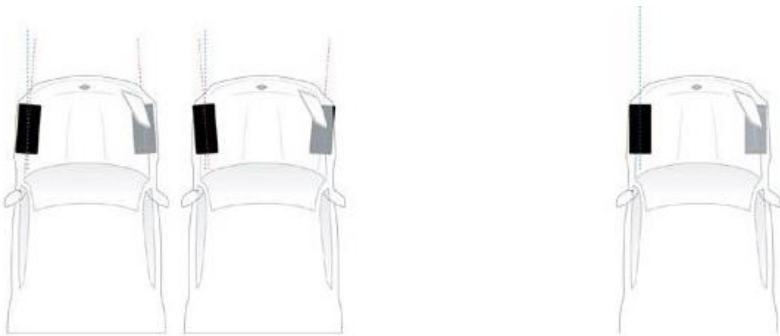


Figure 10: Toe angle is defined as the angle the wheels create off the straight-ahead direction. Toe-in is when the wheels turn in towards the car, while Toe-out is when the wheels turn out from the car [10].

Now when it has been shown how the tyre orientation affects the torque and forces that propagate up to the steering axis, the need arises to find out how the tyre creates those forces in the first place, and how they are affected by steering angle and vehicle dynamics.

Modern pneumatic car tyres are usually constructed as shown in Figure 11, with the following parts [8]:

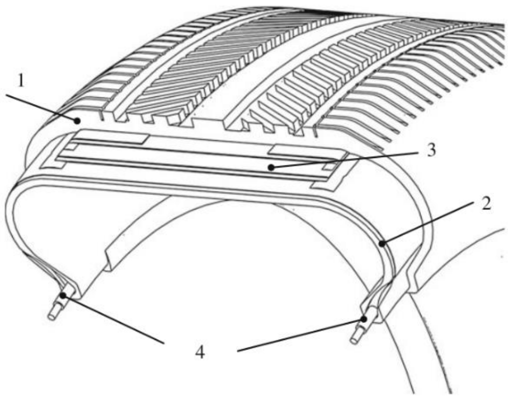


Figure 11: A modern pneumatic tyre [8].

• The tread (1) is made of rubber and contains the tyre tread profile consisting of the tread knobs and tread grooves.

• The carcass (2) consists of tensile surfaces covered in rubber, made up of e.g. artificial silk, nylon, and rayon. The carcass along with the tyre pressure gives the tyre its strength. It runs transversally to the rolling direction, radially from bead ring (4) to bead ring.

• The belt (3) is usually a composite layer of steel that rests on the tread surface of the carcass. It encloses the tyre from the outside and gives the tread its strength.

• The two bead rings (4) ensure a tight fit of the tyre on the wheel and guarantee, along with the enclosed rubber, a seal between the tyre and the rim.

The tyre, unlike a rigid body, deforms under load. This creates a flat area between the road surface and the tyre, where the loads are transmitted from the tyre to the road and vice versa. This area is known as the contact patch (Figure 12).

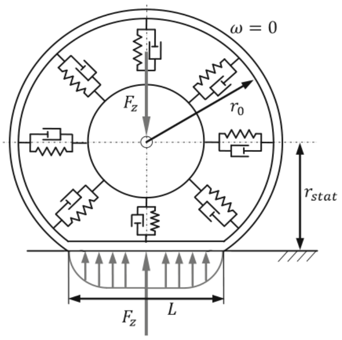


Figure 12: A vertical load F_z leads to tyre deformation, creating a contact patch of length L (not to scale) [8].

Two factors contribute to the tyre-road interaction. Contact friction is due to the intermolecular forces between the tyre compound and the road. This is the dominating factor affecting the amount of force a tyre can provide. The second factor is hysteresis friction, which is caused by the intermeshing of the tyre tread and the road surface. An illustration is shown in Figure 13:

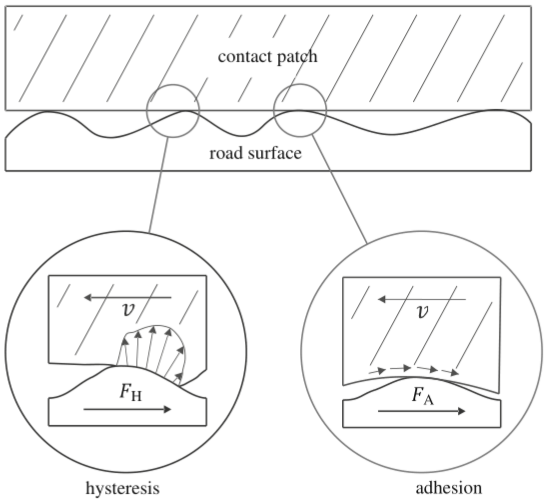


Figure 13: Factors contributing to tyre-road interaction [8].

Both of these effects depend on some relative movement between the tyre and road [8]. This relative movement is called "slip", and due to its direct relation to the forces and torques affecting the tyre, forms the basis of the mathematical tyre modelling.

4. Mathematical modelling

Now that a basic understanding of the components, forces, and effects that create the driver torque feedback has been established, mathematical models must be created for them, in order to simulate their behaviour, and allow for a more formal discussion about the effects.

4.1 Tyres

Modelling tyre behaviour is often done with a semi-empirical model developed by Hans. B. Pacejka, known as the "Magic Formula tyre model" (MF). It is a mathematical description of the relationships between tyre and road. The main inputs to this model are the slip variables as defined in [4]. Two types of slip are defined, longitudinal and lateral. Normally, slip ratios are used instead of pure slip angles, as this normalizes the slip to a value between 0 and 1. Slip ratios for the longitudinal case are defined as:

$$\sigma_x = \frac{r_{eff}\,\omega_w - V_x}{r_{eff}\,\omega_w} \text{ , while accelerating or} \tag{1}$$

$$\sigma_x = \frac{r_{eff}\,\omega_w - V_x}{V_x} \text{ , while braking.} \tag{2}$$

For lateral slip, the following definition can be used:

$$\sigma_y = \frac{V_x}{r_{eff}\,\omega_w}\tan(\alpha) \tag{3}$$

The parameters are defined in the Nomenclature.

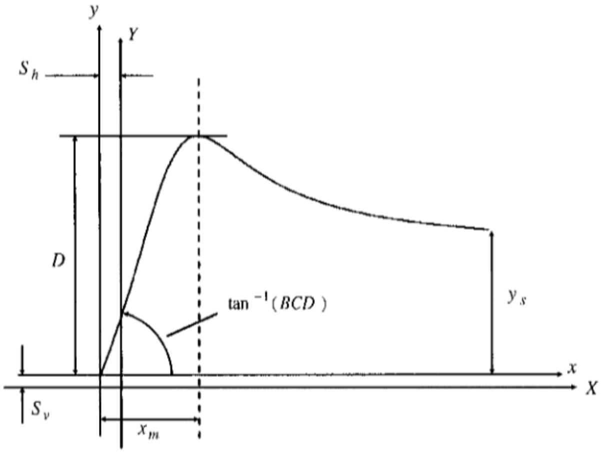
The Magic Formula model is based on the following equation:

$$Y = D \cdot \sin(\mathrm{C} \cdot \arctan(\mathrm{B} \cdot X - \mathrm{E}(\mathrm{B} \cdot X - \arctan(\mathrm{B} \cdot X)))) \tag{4}$$

Where
$$Y(X) = y(x) + S_v$$
$$x = X - S_h$$

X is here the input variable (slip angle or ratio), and Y is the output (torque or force).

The role of the different factors B, C, D, and E is visualized in Figure 14.



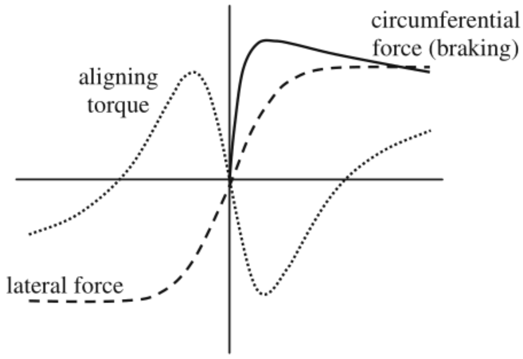


Figure 14: Magic Formula shape parameters [4]. *Figure 15: Generic MF curves [8].*

This produces typically a set of curves of the form shown in Figure 15 [8].

As can be seen, for small slip angles/ratios, the relationship between tyre forces and slip variables is linear, and the formula can be written as Y = (BCD)*X. In this case the factor BCD is known as cornering stiffness C_α or tyre stiffness C_σ , and a linear tyre model is produced. The parameters B, C, D and E are usually fitted to measured data, and vary between tyres. This is a relatively simple model and is therefore widely used.

4.2 Vehicle dynamics

The amount of slip a tyre experiences depends on the vehicles' motion, which necessitates vehicle dynamics modelling. Vehicle models used in simulation vary between relatively simple 2 D.O.F models (such as the single track model used in this thesis) to full-scale multi-body simulation with hundreds of D.O.F's. For a basic introduction to modelling a simple single track model is introduced.

Single-track model

A single track model is a basic vehicle model where the front and back wheel pairs are joined and modelled as a single wheel at the front and back. Due to this, this model is also known as the bicycle model. Figure 16 shows one possible parameterization of a single track model.

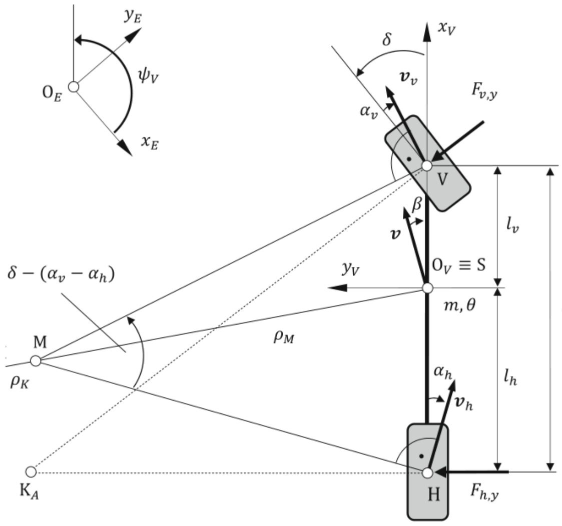


Figure 16: Description of a single track model [8].

Some assumptions are made about the vehicle and its behaviour in order to produce this model:

1. The velocity of the vehicles' centre of gravity (COG) is assumed constant along its longitudinal direction.
2. COG in ground plane.
3. Rolling and pitching motions are ignored due to (2).
4. The wheel load distribution between the front and rear wheel axles is assumed to be constant due to assumptions (1) and (2).
5. Longitudinal forces are ignored, following the constant velocity assumption (1).

These assumptions lead to various constraints on the degrees of freedom in the system, and allow for a fair description of the complete vehicle motion using only two variables: yaw rate and side-slip angle. This model is simple, but provides plausible description of vehicle motion with moderate lateral acceleration, typically up to 0.4 g on dry roads. A state space representation of this model is given in [8] and is presented below.

$$\begin{bmatrix} \ddot{\psi}_v \\ \dot{\beta} \end{bmatrix} = \begin{bmatrix} -\frac{1}{v}\frac{c_{\alpha,v}l_v^2 + c_{\alpha,h}l_h^2}{\theta} & -\frac{c_{\alpha,v}l_v - c_{\alpha,h}l_h}{\theta} \\ -1-\frac{1}{v^2}\frac{c_{\alpha,v}l_v - c_{\alpha,h}l_h}{m} & -\frac{1}{v}\frac{c_{\alpha,v} + c_{\alpha,h}}{m} \end{bmatrix} \begin{bmatrix} \dot{\psi}_v \\ \beta \end{bmatrix} + \begin{bmatrix} \frac{c_{\alpha,v}l_v}{\theta} \\ \frac{1}{v}\frac{c_{\alpha,v}}{m} \end{bmatrix} [\delta] \tag{5}$$

All the variables are defined in the Nomenclature and illustrated in Figure 16.

This model has one input, the steering angle δ, and two outputs: the yaw rate $\dot{\psi}_v$ and side-slip angle β.

From the output, the slip angles for the front and rear wheels can be derived, and used as inputs to the tyre model described previously, in order to obtain the forces acting on the wheels. These forces, when combined with the steering system geometry, are sufficient for a preliminary investigation into steering torque.

The total steering torque produced is approximated here by the tyres self-aligning moment, which is given by the linear relation in Equation 6.

$$M_{sa} = F_y \cdot t_p \tag{6}$$

This is approximately the torque needed in order to turn the wheels. This of course has to propagate up through the system to finally reach the steering wheel. Factors that affect how much of that torque is felt at the wheel include, among others, steering column friction and damping, compliances in the system and the power-steering gain curve. In order to evaluate the steering torque, the aforementioned systems are implemented in MATLAB/Simulink. To keep the model reasonably simple, factors like friction and compliances are discarded, and the focus is instead on the more general dynamic properties of the car and their effect on the steering wheel torque.

5. Simulink implementation of single track model

The single track model discussed in chapter 4 is implemented in Simulink in order to provide a base upon which the steering system will be built.

The car used for the simulation is a VW Golf V, with the data shown in Tables 2 and 3 [9]:

Table 2: Car data

Car	
Mass	1425 kg
Yaw moment of inertia	2500 kgm^2
COG-front	1.03 m
COG-back	1.55 m
Steering ratio	15.9:1

Table 3: Tyre data

Tyres	
Cornering stiffness front	108500 N/rad
Cornering stiffness rear	118600 N/rad

Using these values with Equation 5 yields the state space model in Equation 7.

$$\begin{bmatrix} \ddot{\psi}_v \\ \dot{\beta} \end{bmatrix} = \begin{bmatrix} -5.76 & 28.83 \\ -0.934 & -5.73 \end{bmatrix} \begin{bmatrix} \dot{\psi}_v \\ \beta \end{bmatrix} + \begin{bmatrix} 44.7 \\ 2.74 \end{bmatrix} [\delta] \tag{7}$$

The tyre model is linear as discussed in chapter 4.1 with cornering stiffness taken from Table 3.

The Simulink model used is shown in Figure 17.

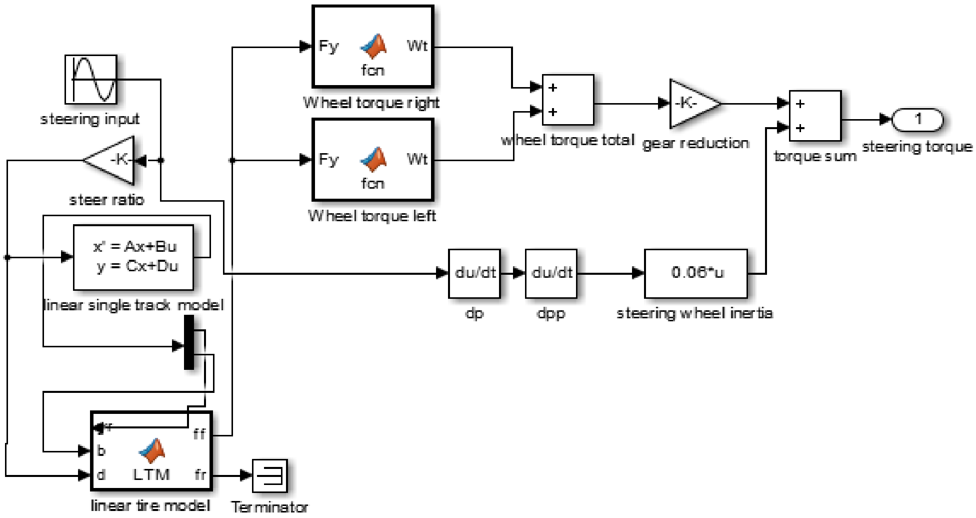


Figure 17: Simulink model.

This model considers steering wheel inertia, but not friction, damping or other properties. The input, for example a sine wave describing the steering wheel movement, is fed into the single track model described by Equation 7, and is also derived twice to obtain rotational acceleration which will be used in the steering wheel inertia calculation. The output from the single track model is then used as input to a linear tyre model which calculate slip ratios and lateral forces for the rear and front tyres. The rear tyre forces are not relevant to this discussion and are therefore neglected. The lateral forces are used in the "wheel torque" block to calculate self-aligning moment according to Equation 6. The reason for using two torque blocks despite the single track model using only one front wheel is to allow the system to be adjusted for more complex dynamic models. For added realism, a power steering system is added to the model, shown in Figure 18.

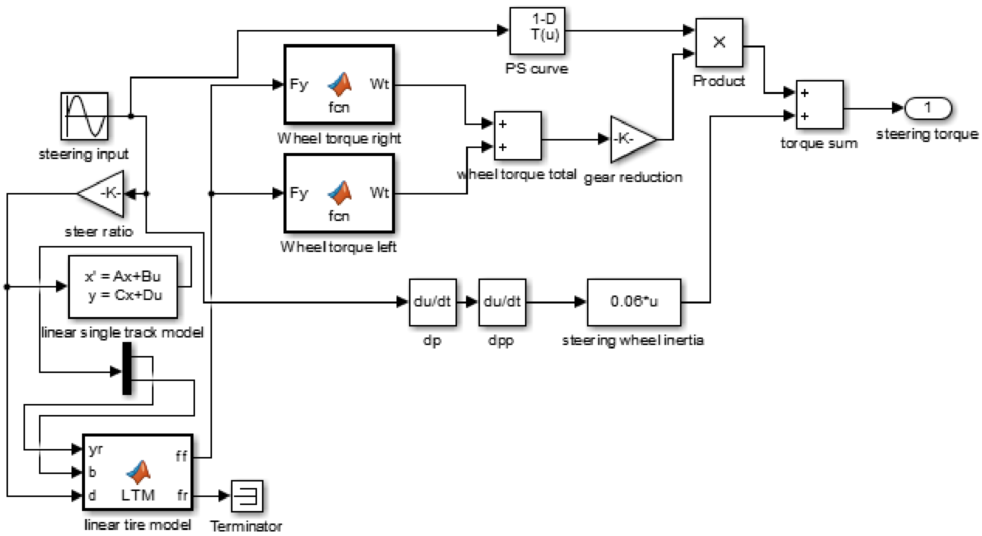


Figure 18: Simulink model with power steering added.

The power steering system here is modelled as gain in the range zero to one, applied to the steering wheel torque. The gain is proportional to the steering wheel angle, with unity gain at zero steer angle, which is described using a lookup table. These are the models that will be used to explore the effects of different parameters on the resulting steering torque.

6. Simulation results

Here, the role of different car properties in the generation of steering torque will be examined. The focus of this simulation will be on the qualitative effect the various subsystems have on the driver torque feedback, rather than on the quantitative one. The parameters that will be varied are the mass of the car, steering ratio, power-steering curves, and the interplay between those. The role of the yaw moment of inertia, and steering wheel inertia was also investigated, and those results are shown in Appendix 1. The steering input is adjusted so that the car retains the same lateral acceleration with different parameters. To establish a baseline, the model without power-steering was run with a vehicle speed of 100 km/h and with the steering input with frequency of 0.2 Hz adjusted to provide around 0.3 g of lateral acceleration. In other words, the Norman on-centre test was simulated, yielding the results shown in Figure 19.

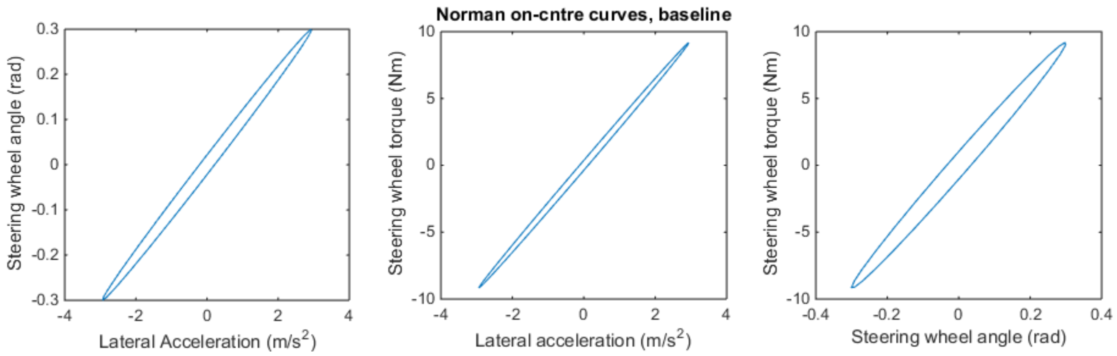


Figure 19: Baseline on-centre test.

As can be seen, the output graphs are of the same general shape and order of magnitude as the graphs shown by Norman in Figures 1 to 3. This is a good sign which shows that even a very simple model can provide useful information. An observant reader might notice that the steering torque recorded here is nearly twice as large as Norman and Higuchi and Sakai present; this is due to the Simulink model not modelling the power-steering system, as stated before. Now that a baseline has been established, the effect of parameter variation can be studied.

6.1 Mass

The first parameter to be examined is mass. The mass of the car, which is nominally 1425 kg, is first increased to 2000 kg and then decreased to 1000 kg, while the steering input is adjusted to keep lateral acceleration constant. The Norman curves are plotted again, and are shown in Figure 20.

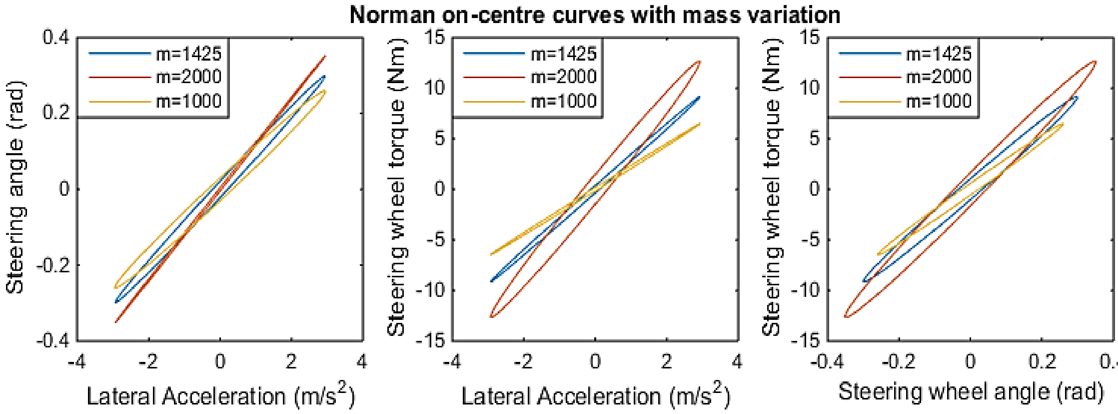


Figure 20: Norman curves for varying mass.

6.3 Steering ratio

The steering ratio is the ratio between the angle of the front wheels and steering wheel angle, and at the same time serves as a reduction factor for forces transferred from the tyres. On a non-power steered car, this parameter is the main way to control steering feel and effort. The steering ratio is first decreased to 14:1 and then increased to 20:1. To calculate the forces at the steering wheel, the inverse of this ratio is used. The resulting Norman curves are plotted in Figure 21.

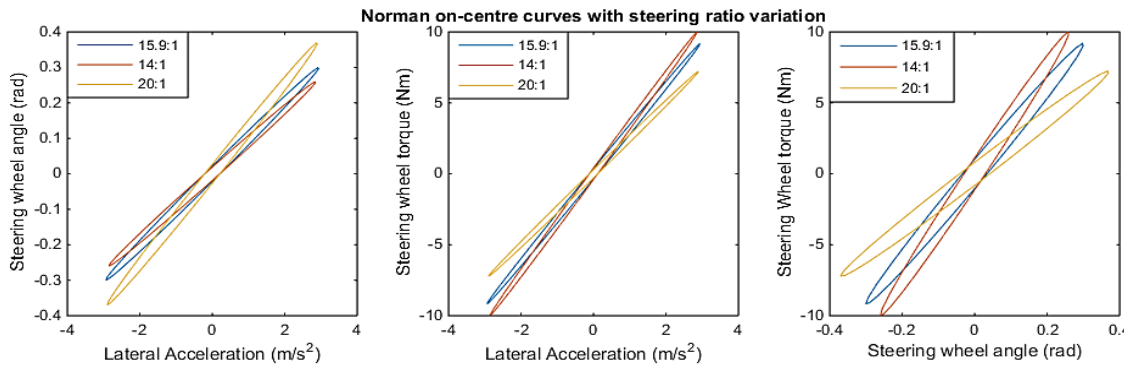


Figure 21: Norman curves for varying steering ratio.

6.4 Power steering system

Modern cars rely on the power-steering system to both provide simple torque reduction and to fine-tune the steering characteristics of the car at different speeds and driving conditions. This is a very important system with a large impact on driver torque feedback. A simple power steering system is added to the model (shown in Figure 18), in which the amount of assist is proportional to the steering angle, with no gain around zero angle. Four different gain curves are evaluated, and are shown in Figure 22. All other car properties are set back to those shown in Tables 2 and 3.

The added power-steering system has no effect on the vehicles behaviour, as it only affects the torque feedback to the driver. As such, the plots relating steering angle input and lateral acceleration are unchanged. The relationship between steering angle and steering torque however is different, and is shown in Figure 23.

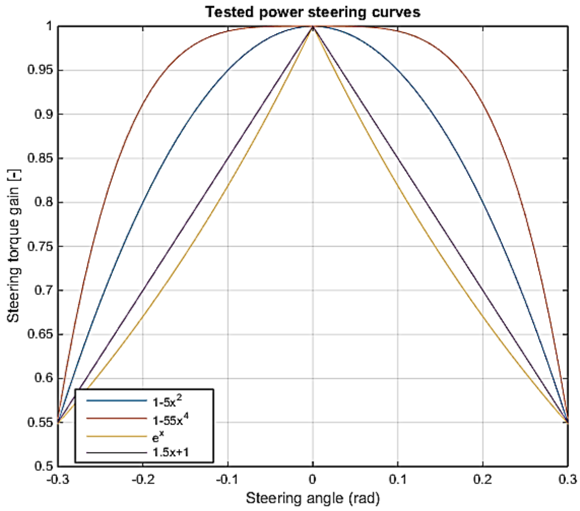


Figure 22: Power steering curves.

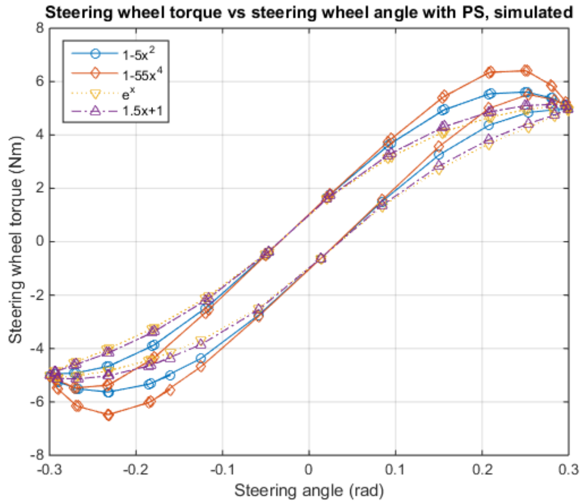


Figure 23: Simulated on centre-test with four different power steering curves.

As nearly all vehicles on the market are equipped with some form of power steering, it can be interesting to run the parameter variation test again, with power steering added. This is shown in Figures 24 and 25, using the first power-steering gain curve, $y = 1 - 5x^2$ where x is the steering angle and y the torque gain factor.

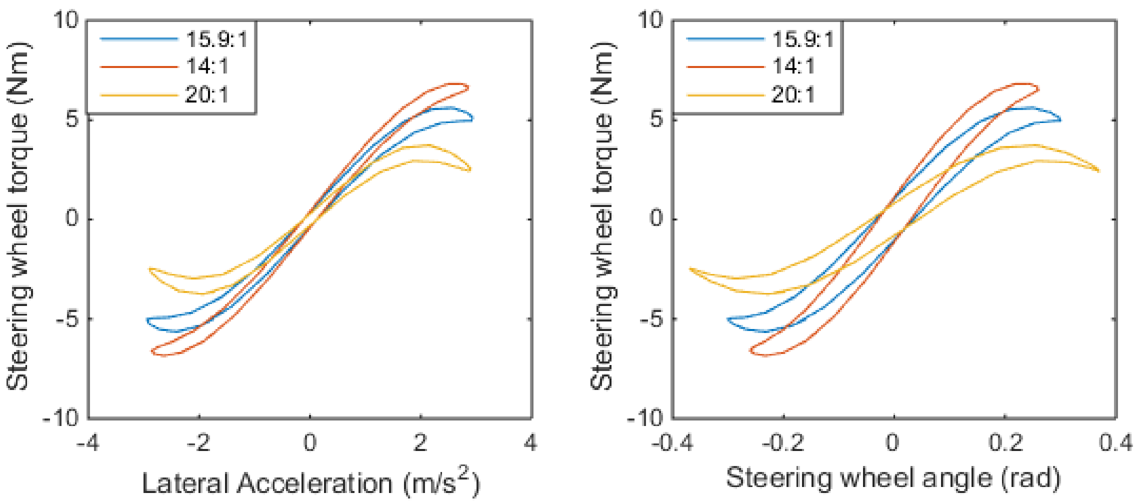


Figure 24: Steering ratio variation, with power steering.

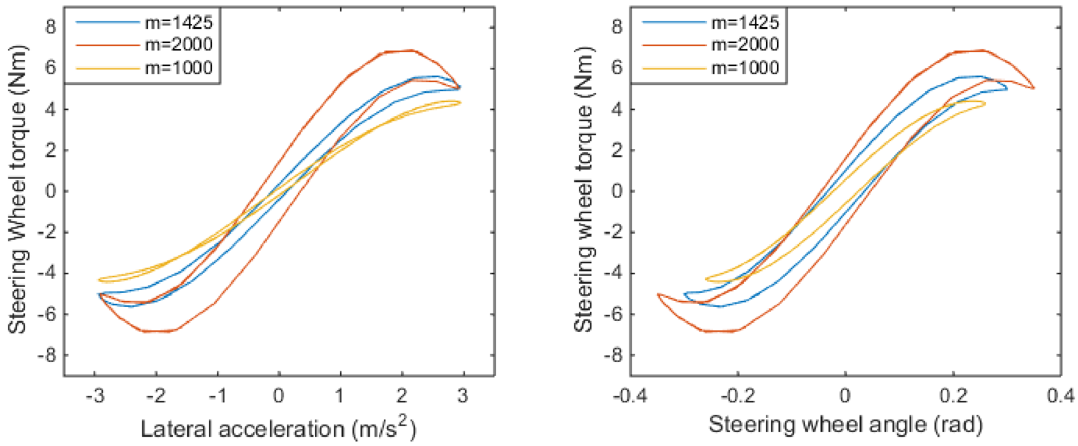


Figure 25: Mass variation, with power steering.

7. Discussion

From Figure 20 it can be seen that a change in vehicle mass affects all three Norman curves. The heavier the car, the higher the peak value of the steering torque, due to the larger force required to turn the car. That in turn increases the steering angle required to make the car turn as desired.

The steering ratio variation is plotted in Figure 21 and like the vehicles mass, affects all three Norman curves, which is why the steering ratio setting can be used to compensate for vehicle mass, at the cost of increasing steering wheel movement. Before the advent of power steering, the steering ratio was the main factor controlling the driving effort. Trucks and heavier vehicles had either very heavy steering, or lighter steering that required large steering wheel angles to turn the vehicle. Smaller and lighter vehicles had more "direct" steering. In modern cars and trucks power steering effectively decouples the steering ratio as it relates to the steering angle, and the torque reduction it provides.

In addition, the effect of changing yaw moment of inertia and steering wheel inertia was studied, and the results are shown in Appendix 1, Figures A1 and A2. The effect of the variation on steering response was small. The yaw moment of inertia affects the lag between the steering input and the car turning. Generally speaking, the lower the yaw moment of inertia the more nimble a car is. However, the inertia also serves to give the vehicle directional stability, so a too small value here can make the car feel "nervous". The small effect on steering torque is likely because it is a product of vehicle mass and geometry, and thus not an independent variable. Varying mass and vehicle wheelbase will affect this inertia as well as the other vehicle dynamic properties and as shown, that has a much larger effect on the steering torque. For the steering wheel inertia, the effect is practically negligible due to the light steering wheel in comparison to the vehicle weight, and in this case could be omitted from the Simulink model to simplify the system further. For steering manoeuvres involving more rapid steering wheel movement and vibrations, this may not be the case, and its effect should be evaluated separately.

Figure 23 shows that the power steering curve has a large effect on the steering torque, especially at larger steering wheel angles. The amount of assist the power steering provides is limited primarily by the strength of the power steering servo motor, and the assist curves can essentially be arbitrary. The power steering system can even be used to decouple the steering torque from the tyre moment and provide a completely constant steering wheel torque. In Figures 24 and 25 it can be seen that for vehicle mass at 2000 kg and for steering ratio=20:1, the selected power steering curve is inappropriate. In Figure 15 the typical shape of Magic Formula tyre forces is shown. It can be seen that after reaching a peak value, the forces decrease. This happens when the wheel starts sliding, and the same shape can be seen in the figures, but this shouldn't happen! The tyre model used is linear, in a moderate lateral acceleration area, away from the tyres sliding region. However, the power steering curve combined with these specific vehicle settings makes the torque response appear as if the vehicle starts skidding. This is false information and can be misleading to the driver. Care must be taken to assist the driver, without hiding or altering the information contained in the steering wheel torque. In sensitive areas, such as for small steering angles, and for the area around the limit of the tyres grip, the power steering system should only provide constant torque reduction, without changing the shape of the torque feedback. This is important to the safe operation of the vehicle. Toffin et al. [7] found that drivers can adapt quite well to different torque feedback curves, as long as these make sense. For no feedback (constant steering wheel torque), or inverted wheel torque (decreasing steering wheel torque when tyre forces increase), no driver could drive the car reliably.

The power steering system eventually proved so successful that even given the added complexity of the system, almost no commercially available car is sold without power steering today, with the Alfa Romeo 4C being a notable exception. The next logical step in the evolution of this system is a pure steer-by-wire system, in which the steering interface, be it a steering wheel or something else, is completely mechanically decoupled from the steering system, and the torque feedback is created by motors. This is comparable to the fly-by-wire systems used by practically all commercial airliners. This provides numerous advantages: eliminating the steering column and tie-rod can reduce the weight and space required for the steering system, which can provide better comfort, performance and fuel economy. Further, it allows for the customising of the steering response to the drivers preferences, which enhances comfort. Finally, not needing a traditional steering wheel allows for a redesign of cabin ergonomics. Without mechanical linkage, the placement, shape and size of the steering device, which no longer needs to be a wheel, can be varied and customized.

7.1 Sustainability aspects in power steering system design

Implementing a power steering system requires adding a means of applying force to the tie rods. This can be done using a hydraulic system, an electric one, or a mix of both. Hydraulic power steering uses a hydraulic pump driven by the vehicles motor to apply pressure to a cylinder, which in turn applies forces to the tie rod. An electro-hydraulic or "hybrid" system, uses a separate electric motor to drive the pump. A purely electric power steering uses the electric motor to directly apply force to the tie-rod. Regardless of the specifics, all of these systems require power to function, power which is not needed on unassisted cars. With increasing focus on vehicle carbon emissions and energy efficiency, the power steering systems energy consumption also needed to be reduced.

Older hydraulic power steering systems typically require around 250W of power [12]. Electrically assisted hydraulic power steering system require much less, typically around 50W. Purely electrical systems do better still, needing just around 20W of power. That is a factor 10 reduction, which is very significant. Reducing the amount of power needed allows the vehicle to achieve better fuel economy and thus lower emissions per km driven. This saving is the greatest advantage offered by electrical power steering systems. In regards to steering feel, however, they are perceived as inferior [11]. Transferring torques and forces back to the steering wheel requires stopping or reversing the electrical motor which provides the assist. As the motor has more inertia compared with a purely hydraulic power steering, the steering torque feedback lags somewhat, leading to less precise steering feel. This will hopefully change in the future when EPS systems improve.

As the mass of the vehicle has the dominant effect on steering effort, an alternative solution is to reconsider the need for power steering on lighter and smaller vehicles. The European B-segment cars are small compact cars with a mass of around 1000 kg. Among the most popular cars of this segment are the Ford Fiesta, Peugeot 208 and Renault Clio, all of which do have power steering. It should be possible with revised steering system settings and more extensive use of lightweight materials to eliminate the power steering system which also adds to the weight and complexity of the car, i.e. both fuel consumption, cost and safety.

8. Conclusions

This thesis set out to explore the concept of steering feel, and has identified the causes, key quantities and geometrical relationships that affect it. The objective was to answer the following questions:

1) What is steering feel?
2) Why is steering feel important?
3) What affects steering feel?

Steering feel is a broad concept and this thesis focused primarily on steering torque as an indication of steering feel. Using simplified, linear models limits the conclusions drawn from the simulation results to be primarily qualitative assessments, not numerical results. With these limitations in mind, the main findings of this thesis are used to answer the three research questions.

1) What is steering feel?

Throughout this thesis, the definition used was “relationships between the steering wheel angle, steering wheel torque and the vehicles dynamic response such as lateral acceleration and yaw rate.” This illustrates that this is not a simple measured quantity as much as an emergent quality of a complex system. An expansion of that definition to include for example steering wheel vibrations or the geometry of the steering wheel is also possible, and can provide a different view of steering feel.

2) Why is steering feel important?

Steering feel carries information about interaction of the vehicle and the road. It communicates to the driver the amount of grip a tyre provides, allowing the driver to feel when the vehicle is about to lose grip and start sliding. A good steering feel also allows the driver to predict how the vehicle will react to a given steering input, allowing for more intuitive and relaxed driving. This makes driving safer, which is important. Another aspect of steering feel is comfort. Transferring every vibration to the steering wheel might be counted as “accurate” steering feel, but is fatiguing to the driver. A well-judged balance between accuracy and comfort increases the subjective quality of the vehicle and its likelihood to sell well.

3) What affects steering feel?

Tyre choice and wheel diameter affect the forces generated by the wheel. These forces both steer the car, and propagate up through the steering system to the steering wheel, providing the driver with information. Steering system and suspension geometry affect Caster, Camber, Toe and steering axis angles, creating lever-arms for the tyre-generated forces to act on, which controls the shape of the steering wheel torque curve for various driving conditions.

In this work, it is shown that the steering ratio is also a very important factor, controlling the steering wheel angle and effort required to steer the car as desired. The mass of the vehicle clearly relates to the driving effort. The heavier the vehicle, the larger the forces needed to turn it, which is reflected back on the steering wheel. The power steering system is the subsystem having the largest effect on the steering feel, by reducing the amount of effort required to steer a vehicle, and shaping the steering wheel torque-steering wheel angle curve.

As it stands today mechanical linkage between the steering wheel and the steering system is still legally required on road vehicles. This is likely to change as more research is being done on this topic and driverless cars, which are an inherently “by-wire” design, gain public interest. Future research on this topic therefore are likely to focus on steer-by-wire systems and simulated force feedback. It is my hope that like Alfa Romeo did with the 4C, more manufactures will re-evaluate the need for SbW or power steering systems on small and light cars, and purely mechanical systems are not forgotten. Consumer cars today are already very complex systems, and adding to this complexity may not be necessary. In the words of Henry David Thoreau: “*Our life is frittered away by detail... simplify, simplify*.”

9. References

[1] Dixon, J. (2009). *Suspension analysis and computational Geometry*. John Wiley & Sons.

[2] Pfeffer, P. and Harrer, M. (2008). "On-centre steering wheel torque characteristics during steady state cornering," SAE Technical Paper 2008-01-0502, doi: 10.4271/2008-01-0502.

[3] Higuchi, A. and Sakai, H. (2001). "Objective evaluation method of on-centre handling characteristics", SAE Technical Paper 2001-01-0481, doi: 10.4271/2001-01-0481.

[4] Rajamani, R. (2012). *Vehicle dynamics and control.* Boston, MA: Springer US.

[5] Norman, K.D. (1984). Objective evaluation of on-centre handling performance. SAE paper 840069.

[6] Pfeffer, P., Harrer, M and Johnston D. N. (2008). “Interaction of vehicle and steering system regarding on-centre handling”, Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility, 46:5, 413-428, doi: 10.1080/00423110701416519.

[7] Toffin, D., Reymond, G., Kemeny A. and Droulez J. (2007). “Role of steering wheel feedback on driver performance: driving simulator and modelling analysis”, Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility, 45:4, 375-388, doi:10.1080/00423110601058874.

[8] Schramm, D., Hiller, M. and Bardini, R. (2014). *Vehicle dynamics: modelling and simulation*. Berlin: Springer.

[9] Thellman, J. (2012). “Vehicle dynamics testing in advanced driving simulators using a single track model.” (Master Thesis). Linköpings universitet.

[10] “More steps to better handling”. (2011). Retrieved March 23rd , 2015, from http://evo123.net/forum/technical/evo-technical/suspension/7425-part-two-more-steps-to-better-handling

[11] “Are we losing touch? A comprehensive comparison test of electric and hydraulic steering assist”. Tech Feature. Retrieved May 21, 2015, from http://www.caranddriver.com/features/electric-vs-hydraulic-steering-a-comprehensive-comparison-test-feature

[12] Pfeffer, P., Lin, J. and Harrer, M. (2011). “Energy analysis of power steering systems during a newly developed driving cycle.” SAE technical paper 2011-01-0230, doi:10.427/2011-01-0230.

Appendix 1: Additional results

Yaw moment of inertia

The yaw moment which is nominally $2500\,\text{kgm}^2$ is first increased to $3000\,\text{kgm}^2$ and then decreased to $2000\,\text{kgm}^2$. The resulting Norman curves are plotted in Figure A1.

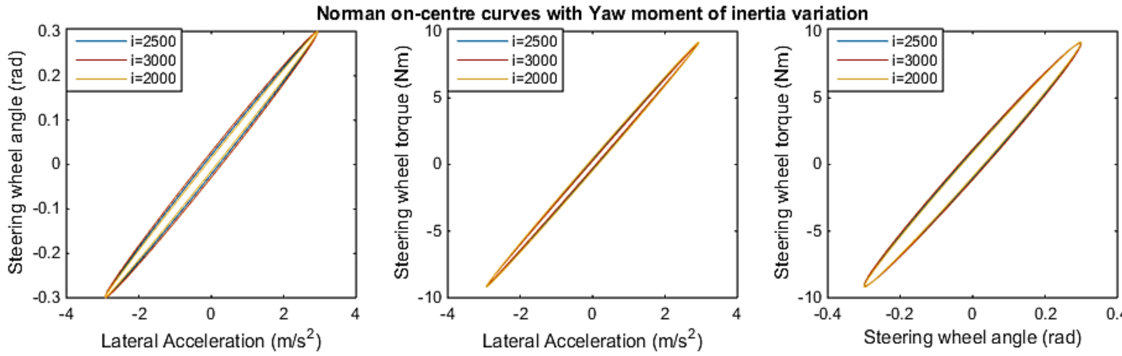


Figure A1: Norman curves for varying yaw moment of inertia.

Steering wheel inertia

The steering wheel moment of inertia used here is around $0.06\,\text{kgm}^2$. This is insignificant when compared to the yaw moment of inertia which is $2500\,\text{kgm}^2$. Increasing or decreasing the steering wheel inertia, even by a factor 10, has negligible effect on the steering torque. This is shown in Figure A2.

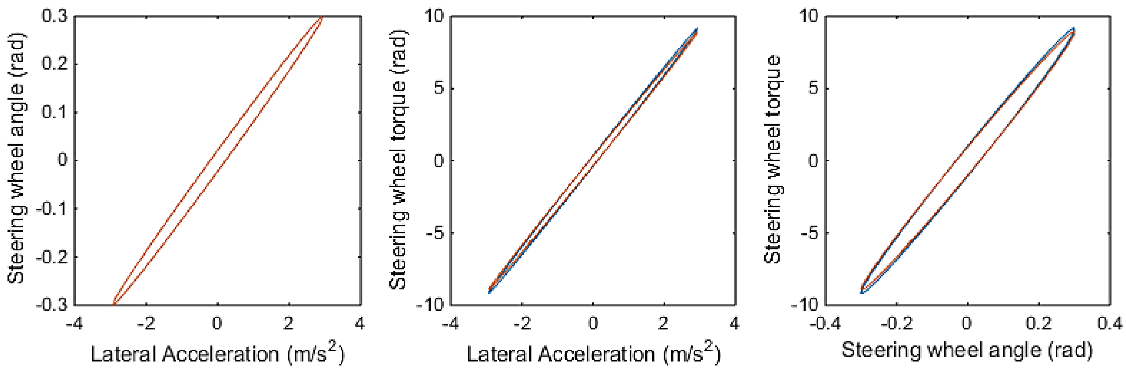


Figure A2: Norman curves for varying steering wheel inertia.

www.kth.se