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Vehicle Stability Control for Roadside Departure Incidents by Steering Wheel Torque Superposition

Developed for and Evaluated in the Chalmers Driving Simulator

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Master's Thesis EX020/2006

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ABSTRACT

The objective of this project was to evaluate the suitability of a system based on active superposition of torque on the steering wheel using the electric power steering. The system should help the driver to keep control of the car in roadside departure incidents.

A description of different scenarios involving this kind of accident suggests the strategies which were considered interesting. The strategies were implemented and evaluated in the Chalmers University of Technology driving simulator. One of them was considered of special interest and the suitability of it as an active safety system was tested by different drivers. Focus was put on providing a realistic and repeatable test environment rather than developing a function ready for production. Such a task is left for future development.

The compatibility of the developed function and existing systems such as the ESP (Electronic Stability Program) was also evaluated. The implementation of the ESP is described in the report.

A new 14-degree of freedom vehicle model was developed and used in the simulator to evaluate the active torque superposition. This work is extensively described in the appendix.

Key words: vehicle dynamics, vehicle modeling, driving simulator, roadside departure, driver steering recommendation, handling.

NOTATIONS

Main report:

β	side slip angle
δ	steering angle
ω	tyre rotational speed
R	tyre radius
v, v_x, v_y, v_z	speed
$\alpha, \alpha_f, \alpha_r$	tyre slip angle
s	longitudinal tire slip
ψ	yaw angle
$\dot{\psi}$	yaw rate
$M_{x,}, M_y, M_z$	tyre torque
$F, F_{x,}, F_y, F_z$	force
n	pneumatic trail
l, l_f, l_r	length, front/rear
F_b	brake force
C_x, C_y	cornering stiffness
FL	front left
FR	front right
RL	rear left
RR	rear right

Appendix:

ρ	air density
A_f, A_l	area frontal/lateral
Z	vertical position
Z_{CG}	centre of gravity vertical position
t_f, t_r	track width front/rear
S	generalised slip
S_x, S_y	longitudinal/lateral slip
φ	generalised slip angle
R_e	effective radius
m	vehicle mass
h_{roll}	roll axle
μ_{brkwhl}	friction coefficient brake pad-wheel
A_{pad}	area brake pad
r_{brk}	brake disc radius
P_{brk}	brake pressure

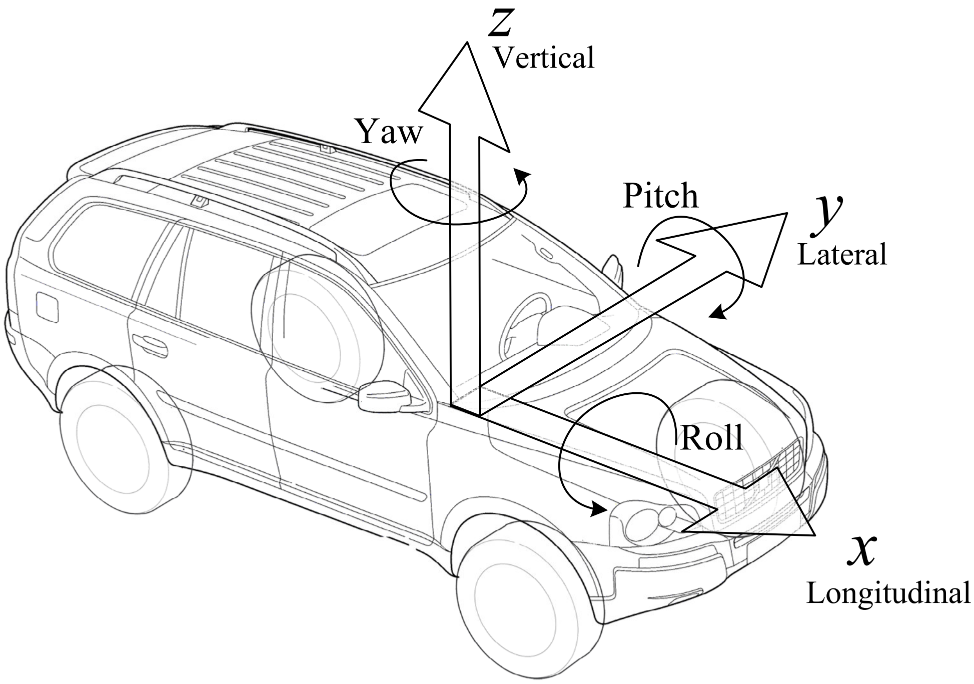

Figure 1: Coordinate system and rotations of the vehicle.

ABBREVIATIONS

ABS.	Antilock Brake System
ESP.	Electronic Stability Program
YCbB.	Yaw Control by Brake
YCbS.	Yaw Control by Steering
LDW.	Lane Departure Warning
LKA.	Lane Keeping Assistance
DSR.	Driver Steering Recommendation
EPAS.	Electric Power Assisted Steering
HIDS.	Honda Intelligent Driver Support System

Contents

1 Introduction

1.1 Background

One of the major efforts of the automotive industry today is improving the safety of the automobiles designed and build. Safety features can be divided into two main areas:

Passive safety aims at reducing the consequences of an accident, when it occurs. It involves proper structural design of the chassis and body in order to limit injuries, existence of areas which can deform in a controlled way and limit the deceleration of the occupants as much as possible.

Active safety involves everything aiming at reducing the possibilities for an accident to happen. This includes ensuring easy and predictable handling of the car, both by improving the vehicle stability and driver response. Vehicle stability can be improved by electronically controlled devices which appeared in the last decades like ABS (Anti-lock Brake System) [39], ESP (Electronic Stability Program) [17]. Driver response and handling can be improved by systems like Driver Alert [43] and LKA (Lane Keeping Assistance) [44], see page 4.

Characteristics not directly connected to safety, such as air condition and driver's comfort also have an influence on the active safety of the vehicle since they can help keeping the driver in a suitable condition for driving.

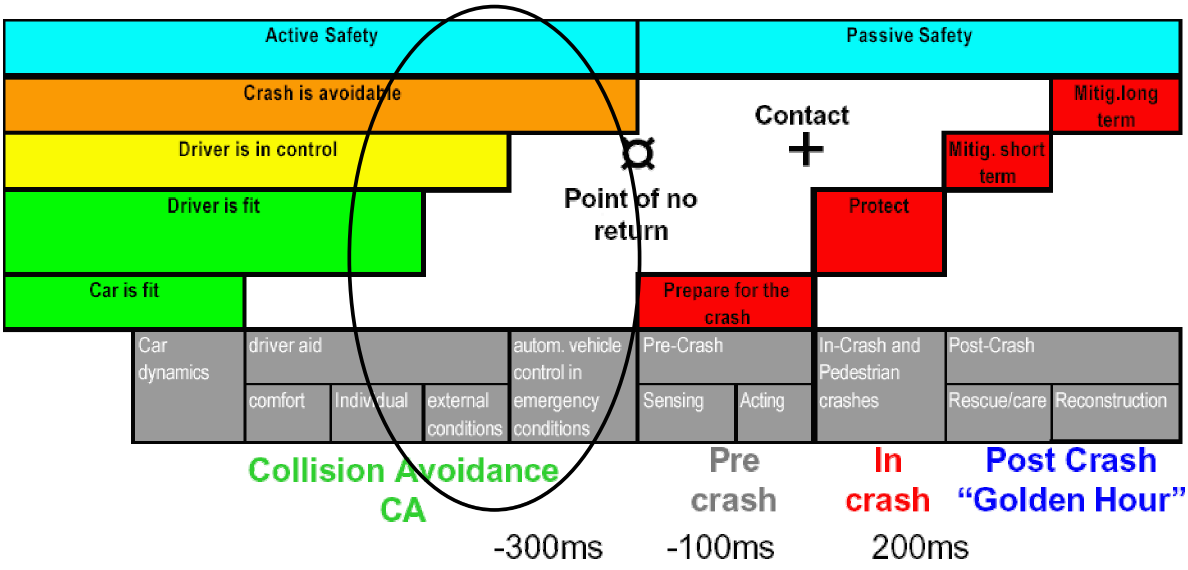

Figure 2: Autoliv safety phase chart, [www.autoliv.com].

This report describes the development and test of an active safety feature in a specific situation. It can be located in the circled area in the Autoliv safety phase chart, Figure 2, since it helps the driver to keep control of the car in a situation which could end up in an accident.

1.1.1 Road side departure accident

Today a common accident is the road side departure accident [45]. It involves situations where the car leaves the road for some reason and looses stability on its way back to the road. Figure 3 depicts possible succession of events, in right side traffic, that can be involved in this situation. The flowchart has been created and discussed during the project together with Jesper Sandin [46].

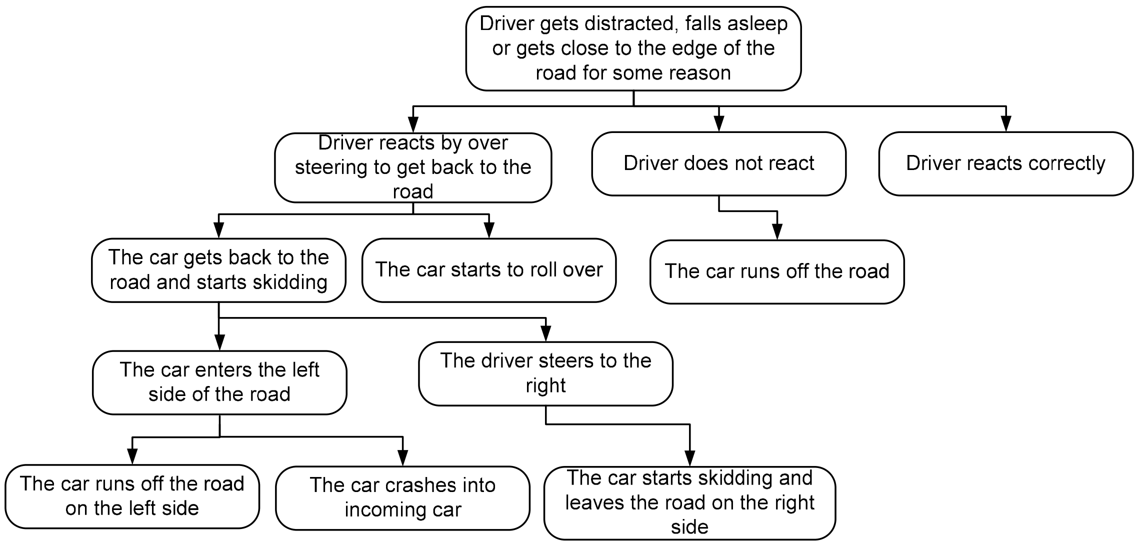

Figure 3: Flowchart of the succession of event during the accident.

Some driver and environment related characteristics of the situation must be kept in mind:

- <u>Surprise</u>. If the reason for which the car is leaving the road is the distraction of the driver, their first reaction in order to restore the situation will most likely be governed by surprise and maybe panic [3], which may lead to an overreaction on the steering wheel.

- <u>Angle of inclination of the ditch</u>. Chances are that the ditch will show a noticeable inclination which, due to the effect of gravity, may prevent the driver from easily steer the car back to the road.

- <u>Lower coefficient of friction in the roadside than on asphalt and most likely asymmetric between left and right tyres, when the car is only partly off the road</u>. The reduced friction limits the ESP's capability of generating a restoring yaw torque that could stabilise the car in this situation. Besides, it will introduce an asymmetry in braking force between left and right tyres, which may induce a strong yaw moment capable of destabilising the car.

A specific case of the accident is shown in Figure 4.

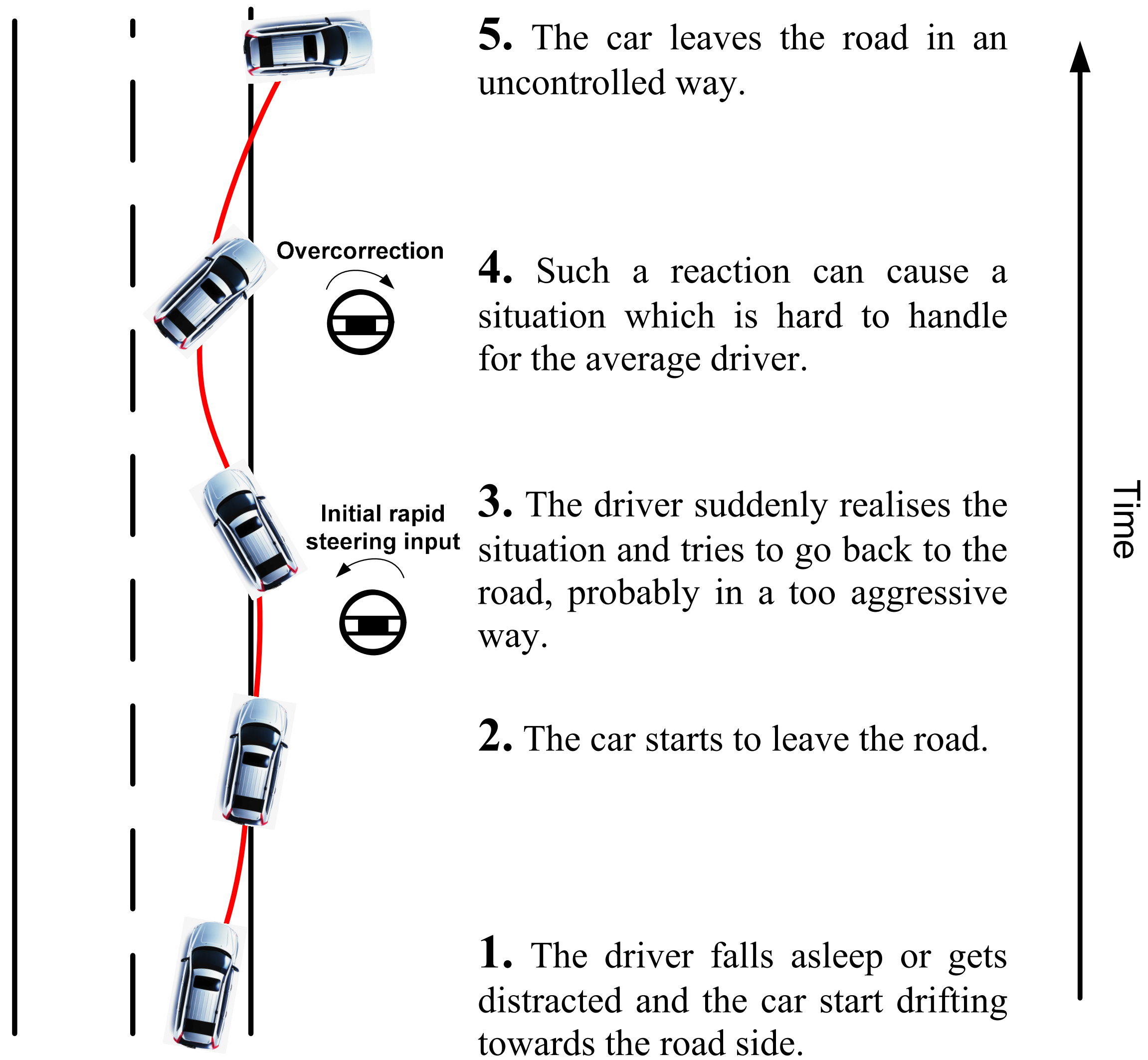

Figure 4: One possible scenario for the road side departure accident.

1.1.2 Introduction to active safety systems

Several systems common in today's cars could help preventing the accident depicted above. Some of them are described below.

Driver alert

Driver alert is a system currently developed by Volvo Cars [43]. It monitors the motion of the vehicle, using a camera and different sensor to calculate how well the driver is planning the path. If the system detect that the driver is not alert it warns them using sound and lights [43].

Lane keeping assistance (LKA)

The Volvo Cars lane keeping system uses a camera to detect lane markings in the road. To help the driver a guiding torque is added to the steering wheel helping the driver to stay in lane [44]. This is also done by Honda in the Honda Intelligent Driver Support System (HIDS). Systems like these can reduce the driver workload and increase the awareness and motivation of the driver [13].

Antilock braking system (ABS)

Today ABS is installed in almost all new cars [40]. The purpose of ABS is to prevent the tyres from locking during hard braking and thereby keeping the steerability of the vehicle. When a tyre becomes locked the amount of lateral force available is practically cancelled and the vehicle looses steerability. Besides loosing steerability the tyre will not gain more braking force by locking. By keeping the tyre from locking the maximum brake force can be used without loosing control of the vehicle [39].

Yaw control by brake (YCbB)

During normal driving a vehicle rarely enters the nonlinear zones of the tyres, see Figure 13. Because of this, it can not be taken for granted that the normal driver has any experience in the behaviour of the car in limit handling situations [17]. In limit situations a properly designed systems can react faster than the driver and help stabilising the car. This makes the car easier to understand and maneuver for the average driver.

Yaw Control by Brake is commonly referred to as ESP in commercial terms. One of the tasks of the YCbB is to limit the vehicle side slip angle and keeping it below a dangerous value while trying to achieve the path that the driver is requiring through the steering wheel [19]. By braking one or more tyres a force is created, which translates into a yaw torque applied over the whole vehicle. The compensating yaw torque will be the applied brake force, F_b, times half the wheel base, see Figure 5.

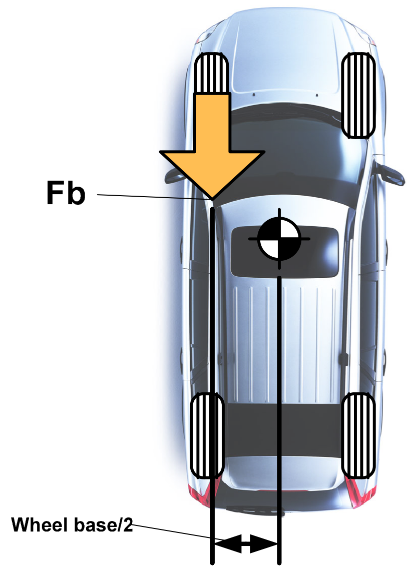

Figure 5: Brake applied to one tyre creating a compensating yaw torque.

YCbB can help in two different situations shown in Figure 6:

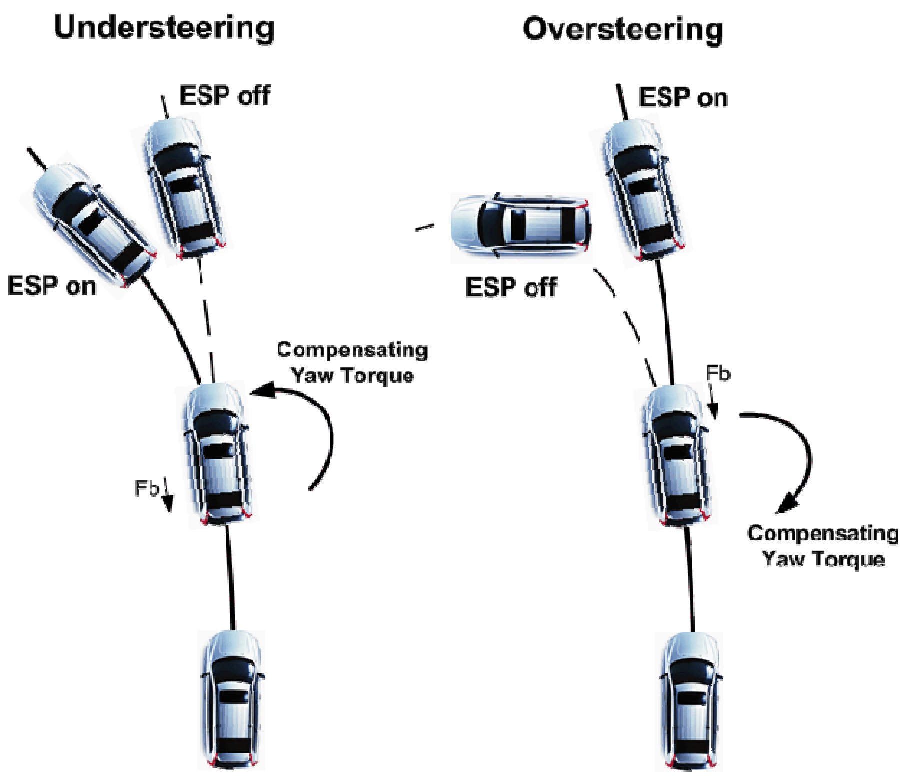

Figure 6: Yaw Control by Brake interference in case of under-steering and over-steering.

- When the car under-steers, the lateral grip is not enough to turn the vehicle as sharp as the driver intended. In this situation the YCbB helps the driver in two ways. The engine torque is reduced to decrease the wheel traction and increase the lateral grip and the inside front and/or rear wheel is braked (F_b in Figure 6) inducing a compensating yaw torque that helps the vehicle turn more.

- When over-steering the engine torque is reduced and the outer front and/or rear tyres are braked to induce a compensating torque helping the vehicle not to over-steer.

The vehicle speed and steering wheel angle is fed to a linear reference model, see chapter 2.5, which gives the YCbB the side slip angle and yaw rate that a vehicle with linear tyres would have in the same situation. The reason for a linear vehicle is desirable is that the common driver expects the vehicle to behave linear, as it is in the linear region of the tyres where the driver is used to drive most of the time [16]. The difference in yaw motion between the reference model and the actual car is used to decide which tyres to brake.

Yaw control by steering (YCbS)

In contrast to YCbB, Yaw Control by Steering systems affects the yaw motion of the car by modifying the steering angle of, generally, the front tyres ([28], [29]). Some effort has been taken into steering the rear tyres as well [25].

The steering angle modification induces a change in lateral force created by the front axle, which is equivalent to adding a yaw moment to the original situation. As it happens with YCbB this change in yaw moment can help to compensate for a wrong attitude of the car.

Yaw Control by Steering systems are commonly known as Active Steering systems. Compared with Yaw Control by Brake (ESP), they have some advantages [30]:

+ By changing the wheel angle a corrective force is applied and the erroneous force created by the tyres is removed.

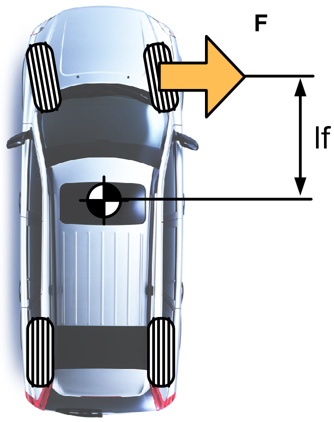

Figure 7: A steering angle creating a compensating yaw torque.

+ The correcting yaw torque that can be created is potentially larger than the torque created by the brakes. The lateral forces of the front tyres are applied with a lever arm which is the distance between the front axle and the centre of gravity of the car, which is larger than that of the ESP which equals to half the track width, see Figure 7.

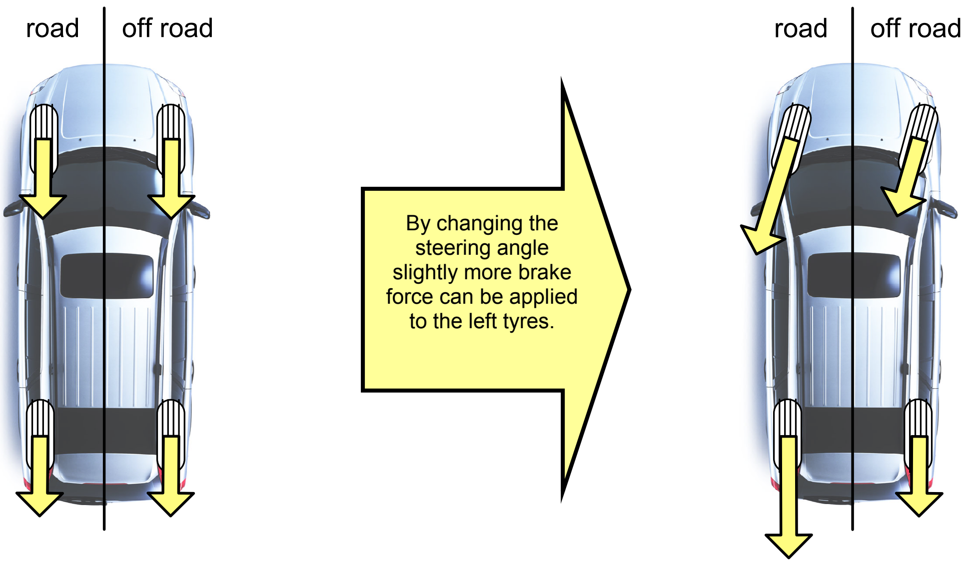

Figure 8: With a small steering angle more brake torque can be applied.

+ It can help reducing the braking distance in a situation with asymmetrical distribution of friction, for example when two tyres are off road. This can lead to a too high yaw moment making the vehicle spin. Normally the lowest force connected to the lowest friction coefficient must be chosen for all wheels in this situation. By changing the wheel angle more brake force can be applied to the left tyres, see Figure 8.

In order to be able to autonomously modify the steering angle of the front tyres, an extra degree of freedom in the steering system is added. This breaks the otherwise fixed relation between the wheels orientation and the steering wheel position.

This degree of freedom permits active systems to add or subtract a certain steering angle to the one requested by the driver. Direct mechanical connection between the steering wheel and the uprights still remains though, as no kind of drive-by-wire systems are still allowed by legislation.

This system provides obvious advantages:

+ It has full and accurate control of the change in the steering angle, without relying on the reaction of the driver.

+ Allows for other functions such as variable steering ratio, which can be used to simplify parking and low-speed maneuvers and reduce steering sensitivity when cruising at high speed.

Its main disadvantage is its cost, which has so far limited its use to expensive vehicles: it needs an exclusive electric motor and other parts which make it more complex and therefore expensive.

Driver steering recommendation (DSR)

Driver Steering Recommendation takes advantage of the hardware of Electric Power Assisted Steering systems (EPAS). The objective of the DSR system is to recommend the driver to steer correct without forcing them; they should always be able to override the applied torque. Its purpose is to use the torque generated by its electric motor not only for easing operation on the steering wheel, but also influencing and improving the driver reaction.

In comparison with Active Steering, explained below, the DSR system has some advantages:

+ It has a lower cost as no extra hardware is needed, provided that the car is already fitted with EPAS.

+ It can actually influence in driver actions, which allows for useful synergies with other active systems such as Lane Keeping Assistance [11], [12].

Unfortunately, it has some disadvantages:

- DSR has the driver in the loop and the effectiveness relies on the reaction of the driver which introduces a big uncertainty. It is hard to determine whether a surprised and unskilled driver will let go and follow the systems suggestion or interpret the change in steering torque as a natural reaction of the car which has to be overcome in order to impose their will.

- As a consequence of the latter, DSR systems are harder to tune than Active Steering systems. Margin for a great variability in reactions must be considered and a solution that will not lead to a worse driver operation on the steering wheel must be selected, see Figure 9.

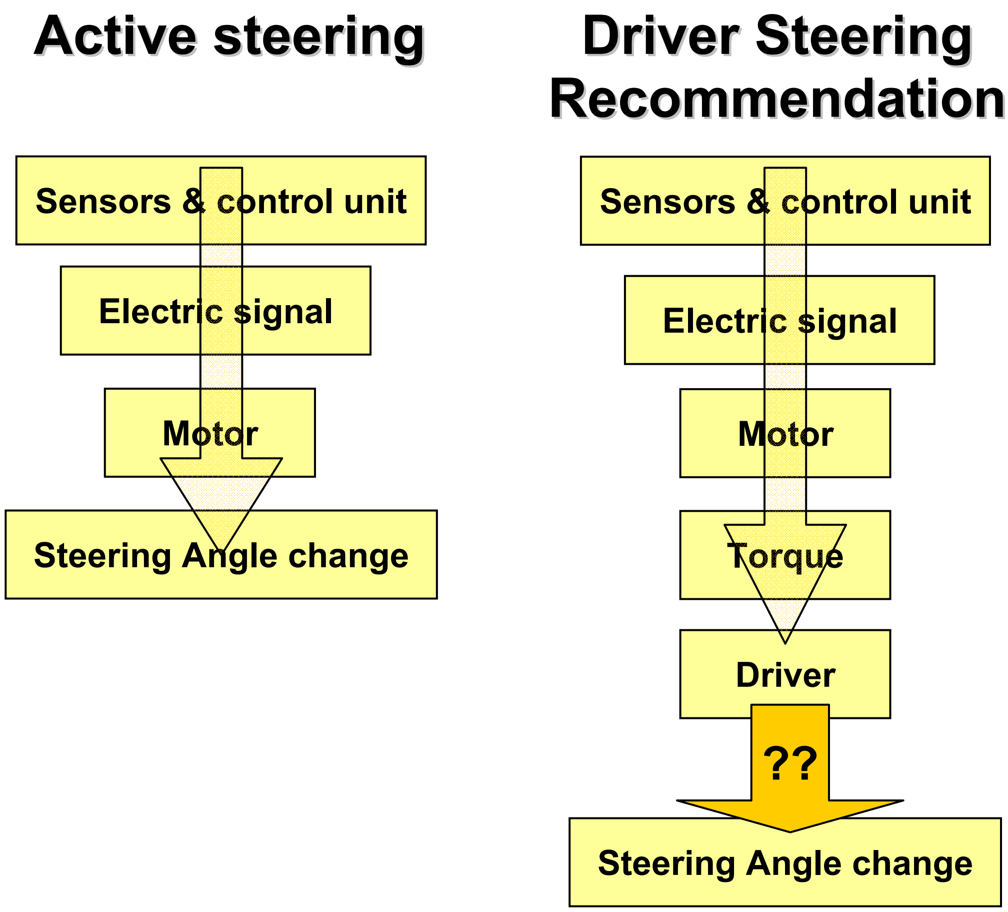

Figure 9: Operational comparison between Active Steering and Driver Steering Recommendation.

Different opinions exist if DSR is an Active Steering system or not. If the only requirement for a system to be considered an Active Steering device is to be able to influence on the steering angle by any means, DSR would be one of them. If the fact that it has to be able to do it autonomously, it wouldn't. DSR tries to affect the driver's inputs on the steering wheel, but the front tyres will always follow their orders, as no active hardware exists on DSR systems.

1.2 Active steering and driver steering recommendation in the market

Both Active Steering and DSR systems have been available in the market for a few years now. The following is a brief description of different models equipped with at least one of them. Unfortunately accurate technical data for these systems is not available.

Models with active steering

The first model to fit Active Steering was the Toyota HDJ 100 [26], the system was only offered in Japan or at least a limited number of markets.

The first European model to offer Active Steering was the 2003 BMW 5 [27]. The system was manufactured by ZF-Lenksysteme and was focused on comfort rather than safety. It provided a variable steering ratio which made low-speed maneuver fast and easy, and gave a high-speed stability.

The active safety side of the system came from its ability to steer front wheels in order to compensate for a wrong attitude of the car, but the amount in which it could modify wheels' orientation was most likely no bigger than 1º [26].

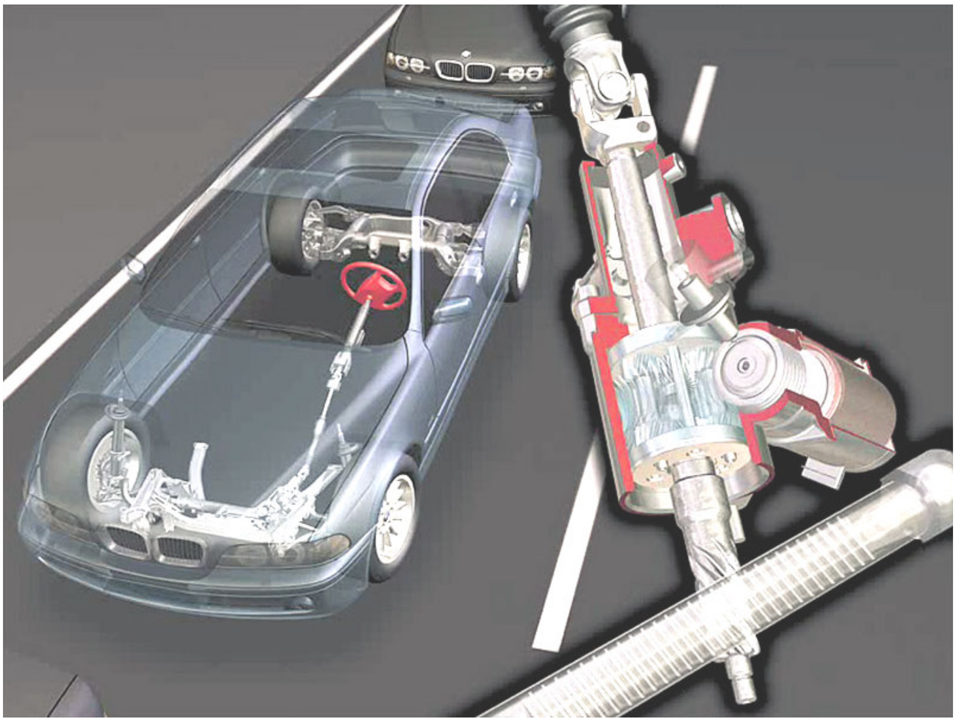

Figure 10: BMW 5-Series Active Steering System.

A newer model fitting active steering is the 2004 Lexus GS, which is able to steer the front wheel up to 3º to each direction. For better visualisation it can be thought that with a usual steering ratio of 1:16, this is equivalent to a rotation of the steering wheel of one fourth of a revolution. The system also fits DSR strategies.

1.2.1 Models with driver steering recommendation system

The first model which fitted a DSR system, at least in Europe, was the second generation of the Toyota Prius. Its purpose was to improve the stability of the vehicle by applying a torque of 2 Nm to the steering wheel. No information on when this torque is applied can be found in the literature.

Other models rely on Bosch's ESP version 8 to fit this capability [17]. Today Volkswagen AG group is offering it as an option in Volkswagen Golf, Jetta and Passat, the Seat Leon and the Skoda Octavia. As in the case of the Prius, it seems to be able to apply up to 2 Nm to the steering wheel.

1.3 Objective

The objective of this project is to assess the feasibility of helping the driver by means of controlling the torque applied by an Electric Power Steering System (EPAS, [22]) to the steering wheel, in the situation in which the car leaves the road as a result of the distraction or sleepiness of the driver.

The developed functions could not be tested using off-line simulations because of the fact that the effectiveness of the functions depended on the driver reaction. Due to the difficulty and danger of testing such a situation in an open road, the main tool used in the project was the driving simulator developed and built at Chalmers University of Technology. The simulator would allow for safe testing of the different functions developed.

When developing such a function several ideas have to be kept in mind:

- The driver must always be able to overcome the system's actuation and impose their will. It could happen that the driver is actually trying to leave the road in order to avoid a worse accident – for instance, a collision with a car stopped in front of them, fact that such a function is not able to recognise.

- As a consequence of the later, the torque that the system can apply should be low enough for all drivers being able to overcome it.

The project was aimed at answering questions such as:

- Is such a function really useful in roadside interaction situation?

- How does the driver experience such an intervention on the wheel? Will they get scared and counteract in order to overcome it? Or will they actually recognise it as something useful which could actually guide them to control the car?

- If helpful, will the driver need any kind of training to take advantage of the function or will they unconsciously follow its guidance without even knowing about its existence?

In other words, the desired output of the project was not a complex and fully developed function, capable of adapting itself to different situations and driving styles. This task is left for further work.

Instead, the main objective was to provide Volvo Cars with a tool for easy and safe assessment whether such a function would be really useful in a specific situation, collecting as much information as possible about how the driver would feel such an intervention. All this should be useful for deciding if further development towards final implementation is worth it and, if so, outline some guidance how to continue the work.

1.4 The simulator as the main tool for the project

The Chalmers driving simulator is the main tool available in order to test the functions developed in a safe and cost effective way.

A vehicle model to be used in the simulator was implemented during this thesis. A comprehensive description of the model is included in Appendix A. Together with all the comments included in the model, it should allow for anybody involved in further development to understand it and carry on with work from where this project has left it.

1.5 Report outline

A brief introduction to vehicle dynamics is included in Chapter 2.

Chapter 3 includes an analysis of the roadside recovery problem and definition of suitable safety functions for helping the driver react in a correct way.

Chapter 4 contains a general description of the physical model developed for the simulator during this project. The model together with active systems developed during the project is validated against data from Volvo Cars simulations. A more in-depth description of the vehicle model, meant for those who plan to modify it, can be found in Appendix A.

Chapter 5 describes all the driver steering recommendation strategies implemented during the project followed by comments on their functionality, effectiveness and how they felt when tried in the simulator. Two of them are finally selected as the most effective and offered for different people to try. Feedback from these people and personal experience of the authors are included in this chapter.

In appendix B a short description of the Chalmers driving simulator together with a start-up guide can be found.

Appendix C includes a table of commercial vehicle models more or less suitable for the Chalmers driving simulator.

2 Vehicle Dynamics theory

In the following chapter the basic vehicle dynamics concepts used in the project are presented. Such as coordinate systems, angles, rotations, tyre dynamics and basic concepts.

2.1 Coordinate systems

A number of different coordinate systems are used to model a vehicle. In this report the ISO coordinate systems are used which are based on the following seven coordinate systems.

- **Earth.** The global coordinate system, *X, Y, Z*.
  The only coordinate system that does not move during travel. It is used as the reference for the position of the vehicle.

- **Vehicle**. The centre of gravity coordinate system, *x, y, z*.
  The origin of this coordinate system is fixed to the centre of gravity of the vehicle. The *x*-axis is parallel to the longitudinal axis of the vehicle and points to the front, see Figure 11.

- **Path**. The velocity coordinate system, x_p, y_p, z_p.
  The origin of the velocity coordinate system is located in the centre of gravity of the vehicle. The difference from the centre of gravity coordinate system is that the *x*-axis follows the velocity vector of the vehicle.

- **Wheel.** The wheel coordinate system, x_w, y_w, z_w.
  Each wheel has one coordinate system. The origin is located in the centre of each wheel and the *x*-axis follows the heading of the wheel.

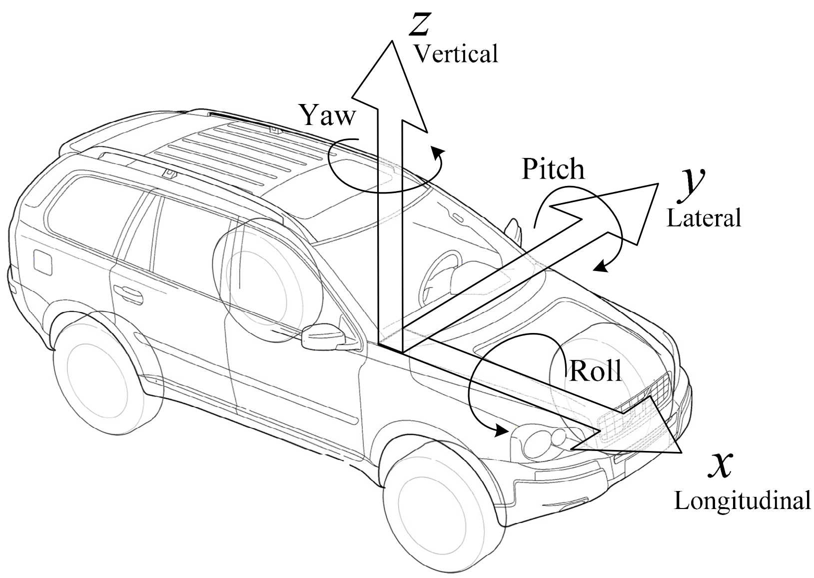

Figure 11: Coordinate system and rotations of the vehicle.

2.2 Roll, pitch and yaw of the vehicle body

Roll, pitch and yaw are the rotations about the three axes of the centre of gravity coordinate system. All three rotations are shown in Figure 11.

- Roll is the rotation about the longitudinal axis, x. Roll can be felt during lateral acceleration, for example during a turn or a lane change.
- Pitch is the rotation about the transversal axis, y. It can be felt when the car is braking or accelerating.
- Yaw is the rotation about the vertical axis, z. This rotation can be felt during cornering or skidding.

2.3 Terminology used in the report

For the understanding of the rest of this report some vehicle dynamics terminology is explained below. Figure 12 depicts the most important geometries and angles.

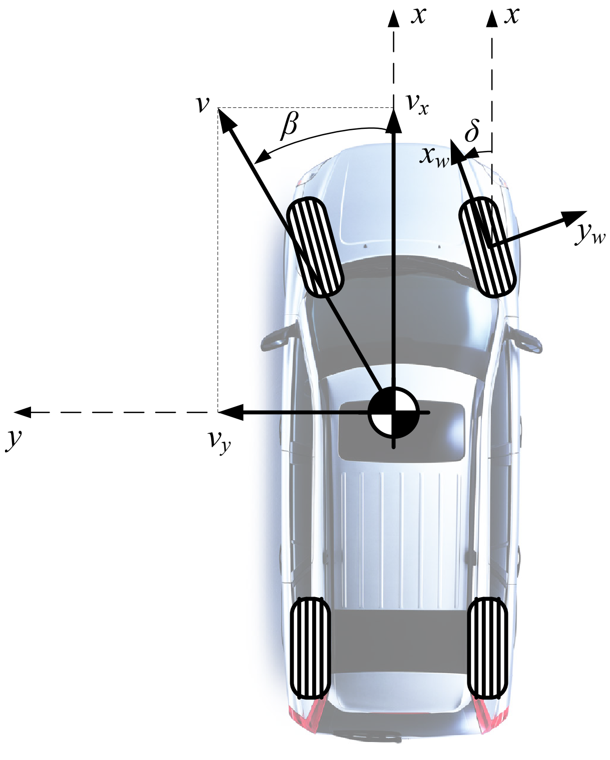

Figure 12: The centre of gravity coordinate system and the main quantities.

Side slip angle

The side slip angle, β, is the angle between the fixed x-axis of the centre of gravity coordinate system and the x_p-axis of the path coordinate system. The angle can be used as a measure of how much the vehicle is sliding and is therefore important in the active safety area.

Steering wheel angle
The steering wheel angle, δ, is the angle between the x_w-axis of the wheel and the x-axis of the centre of gravity coordinate system.

Under-steering
The term under-steering is defined in this report as: the vehicle develops less side slip angle than intended by the driver when turning the steering wheel. This means that the car turns less than intended.

Over-steering
The term over-steering is defined in this report as: the vehicle develops more side slip angle than intended by the driver when turning the steering wheel. This means that the car turns more than intended.

2.4 Tyre modelling

Since the tyres are the only contact between road and vehicle it is here forces have to be generated in order to influence the motion of the vehicle. These forces also provide information to the driver about vehicle behaviour and road condition through the steering wheel. How these forces are developed is complex and combines many components. The most basic ones are explained below.

2.4.1 Longitudinal forces

When accelerating and braking the tyre must deform to develop the desired force in the contact patch between tyre and road, this can only be accomplished if there is a difference in velocity between the tyre and road. This difference is called longitudinal slip or slip rate, s, and can be defined as a dimensionless relation between the rolling speed of the tyre and its speed of vehicle. Two different equation is used in the model, one for acceleration, $\omega \cdot R > v$, and one for braking, $\omega \cdot R < v$, equation 1.

$$s = \begin{cases} \dfrac{\omega \cdot R - v}{\omega \cdot R} & , \text{if } \omega \cdot R > v, \\[2ex] \dfrac{\omega \cdot R - v}{|v|} & , \text{if } \omega \cdot R < v, \end{cases} \tag{1}$$

where ω is the rotational velocity of the tyre, v the velocity of the vehicle and R the radius of the tyre.

When the slip is non-zero the tread elements in the tyre will deform and produce a force that will make the vehicle move.

When the slip increases the force will also increase until it reaches the maximum force, see Figure 13. The slip value where maximum longitudinal force is produced depends mostly on the construction and compound of the tyre, the road condition and the vertical force applied on it. Common values for slip rates for maximum force on dry asphalt are in the region of 0.15-0.2 [16].

For low slip rates, the relation between force and slip is linear and its slope is called longitudinal tyre stiffness. As slip increases the longitudinal force increases until it reaches its maximum. The reason for the increasing force is that more and more of the thread element will be deformed and help creating the longitudinal force. After the maximum the longitudinal force decreases because the elements starts to become saturated and unable to develop more force until all elements are saturated and the tyre is locked, slip value equals one.

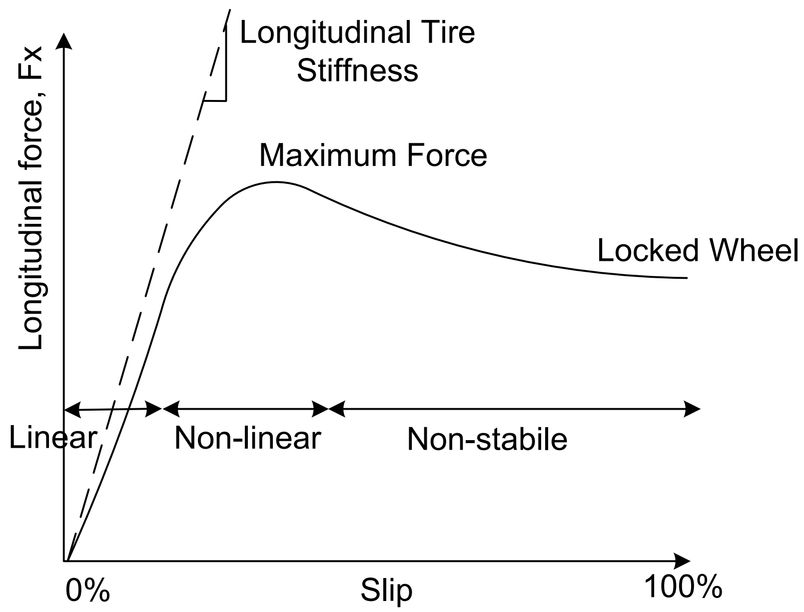

Figure 13: Longitudinal force vs. slip in the tire coordinate system.

2.4.2 Lateral forces

Similarly to longitudinal slip, lateral slip or slip angle, α, can be defined as the angle formed by the *x*-vector of the tyre coordinate system and its own velocity vector. This angle develops, for instance, when the driver steers and forces the tyre to have a different heading than the travel direction of the vehicle.

As slip increases the lateral force will build up until it reaches its maximum in the same way as the lateral force. In the lateral case the initial slope of the force vs. slip graph is called the cornering stiffness.

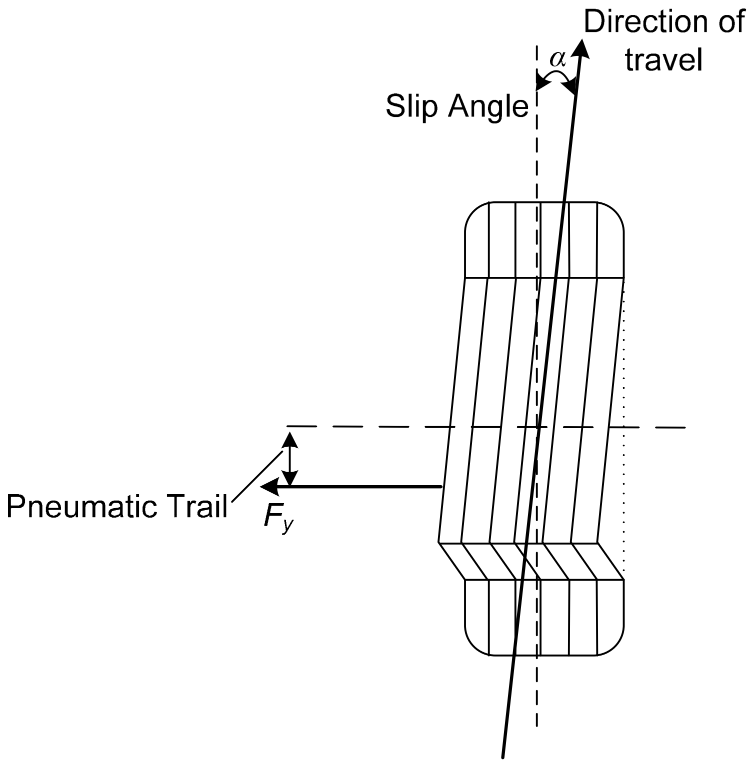

Figure 14: Lateral slip and lateral force in the tire coordinate system.

The slip angle can be expressed as:

$$\alpha = \arctan \frac{V_y}{|V_x|}, \tag{2}$$

where V_x and V_y are the components of the velocity of the tyre projected on the tyre's coordinate system.

2.4.3 Combined longitudinal and lateral slip

There is a maximum traction that a tyre can generate. No matter if this force is pointed in longitudinal, lateral or combined the maximum available force remains approximately the same. Therefore it is not possible to extract both maximum longitudinal and maximum lateral force simultaneously from one tyre. This is called the friction circle or ellipse and is the reason of phenomena such as a vehicle loosing its steerability when braking hard or a rear wheel drive car over-steering when accelerating at the exit of a corner. Figure 15 shows how the longitudinal and lateral forces interact.

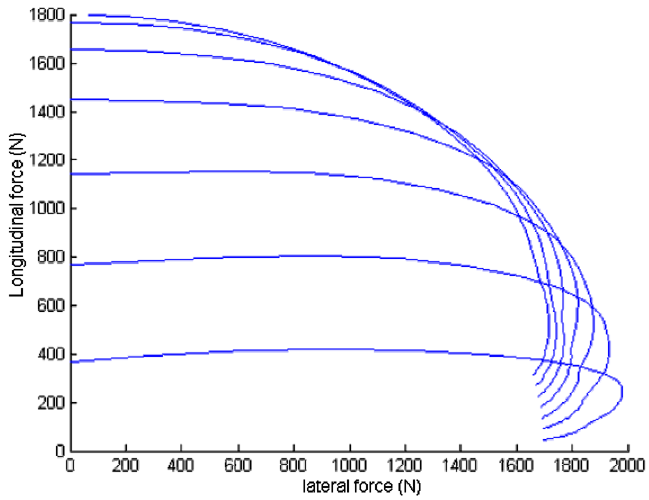

Figure 15: Plot showing combined lateral and longitudinal force.

2.4.4 Self aligning torque

The resulting lateral force is not centred to the hub of the tyre or the steering axis (the one about which the tyre is steered, defined by the linking points of the upright). The longitudinal distance between the resulting force and hub's vertical projection is called the pneumatic trail, *n*, see Figure 14, and acts as a lever that creates the self aligning torque, *Mz* [2]. The self aligning torque is felt in the steering wheel and will in every normal situation always attempt to steer the tyre in the direction it is travelling [1]. That is, the driver will always feel a torque tending to self-align the steering wheel.

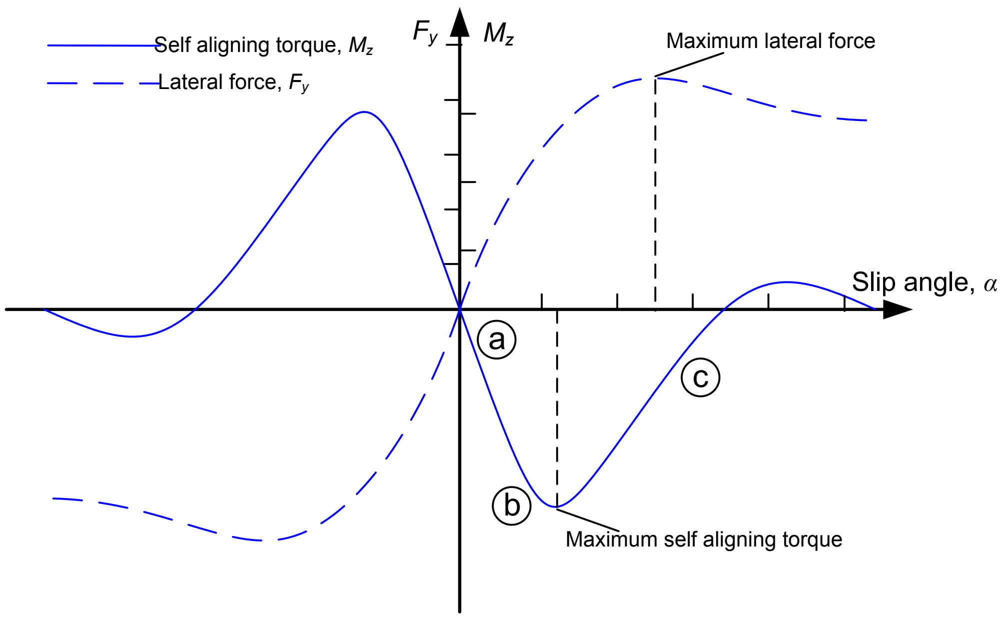

Figure 16: Lateral force & self aligning torque vs. tyre slip angle in the tire coordinate system. a, b and c refers to figure 17.

Figure 16 shows how the lateral tyre force and the self aligning torque depend on the slip angle. It shows that the maximum self aligning torque occur for a lower slip angle than the maximum lateral force. This means that before maximum lateral force is reached a decrease in self aligning torque will occur. An experienced driver can take advantage of this to drive close to the handling limits of the vehicle.

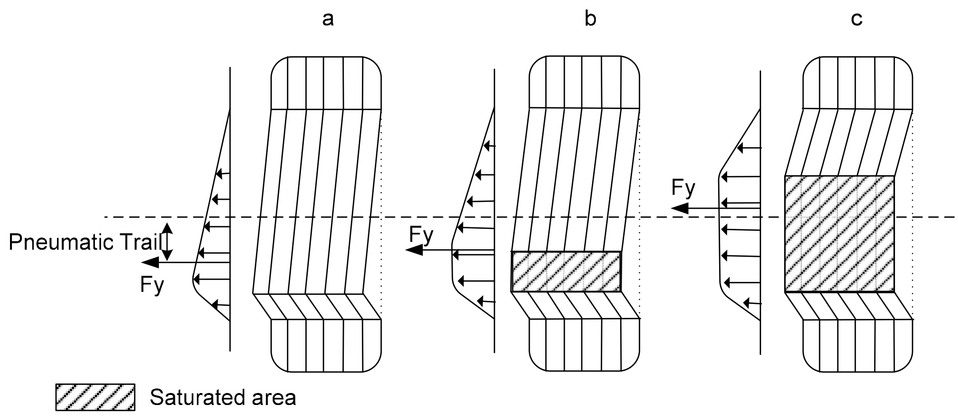

Figure 17: Lateral slip angle and the resulting force in the tire coordinate system.

The reason for the decrease of torque before reaching maximum force is shown in Figure 17. The self aligning torque can be calculated as:

$$M_z = F_y \cdot n. \tag{2}$$

Both the resulting force, F_y, and the pneumatic trail, n, change with the tyre slip angle. When the slip is increasing from zero, Figure 17a, the resulting force increases. The tyre is deforming more and more. This makes the self aligning torque increase fast.

In Figure 17b the slip has increased and the deformed patch have become saturated which forces a larger part of the tyre to deform. The pneumatic trail is starting to decrease and the maximum self aligning torque will soon be reached. However there is still more force available from the tyre that can be accessed by deforming a larger part of the tyre. Since both the pneumatic trail and the resulting force are changing the curve will now become nonlinear. When this happens the pneumatic trail will decrease resulting in a decreasing self aligning torque that even can become negative for very high slip values, Figure 17c. After maximum lateral force has been reached, both force itself and pneumatic trail decrease. As a result, self-aligning torque decreases much, with the possibility of even becoming negative.

2.5 Bicycle model

In many cases a linear vehicle model of is a very good approximation of the dynamics of the vehicle [1]. One reason that makes the model linear is that a linear tyre model is used. This means that only the linear zone of the slip/force curve, see Figure 13, is considered in calculations of tyre forces. The bicycle model is commonly used in controllers for example in the ESP. It can also be a simple way to understand the basics of vehicle modelling.

In the following section the equations of the bicycle model is presented. Figure 18 shows the bicycle model and the basic quantities used.

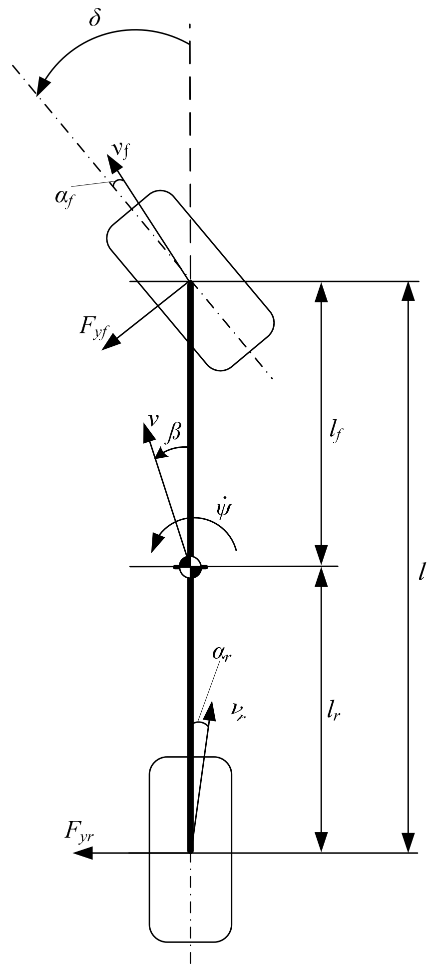

Figure 18: The Bicycle model.

First the forces from the tyres are calculated. Since the model is linear the cornering stiffness is the only parameter used to model the tyre. The approximation will give a god value of the tyre forces for small tyre angles.

$$F_{yf} = C_f \cdot \alpha_f \qquad \text{, front lateral forces,}$$

$$F_{yr} = C_r \cdot \alpha_r \qquad \text{, rear lateral force.} \tag{3}$$

To calculate the side forces the slip angles need to be known. They can be calculated using the steering angle, δ, side slip angle, β, yaw rate, $\dot{\psi}$, and the speed, *v*, of the vehicle as follows:

$$\alpha_f = \delta - \beta - \frac{\dot{\psi} \cdot l_f}{v} \qquad \text{, front slip angles,}$$

$$\alpha_r = -\beta + \frac{\dot{\psi} \cdot l_r}{v} \qquad \text{, rear slip angle.} \tag{4}$$

The tyre forces are then used to calculate the torque and force balance of the complete vehicle. Since it is a linear model and the angles are relatively small the forces are added assuming $\cos(\varphi) = 1$ and $\sin(\varphi) = 0$.

$$\sum T = 0 \Rightarrow J_z \cdot \ddot{\psi} = F_{yf} \cdot l_f - F_{yr} \cdot l_r \qquad \text{, torque balance,}$$

$$\sum F_y = 0 \Rightarrow m \cdot v \cdot \left(\dot{\psi} + \dot{\beta}\right) = F_{yf} + F_{yr} \qquad \text{, lateral force balance.} \tag{5}$$

3 Analysis of the Roadside Departure Accident

This chapter gives a general background for the development of assisting strategies for the road departure accident. Possible events towards both complete loose of control or final recovery is covered. Those who are not interested in this can jump directly to Chapter 4 without missing any big contributions of the thesis work. This chapter was considered useful since it was valuable for the authors to work on it and helped them to have a broad view of the problems involved in this kind of accident. It will therefore also be useful for anyone who wants to further improve the strategies yielded by this project. Besides, it would make it easier for such a person to track the logical steps which guided the authors to end up with the proposed function, and therefore help them to better take advantage of the work done so far.

Description of the accident

Figure 19 below shows the course of events, different possible reactions of the driver and the actuation of active systems like yaw control by brake (ESP) and Driver Steering Recommendation (DSR). Rectangles represent events, situations or states; ellipses actuation of the two active safety systems and arrows reactions by either driver or car. The scenario considers an accident where the vehicle leaves the road on the right hand side in right hand side traffic. No cases involving objects or meeting cars are considered.

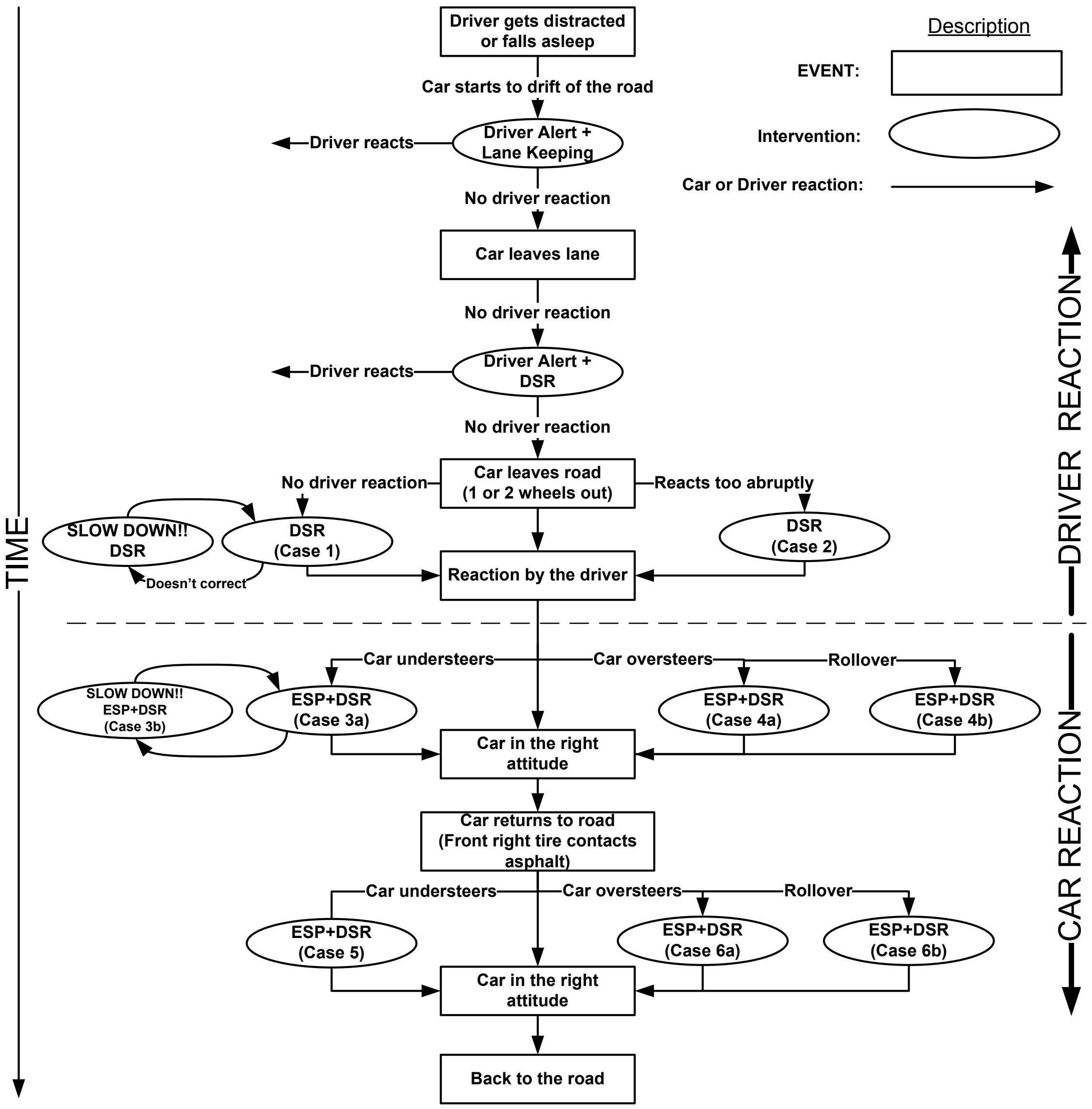

Figure 19: The accident scenario and possible interventions.

Two main stages can be distinguished, indicated by the arrows on the right side of Figure 19. In a few words: first the driver should react correctly (react without overreacting). Then the car should react correctly to what the driver asks for.

1. Driver reaction. In the first stage the driver should react correctly to the situation in which the car starts to leave the road. This stage may extend a few milliseconds, see Figure 2, before the car has developed a wrong attitude (under-steering / over-steering / rollover). This is why the ESP is not considered in this stage.

2. Car reaction. The second stage involves trying to maintain the car in the correct attitude. This means to prevent the car from sliding, spinning or rolling over. In this stage the ESP can help keeping its path, but its help may be limited. The fact that one or more tyres is off the road on a low friction surface prevents the ESP from braking the off-road tyres as much as would be needed to correct the vehicle path.

3.1 Possible countermeasures

In the following section each intervention, ellipses labelled (Case X) in Figure 19, are analysed and discussed.

Figure 20 shows how the assisting functions that should help the driver to steer in a correct way are presented. The same axle is used to represent two related but different concepts: steering angle and tyre slip angle. Same goes for the axel representing both self aligning torque and lateral force. It is done so to display the conceptual characteristics behind the functions in a simple way, even if these two variables represent different, thought related, magnitudes. Numbers in the figures are referred to in the text. The proposed functions are shown with dotted lines which represents the resulting torque. That means the torque that the driver should feel in the steering wheel (sum of proposed function and self aligning torque).

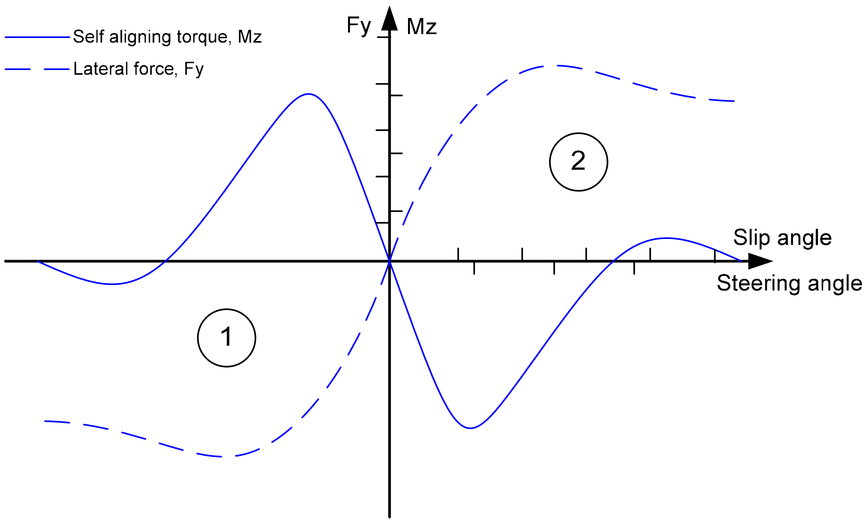

Figure 20: lateral force and self aligning torque.

Point 1. If the driver steers to the right (negative *x*-axle) a negative lateral force is created in the front wheels and a positive opposing self aligning torque will be felt in the steering wheel.

Point 2. If the driver steers to the left (positive *x*-axle) a positive lateral force is created in the front wheels and a negative opposing self aligning torque will be felt in the steering wheel.

3.1.1 Driver reaction

Before the car leaves the road

First of all, an accident should always, if possible, be prevented well before the driver is involved in the dangerous situation of road departure. Active safety systems currently developed by Volvo Cars such as Driver Alert and Lane Keeping Assist could be effectively applied here [43], [44].

Before the vehicle starts to leave the road the driver is probably tired or not concentrated on the driving task and as a result the path of the vehicle will be worse planned. This could be detected by the Driver Alert-system and the driver could be warned in time.

If the vehicle starts to drift away from the centre of the lane the Lane Keeping Assistance system can detect that and try to prevent this. To help the driver a guiding torque is slowly added to the steering wheel by the system helping the driver not to leave the lane.

However, these systems can fail. Snow or dirt can cover the camera and lane markings or maybe the road does not have any lane markings. The roadside can not be detected in a reliable way by any system existing today [44].

Car leaves road

Case 1: No driver reaction

The car has started to leave the road but the driver does not react. One or two wheels are off the road. This means that both the Lane Keeping Assist and Driver Alert have failed and that the vehicle now needs to recover in a safe way.

The driver should steer to the left to go back to the road. A counter clockwise torque should be applied in the steering wheel, see point 1 in Figure 21, to help the driver steer back to the road. This help should tend to disappear soon after the driver has started to react to prevent over reaction from the driver. In this case vibrations in the steering wheel could also help the driver to wake up. It could happen that normal vibrations, consequence of the tyre rolling on soil, are enough to make the driver recover their awareness. If the driver doesn't react, the helping torque should be increased over time.

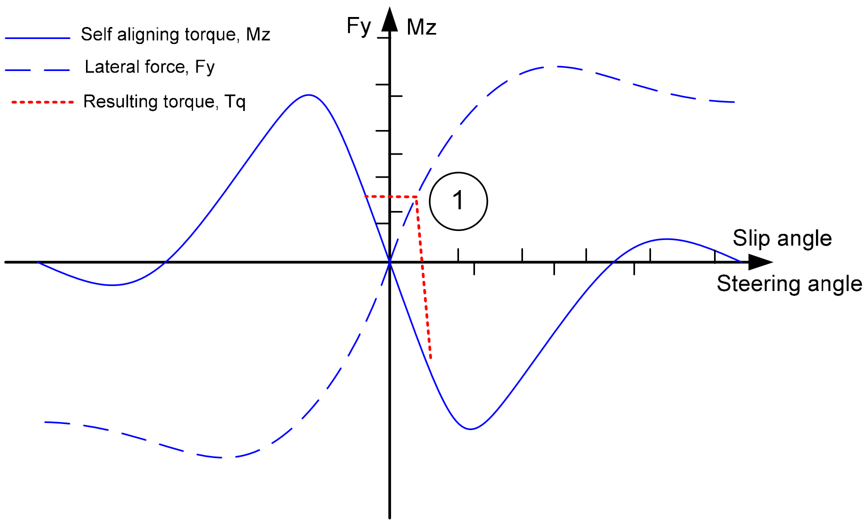

Figure 21: When the driver does not react, a torque, point 1, should guide them to react correctly.

Case 2: Reacts too abruptly

The car has started to leave the road. The driver suddenly realises this, overreacts and steers too much.

This reaction is both useless and erroneous. Useless, because it is probably inducing a too high tyre slip angle, exceeding the one for which maximum lateral force is achieved. Erroneous, because it might be exerting a too high step increase in front axle lateral force which may lead to over-steering. In order to mitigate this reaction a resisting torque should make the steering harder, see point 1 in Figure 22. This would be felt by driver as an indication that what they are doing is wrong.

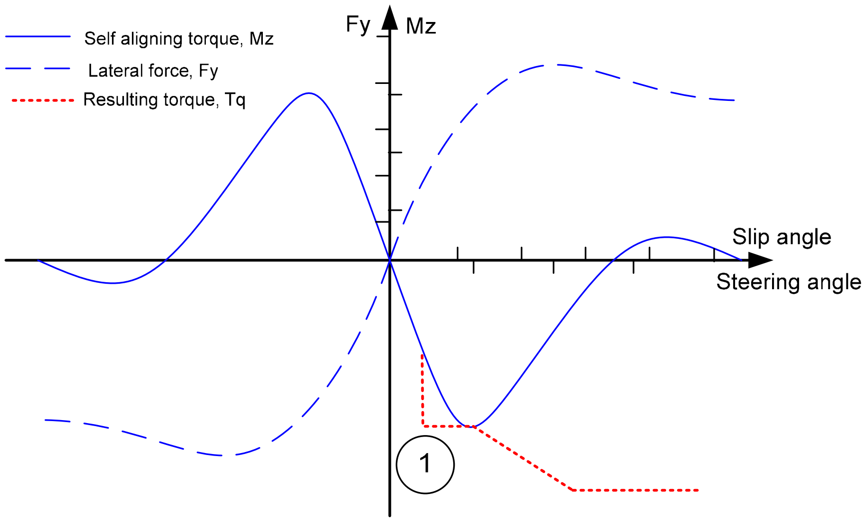

Figure 22: If the driver reacts too abruptly a resisting torque should guide them to steer less.

3.1.2 Car reaction

Case 3: Car under-steers

The car under-steers for some reason; there are two scenarios found that can lead to this situation which could be corrected.

Case 3a: Steering too much

One of these reasons is that the driver is steering too much leading to too high slip angles in the front wheels. To make the driver steer less an increased helping torque should appear in the steering wheel for too high steering angle; see point 1 in Figure 23. The helping torque should increase as time passes without correction.

ESP can be useful in this situation by braking one or both left tyres, still in the road. This would induce a counter-clockwise yaw moment which would help to re-orientate the car in the direction of the road.

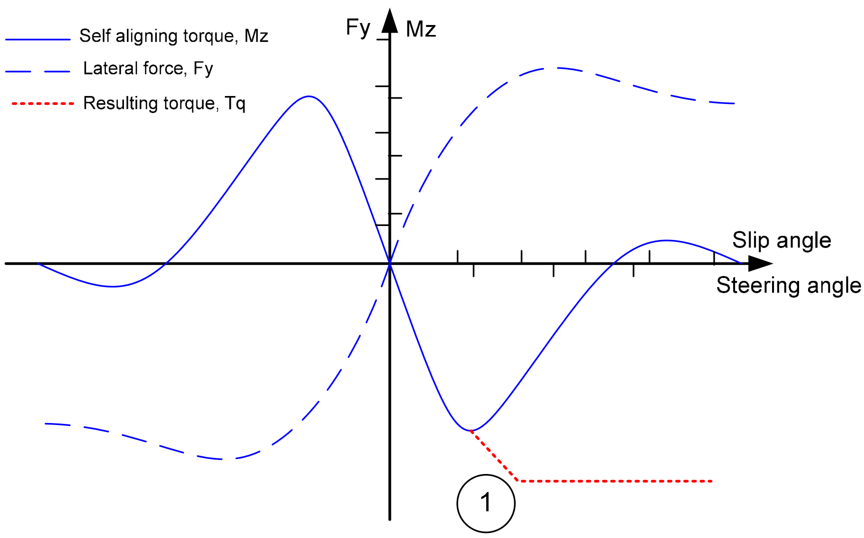

Figure 23: If the driver steers too much a resisting torque should guide them to steer less.

Case 3b: The car is unable to go back to road

The second reason for under-steering to appear is that a certain slope exists on the roadside that prevents the vehicle from going back to the road, even with the maximum lateral force available. The only resource here is to try to slow down the car until it is possible to regain control of the car, or at least to reduce the damage in an eventual collision or rollover.

Maximum braking force should be applied, but the difference in friction between the left and right tyres may lead to a too high yaw moment. More brake force could be applied if the driver steered slightly to the right. This function is already implemented in some vehicles in the market (See Chapter 1.2).

A resisting torque should appear when the driver is steering to the left, and maintained even if they (correctly) starts steering to the right, see Figure 24. Point 1, where the sum of torques changes sign, will be the recommended steering wheel angle. This would allow for greater braking forces in the left tyres and therefore a more efficient deceleration.

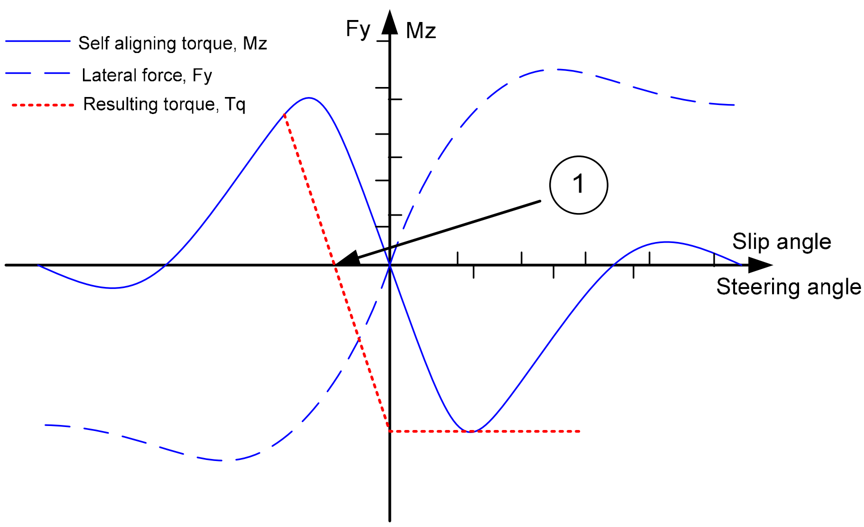

Figure 24: A guiding torque should invite the driver to counter-steer to a correct steering angle, point 1.

Case 4a: Car over-steers

If the car is starting to over-steer there is not much that the ESP can do, as it is the right tyres that should be braked in order to compensate for the over-steering and both of them are on a low friction surface [45]. Lateral force exerted by front tyres should be reduced. In other words, driver should steer less. A resisting torque at high slip/steering angles should help the driver to steer less, see point 1 in Figure 25.

If the over-steering becomes worse, that is, a too large side slip angle is developed, counter-steering should be induced. In this case the helping steering wheel torque should lead the driver to steer to the other direction, see point 2 in Figure 25.

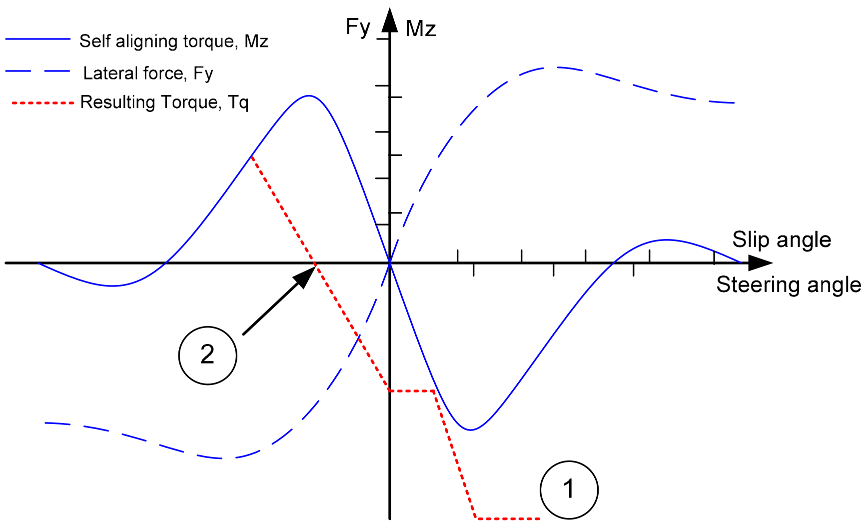

Figure 25: A resisting torque should prevent the driver from steering too much, point 1, and induce counter-steering, 2, if necessary.

Case 4b: Rollover

This is the same situation as above except that the car has developed an even larger side slip angle. Here the same strategy as in case 4a should be applied except using more torque recommending the driver to counter-steer more.

Car returns to road

The vehicle returns to the road and the front right tyre regain contact with asphalt. This will induce a rapid change from low friction to high friction for this tyre resulting in a sudden increase in lateral force. The lateral force will induce a yaw torque capable of rotating the car, see Figure 26.

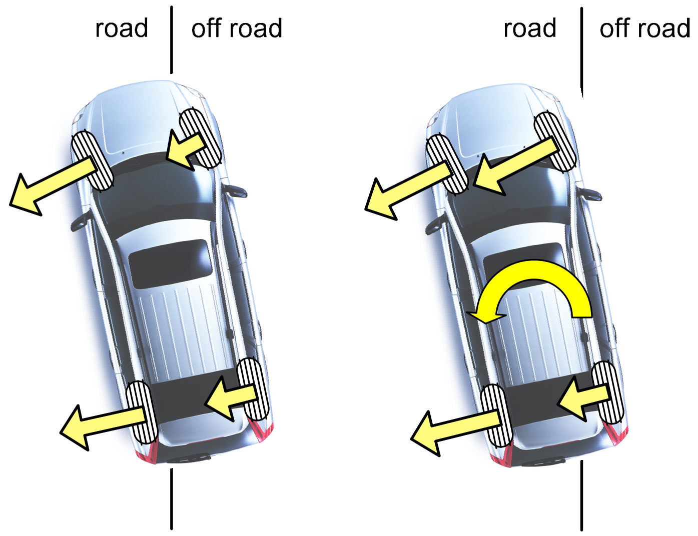

Figure 26: Car moving from low to high friction surface.

Case 5: Car under-steers

The car under-steering in the situation is very unlikely since the lateral force step will help the car rotate even more.

Case 6a: Car over-steers

This increase in yaw torque could be especially critical if the side slip angle of the car already is large. In this situation that tyre will be subjected to a high slip angle, probably regardless what the driver does with the steering, as the whole car already has a large side slip angle. Therefore it is desirable to keep the side slip angle as low as possible at all times.

Figure 27 shows how the self aligning torque and lateral force changes when the tyre moves from gravel to asphalt. Both lateral force, point 1, and self aligning torque, point 2, will increase fast. The change in right front wheel lateral force and self aligning torque should be as small and smooth as possible. This can be accomplished by reducing the steering angle inducing a counter torque recommending the driver to steer less.

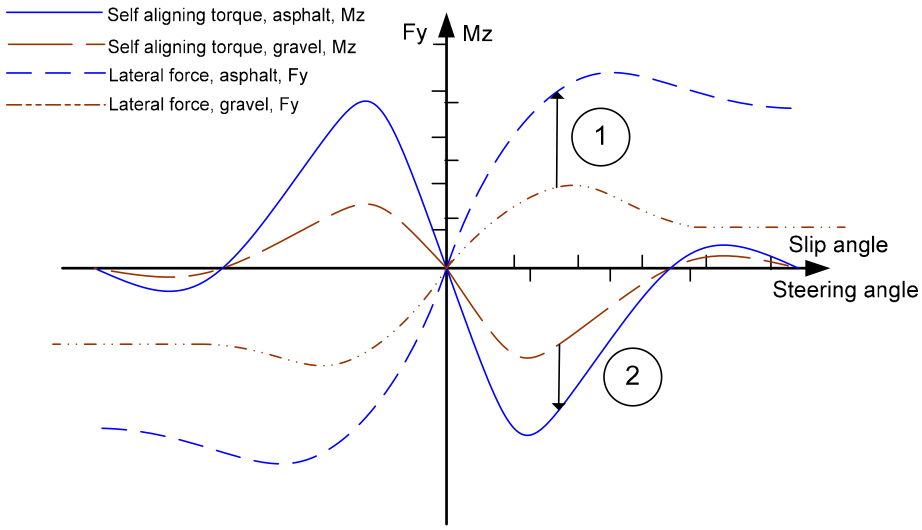

Figure 27: When a tyre moves from gravel to asphalt lateral force, point 1, and self aligning, point 2, torque will increase.

Case 6b: Rollover

This is the same situation as 6a except the car has developed a too large side slip angle. This would probably lead to tripped rollover. What can be done is to recommend the driver to steer less and even counter steer in the same way as case 6a except using more torque recommending the driver to counter-steer more. In general, once that the car has developed enough side slip angle for a tripped rollover to happen, the driver has completely lost all the steerability and is unable to recover the car. Therefore, this situation should be prevented before it happens.

3.1.3 Summary of the countermeasures

From the functionality point of view, many of the proposed strategies above show more similarities than differences. They can be summarised in the following demands for a system implementing them:

1. The system should invite the driver to move the steering wheel to a desired position.

2. The bigger the difference between that desired position and what the driver is doing, the stronger the torque applied by the function should be.

3. The longer the driver takes to react correctly to the situation, the stronger the torque applied should be.

4. If the driver is moving the steering wheel too fast, a torque trying to limit this should be applied.

4 Dynamic Vehicle Model

When the project begun the fact that no working vehicle model for the simulator existed gave two options, either to buy a commercial model or develop one. Therefore a survey on existing commercial model was made, but the lack of economic resources for a model lead to the decision of developing a new vehicle model in **Matlab/Simulink**.

4.1 Dynamic model

The dynamic model is a set of equations that describe the motion of the vehicle according to a series of inputs.

The model covers the whole structure of a vehicle, setting out the interactions of all elements.

4.2 Commercial models

In the beginning of the thesis a survey of different commercial models was performed. In order to integrate the model with the simulator the following demands where set up.

- **Compatible with Simulink**
  The model should be implemented in Simulink or at least be compatible with xPC Target.

- **Tuneable/Flexible model**
  The model should be easy to tune and it should be possible to change entire subsystems, i.e. if a different powertrain is tested. This demands an open structure with as few precompiled S-functions as possible.

- **Real Time**
  The model must be able to be run in real time.

- **xPC Target compatible**
  Since an xPC Target real time card is currently installed in the simulator the model has to be compatible with it. It was not possible to change the real time card during the thesis.

- **Possible to use an extern road profile**
  To be able to run the model together with the existing graphical environment it must be possible to feed it with signals from this environment. This includes individual *Z*-displacement and friction for each tyre.

- **Inputs**
  To control the model from the simulator car the following inputs are needed:
  - Accelerator pedal.
  - Brake pedal.
  - Steering wheel angle.

- **Access to output variables**
  Minimum requirements to make the platform move are:
  - Centre of gravity in *X, Y* and *Z* accelerations.
  - *Roll, Pitch, Yaw* angle of the vehicle.

- **Realistic tyre model**
  The tyre model should be realistic and easy to tune in order make the vehicle behave in a realistic way and reproduce the different characteristics of asphalt and soil.

- **Realistic steering wheel torque**
  The steering wheel torque is needed for the force feedback in the steering wheel.

A list of commercial models which could be interesting for the simulator can be found in appendix C. The most interesting solutions were **veDyna** from **Tesis** because it is currently used by Volvo Cars and **Adams/Car Real Time** where Chalmers have an existing license.

Common for all solution was the high price, around 100 000 SEK, and that none of them had a completely open structure. The **Adams/Car Real Time** model had to be dropped because it was not compatible with xPC Target.

As a result of their high prices, the commercial model approach was discarded.

4.3 The developed vehicle model

The model, developed in Matlab/Simulink to be used in the Chalmers driving simulator, has the following 14 degrees of freedom:

- Translations *X, Y, Z.*
- Rotations *Roll, Pitch, Yaw.*
- Four wheel speeds, ω.
- Four wheel vertical translations, z.

It is supposed to behave like a Volvo XC90. The reason for this is that an XC90 has a high centre of gravity which makes it particularly dangerous in roadside accidents where roll over can occur. Obviously, much of the effort was put on developing the most important sub-systems for the needs of this project, such as the tyres, leaving refinement of others for future development.

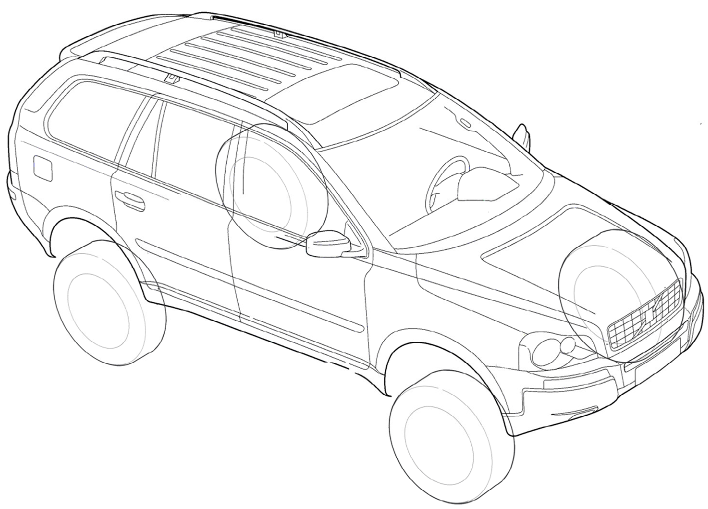

Figure 28: Volvo XC90.

In the model data from the XC90 is used for:
- Springs and dampers.
- Engine.
- Transmission.
- Geometry and mass.
- Power steering.
- Tyres.

In order to validate the interaction between already existing active functions and the functions developed the following functions was developed and integrated in the model.

- ABS.
- Yaw control by brake, ESP.
- Cruise Control.

A complete description of the model can be found in appendix A.

4.4 Validation of the developed model

When a model is used in a vehicle simulator it had to provide a realistic experience for the driver, good enough for them to be able to assess the effectiveness of the steering functions which was developed. Therefore, it wasn't a prime requirement to ensure accurate numerical results. Even though, the model was validated against a commercial package, **veDyna**.

A standard maneuver was simulated and the result compared with simulation data from a commercial model from **Tesis**, www.tesis.com, called **veDyna** with parameters for the Volvo XC90 received from Ford in Dearborn, Michigan.

4.4.1 The fish hook maneuver

The fish hook maneuver is a test developed by NHTSA (National Highway Traffic Safety Administration) that subjects the vehicle to a high speed collision avoidance maneuver. It is shown in Figure 29 and consists of a sharp turn at speeds about 80 kph, followed by an overcorrection of the same magnitude to the other direction. The present Fishhook test developed by NHTSA includes roll rate feedback in order to time the overcorrection to coincide with the maximum roll angle of the vehicle. This is not done in the simulations performed in this report; instead, the steering input used is the steering input received from Volvo Cars. The accelerator pedal is released the second before the steering begins.

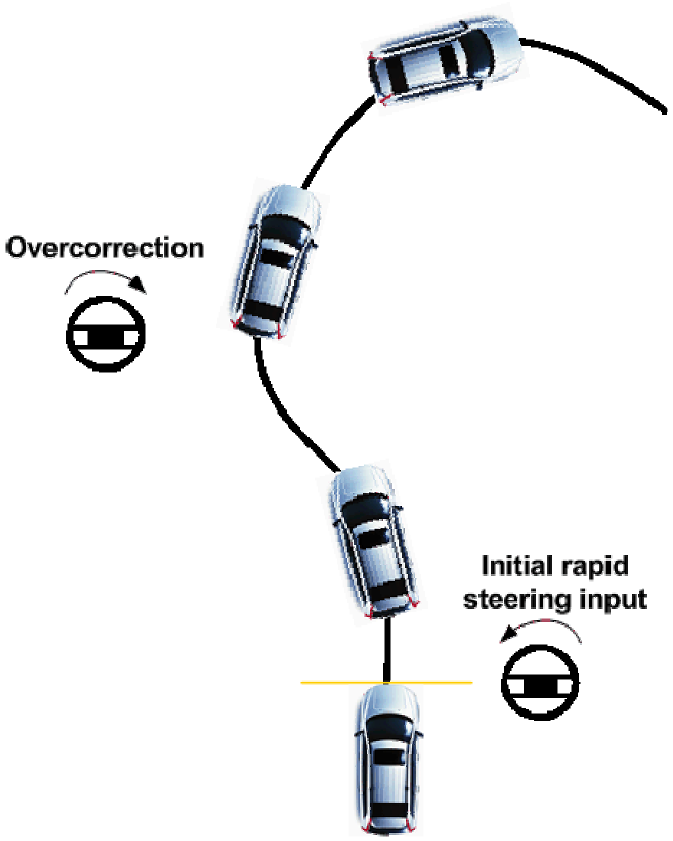

Figure 29: The fish hook maneuver.

The results from simulation are shown in Figure 30. The first plot shows the steering wheel angle input and the following the motion of the vehicle. Dotted lines represent the **veDyna** model with Volvo XC90 parameters from Volvo Cars and solid lines represent the created model. As can be seen the both models react similar to the input in yaw rate and velocity. Small differences can be seen in side slip angle and rotation. The results are still good enough to conclude that the model is accurate and will be suitable for the simulator.

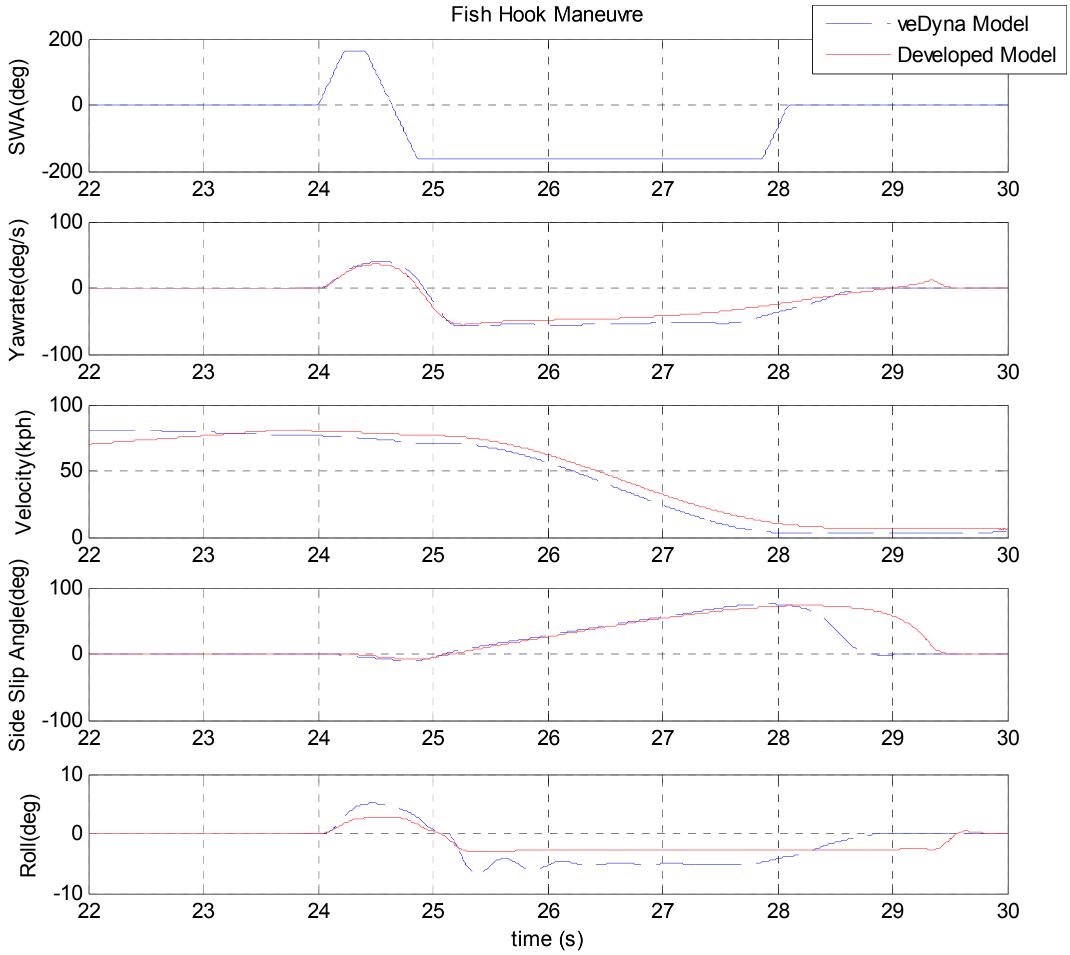

Figure 30: Comparison between veDyna simulation data and the model created.

4.5 Validation of developed active safety systems

4.5.1 Validation of the yaw control by brake model.

To validate the YCbB model the result was compared with results from an advanced yaw control using the Fishhook maneuver. The result is shown in Figure 31. It can be seen that the developed YCbB controls the vehicle in a similar way as the advanced yaw control does on the **veDyna** model. One reason for the differences between the simulations is that the advanced system controls both yaw and roll. The roll is not controlled in the developed system.

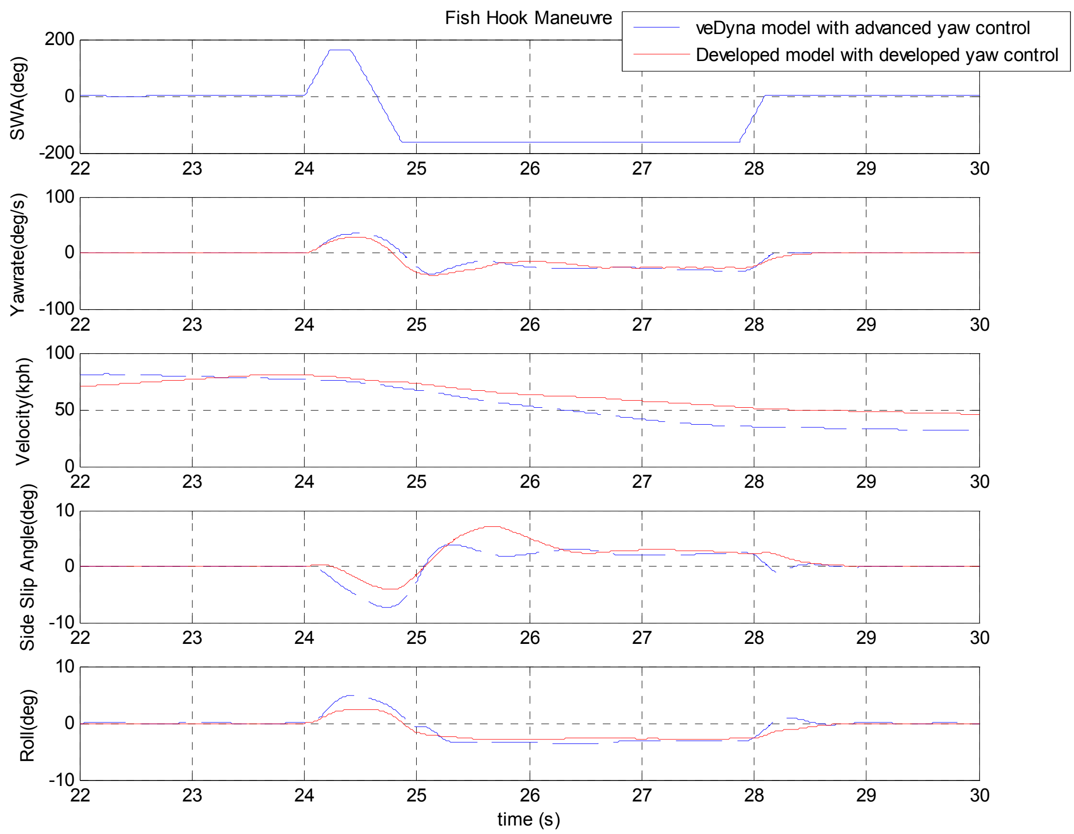

Figure 31: Comparison between advanced yaw control and the developed yaw control.

The brake torque used by the developed YCbB during the fishhook maneuver is shown in Figure 32. The dotted line is the requested brake torque and the solid line is applied brake torque. The difference in requested and applied torque is due to delays in the hydraulic brake system used in cars.

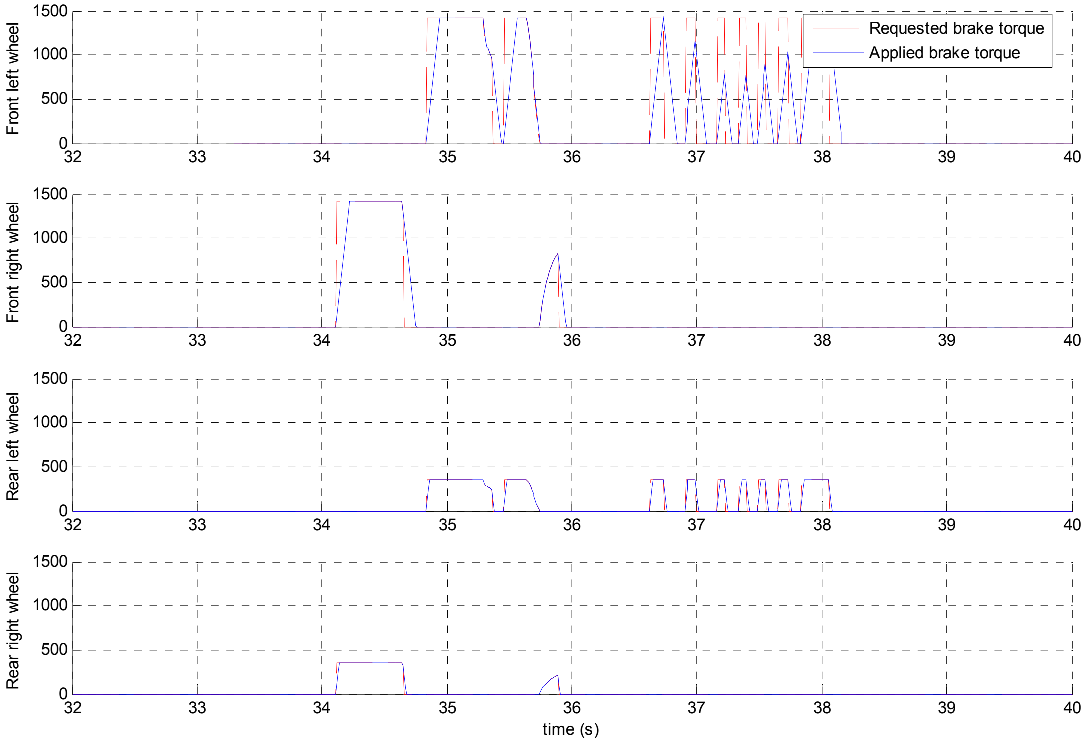

Figure 32: Requested and applied brake torque by the developed YCbB system for each wheel during the fishhook maneuver.

Figure 33 shows the difference in the path between the created model with and without YCbB when the steering wheel input from the fishhook maneuver is applied. Without YCbB, black car, the car starts to spin and completely stops in the end of the simulation. With YCbB, white car, the vehicle maintains both stability and speed.

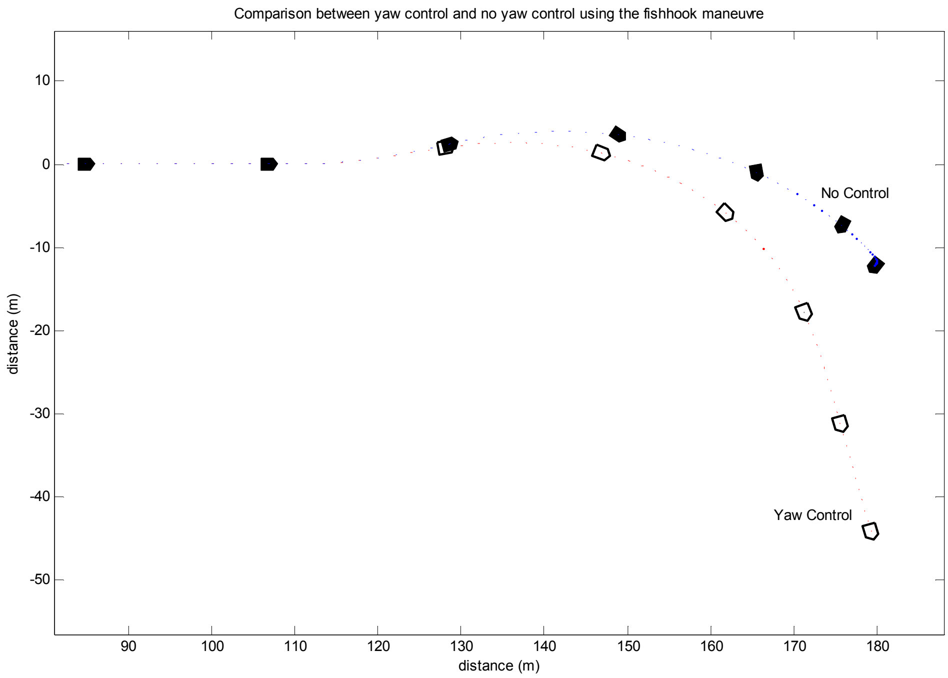

Figure 33: Comparison between vehicle with and without the created yaw control for the fish hook maneuver.

4.5.2 Validation of the antilock brake system model.

Figure 34 shows the wheel speed and wheel slip rate for one wheel during a full brake maneuver from 80 kph to full stop with and without ABS. It can clearly be seen that the slip rate is held to a stable value without locking the wheel. A slip rate equal to minus one represents a locked wheel.

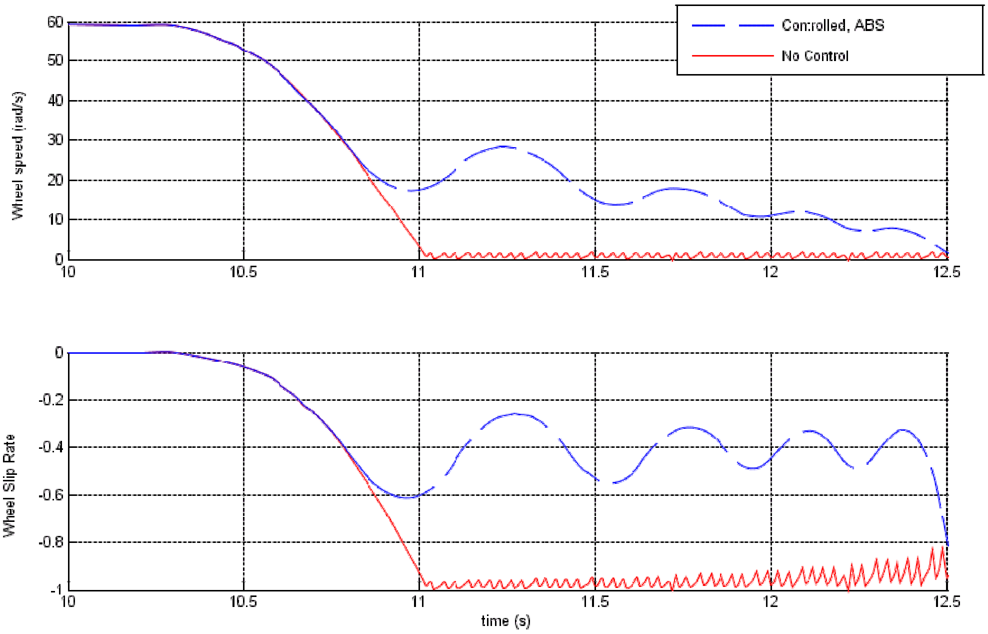

Figure 34: The Wheel speed and Slip rate with and without ABS during full brake from 80 kph.

5 Evaluation of Driver Steering Recommendation functions

Five different strategies were implemented and tested in the simulator during the project. Even though it was clear from the beginning that some of them would be of very limited help, they could give useful feedback and guidance for on-going work.

All the functions share the same structure. They receive one or more signals related to the dynamics of the car, uses it to compute a resulting torque which is applied to the steering wheel helping the driver to control the car. This torque is added to the base steering feel (the torque coming from the tyres) and the power steering, see Figure 35. A detailed description of how the base steering feel is calculated is included in Appendix A.

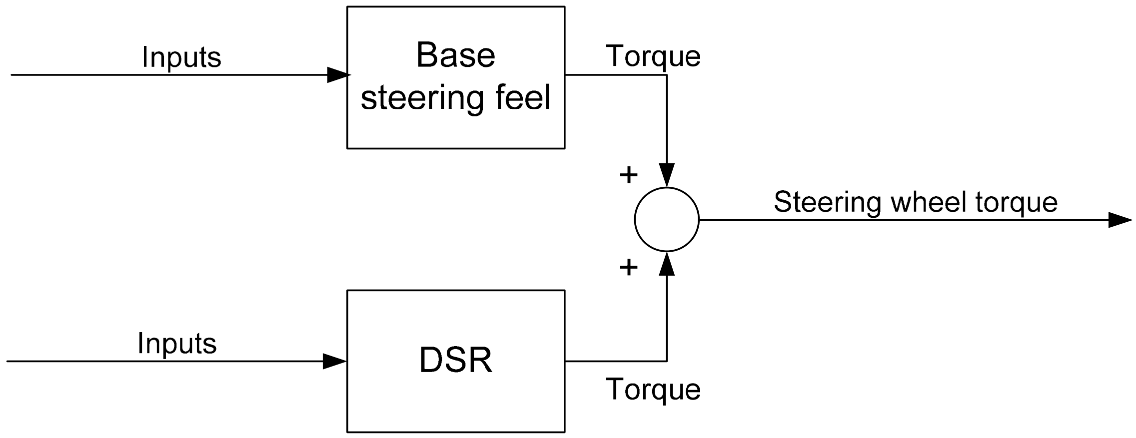

Figure 35: DSR added to the base steering feel.

The proposed strategy is presented in Appendix D.

5.1 Description of the developed functions

In the following section some developed functions will be explained and discussed. The comments included are the authors' and different test drivers' opinions after testing the functions in the simulator. The test drivers were instructed to pretend to fall asleep, suddenly wake up when off the road and then over react by turning the steering wheel too much. Some knew about the functions before driving and some did not know about its existence.

5.1.1 Limiting steering wheel angle.

The steering wheel angle is limited by a steadily increasing resisting torque applied to the steering wheel when it exceeds a certain angle in any direction. Figure 36 shows the basic structure of the function. Even if it is effectively capable of limiting the angle that the driver applies to the steering wheel, it's unable to do it with enough anticipation as it does not take into consideration how fast the driver is working on the wheel. Due to the speed of the steering a resisting torque, applied when a predefined angle is reached, won't be noticed by the driver. Who will not have time to perceive it and only notice it when they stop steering. Besides, it doesn't help the driver to actually control the car once it has started to slide

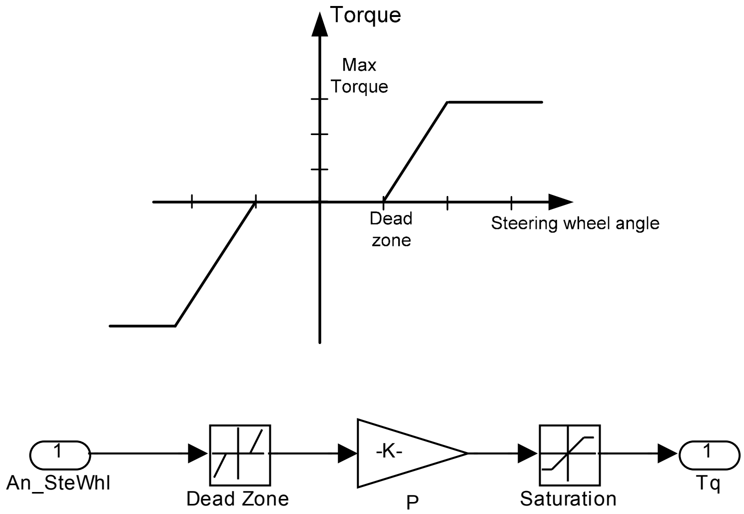

Figure 36: Function limiting steering angle.

5.1.2 Limiting steering wheel rotational speed.

The function limiting steering wheel rotational speed is similar to the previous function, but instead triggered when the rotational speed of the steering wheel exceeding a given value. It was found during simulator tests that it's possible to set a threshold which differentiates panic situations from those in which the driver is just moving the wheel normally fast, even when sensibly counter-steering in order to regain control of the car. It can help to mitigate the very first and probably wild reaction of the driver when they realises they have left the road. Despite this it is not able to guide the driver towards full recovery of a sliding car. The function is shown in Figure 37.

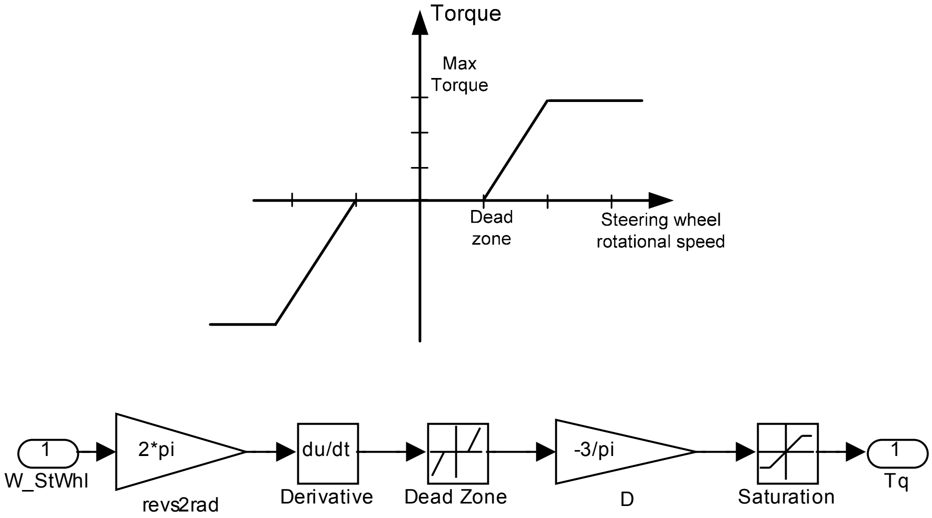

Figure 37: Function limiting steering wheel rotational velocity.

The idea behind it is that, once that the driver has taken the decision to suddenly steer the wheel and given the order to their arms, it will take a very short time for the wheel to be set in the desired orientation. Because the function is triggered by steering wheel speed it will act immediately. As a result, the movement is effectively damped and the final steering wheel angle is reduced. A first overreaction of the driver can this way be mitigated.

5.1.3 Limiting the tyre slip angle.

Limiting the tyre slip angle is quite similar to the first strategy in practise, with the advantage of directly reacting to the source of the front lateral forces which are to be limited. As an attempt to improve the function it reacts to both the tyre slip angle and the tyre slip angle time derivatives as shown in Figure 38. It was done so to make the function sensitive both to high slip angle angles and fast changes in slip angle.
But the function could not be tuned to be helpful, and what is more important, it was the most unpractical solution as it would require a sensor capable of measuring tyre slip angles. This sensor does not exist today.

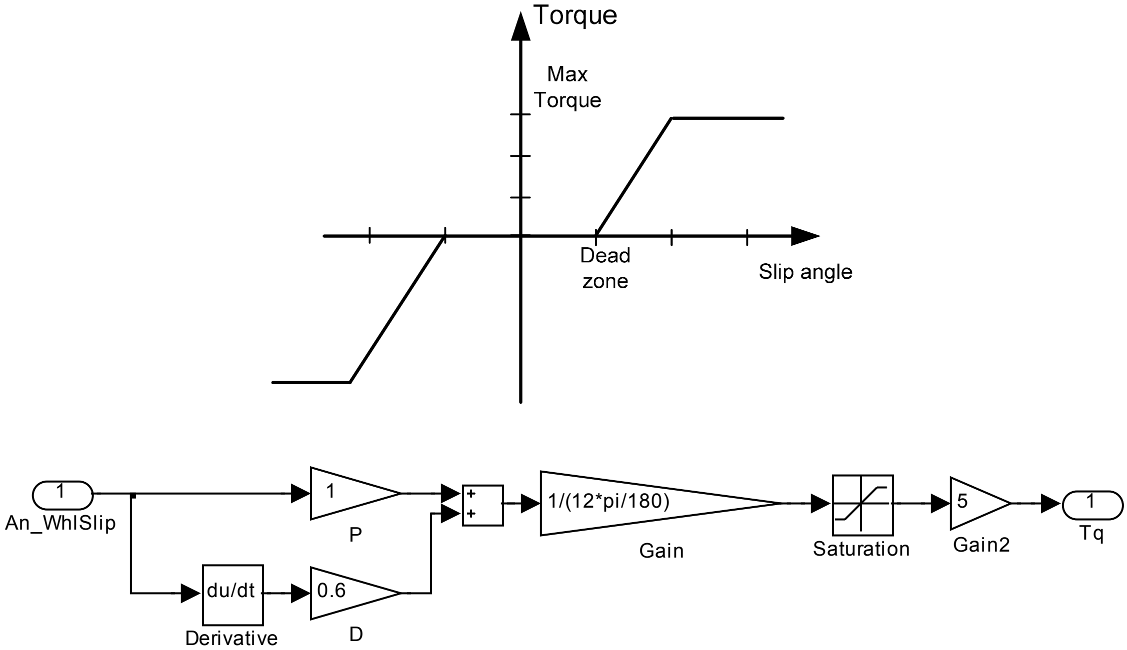

Figure 38: Function limiting front tyres slip angles.

5.1.4 Functions stabilising the vehicle

Two more functions where developed and tested with success. These functions can be seen in Appendix D.

These functions showed to be the most effective ones. Almost every driver who tried the simulator, none of them professional drivers or with racing abilities but at least car enthusiasts in most of the cases, tended to react too late when the car skidded. This wrong reaction can trigger a succession of alternate skids of increasing amplitude. This has to be controlled by a resolute counter-steering which cancels the kinetic energy associated to the yawing car well before the maximum angle is reached in a given oscillation. Common drivers seem to only react to the absolute magnitude of the skid, this could be compared to a harmonically excited oscillation, with a fatal result. With the help of the function all drivers were able to go back to the road in a controlled way.

5.1.5 Vibrations

Combinations of each of the functions above and added vibrations, see Figure 39, were also tested. The reason was to help the driver to differentiate between normal steering wheel torque generated by the tyres and steering system, and torque purposely generated by the system to help or warn them.

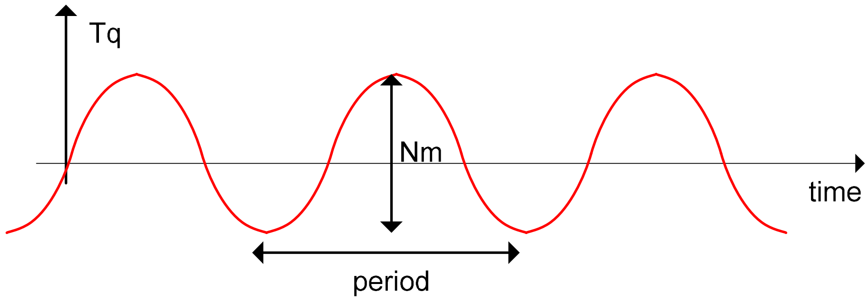

Figure 39: Vibrations added to the steering wheel.

Different amplitudes and frequencies were tested without any satisfactory results. It did not feel helpful and it could confuse the driver, forcing them to process yet another bit of information in a situation which could most likely be governed by panic. Nonetheless, this feeling could also be conditioned due to two limitations of the simulator:

- The physical connection of the steering column to the electric motor showed a certain play. It was fixed and reduced to such an extent that had a negligible influence for normal driving, but it was probably still bad enough to distort the generation of the vibrations.

- The electric motor had a limited torque of around 7 Nm. This was enough to simulate the steering wheel torque including the DSR function, but was easily saturated if a ±2 or ±3 Nm oscillating torque was superposed, ruining the haptic perception of the vibrations.

This means that the authors can not discard vibrations as a helpful tool for guiding the driver. But it is the authors' opinion that vibrations could scare the driver more than it would help in a panic situation.

5.2 Driver feeling of proposed function

The function can be felt in the simulator even if the car has not left the road. Though the original purpose of the function was to help the driver if the car leaves the road, the function was helpful also during normal driving.

Its activation is not felt as a sudden change in steering wheel torque which might scare the driver. As a result, the function is not felt as a weird effect of the car. Chances are that many drivers would feel it as something completely natural, a result of the normal operation of the car.

It was found that it helps keeping control of the car in the situation studied. Several drivers showing different driving skills and technical knowledge were able to recognise its effect and agreed it made it easier to keep control of the car. None of them felt its intervention as too intrusive or brusque. Actually, its effect was smooth enough for the driver not to need being warned of the presence of the function or trained to take advantage of its help.

5.3 Countermeasures by the proposed function

Below the countermeasures by the proposed function are discussed for cases, previously consider in Chapter 3, where the function actuates.

5.3.1 Driver reaction

Before leaving the road

Case 2: Reacts too abruptly

If the driver reacts too abruptly, moving the steering wheel too fast, a damping torque will be added to the normal torque felt in the steering wheel. The torque will increase with steering wheel speed and prevent the driver from applying a too large steering wheel angle. Figure 40 shows how the damping torque is applied, note that one *x*-axis represents the steering wheel rotational speed.

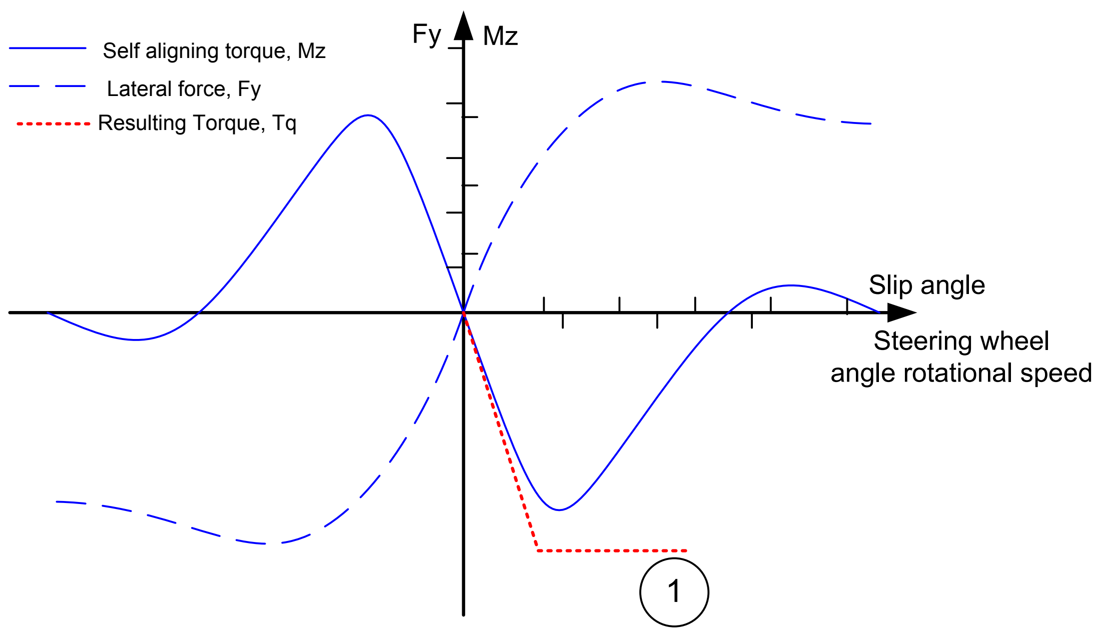

Figure 40: Function limiting the steering wheel rotational speed by applying a resisting torque for high steering wheel rotational speeds.

5.3.2 Car reaction

Case: 4a, 4b, 6a, 6b, Car over-steers or rolls over

For all the cases listed above the proposed function will actuate in similar ways. If the car over-steers or is about to rollover the function will apply a counter torque recommending the driver to decrease the steering wheel angle and in most cases even counter-steer. The driver will be recommended to steer in a, by the system decided, correct direction which stabilises the vehicle. Point 1, Figure 41, will constantly change position according to the state of the vehicle.

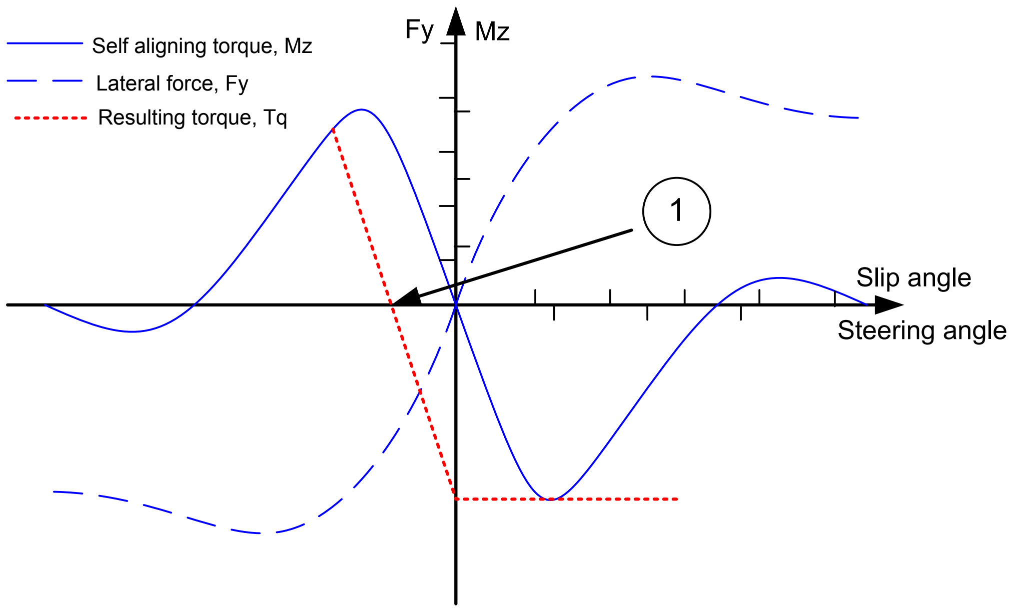

Figure 41: Function keeping the car stable by guiding the driver to counter-steer.

5.4 Summary of the proposed function

The proposed function is now summarised and compared with the demands set up in Chapter 3.1.3.

1. **The driver should be invited to move the steering wheel to a, by the system, desired position.**
   The proposed function will direct the driver to steer in a correct direction decided by the system.

2. **The bigger the difference between the desired position and what the driver is doing, the stronger the torque applied by the function should be.**
   The proposed function will increase the helping torque if the position of the steering wheel is far from the desired position.

3. **The longer the driver takes to react correctly to the situation, the stronger the torque applied should be.**
   The proposed function will increase the helping torque if the driver steers in a way that does not help to stabilise the vehicle.

4. **If the driver is moving the steering wheel too fast, a torque trying to limit this should be applied.**
   The proposed function will apply a resisting torque proportional to the steering wheel rotational speed preventing the driver from moving the steering wheel too fast.

6 Conclusion

This project has fulfilled the objective of providing Volvo Cars with a realistic and cost effective tool to asses the feasibility and usefulness of steering wheel torque superposition as a way to assist the driver in road departure incidents.

The degree of fidelity achieved in the simulator was satisfactory and allowed evaluation of the way such a function is felt by the driver.

It was found that it helps keeping control of the car in the situation studied. Several drivers showing different driving skills and technical knowledge were able to recognise its effect and agreed it helped to keep control of the car. None of them felt its intervention as too intrusive or brusque. Actually, its effect was smooth enough for the driver not to need being warned of the presence of the function or trained to take advantage of its help.

It can be concluded that the experience with the simulator points to a very certain viability of the system. Even assuming the big differences between the experience provided by the simulator and the act of driving a real car, the results from this project proves further development of the proposed function to be profitable and highly advisable.

Furthermore, the physical model developed for the Chalmers driving simulator remains as a useful tool for similar projects. With its modular structure and documentation, it can be considered as a solid base for future improvement.

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A The vehicle model

What follows is not an in-depth description of the model, but theoretical explanations of the approach followed in each of the sub-systems. Therefore, the authors would advice the reader to read them first before going through the file in Simulink, so that the reader understands the basic structure and the way it works. Extensive comments are included in each block in the Simulink file with further explanations.

Development tool

The model was created in Matlab/Simulink making all equations easily implemented in an object-oriented graphical environment and later converted to C-code. This makes the model easy to understand for new users with signal flow that is easy to follow. Simulink is also currently used in the simulator and has a tight connection to xPC Target.

Structure

The model is divided into several main blocks where inputs enter to the left and outputs exits to the right see Figure 42. All signals are transported in two busses, one feed-forward and one feed-back bus. This makes the signal flow easy to understand and every signal can be accessed from any part of the model. This also makes the exchange of whole blocks of the model easier, if for example a new powertrain or a more accurate brake hydraulics model should be tested. As long as the input and output bus structure and units are respected, the whole block can be completely modified without worrying about the other subsystems.

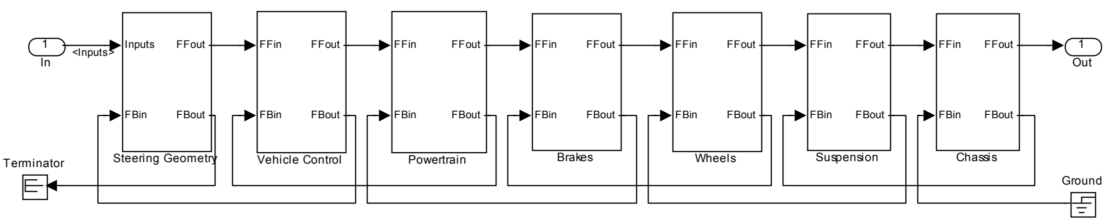

Figure 42: Overview of the vehicle model.

The model is compiled to executable code in xPC Target. It is run with a time step of 1.67 ms, that is, approximately 600 Hz. It was originally tested at 1000 Hz, but it is recommended by the suppliers that the communication with the platform is performed at 60 Hz. To be able to combine different sample times in the model they have to be multiples of each other, so a multiple of this value had to be chosen. The xPC Target computer shows the average, maximum and minimum execution times. The average time is about 0.2 ms, that is, in average terms it would be able to run the model up to 10 times faster. Nonetheless, it’s enough that one single loop requiring a integration time higher than the specified execution period is enough to make the simulation fail and stop, so that it is the maximum execution time which must be taken into consideration in order to make sure that the xPC Target is capable of handling it.

Common values for this maximum execution time are around 3.5 ms, which means that it would be possible to run the simulation several times faster without any problems. Nonetheless the model probed stable and reliable at 600 Hz, so that was the frequency finally chosen.

Signals and data flow within the model

Every data included in the model is given a name according to its characteristics. Those sharing particular characteristics (usually their origin) are grouped.

Inputs and outputs

In the communication between the different parts of the simulator the following inputs and outputs to and from the model are used.

<u>Inputs to the model:</u>

From Simulator:

Steering wheel angle, (revolutions).
Accelerator pedal, position (0-1).
Brake pedal, position (0-1).
Gears stick, gear (-1-6).
Hand brake, position (0-1).

From Virtual environment:

The road friction under each tyre.
Vertical-displacement under each tyre, (m).

<u>Outputs from the model:</u>

To Virtual environment

X, *Y* and *Z* position of the vehicle, (m).
Yaw, *Roll* and *Pitch* angles, (radians).

To Platform

Steering wheel torque, (Nm).
Yaw, *Roll* and *Pitch* angles (radians).
Positions *X*, *Y* and *Z* (m).

Signal names

As an attempt to make the model easy to understand and rearrange a naming standard from Volvo Cars/Ford called the NPA Naming Standard has been used as far as possible. This makes the model easy to change and reuse beyond the original purpose. The standard includes well defined rules for creation of signal names. The basic structure of the names is:

Quantity_SourceDescription_(units if not standard).

This means that the first part of the name, the Quantity, explains what kind of signal it is, for example torque, angle or velocity. The quantity is followed by an underscore and the source and description of the signal. The source means from where signal arises or where it is used. The description is a short description of the signal. The last underscore and unit is optional and should only be used when a non standard unit is used for the signal. For all names the standard abbreviations used by Volvo Cars are used a far as possible.

Examples:

The lateral slip angle of the front left wheel:
An_FlWhlSlipLat.

The steering wheel torque that should be fed back to the driver:
Tq_SteWhlFdbck.

The different quantities used in the model are listed in Table 1 below.

Table 1: Signal units

Quantity	Notation	Unit
Position:	Ps_	
Angle:	An_	radian
Angular velocity:	W_	radian/s
Force:	F_	N
Torque:	Tq_	Nm
Length/Distance:	L_	m
Velocity:	V_	m/s
Acceleration:	A_	m/s^2
Bus containing Aa signals:	Aa_.....Bus	
Bus containing mixed signals:	Ms_.....Bus	
Mass:	M_	kg
Inertia:	J_	kgm^2
Ratio:	Rt_	

Chassis

Chassis spatial location

The chassis has six degrees of freedom: three translations and three rotations. The orientations of the axles in the cassis are shown in the in Figure 43.

The forces generated by the tyres, translated into the chassis coordinate system, are fed into the chassis block (Figure 44), and linear and rotational accelerations are computed. These accelerations are then projected to the world coordinate system before being integrated to obtain the corresponding velocities and positions for location of the car.

Accelerations are provided in the chassis reference system, a form in which it can be used by the motion platform in order to generate the feel of acceleration. Velocities are found in both chassis (used in most of the calculations involving suspension and tyres) and world coordinate systems (for locating purposes). Position of the chassis is only calculated in the world coordinate system, as it is only used for positioning the car in the world.

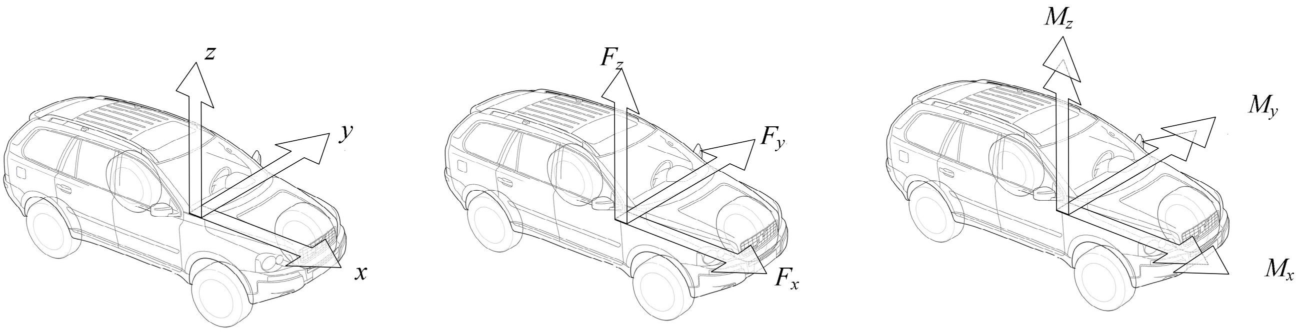

Figure 43: Vehicle's body coordinate system and forces/torque acting on the vehicle.

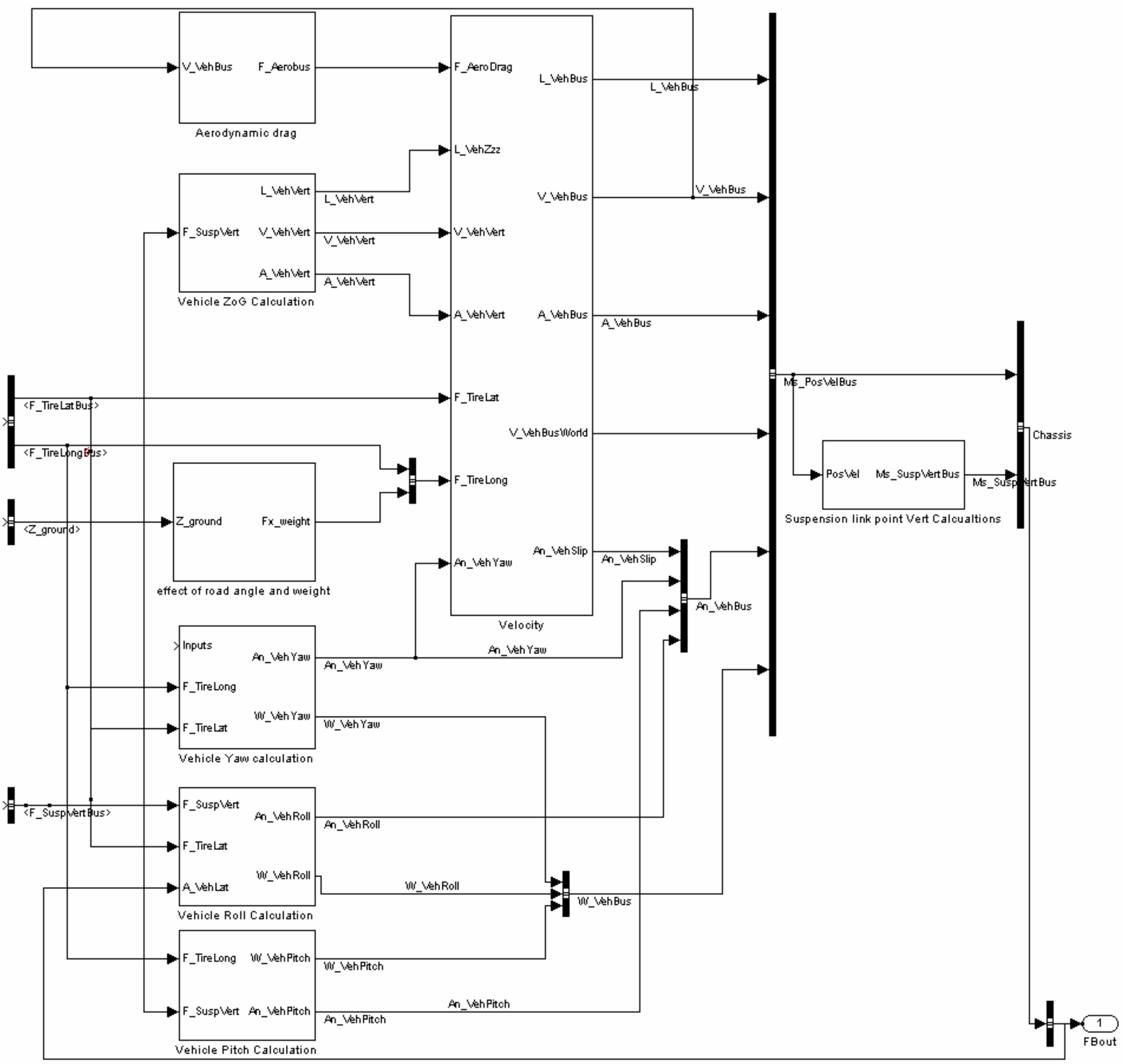

Figure 44: Chassis block.

Aerodynamic drag

Both frontal and lateral aerodynamic drag forces are computed from the longitudinal and lateral velocities. They are calculated as follows:

$$\begin{aligned} F_x &= -\frac{1}{2} \cdot \rho \cdot A_f \cdot C_x \cdot v_x^2, \\ F_y &= -\frac{1}{2} \cdot \rho \cdot A_l \cdot C_y \cdot v_y^2, \end{aligned} \tag{6}$$

where ρ the density of air, C_x and C_y the frontal and lateral aerodynamic coefficients, A_f and A_l are the total frontal and lateral area of the car and v_x and v_y are the longitudinal and lateral velocities of the car defined in its own reference coordinate system.

Effect of gravity on sloped surfaces

The effects of the gravity are felt by the driver in the simulator. It is implemented as follows:

- Gravity can effectively change the orientation of the car when it leaves the road and enters a tilted ditch. This was important for our purposes, as it could prevent the car to return easily to the road and have a big influence in the time that it remains in the ditch and the reaction of the driver.

- Gravity should accelerate or slow down the car when going up or down a hill. This would have been irrelevant when coming to test our DSR functions as testing would only be done in flat roads. But it was still taken into account for future applications.

These two effects are computed from the vertical position of each single tyre. When a difference in the vertical position between tyres in the same axle or in the same side of the car appears, lateral or longitudinal forces are applied to the chassis.

For the first effect, weight of each single axle is projected laterally according to the angle between the vector between left and right tyres and the horizontal plane of the world (*XY*). The effect is represented for the front axle in Figure 45**.** Front right tyre is in rebound, and the difference in height with the left one makes a lateral force to appear which may influence the yaw of the car.

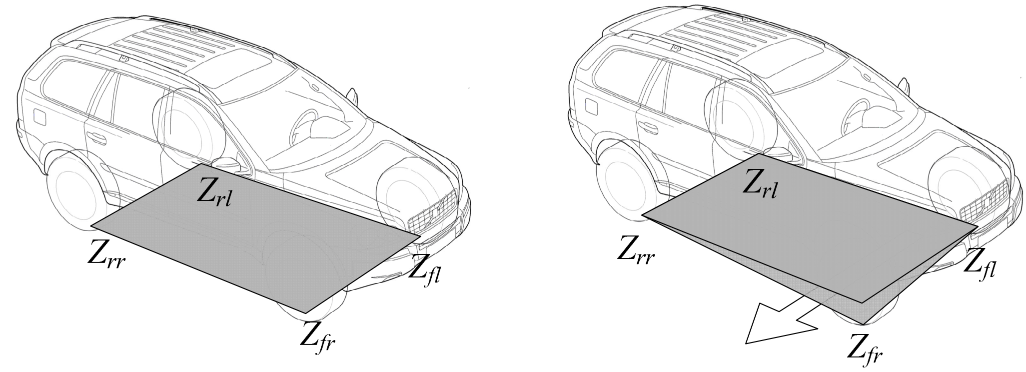

Figure 45: Lateral force created by gravity.

A similar calculation is performed when different height position appears between the front and rear tyres, which creates the effect of the car loosing or gaining speed when going up or down a hill.

Vertical position of suspension links

The distance between the upper and lower ends of the springs yields their extension relative to stand still position, and therefore the force they exert.

The lower end is calculated according to the height of the terrain below the corresponding tyre. The upper ends depend on the vertical position of the centre of gravity of the car and the roll and pitch angles. They are approximated as follows:

$$\begin{aligned}
Z_{FL} &= Z_{CG} + t_f \cdot \sin(roll) - l_f \cdot \sin(pitch), \\
Z_{FR} &= Z_{CG} - t_f \cdot \sin(roll) - l_f \cdot \sin(pitch), \\
Z_{RL} &= Z_{CG} + t_r \cdot \sin(roll) + l_r \cdot \sin(pitch), \\
Z_{RR} &= Z_{CG} - t_r \cdot \sin(roll) + l_r \cdot \sin(pitch).
\end{aligned} \tag{7}$$

Velocities of these points are calculated similarly, and are used as inputs for calculating the forces created by the dampers.

Tyres

All the calculations involving the tyres are found in Vehicle/Wheels (Figure 46). It contains one subsystem for each of the tyres plus another one for computing the torque felt by the driver in the steering wheel. All the four subsystems for the tyres are functionally identical to each other, the only differences involving the relative position of each tyre within the car. Even if the vehicle has non-steerable rear wheels, steering wheel angle calculations are also present in the rear tyres' subsystems. This allows for toe-in changes related to suspension compression and possible future work on four wheel steering systems. The four tire blocks are links to TireLib.mdl. It is done so to avoid several copies of the same code in the model and to make changes in the tire model easier.

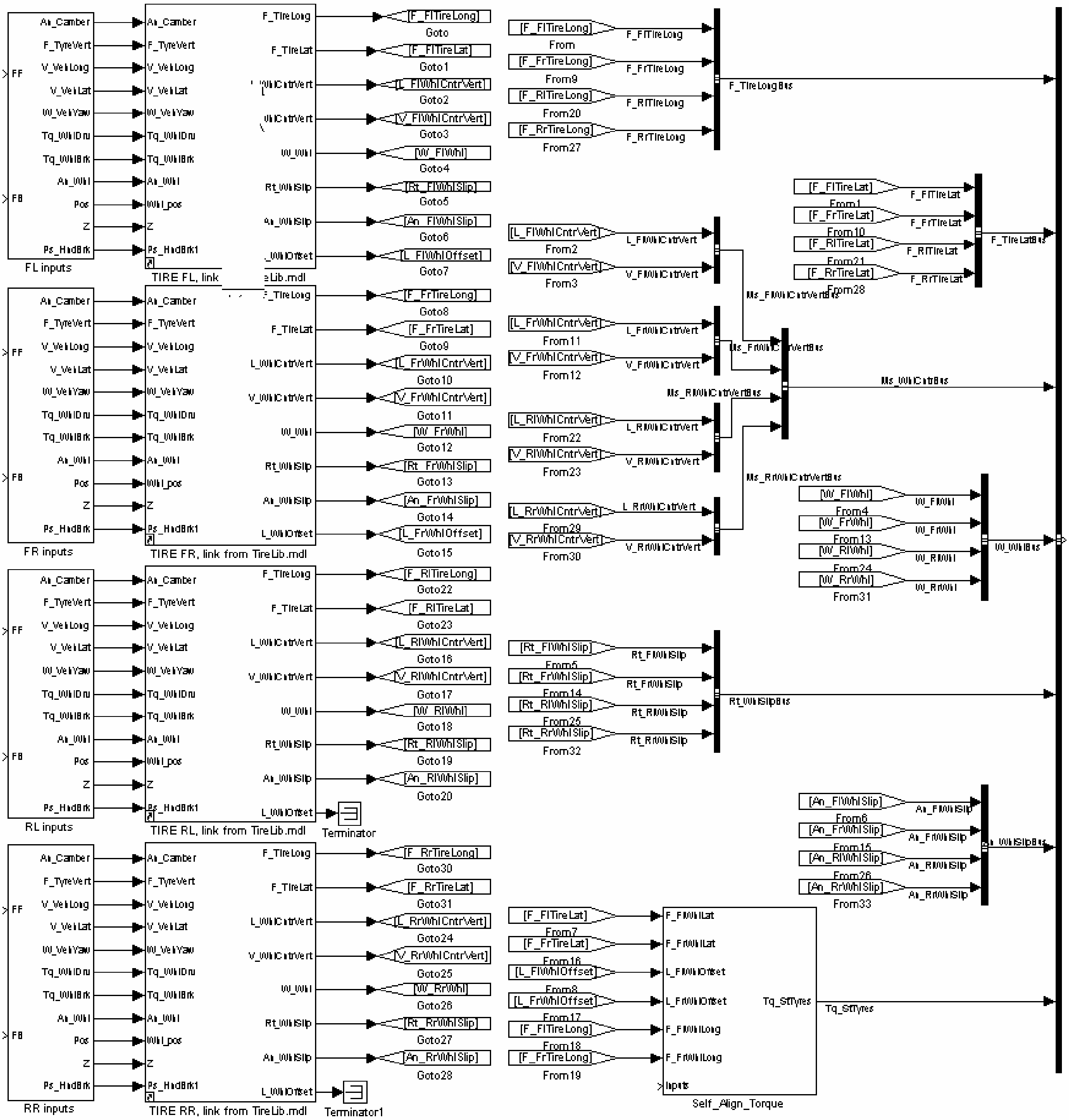

Figure 46: Main wheels block.

The selected model needs three inputs in order to compute the forces developed by the tyres: the vertical weight resting over it, the slip rate and the slip angle. Only the two latter are computed within the wheels block, as the weight is obtained from the suspension.

Slip rate calculation

As described in Chapter 2**,** slip rate involves rotational and longitudinal speed of the tyre. On the other hand, slip angle is obtained from longitudinal and lateral speeds. All of them are calculated as follows:

- Rotational speed. It is obtained from a moment balance involving driving torque, braking torque, rolling resistance, inertia of the tyre and the resulting force which effectively accelerates or decelerates the car. All these torques and forces are shown in Figure 47. Integration of the rotational acceleration α yields to the rotational velocity.

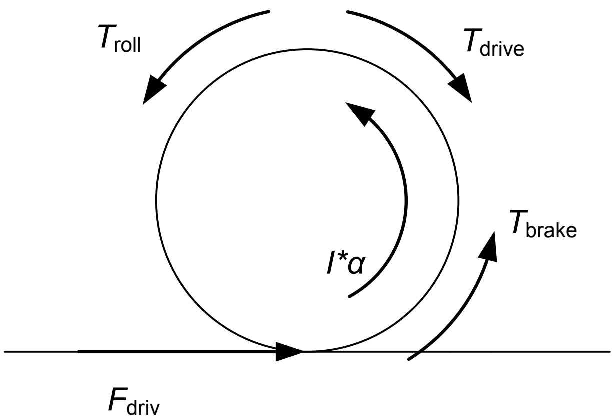

Figure 47: Moment balance on a tyre.

- Longitudinal speed. As it has to be calculated in the hub of the wheel, it will be a function of the velocity of the car but also of the geometrical position of the wheel and the yaw rate of the body. The individual velocities are calculated as follows:

$$
\begin{aligned}
Vx_{FL} &= Vx_{CAR} - \dot{\psi}_{CAR} \cdot t_f, \\
Vx_{FR} &= Vx_{CAR} + \dot{\psi}_{CAR} \cdot t_f, \\
Vx_{RL} &= Vx_{CAR} - \dot{\psi}_{CAR} \cdot t_r, \\
Vx_{RR} &= Vx_{CAR} + \dot{\psi}_{CAR} \cdot t_r,
\end{aligned} \tag{8}
$$

  where t_f and t_r are the front and rear tracks.

- Lateral speed. Similarly, position of the wheel and yaw rate of the car will also influence the individual lateral velocities for each tyre:

$$
\begin{aligned}
Vy_{FL} &= Vy_{CAR} + \dot{\psi}_{CAR} \cdot l_f\,, \\
Vy_{FR} &= Vy_{CAR} + \dot{\psi}_{CAR} \cdot l_f\,, \\
Vy_{RL} &= Vy_{CAR} - \dot{\psi}_{CAR} \cdot l_r\,, \\
Vy_{RR} &= Vy_{CAR} - \dot{\psi}_{CAR} \cdot l_r\,,
\end{aligned}
\tag{9}
$$

where l_f and l_r are the distances between the front and rear axle to the centre of gravity of the car. V_x and V_y refers to the velocity of the in the car coordinate system.

These values are substituted in (1) in order to obtain the slip rate.

$$
slip = \begin{cases} \dfrac{\omega \cdot R - V_x}{\omega \cdot R} & \text{, if } \omega \cdot R > v\,, \\[2ex] \dfrac{\omega \cdot R - V_x}{V_x} & \text{, if } \omega \cdot R < v\,. \end{cases}
\tag{10}
$$

Similarly, slip angles are computed according to

$$
\alpha = \arctan \frac{V_y}{|V_x|}\,.
\tag{11}
$$

Unfortunately, these formulas show serious instability problems when the tyre's longitudinal or rotational speeds are close to zero, as they appear in the denominators. This is not an isolated problem in the model, but a common challenge that the designers have to face when a model which must respond in all situations is being developed.

The solution used to solve this issue was to manipulate the denominators for low car or tyre speeds so that they never became lower than a certain value, low enough for ensuring stability, but high enough to maintain accuracy in most of the driving conditions. It was found that a value of 2 was valid for the longitudinal slip calculations, and a value of 1 in the case of the slip angle.

All the calculations described above are included in the blocks *Long Slip Calculation* and *Slip Angle Calculation* within each tyre subsystem.

TMEasy model

The physical model for the tyres is the most critical factor in the final result of a vehicle model, at least from the handling characteristics point of view. Almost everything else can be mathematically described in a quite neat and straightforward way, and therefore simulated quite accurately.

On the other hand, tyres show a greatly variable behaviour depending on their construction characteristics and the environment. There are big differences between different tyre models, and what is worse, the same set of tyres will behave different depending on their temperature, inflation pressure, age, thread wear, etc. [38]. As a result, when it comes to real-time simulations, a compromise must be found between accuracy and complexity. In the search for that compromise in this project, the TMEasy model was chosen for the following reasons:

- Despite the amount of “code” needed it is a straight-forward implementation.
- It takes into consideration the non-linear response of the tyre to the change in vertical load.
- The characteristics of the simulated tyre are physically understandable constants like forces and angles.
- It is being used by Volvo Cars, which allowed access to some validated data for the tyres.

The TMEasy model is implemented as described in [35]. The main idea behind the model is to characterize the force vs. slip relation by the use of two sets of five and an optional sixth constant. Each set contains:

- Cornering stiffness.
- Maximum force that the tyre is able to generate.
- Slip rate or slip angle for which this maximum force is achieved.
- Force generated by the tyre when full sliding or spinning.
- Slip rate or slip angle after which it is considered that full sliding or spinning conditions have been reached.
- (Optional). Rate at which the tyre looses force for even higher slip or slip angles. It’s not used in the basic definition of the TMEasy model as described in [35], but it can improve accuracy of the model for very high slip angles.

The two sets of the constants described above reflect the response of the tyre for two vertical loads, normal load and double normal load.

The model also takes into consideration the fact that the tyre is able to develop both longitudinal and lateral forces, which are not independent but limited by the so-called circle or ellipse of friction. Therefore, another two sets must be included in order to account for the lateral behaviour of the tyres, which won’t necessarily be the same as the longitudinal.

Therefore, four sets of five or six parameters are all the data that the TMEasy model needs to simulate a tyre rolling over a given surface. But the simulations performed in this project involved at least two surfaces: one for the asphalt and one for the soil. Therefore, a total of eight sets of parameters were needed.

Table 2 collects the names used for all the 48 constants. Sub indexes 1 and 2 refer to the weight resting on the tyre, while A and B differentiate between asphalt and soil**.** The corresponding set of data is provided to the model according to the surface over which the tyre is rolling.

Table 2: TMEasy parameter names.

ASPHALT				
	Long (*Fz*=3000N)	Lat (*Fz*=3000N)	Long (*Fz*=6000N)	Lat (*Fz*=6000N)
Initial slope	*dFx_1_A*	*dFy_1_A*	*dFx_2_A*	*dFy_2_A*
Slip for max. force	*Sx_max1_A*	*Sy_max1_A*	*Sx_max2_A*	*Sy_max2_A*
Max force	*Fx_max1_A*	*Fy_max1_A*	*Fx_max2_A*	*Fy_max2_A*
Slip for full slide	*Sx_slide1_A*	*Sy_slide1_A*	*Sx_slide2_A*	*Sy_slide2_A*
Force at max slide	*Fx_slide1_A*	*Fy_slide1_A*	*Fx_slide2_A*	*Fy_slide2_A*
Slope for higher slips	*dFx_slide1_A*	*dFy_slide1_A*	*dFx_slide2_A*	*dFy_slide2_A*
OFF-ROAD				
Initial slope	*dFx_1_B*	*dFy_1_B*	*dFx_2_B*	*dFy_2_B*
Slip for max. force	*Sx_max1_B*	*Sy_max1_B*	*Sx_max2_B*	*Sy_max2_B*
Max force	*Fx_max1_B*	*Fy_max1_B*	*Fx_max2_B*	*Fy_max2_B*
Slip for full slide	*Sx_slide1_B*	*Sy_slide1_B*	*Sx_slide2_B*	*Sy_slide2_B*
Force at max slide	*Fx_slide1_B*	*Fy_slide1_B*	*Fx_slide2_B*	*Fy_slide2_B*
Slope for higher slips	*dFx_slide1_B*	*dFy_slide1_B*	*dFx_slide2_B*	*dFy_slide2_B*

The TMEasy calculations can be split in four steps:

- Step 1: In this first step all the constants are interpolated according to the current load resting on the tyre. This defines two new force vs. slip curves (one for longitudinal and one for lateral forces) which define the behaviour of the tyre for the current vertical load. The S_{max} and S_{slide} for the given weight are calculated by a linear interpolation. All the others (F_{max}, F_{slide}, dF and dF_{slide}) are calculated using quadratic interpolations.

- Step 2: One of the main reasons of the success of the model is its ability to accurately simulate the interaction between lateral and longitudinal forces. The way to do is to calculate a so-called *generalised slip, S,* which combines the effect of both longitudinal and lateral slips, as well as keeping track of which is the relative effect.

  The generalised slip is calculated as follows:

$$S=\sqrt{\left(\frac{S_x}{\widehat{S}_x}\right)^2+\left(\frac{S_y}{\widehat{S}_y}\right)^2}\,, \tag{12}$$

  where,

$$\begin{aligned}\widehat{S}_x&=\frac{Sx\_\max}{\sqrt{(Sx\_\max)^2+(Sy\_\max)^2}}+\frac{\dfrac{Fx\_\max}{dF_x}}{\sqrt{\left(\dfrac{Fx\_\max}{dF_x}\right)^2+\left(\dfrac{Fy\_\max}{dF_y}\right)^2}}\,,\\ \widehat{S}_y&=\frac{Sy\_\max}{\sqrt{(Sx\_\max)^2+(Sy\_\max)^2}}+\frac{\dfrac{Fy\_\max}{dF_y}}{\sqrt{\left(\dfrac{Fx\_\max}{dF_x}\right)^2+\left(\dfrac{Fy\_\max}{dF_y}\right)^2}}\,.\end{aligned} \tag{13}$$

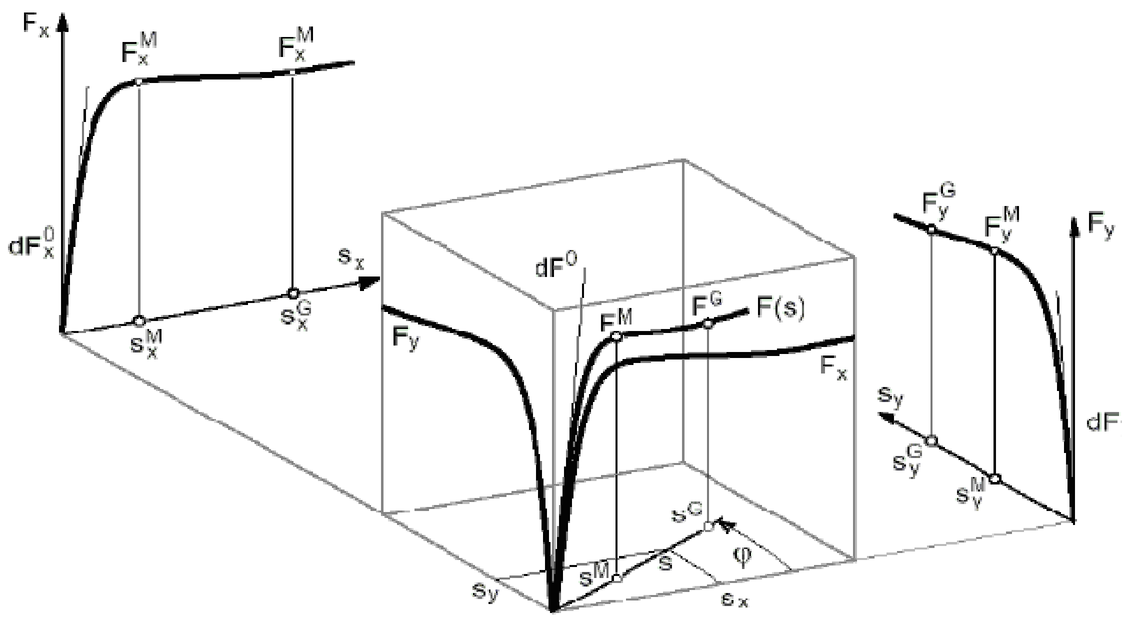

Figure 48: Longitudinal and lateral slip combination. [35]

- Step 3. The same way the longitudinal and lateral slips have been combined in the previous step, so is it done now with the constants which define the longitudinal, on one side, and the lateral, on the other, characteristic curves of the tyre:

$$
\begin{aligned}
&dF = \sqrt{\left(dFx \cdot \widehat{S}x \cdot \cos(\varphi)\right)^2 + \left(dFy \cdot \widehat{S}y \cdot \sin(\varphi)\right)^2}, \\
&S\_\max = \sqrt{\left(\frac{Sx\_\max}{\widehat{S}x} \cdot \cos(\varphi)\right)^2 + \left(\frac{Sy\_\max}{\widehat{S}y} \cdot \sin(\varphi)\right)^2}, \\
&F\_\max = \sqrt{\left(Fx\_\max \cdot \cos(\varphi)\right)^2 + \left(Fy\_\max \cdot \sin(\varphi)\right)^2}, \\
&S\_slide = \sqrt{\left(\frac{Sx\_slide}{\widehat{S}x} \cdot \cos(\varphi)\right)^2 + \left(\frac{Sy\_slide}{\widehat{S}y} \cdot \sin(\varphi)\right)^2}, \\
&F\_slide = \sqrt{\left(Fx\_slide \cdot \cos(\varphi)\right)^2 + \left(Fy\_slide \cdot \sin(\varphi)\right)^2}, \\
&dF\_slide = \sqrt{\left(dFx\_slide \cdot \widehat{S}x \cdot \cos(\varphi)\right)^2 + \left(dFy\_slide \cdot \widehat{S}y \cdot \sin(\varphi)\right)^2},
\end{aligned}
\tag{14}
$$

where,

$$
\begin{aligned}
\cos(\varphi) &= \frac{\frac{S_x}{\widehat{S}_x}}{S}, \\
\sin(\varphi) &= \frac{\frac{S_y}{\widehat{S}_y}}{S}.
\end{aligned}
\tag{15}
$$

These computed set of constants define a new force vs. slip curve. In this case, it is the generalised slip and the total force which are related.

- Step 4. The next step is to find the exact operating point in this curve and obtain the total force. It's calculated by interpolation, using one of the following three formulas depending on the value of the generalised slip:

$$F = \begin{cases} S\_\max \cdot dF \cdot \dfrac{\sigma}{1+\sigma\cdot\left(\sigma + dF\cdot\dfrac{S\_\max}{F\_\max} - 2\right)} & , \sigma = \dfrac{S}{S\_\max}, 0 \le S \le S\_max, \\ F\_\max - (F\_\max - F\_slide)\cdot(\sigma)^2\cdot(3-2\cdot\sigma) & , \sigma = \dfrac{S - S\_\max}{S\_slide - S\_\max}, S\_max \le S \le S\_slide, \\ F\_slide & , S \ge S\_slide. \end{cases} \tag{16}$$

- Step 5. The last step is to project this total force longitudinally and laterally, according to the importance of the slip rate and slip angle within the generalised slip:

$$\begin{aligned} Fx &= F\cdot\cos(\varphi), \\ Fy &= F\cdot\sin(\varphi). \end{aligned} \tag{17}$$

Apart from the forces generated by the tyre, the TMEasy model also provides an estimation of the pneumatic trail or dynamic offset of the lateral force that the tyre will show. Its calculation follows two steps, which are analogous to steps 1 and 4 used for the forces, as torque is supposed to be applied only in one direction. This value will be used to calculate the torque that the steering system will induce to the steering wheel.

Suspension

The model has independent suspension both front and rear. Nonetheless, a very simple model has been used. The main simplifications are the absence of calculations for the change in toe and camber angles for the wheels, which might have been interesting to have in order to reproduce effects such as bump or roll steering, but they were discarded early in the development of the model because of:

- The lack of real data for the XC90. Some guessing could have been done here, but that would have meant implementing some unknown parameters which may negatively affect the final results.

- Implementation of an improved suspension model would have led to more accurate results in the model. But it would probably not affect the performance of the model in the simulator and no big effects were expected in the way that the driver would feel the DSR functions.

The calculations for the suspension are found in *Vehicle/Suspension*. From the data describing the roll angle of the body (which also defines the position of the linking points between the body and the struts) and the lateral acceleration of the car it computes the weight transfer and the individual vertical load resting on each tyre. Three sources for the weight transfer are considered and then summed up:

- Weight transfer of the unsprung mass.
- Weight transfer of the sprung mass through the linking parts of the suspension.
- Weight transfer of the sprung mass through the springs and dampers.

Weight transfer of the unsprung mass

The unsprung mass accounts for the mass of the tyre and part of the suspension elements of one axle. The weight transfer of these masses can easily be calculated by:

$$F_{unsprung} = \frac{2 \cdot mass \cdot R_e}{Track}, \tag{18}$$

where *mass* is the weight for each tyre and associated suspension, R_e is the effective radius for the tyre and *Track* the track for the given axle.

Weight transfer through the linking parts of the suspension

The calculations of the weight transfer of the sprung mass are very similar to that of the unsprung mass:

$$F_{sprung\_links} = \frac{mass \cdot h_{roll}}{Track}, \tag{19}$$

where *mass* is the mass resting on the given axle and h_{roll} is the height of the roll centre.

Weight transfer through the elastic elements

The vertical position of the wheels and the vertical position of the linking points of the springs connected to the body gives the linear elongation of the springs (which will give the force exerted by the springs) and its rate (which yields the force created by the dampers).

Here another limitation of the model appears: the installation ratio is taken as constant for every position of the wheel respect to the body which might induce some differences to the results from Volvo Cars, despite using proper characteristic curves for both the springs and dampers.

Force created by the anti-roll bars is also taken into consideration, according to the difference between the compressions in both sides of the same axle. The forces from springs, dampers and anti-roll bars are summed up for each tyre to the total force as follows:

$$F_{total} = F_{spring} + F_{damper} + F_{anti\_roll\_bar} \,. \tag{20}$$

These resulting forces will be used in other blocks mainly to:

- Compute the roll acceleration of the body. Only forces transferred through the elastic elements are taken into consideration, as the resultant of those transferred through the suspension elements will be applied on the roll centre which, by definition, will create no rolling moment on the body.

- Feed the TMEasy model in the Wheels block with the vertical load resting on the tyres, in order to calculate the longitudinal and lateral forces created by them. In this case, the weight transferred through the suspension elements is also included.

Powertrain

The vehicle simulated by the model is equipped with a 2,5 litre engine and the corresponding automatic transmission from Volvo Cars. It can be found in the *Vehicle/Powertrain* subsystem. All parameters and look up tables are given by Volvo Cars.

From the wheels speed, ω, the engine speed is calculated using the current gear ratio as follows:

$$EngineSpeed = \omega \cdot GearRatio\,. \tag{21}$$

The automatic transmission shifts up and down when the engine speed is crossing predefined values, one value for shifting up and one for shifting down. The torque output from the engine is calculated using a look-up table with throttle position and engine speed as inputs, see Figure 49. The torque output from the powertrain is then calculated:

$$EngineTorque = f\left(TrottlePosition, EngineSpeed\right) \cdot GearRatio\,, \tag{22}$$

where the function *f is the output from the look up table in* Figure 49.

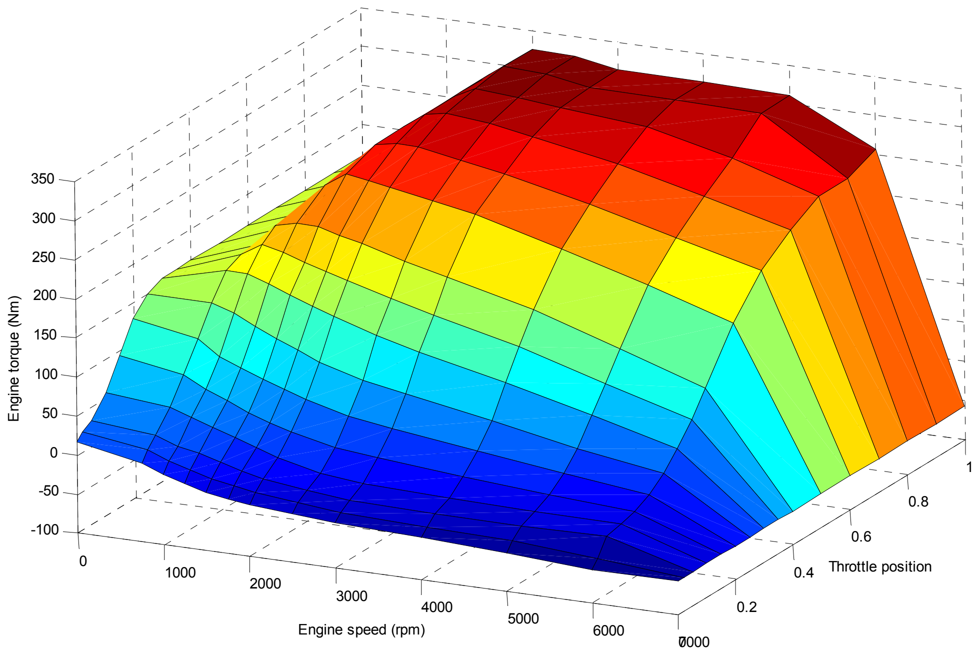

Figure 49: Engine torque vs. engine speed and throttle position.

Brakes

A simplified brake system has been modelled in the Vehicle/Brakes subsystem; it contains brake hydraulics and equations for the disc brakes used.

To model the brake hydraulics a blocked limiting the rate of the pressure have been used preventing the brake pressure from rising and falling faster than in a real brake system.
The brake pressure is applied to a piston pressing the brake pads to a rotor attached to the wheel, see Figure 50.

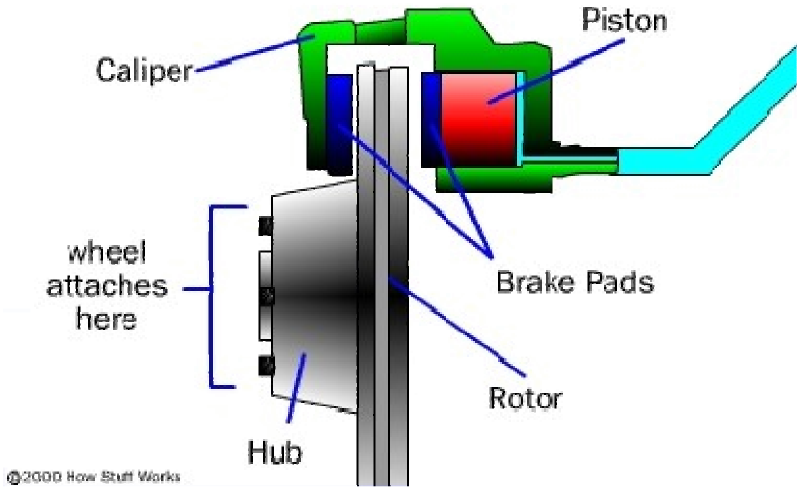

Figure 50: Picture showing the basics of a disc brake, from www.howstuffworks.com, 060115.

The actual brake torque applied to the brakes is calculated using the following equation for each tyre,

$$T_{brk} = 2 \cdot \left(\mu_{brkWhl} \cdot A_{pad} \cdot r_{brk} \cdot P_{brk} \right), \tag{23}$$

where μ_{brkWhl} is the friction coefficient between the brake pad and the brake rotor, A_{Pad} is the area of one brake pad, r_{brk} is the trail arm for the force (approximately the mean radius of the brake rotor) and P_{brk} is the brake pressure applied to the brake pads by the piston.
The equation is multiplied by two because there are two brake pads, one on each side of the rotor, see Figure 50.

Steering geometry

Steering geometry calculations are found in Vehicle/Steering Geometry. Real steering ratio from the XC90 is used to calculate the wheel angle. It is then is modified in order to approximate the characteristics of an Ackermann steering system, so that inner wheel is always steered slightly more.

These calculations are found in *Vehicle/Steering Geometry*.

Steering torque

One important factor that the model had to be able to reproduce is the torque that the driver would feel in the steering wheel as a consequence of the tyre forces and steering geometry. This torque is filtered through the power steering sub-system, *Vehicle/Vehicle Control/EPS & DSR,* and then summed up to the torque generated by the DSR function. It can be found in *Vehicle/Wheels/Self_Align_Torque*.

For the basic feel of the steering wheel, the one that could be found if the car had no power steering, six sources were taken into consideration.

- Lateral forces. Torque due to lateral forces in the tyres, and the fact that the point of application of this forces is generally slightly behind the intersection of the axle about which the tyre is steered and the ground. This distance, which in this case acts as a lever arm, is the sum of the so-called static offset (a consequence of the steering geometry of the car), which is considered to remain constant in the model, and the pneumatic trail, which changes according to the slip angle of the tyre [1].

- Longitudinal forces. Torque due to longitudinal forces in the tyres, that is, when the car accelerates or brakes. Similarly as the previous case, it is caused by the existence of a moment arm which translates these forces into a torque in the steering wheel. The distance responsible for this in this case is the scrub radius. Even if the blocks responsible for the calculation of this effect remain in the model, the scrub radius is set to zero so these forces have no effect on the steering wheel. The main reason for this was to avoid undesirable interactions with the DSR functions when the ESP was activated, due to the braking forces that it generates.

- Front lift. Torque proportional to steering wheel angle. As the front wheels are steered, front right and front left ends of the car are lifted or lowered according to the caster and KPI angles [1]. The effort to do this must be provided by the driver (in conjunction, if it exists, of the power steering system), which can mean an increased resisting steering torque. Proper calculation of this effect needs accurate data from the steering geometry and the rotation-translation-rotation conversion imposed by the wheel carrier-rack-pinion mechanism. This is not included in the model.

- Linear damping. An opposing torque is applied which is linear with respect to rotational speed of the steering wheel.

- Dry friction. A constant friction is applied whenever the wheel is steered.

- Inertial effects. A detailed study of the mass and kinematics properties of steering column, pinion, rack, wheel carriers and hubs would lead to inertial (and gyroscopic) effects on the steering wheel which would increase the resisting torque the higher the driver is accelerating it. This would be especially noticeable in rapid movements such as evasion manoeuvres or an over-reaction in the accident focus of this project. Such a detailed simulation of the steering system was not performed, but still the influence of inertia had to be included somehow. The last source of steering torque accounts for this, and is calculated as the steering wheel acceleration (which would be somewhat proportional to the acceleration of all the other parts involved in the steering) times a constant factor which would account for all those inertial effects.

All parameters where estimated mostly by experimental tuning in the simulator.

Power steering

The function of power steering is to reduce the physical effort required from the driver to operate the steering wheel, especially when manoeuvring at very low speeds.

Figure 51 shows the schematics of a common steering system. The effort created by the driver is used as an input, from which a resulting assisting torque is calculated. This torque is then subtracted from that created by the tyres, either by a hydraulic circuit (in a classic hydraulic power steering) or an electric motor (electric power steering).

The law which defines the assisting torque is referred to as boost function and usually depends on other variables such as the speed at which the car is travelling [34].

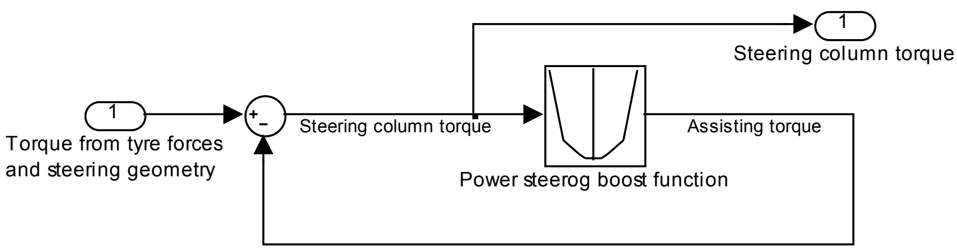

Figure 51: Power steering layout.

A problem arises when implementing this layout in Simulink. The input for the system (the steering column torque) is at the same time a direct function of the output (the assisting torque). This defines an algebraic loop, which requires a higher computing time. As a result, it's able to manage it in off-line simulations (considerably increasing the solving time), but it is not possible to solve it in real-time applications.

Consequently, a simplified layout was implemented in the model, as shown in Figure 52. The torque generated by the tyres and the steering geometry is directly used as an input for a look-up table, which calculates the resulting torque. What the driver finally feels in the steering wheel is the addition of this torque and the one generated by the DSR intervention.

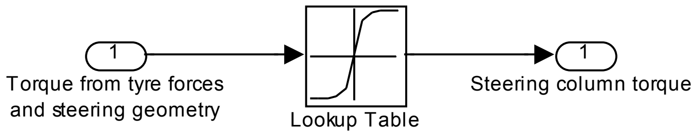

Figure 52: Implemented power steering.

Modelling the yaw control by brake (YCbB)

Information about the YCbB was gathered through [16], [19] and [20], and competence at Volvo Cars. As a result, a model of the YCbB has been implemented and it can be found in Vehicle/Vehicle Control/ESP & ABS/ESP. Figure 53 shows how the model is set up.

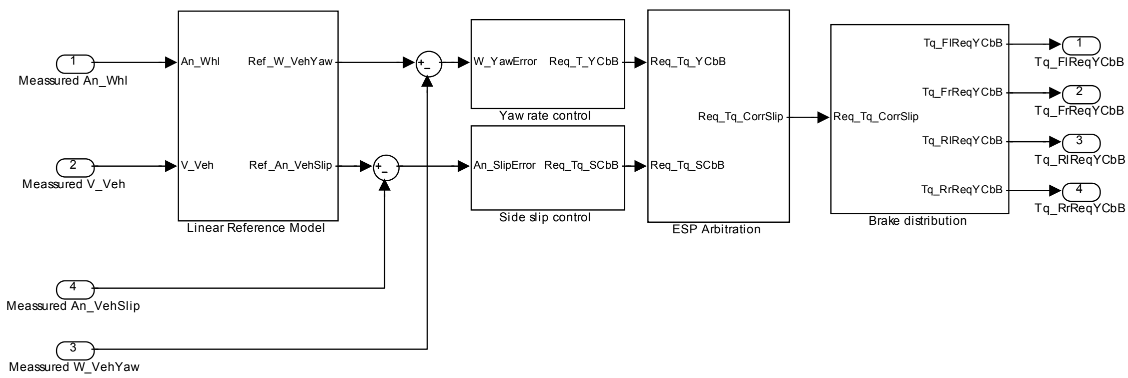

Figure 53: YCbB Simulink model.

It is divided in the following parts:

- <u>Linear reference model</u>

The linear reference model is a bicycle model, as explained in the vehicle dynamics chapter. It takes the wheel angle and the speed of the vehicle as inputs. These inputs are processed and the outputs are the yaw rate and the side slip angle that a linear vehicle would have for the same inputs.

- <u>Yaw rate control</u>

The yaw rate control is a PD controller fed by the difference of the measured yaw rate and the reference yaw rate. The controller outputs a requested torque needed to correct the vehicle.

- <u>Side slip control</u>

The side slip control is a PD controller fed by the difference of the measured side slip angle and the reference side slip angle. The controller outputs a requested torque needed to correct the vehicle.

- <u>ESP arbitration</u>

Since there are two controllers, trying to control two different variables with the same actuator, arbitration between them has to be done. This is solved by letting the controller prioritize the highest correcting torque. When requesting equal or close to equal torque the mean value of the torques is used and the transition between them is done in a smooth way, see Figure 54. The output is the torque needed to correct the vehicle. This is probably not how it is done in real systems but a simplification that works in the simulator.

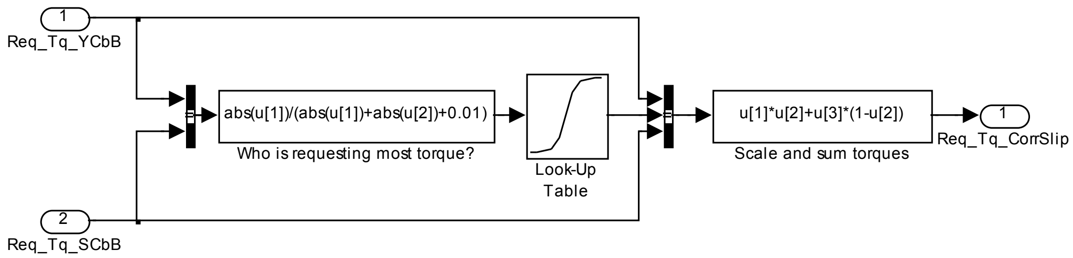

Figure 54: The arbitration between the requested yaw and side slip corrections.

- <u>Brake distribution</u>

The requested torque is here calculated to a requested wheel braking torque by the following equation.

$$T = \frac{T_{vehicle}}{Track/2} \cdot R. \tag{24}$$

The requested torque is divided by half the wheel base to get the force needed from the tyre. This force is then multiplied by the trail arm, the radius of the wheel, to get the brake torque needed.

The decision of which wheel to brake is done by selecting the left wheels if the requested torque is above zero and the right wheels if below zero. Most of the torque is applied to the front wheel, 80 %, and the remaining 20 % is applied to the rear tyre. The real ESP systems continuously calculate the distribution of the braking force between the different tyres when more than one tyre is braked. The fixed 80/20 distribution in the model was chosen in order to reflex the ability of real systems to involve more than one tyre in trying to control the car, though a strong bias to the front was maintained. Too much rear actuation would be undesirable in over-steering situations.

Modelling the antilock brake system (ABS)

To be able to validate the created model in an accurate way with ESP and created steering function an ABS system was modelled. The information about the ABS system was found in [39] and in the Matlab/Simulink tutorial files.

The basic principle of the system is that when the brakes are applied and a slip rate occurs the system keeps the slip rate to a limited value, where the tyre is still able to create some lateral force which ensures certain steerability of the car. In the model this is done by a PI controller fed by the difference in slip and the desired slip value, in this case 0.4. When the slip exceeds the desired value the brake pressure is decreased until the slip is below the value. Then the brake pressure is increased again until the slip again exceeds the desired value. By doing this the slip is held oscillating close to the desired slip rate value giving the vehicle maximum brake with maintained steerability. Because the ABS model is used in a simulator a simplified version of the real system could be used since the slip values are known in the vehicle model. The ABS Simulink model for one tyre is shown in Figure 55.

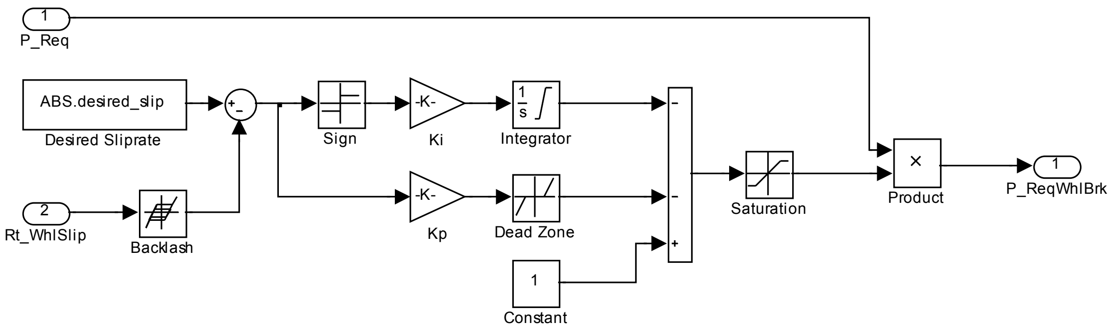

Figure 55: The ABS Simulink subsystem for one tyre.

Result and tune of the feel of the vehicle model

The model provided a satisfactory feel in the simulator but is showed one noticeable defect: The car felt a bit too loose in the rear axle, which lead to instability when driving at speeds over 100 km/h. Any actuation on the steering wheel would lead to too high slip angles in the rear, which the drivers tended to overcompensate and therefore make the car wander until control was lost. It was possible to control the car if the driver showed enough anticipation on the wheel, but its basic attitude would feel quite strange.

The reason seemed to be the need for the car to develop enough side slip angle for the tyres to be able to compensate for the lateral forces in the front. Assuming that the rear tyres don’t steer at all (which is not completely true when suspension kinematics is taken into consideration), whole body side slip angle is the only way for them to develop slip angle and therefore lateral force. The equilibrium between front and rear axles was not reached until a higher than expected side slip angle was reached, and by that time the surprised driver was probably counter-steering and starting an unstable sequence of oscillations in the behaviour of the car.

Though this phenomena appeared early in the development of the model, no concrete problems where found which could cause it. It's more likely a consequence of the simplifications of parts of the model.

In order to mitigate this effect, a correction where included in the model to limit the effect felt in the simulator. The correction can be considered as non-realistic tuning from the physical point of view and it should be removed before any further development is done on the model. But it proved to be effective and perfectly compatible with the final purpose of the model, which was ensuring a realistic experience to those who drove the simulator.

The correction is:

> Inducing some toe-in in the rear tyres. In a similar way as it would be done by the suspension kinematics, rear steer angle is modified according to the vertical position of the wheel relative to the body. But it's included in Vehicle/Steering Geometry instead of Vehicle/Suspension, as it may have been expected.
>
> As soon as the car steers in one direction, the weight transfer compresses the outer wheel suspension. The corresponding rear tyre will then lean inside, inducing a slip angle that will increase its lateral force without the need of developing so much side slip angle. This correction effectively corrected the behaviour of the car but at the expense of using a too aggressive toe-in angle increase which led to a sharp non-linearity, as it is done so that no toe-out is created when the spring is extended.

Future improvements of the vehicle model

a. Review of tyre model with other parameters. Other parameters for the TMEasy model could be tested.

b. Inclusion of tire deflection which would account for the phenomena of tyre lag ([36]), more information can be found in [37] and [38].

c. Proper suspension kinematics calculations, with realistic camber and toe angles simulation and authentic compression rates of the springs and damper related to the wheel travel. A thesis work in this topic can be found in [41].

d. Improvement of the simulation of the contact between the road and the tyre, properly accounting for wheel lift, when the car jumps or under extreme cornering. So far such condition is not accurately covered. Force created by the suspension is just cancelled when the wheel travel reaches a certain limit.

More detailed calculation of the torque generated in the tyres and the steering system. An example of such calculations implemented in another driving simulator can be found in [32].

Another improvement would mean a complete review of the model, taking advantage of the Multibody System Dynamics approach [42]. That would account for inertial effects of different individual parts of the car (suspension elements, for instance) and a more effective simulation of parts interaction (deflections, elastic joints, etc, etc).

B Chalmers driving simulator

This appendix gives a short overview of the vehicle simulator at the Department of Signals and Systems at Chalmers University of Technology. A brief user guide is also presented.

The simulator system, shown in Figure 56, consists of three main parts; a motion base platform, a virtual environment and a dynamic vehicle model.

Figure 56: Chalmers driving simulator.

Motion base platform

The motion platform is used for motion feedback to the driver. It is a hexapod with six degrees of freedom.

In Table 3 below the specifications for the platform can be found.

Table 3: Motion platform specification.

DOF	Movement	Velocity	Acceleration
Roll	±22º	±30º/s	±500º/s^2
Pitch	+25º/-23º	±30º/s	±500º/s^2
Yaw	±23º	±40º/s	±400º/s^2
X	±0.27m	±0.25m/s	±0.6g
Y	±0.26m	±0.25m/s	±0.6g
Z	±0.18m	±0.18m/s	0.5g

A quarter of a Volvo Cars S80 is mounted on the motion base platform and handles the interaction between the driver and the simulator system. From here it is possible to control the steering wheel, gas and brake of the simulated car. When executing the simulator system the motion base platform gives an experience of the elevation of the ground in the visual environment and the acceleration of the simulated car. The CAN bus of the Volvo Cars is also available but not used today.

xPC Target

xPC Target is a real time operating system from **MathWorks** that enables the connection of Simulink models to the physical systems and executes them in real time. It is in xPC Target all the communication with the platform and the car are performed.

Virtual environment

A projector displays the virtual environment on a screen in front of the driver. The projector is located below the Volvo Cars to the right and it projects the image to the screen via a mirror. Receiving the current position of the car from the xPC Target computer the visual environment renders and displays the world. It also calculates the height of the ground and surface under each wheel and sends this to xPC Target.

Dynamic vehicle model

The dynamic vehicle model, developed in Simulink, is executed on the computer running the xPC Target operative system.

Communication

The simulator uses four computers to run, a diagram of the communication is shown in Figure 57.

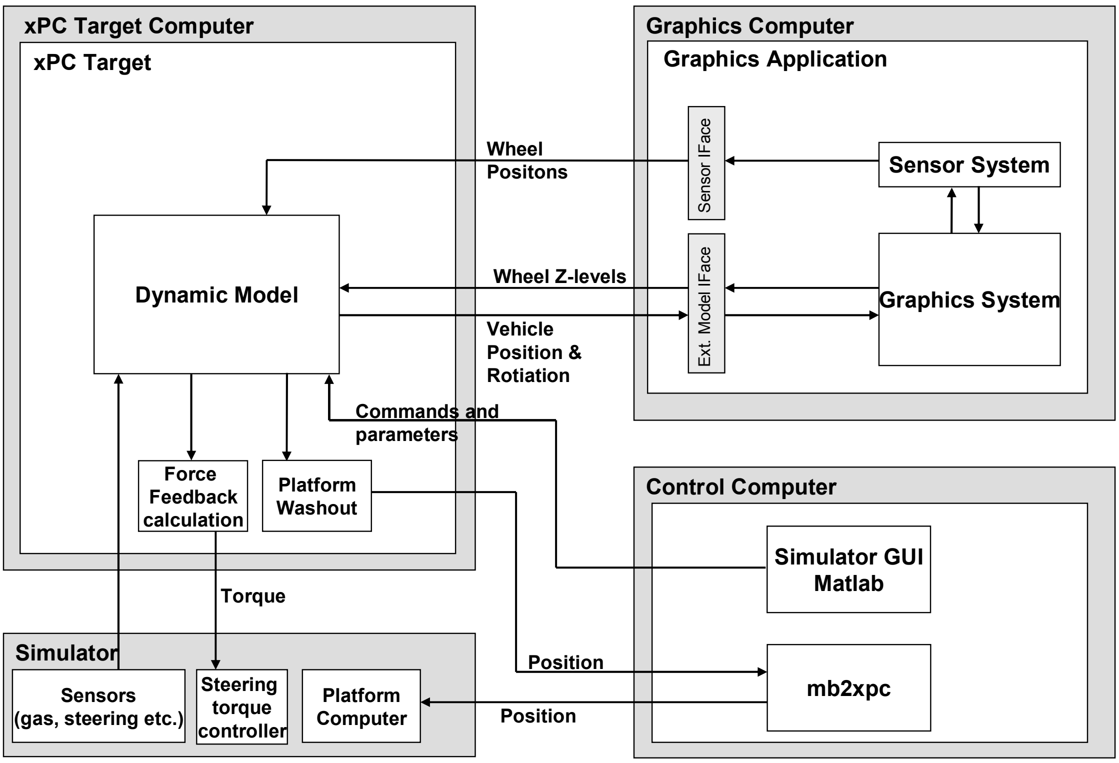

Figure 57: Simulator communication.

- Control computer
The control computer handles the communication between xPC Target and the platform. This is also where the Simulink models are developed, compiled and transferred to xPC Target.

- xPC Target computer
The xPC Target computer is where the xPC Target real time system is run. All inputs to and from the simulator go via the xPC Target computer.

- Platform computer
The platform computer shows the status of the platform. It is also possible to move the platform manually from here.

- Graphics computer
The graphics computer is where the virtual environment is run.

The different computers in the network communicate via an UDP protocol, (User Datagram Protocol), and the switch located on top of the xPC Target computer.

The communication speed between the xPC Target and the platform is set to 60 Hz. If data is sent slower or faster the platform may loose the communication and return to park position. The graphics computer both sends and receives at 100 Hz.

Simulator start-up guide

This is a guide through the steps required starting the simulator and running a simulation. Figure 58 shows the simulator and the control computers.

1) Simulator Control Computer *2) xPC Target Computer*
3) Platform Computer *4) Graphics Computer*

Figure 58: Computer setup.

1. Boot up

Boot up the computers in arbitrary order:

- Log in on the Simulator Control Computer as *Simulator*.

- Boot xPC Target by inserting the floppy disc in the xPC Target computer and start it up. It should boot up showing the xPC Target start screen.

- Log in to the Graphics Computer as *Graphics Computer,* the password is *simulator2004*.

- Turn the switch on the wall in front of the simulator in order to start the Platform Computer.

- Turn on the platform computer screen. A switch in the rear.
  It should boot up showing the platform status screen with the message *need host communication* in the upper left corner.

- Turn on the projector.

3. Start the graphics application

Do the following on the Graphics Computer:

- Start CarSim.exe (shortcut on desktop).

- From the start-up screen, (Figure 59), select *Wheel Position Sensor* and *Simulator – External Model*. This way the model are set to send the data about the position of the tyres to the vehicle model, and use the model running on the xPC Target computer.

- Finally click *Load Environment* and select an environment file (.env).
  Suggestions: StraigtRoad.env or TestTrack.env are good environments for testing.

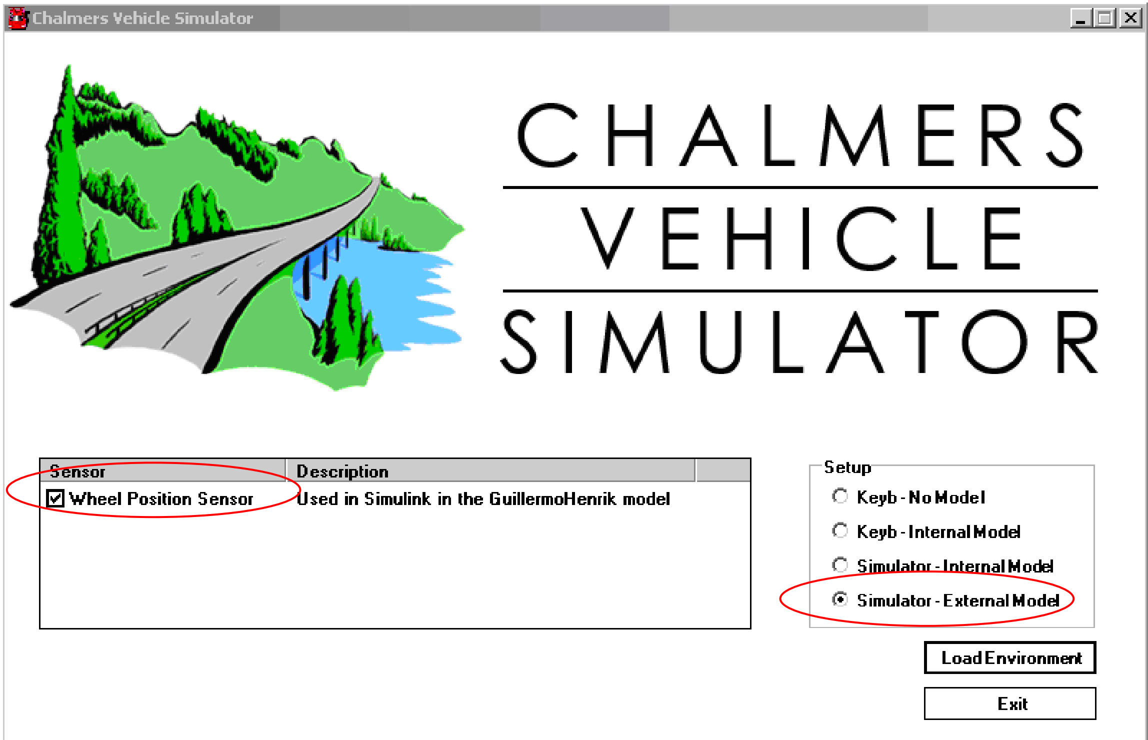

Figure 59: Start up screen of the graphical environment.

2. Load model into xPC Target

Do the following on the Simulator Control Computer:

- Start Matlab 6.5.1 (shortcut on desktop).

- Load a precompiled model
    - Go to the directory containing the model.
      Example: C:\simulator_use_models_in_here\Simulator.

    - Type *xpcload* in Matlab on the Control computer and then select which model.

      xpcload loads the model into xPC Target, opens she simulator GUI and starts mb2xpc.exe. mb2xpc acts as a proxy between the xPC Target computer and the platform.

4. Run simulation

Now the driver can take place in the simulator. Make sure the steering wheel is centred; the initial angle will be used as the centre. When the steering wheel is centred it should be possible to turn it two revolutions in each direction.
Start the simulation:
The simulation can be started in a several ways; the first way in the list below is the recommended way.

- After typing xpcload in the step before the simulator GUI should have appeared (Figure 60). If not, type *"simulator" in* Matlab at the Simulator Control Computer and then:
    - Click Start in the Simulation frame of the GUI in the Simulator Control Computer.

Or

- type *"+tg"* in Matlab at the Simulator Control Computer.

Or

- Type *"c"* on the xPC Target computer to go to the command line.
    - type *"start"* then enter.

The Platform Computer should no longer say "*need host communication"*.

5. Start platform

If using the simulator GUI:

- Click “Start” in the Platform frame.

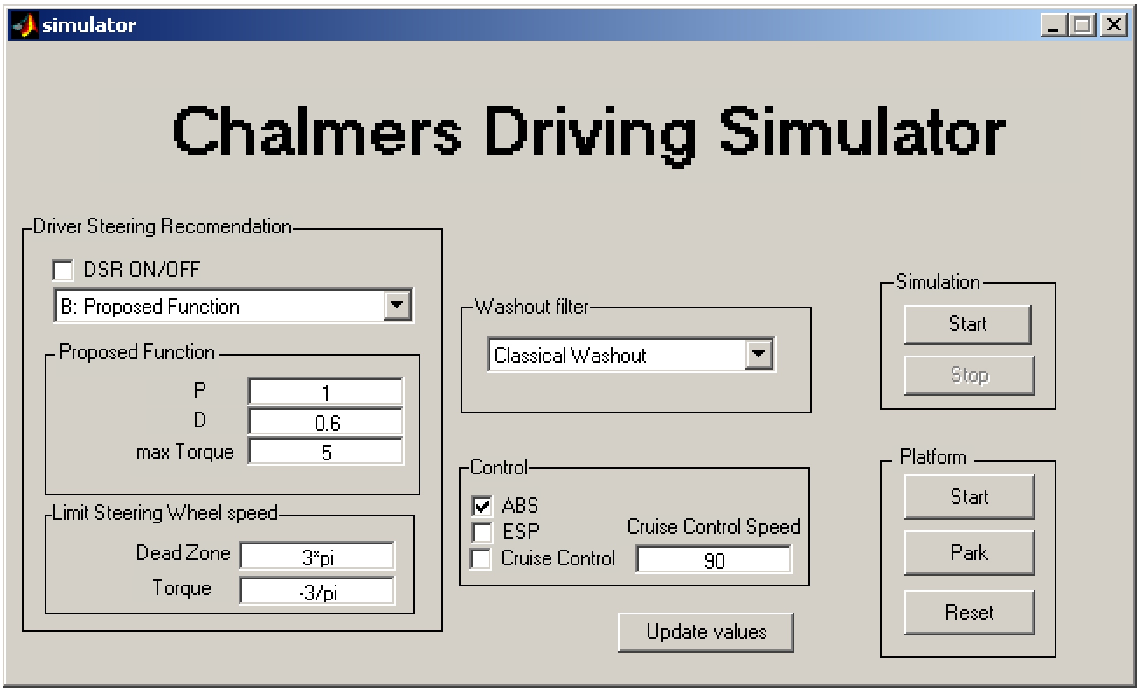

Figure 60: The Simulator GUI.

4. Stop simulation

- Stop Platform by clicking *Park* in Platform frame in the Simulator GUI in the Control Computer.

- Stop simulation.

  - Click Stop in the Simulation frame of the GUI in the Simulator Control Computer.

  Or

  - type *“-tg”* in Matlab at the Simulator Control Computer.

  Or

  - Type *“c”* on the xPC Target computer to go to the command line. Only necessary the first time.
  - type *“stop”* then enter.

Common problems

- *Matlab shows error message "TCP/IP Error" while compiling.*
  This happens sometimes, especially first time after boot up. Just load or compile the model again. If this does not work, reboot xPC Target and/or restart Matlab.

- *Platform shows error message "loaded battery test failed".*
  The platform battery needs to be charged. Keep the Platform Computer on for 10 hours or so to charge it.

- *Platform shows error message "not at base position".*
  Sometimes the platform fails to return to base position, pushing it down on the left rear corner should move it to the correct position.

- *Platform has tilted out of base position.*
  If the platform for some reason has tilted out of base position it has to manually be moved back by lifting in the corner that is tilted down while pushing down the opposite corner. Just pushing down one corner is not enough due to the singularity. Three or more people might be needed.

- *Platform computer shows following error message during start-up:*

  ERROR: HIMEM.SYS has detected unreliable XMS memory at address XXXXXXXXh.
  XMS Driver not installed.

  Press enter. The platform should function anyway.

- *The steering wheel is stuck.*
  The steering wheel stop is probably at its end position. Return the steering wheel to its centre position. It should be possible to turn it two revolutions in each direction.

- *The graphics application looks strange when switching to full screen.*
  Press f some times and eventually it will look good again.

- *There is no force feedback in the steering wheel.*
  The power cable is probably disconnected from the main power supply box. Connect it.

- *Strange Error somewhere*
  Restart Matlab and/or the xPC Target computer.

Graphical environment user guide.

The virtual environment can be controlled using the keys found in Table 4 on the graphics computer.

Table 4: Graphical Environment user guide.

Key	**Action**
1	Change to Option mode, screen will freeze
2	Change to Drive mode (default)
Option mode, (press 1 first)	
j	Switch speedometer on/off
f	Switch between full and normal screen
g	Switch info on/off, xyz, etc
Camera manipulation	
Home	Place camera in driver view
Insert	Pitch camera up
Delete	Pitch camera down
Page up	Move camera up
Page down	Move camera down
Left arrow	Move camera left
Right arrow	Move camera right
Up arrow	Move camera up
Down arrow	Move camera down

Sensors in the graphical environment

The following appendix will give a short description of sensors in the graphical environment and also give an example of one.

A sensor is a plug-in programmed in C++ used in the graphical environment. A sensor can reach almost all data from the graphical world such as positions and directions of the car. It can process the data and send the result to the xPC Target computer.

To use a sensor the check box in front of the sensor name must be checked in the graphical Environment start up screen, see Figure 59. When a sensor is running the update function, see below, of the sensor is run at 100 Hz during simulation. The return data from the update function is then sent to a predefined port in xPC Target. The port number can be found and changed in the file Globals.h.

If more than one sensor is running the out data from all sensors is sent on the same port one after each other. In that case xPC Target must be set to receive the correct number of floats, 4 times the number of sensors.

The easiest way to create a new sensor is to open an old sensor, change the name and replace the old code.

The sensor below was written during the project in order to get data about the wheel positions from the graphical environment. It is used to calculate the friction under each wheel during simulation. It sends four floats to xPC Target, one for each wheel, in the following order: front left, front right, rear left, rear right. If a wheel turns out to be on the road during update the number one is returned in the corresponding float, a number less than one otherwise.

WheelPosSensor.cpp

```
#include <stdio.h>
#include <communication/TransmitterPool.h>

extern "C"
{

__declspec(dllexport)
bool update(float carPos[3], Communication::TransmitterPool& transmitterPool, float output[4])
{
	// The sensor returns 4 floats, one for each tyre
	memset(output, 0, sizeof(float)*4);

	Environment::Road::Profile::laneType laneType;

	// for all 4 tyres
	// Order: FL, FR, RL, RR
	for (int i = 0; i < 4; ++i)
	{
		// Get present lane type for the tyre
		laneType = transmitterPool.getLaneDirection(transmitterPool.getSimCar()-
			getWheelROC((Environment::ObjectNodes::Car::wheelIndices)i),
			transmitterPool.getSimCar()->getDirection());

		// If the tyre is on the road return 1 else return a number less than 1
		switch (laneType)
		{
			case Environment::Road::Profile::RIGHTLANE:
			case Environment::Road::Profile::LEFTLANE:
				output[i] = 1.0f;
				break;

			case Environment::Road::Profile::LEFTSHOULDER:
			case Environment::Road::Profile::RIGHTSHOULDER:
				output[i] = 0.9;
				break;

			case Environment::Road::Profile::LEFTFORESLOPE:
			case Environment::Road::Profile::RIGHTFORESLOPE:
				output[i] = 0.5f;
				break;

			case Environment::Road::Profile::DIVIDER:
				output[i] = 0.75f;
				break;

			// The tyre is off the road
			default:
				output[i] = 0.0f;
		}
	}

	return true;
}

// Functions used to print information about the sensor
__declspec(dllexport)
const char* name()
{
	return "Wheel Position Sensor";
}

__declspec(dllexport)
const char* description()
{
	return "Used in Simulink by the GuillermoHenrik model";
```

The anatomy of an environment file

The road and environment used in the graphical environment are defined in ".env" files. The files can be viewed and altered in a text editor. An example of the structure of an environment file is shown below.

Define the graphical environment, sky, sun etc.
Skybox {
  Dimensions: 2000 2000 2000
  ZDisplacement: 0
  Texture: Front ../textures/skyboxes/terragen_afternoon/front.png
  Texture: Back ../textures/skyboxes/terragen_afternoon/back.png
  Texture: Left ../textures/skyboxes/terragen_afternoon/left.png
  Texture: Right ../textures/skyboxes/terragen_afternoon/right.png
  Texture: Top ../textures/skyboxes/terragen_afternoon/top.png
  LightDirection: 0 1 1 0
  LightSunDiffuseColor: 0.9 0.9 0.9 1
  LightSunSpecularColor: 0.9 0.9 0.9 1
  LightSkyDiffuseColor: 0.6 0.6 0.7 1
  LightSkySpecularColor: 0.6 0.6 0.7 1
  LightGroundDiffuseColor: 0.2 0.2 0.2 1
  LightGroundSpecularColor: 0.2 0.2 0.2 1
  LightAmbientColor: 0.1 0.1 0.18 1
}
Ground {
  Define the road profile, from left to right side of the road.
  RoadTextureLength: 12
  RoadProfile
    Lane: 30 2 1 0 0.311 0.851
    Lane: 10 0.55 1 0 0.851 1
    Lane: 0.0199973 2 2 3 0 1
    Lane: 0.0199973 3.75 3 20 0 1
    Lane: 0.0199973 3.75 3 21 0 1
    Lane: 0.0199973 0.5 2 6 0 1
    Lane: -0.0199973 0.5 6 5 0 1
    Lane: -0.0199973 3.75 5 20 0 1
    Lane: -0.0199973 3.75 5 21 0 1
    Lane: -0.0199973 2 6 4 0 1
    Lane: -0.0199973 0.5 6 5 0 1
    Lane: -0.0199973 3.75 5 20 0 1
    Lane: -0.0199973 3.75 5 21 0 1
    Lane: -0.0199973 2 6 4 0 1
    Marker: 0 0 1.25 3.1415926535
    Marker: 0 1 26.1 0.0
    Marker: 2 1 12.65 3.1415926535
    Marker: 2 0 14.7 0.0
  Define the road; vertex is the coordinates of the road segments. To create a straight road two road segments is not enough, a couple of segments in between must be defined in order to make the simulation run smooth.
  RoadSegment
    Vertex: 0 0 0
    Normal: 0 0 1
    Profile: 0
  RoadSegment
    Vertex: 5166.14 5613.2 0
    Normal: 0 0 1
    Profile: 0
  Define the terrain surrounding the road
  TerrainTexture: ../textures/ground.tga
  TerrainProfile
    Fence: 0
    Length: 1000
    Height: 1
    PreBlendDistance: 0
    PostBlendDistance: 100
    Color: 0.7 0.9 0.4 1
    ColorVariance: 0.06
    Side: Left
}
Place trees, speed signs etc.
Objects {
}

C Commercial models

Name	Company	web page	Simulink Compatible	Real Time	xPC Target Compatible	Evaluation Version	Tyre model	Comments
ADAMS/Car Real Time	MSC.Software	www.mscsoftware.com	X	X	N		Pacejka 2002	Doesn't support xPC Target yet. Existing Chalmers license. Interesting in the future.
ADVANCE	tno automotive	www.automotive.tno.nl	X	X		X	?	Open structure. Forces as inputs, velocities as outputs.
carMaker	IPG Automotive	www.ipg.de	X	X		X	their own model	simple steering model, only steering ratio
Carsim	Mechanical Simulation Corporation	www.carsim.com	X	X			?	Used in the VTI vehicle simulator
dSPACE	dSPACE	www.dspaceinc.com	X	X	N	N	MagicFormula and TMEasy	Very high hardware requirements.
Labcar-vdym	ETAS	en.etasgroup.com	X	X	?	X	HSRI tyre model	
Modelica	DYNASIM	www.dynasim.com	X	X		X	Linear, Rill model, Pacejkas MF	No working version today. Open structure in Dymola.
Open dynamics engine	Open source	www.ode.org		X			?	Very general rigid body dynamics. Previously implemented in the Chalmers simulator, Free licence
Raptor	Southwest Research Institute	www.raptor.swri.org	X	X		X	Simple rolling resistance tyre model	.
SimVehicle	Realtime Technologies	www.simcreator.com	X	X	?	X	Magic Formula	Inflexible standard version
VDMS	Milliken	www.millikenresearch.com	X	X		N	MRA's Non-dimensional Tyre Model	No steering torque calculations
veDyna	Tesis Dynaware	www.tesis.de	X	X	X	X	VeDyna	Used by Volvo Cars. Inflexible, a lot of S-functions

D The proposed function

This chapter is removed because the proposed function is undergoing a patent process.