Department of Precision and Microsystems Engineering

Influence of active camber control on steering feel

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Delft University of Technology

Precision and Microsystems Engineering

Challenge the future

Influence of active camber control on steering feel

Master Thesis

by

Daan Roethof

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Voorwoord

Na de jaren in de bachelor werd het tijd om de master serieuzer aan te gaan pakken. Met de master track Precision and Micro systems engineering wilde ik meer diepte creëren in mijn brede basis van de werktuigbouwde bachelor. Vooral de vakken zoals voertuig dynamica en mechatronica spraken mij erg aan. We kregen vooral de kennis mee hoe het ontwerpproces in elkaar zat en welke kennis er nodig is om deze verder te kunnen ontwikkelen. Met practica en opdrachten ontdekten wij hoe dit in elkaar zat. Van het afstemmen van een regelaar tot het optimaliseren van een actieve voertuig vering. Door middel van de combinatie van vaak taaie stof en interessante theoretische en praktische opdrachten was het erg leerzaam en interessant om jezelf te verdiepen in de stof.

Om de master fase met een goed gevolg te kunnen afsluiten ben ik op zoek gegaan naar een opdracht in de automobiel vakgroep. De stap naar de industrie was de eerste gedachte, maar aangezien de professoren, docenten en andere onderzoekers op de universiteit hun leven maken van onderzoek en lesgeven was ik snel overtuigd dat ik hier mogelijk betere begeleiding zou kunnen krijgen. Daarom kwam ik uit bij Shyrokou en Arat, wie post-doctoral researcher zijn en waarvan ik vakken in voertuig veiligheid en voertuig dynamica had gevolgd. Zij kwamen met verschillende voorstellen op het gebied van voertuig dynamica en banden modellen. Na een zorgvuldige keuze te hebben gemaakt was het antwoord om onderzoek naar actieve camber oriëntatie te gaan doen. De camber hoek is het kantelen van het wiel in de bocht, zoals een motorfiets zou hangen in de bocht. Het voertuig blijft horizontaal en de wielen hangen in de bocht.

Met de keuze van deze master zou het een logische stap zijn om er een ontwerpopdracht van te maken en een mechanisch ontwerp van zo een systeem te maken, maar na een aantal weken literatuuronderzoek kwam ik er al achter dat er al tal van ontwerpen gepubliceerd waren. Opvallend was dat de onderzoeken alleen gericht waren op voertuigdynamica en de persoonlijke interactie met de bestuurder slechts een enkele keer in het onderzoek werd meegenomen. Dat leidde mij in de richting van het onderzoeken naar wat er zou gebeuren als je de bestuurder laat interacteren met een dergelijk systeem. In het laboratorium was een voertuig rij simulator beschikbaar welke ik kon aanpassen tot een actief controleerbare wiel oriëntatie. Op deze manier heb ik mij zowel beziggehouden met het ontwerp van het mechanisme als hoe de mens het ervaart bij het reguleren van de wiel oriëntatie.

Tijdens dit onderzoekstraject ben ik door Barys Shyrokou en Mustafa Ali Arat begeleid. Ik kon altijd bij hun terecht voor vragen en zij hebben mij de vrijheid gegeven zelf het onderzoek richting te geven. Zij hebben mij gestimuleerd een paper te schrijven samen met Tarik Sezer. Dit was in de eerste instantie een lastige opgave, maar door het schrijven ben ik mij bewuster geworden wat de waarde van een publicatie is en hoe dit proces in zijn werk gaat. Daar ben ik Barys en Ali zeer dankbaar voor. De studiegroep waarmee we het eerste jaar de vakken samen mee hebben gedaan ben ik ook erg dankbaar, zonder hen zou het stuk saaier en zwaarder zijn geweest. Gelukkig hebben we dit zowel in China als Japan mogen vieren in twee verschillen studie reizen georganiseerd door studievereniging Taylor.

Daan Roethof
Delft, donderdag 31 december, 2015

PREFACE

After a couple of years in the Bachelor it became time to put some serious effort in the Master. My goal was to create more depth in my wide bases of the Mechanical Engineering Bachelor. The master track Precision and Microsystems engineering would be the best choice to accomplish this. The courses I enjoyed the most were vehicle dynamics and mechatronics. We received the knowledge which is required for the design process and how to develop. With practical training and assignments, we were challenged to gain this knowledge from tuning a controller till optimizing an active suspension. The combination of tough course material and interesting theoretical and practical assignments made it very informative and interesting to challenge yourself.

The automotive group was from the start interesting for me and it was the best choice to do my final thesis in this group. First, I had the idea to do this thesis in the industry, but the fact that the professors, teachers and researchers make their work of research on the university made me change my mind. In the university the supervision would be much better than industry. I followed the courses in vehicle dynamics and vehicle safety with Shyrokou and Arat, who are post-doc researchers and they proposed me some thesis assignments in the area of vehicle dynamics and tire models. After a careful decision I went for the research of active camber control. The camber angle is tilting the wheel, like a motorcycle would lean into the corner. The vehicle maintains its horizontal position and the wheels are leaning into the corner.

With the background of my master track it would make sense to do a design assignment of the mechanism and actuation, but after a few weeks of literature research I found out that there where a variety of designs already published. These investigations were all focused on vehicle dynamics and the human interaction was noticed only once in a publication. This motivated me to investigate what would happen when taking the human into the loop. The driving simulator of the intelligent automotive systems laboratories was available to be modified to an active camber control system. In this manner I investigated the design of an active camber mechanism and how this would be perceived by the driver.

During this research project I was supervised by Barys Shyrokou and Mustafa Ali Arat. They were almost always available and gave me the freedom to let me decide which direction I should take this investigation. They encouraged me to write a paper with Tarik Sezer, which was a tough task, but it made me aware what the value is of a publication and how much effort is needed to write it. I am grateful to them to give me this opportunity. I am also grateful to the study mates of the master, because without them there was a whole lot less fun and it would be much tougher. Luckily, we celebrated this in two different study trips to China and Japan organized by the study association Taylor.

The last two and half year where very special and I can say I am pleased to present the thesis. I invite you to read this thesis and share my findings of the last year with you.

Daan Roethof
Delft, Thursday 31st December, 2015

SUMMARY

In literature many investigations regarding camber control are published and show improvements regarding vehicle performance and safety. These investigations are mainly focused on vehicle dynamics and active camber mechanism concepts. For these studies the driver is left out of the loop, while he/she is of significant importance for controlling the vehicle.

A study with varying camber angles in multi-body simulation software (MSC.ADAMS/Car) showed increased lateral forces and a decrease in overshoots and settling time of the vehicle dynamics for camber angles leaning into the corner. This confirmed the published investigations and it gave insight in the changing understeer gradient when varying the camber angle on the front or rear axle.

Several concepts regarding active camber control mechanism were summarized and a benchmark of these concepts are presented showing the kinematic characteristics per actuation type. The actuation can be for example on the upper or lower wishbone, which have opposing characteristics regarding the wheel displacements. The lower one has more friction with the road and more force taken in the actuator, while the upper one has more conflicts with the mudguard and damper.

The control system should assist the driver in the trajectory and it should always behave regarding the driver's inputs. A desired control scheme would be one that increases the lateral forces when needed. During cornering the lateral accelerations increase, which is directly linked to the required lateral forces. Considering the fact that the centripetal forces of the vehicle's body increase when the lateral accelerations are higher, which makes the lateral accelerations an effective input and therefore three out of the four designed controllers of the camber angle uses the lateral acceleration as an input. The fourth controller compares the desired yaw rate with the actual yaw rate, which is mostly determined by the tire slip angle. By increasing the camber angle dependent on the yaw rate error the tire slip angle decrease, which decreases the yaw rate error.

These control schemes were implemented in the dSPACE ASM simulator. A fixed-base driving simulator with an extensive vehicle model which can be modified to perform the desired experiments. First a proof of concept was conducted, which was driving the track with a constant camber angle applied on all four wheels leaning into the corner. The track started with a straight part to reach a constant speed continued with a circle for constant cornering. From these initial experiments it became clear that the lateral forces were increased for the configurations with additional camber angle leaning into the corner. These changes in forces induced different vehicle behavior responses. The reference vehicle without additional camber angles was not able to deliver the required lateral forces to drive the track and a large lateral position error was the result, while the cambered vehicles were able to reach higher lateral accelerations and a higher yaw rate which resulted in a smaller position error. The overshoots and settling time also decreases with the cambered vehicles as was found before with the MSC.ADAMS/Car simulations. These improvements were in general higher graded by participants.

After the successful proof of concept, the track was extended to an eight shape like track with two different corner radii. This prevented the drivers to get used to a specific steer input and this infinite loop makes it possible to negotiate the corners multiple times per session. The two corner parts were connected with straight parts to prevent distortion of the transient behavior of the previous corner. Furthermore, these straight parts levels out the velocity changes and prevented combined slip in the corner. The reference configuration and the four control systems were clearly distinguished, while the difference between the controlled camber systems showed similar results. The reference configuration showed large overshoots and a long settling time, which leaded to a large lateral position error. This behavior resulted in a low grade for the vehicle control, response time and safety. Especially the steering wheel angle had many large corrections and therefore it was graded as a high steer effort. On the other hand, the control systems had small overshoots and short settling times, which leaded to higher grades in vehicle control and feel of safety. Furthermore, less steering wheel correction were needed and resulted in a smaller position error. A higher grade for steer response and less steer effort were given by the drivers.

Active camber control shows an increase in lateral forces, which resulted in improvements in vehicle performance. The participants gave higher grades in vehicle control, steer response, steer effort and safety experience.

As a recommendation a more detailed description of the suspension dynamics and an improved tire model would be desired for a more realistic suspension response for the vehicle behavior. Furthermore, additional motion cues would deliver more feedback to the driver for a more realistic experience.

Contents

1

Introduction

"He who follows another is always behind."

- Anonymous

1.1. Introduction

It is well known that the forces at the tire-road contact play a major role in a vehicle's safety and driving performance, which have always been prominent subjects in the automotive industry. Numerous studies [1], [2], [3] and [4] have shown that the camber angle of the wheel substantially effects these forces and thereby the range of a vehicle's traction, which is of critical value especially in emergency and evasive maneuvers.

A considerable number of active-camber design concepts have been proposed to evaluate possible improvements on vehicle response. Most of which refer to open-loop maneuvers with the focus on vehicle response and therefore leave the driver out, while the driver's perception can be critical to alter the control and response of the vehicle.

The driver is still responsible for the control of the vehicle, until the era of autonomous vehicles. When including an active camber control system, the driver has to co-operate with this new system. The driver should understand the vehicle's response and he/she should be able to predict the behavior after a steering input. Primarily, the influence of the wheel camber on the steering feel is investigated and therefore on the overall control of the vehicle; which in turn redefines the safety experience.

The usage of a steer robot is a robust way to repeat the same test, which is perfect to compare the responses of vehicle dynamics, but this is different than taking the driver into the loop. A driving simulator with a high fidelity steering model was utilized to realize the effect of camber variations specifically on lateral force response, which is starting at the tire-road contact through the suspension and steering linkages. The vehicle model is modified to change and control the camber angle of the wheels. Several controllers are introduced and implemented in this model, because the controller can change the predictability of the vehicle's response and it should be excluded that a specific controller design disturbs the results. This allowed us expanding this investigation ahead of driving performance (e.g. [5]) and include specifically the steering feel and driver experience on vehicle control systems.

This is not only interesting for the racing industry where the tire force limits play a crucial part in vehicle control. Especially for the daily use of vehicles the tire forces can make the difference in an emergency situation. By increasing the tire forces, the driver has the control to steer away of the hazard or by decreasing the tire forces to change the stability of the vehicle to prevent a roll-over or a spin.

1.2. THESIS OBJECTIVES

This research includes an analysis of the design of an active camber mechanism and examine its effect on the human driver using a driving simulator test platform. The following objective will be a guide line of this research project.

- Evaluation of active camber mechanisms by kinematics and dynamics evaluation.
- Design of a control system for the active camber mechanism.
- Real-time implementation of the active camber model in a virtual vehicle environment.
- Subjective investigation of the active camber design on driving simulator.

1.3. THESIS STRUCTURE

Based on these objectives, the thesis is structured as follows. First, the effect of camber on tire characteristics and vehicle dynamics is explained and shown with multi-body simulation software MSC.ADAMS/Car [6]. Then the camber mechanism will be treated, with a review of the published camber actuation systems and the designed control systems. Followed by an introduction to the simulator. Finally, the results are presented with conclusions on the numerical and experimental tests and with corresponding recommendations.

2

Effect of Camber

"We cannot solve our problems with the same thinking we used when we created them."

- Albert Einstein

2.1. Introduction

In order to make use of the full potential of active systems the effects of the camber angle need to be investigated. Studies [1], [2], [3] and [4] have shown that the camber angle can change the contact area and increase the lateral forces of the tire. In this chapter an investigation to the tire forces is presented. Starting with which angle the camber angle is and how this affects the tire characteristics. And as a result on those tire characteristics it affects the vehicle dynamics. With the use of a multi body vehicle dynamics software program the characteristics are validated. Finally, some tire concepts and a conclusion is presented.

2.2. The camber angle

In order to find out the influence of active camber on the steering feel it is important to understand what the effect of camber is. This starts with the tire axis frame. In this thesis the SAE tire axis frame [4] is adopted as shown in Figure 2.1. The camber angle is the angle of the tire around its X-axis, shown in Figure 2.1 as the γ angle (like a motorcycle leaning into the corner). In the SAE standard the camber angle is defined as "the angle between the wheel and the vehicle body, while the inclination angle is the angle between wheel and perpendicular to the road" as shown in Figure 2.2. The inclination angle is used for the tire data, while the camber angle is used on the vehicle, thus the signs must be converted to use them. The sign of the

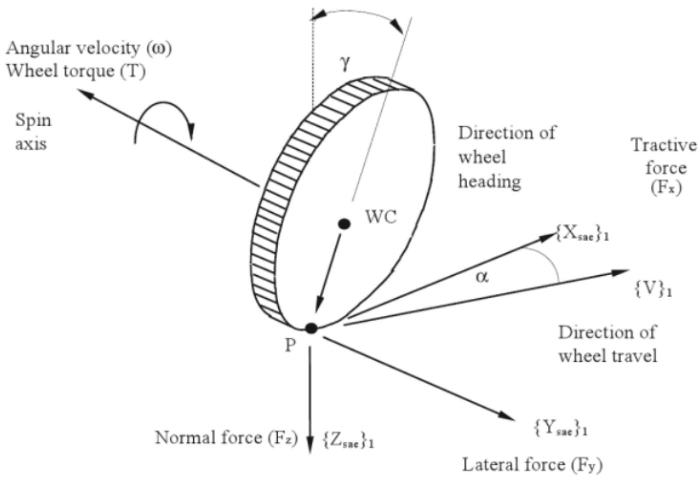


Figure 2.1: Tire axis frame, the SAE standard [7]

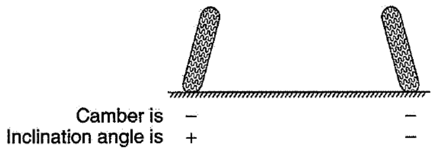


Figure 2.2: Camber inclination definitions viewed from the rear of the vehicle [4]. The camber angle is the angle between the vehicle body and the wheel, while the inclination angle is the angle between the tire and road.

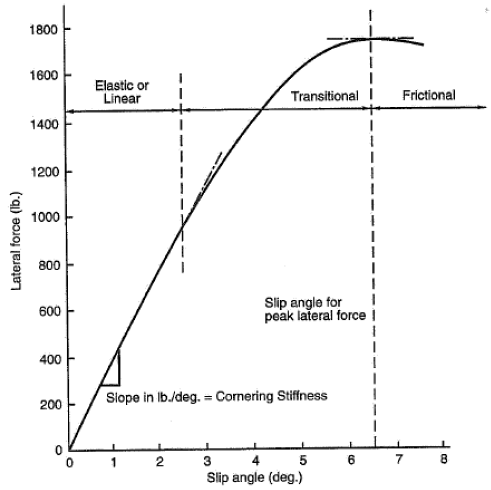


Figure 2.3: The lateral force tire characteristics [4]. It is divided in three parts. First the linear elastic part, then a transition to the peak force and finally a frictional part where the slip angle is too large and the lateral forces decrease.

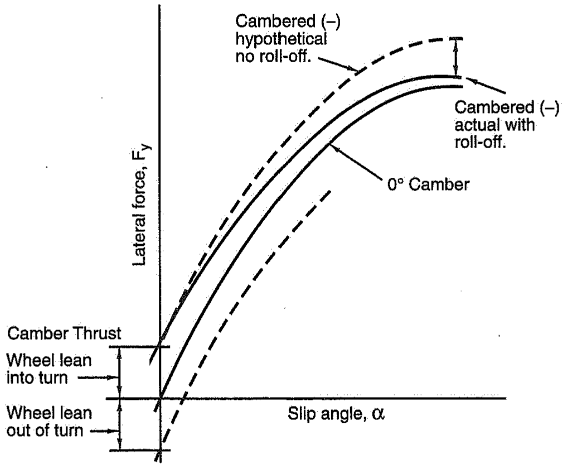


Figure 2.4: The camber thrust [4]. When leaning the wheel into the corner the lateral force increases.

camber angles is mirrored on the vehicle, because a negative camber angle is when the wheel is leaning inwards the vehicle, while a positive camber angle is when the wheel is leaning outwards the vehicle. This means if the tires are leaning into the corner a positive camber angle is used for the inner corner wheels, while a negative camber is used for the outer corner wheels.

2.3. EFFECT OF CAMBER ANGLE ON TIRE CHARACTERISTICS

In existing vehicles, the tire produces a lateral force due to the slip angle α, with characteristics as shown in Figure 2.3. The lateral force starts increasing linear, then it reduces the growth and it reaches its maximum lateral force. When the slip angle become too large the lateral forces decreases again. By changing the camber angle of the tire the lateral force can be shifted. When leaning the wheel into the corner a higher lateral tire force can be produced, while when leaning out of the corner it decreases the lateral tire force. This is called camber thrust. The reason for this camber thrust is due to tire properties like stifness, shape, material use, etc and is explained in the next paragraphs.

If the linear range is exceeded the additive effects of camber inclination decreases. This effect is called the roll-off [4] which is also shown in Figure 2.4. This is a characteristic due to the shape of the tire and the combination of camber angle and slip angle.

The contact area of the tire is called the tire print. The deformation of the tire print leads to the generation of the lateral tire forces. To gain a better understanding of the tire print distortion during cornering we take a look at Figure 2.5. During cornering the steer angle change the orientation with respect to the road. At the front side of the tire print the tire comes in contact and sticks to the road. When rotating the wheel, the tire is pushed sideways due to the inertia of the vehicle body and the tire starts deforming. These deformations increase during the rotation of the wheel till the vertical force decreases and the tire wants to get back to its original plane of the tire. From this point to the end of the print the tire is sliding and the distortion decreases again. This distortion is the displacement u in Hooke's law ($F = -k \cdot u$) and therefore the distortion pattern is equal to the lateral force distribution as shown in Figure 2.6. These

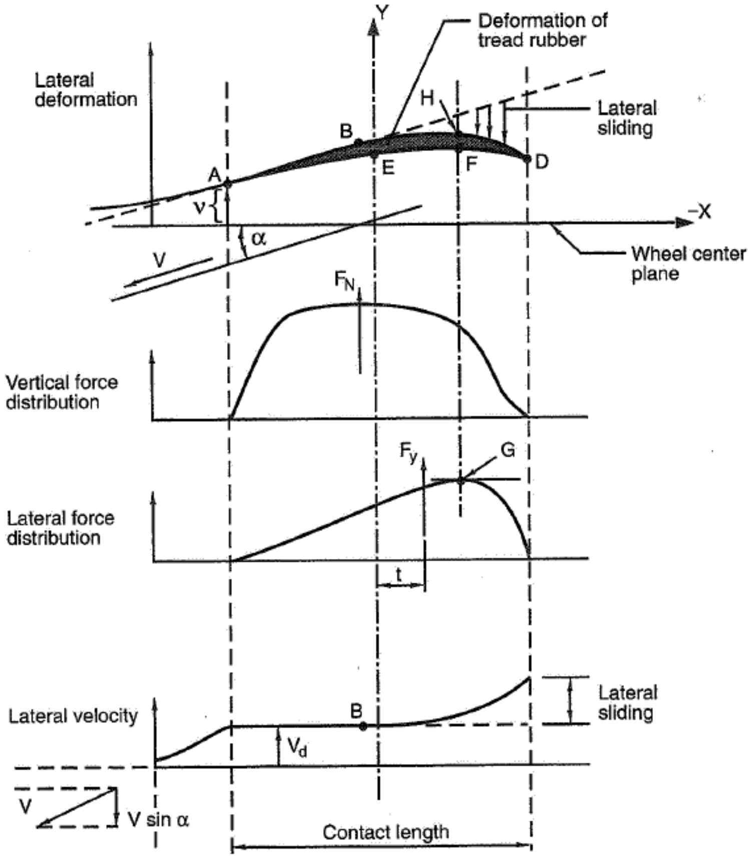


Figure 2.5: The tire print deformation, distribution of the forces and lareral velocity over the contact length [4]. This figures shows the development of the tire forces on a schematic view for a corner to the right. The vehicle is turned to the right with angle α, but the vehicle velocity is still out of the corner. In point A the tire comes is contact with the road and sticks to road as the wheel turns. Due to the inertia of the vehicle the tire deforms and point A follows the velocity line to point B, where the wheel center is shifted to point E. Thus the tire sticks to the road and follows path ABHD and the wheel center follows path AEFD. Therefore, the deformation and thus the force increases over the contact length. Finally, the resultant force of the lateral tire forces is positioned after the wheel center with distance t, which leads to the aligning torque of the wheel.

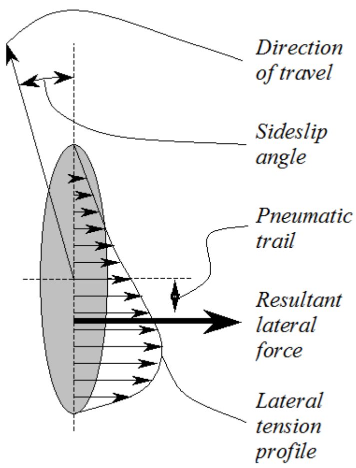


Figure 2.6: Tire deformation patern due to slip [8]. This is a repetition of the previous figure and clearly shows the tire contact area with the distortion partern of the lateral forces.

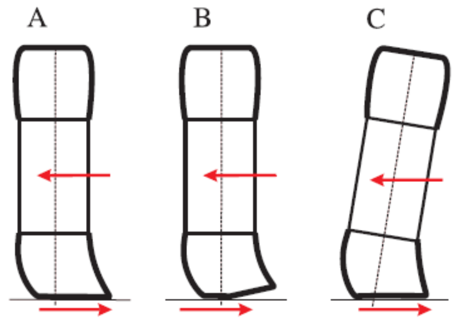


Figure 2.7: Tire deformation during cornering [9]. Due to the lateral forces the tire deforms (A). In extreme cases this leads to extreme deformation and the contact area is decreasing (B). When tilting the tire into the corner the contact area can be recoverd.

characteristics are a result of the tire slip angle. Instead of steering the wheel, a camber angle can be used. This creates an almost symmetric tire print, because the plane of the tire is parallel to the plane of the tire sticking to the road and there is almost no development of slip angle. Usually only camber angle is not sufficient to make a corner, therefore a combination of slip angle and camber angle is used. The addition of the camber angle changes the distortion pattern and shifts the resultant lateral force towards the center of the tire. The distance from the resultant lateral force to the tire center is the pneumatic trail and has influence on the aligning torque as is discussed in the next Section.

The deformation of the tire shape due to the applied forces is another tire property related characteristic. In the neutral camber position of the wheel a large moment will be created due to the applied lateral forces, while the resultant forces of a negative tilted wheel will be mostly loaded in axial direction. This means that the tire is less depended on the cornering stiffness of the wheel in tilted orientation. For a small tire with low cornering stiffness, the camber angle will increase the contact area due to the deformation as shown in Figure 2.7. In "A" the tire is deformed due to lateral force, but "B" shows what actually happens due to the lateral force. In "C" a re-establishment of the contact area is done by tilting the tire. For a wider tire with high cornering stiffness, a large camber angle will result in a lower contact area, due to its stiff square cross-section, it will run on the tire edge. This leads researchers to design and to use rounded cross-section tires e.g. [10]. This design allows maintaining a sufficient contact area on the road surface while tilting the tire. Nevertheless, the shape and stiffness are also very important properties for the tire design of a camber actuated suspension system.

The contact area of the tire is essential for the tire forces, because the friction coefficient of the tire is load sensitive in such a way that a lower load is more efficient than a higher load, which leads to wide tires for example for racing cars. A wider tire distributes the load over the tire and a higher friction coefficient can be established, therefore it is useful to aim for the largest possible tire contact area. For example the normalized lateral force with respect to the vertical load is presented for a race tire in Figure 2.8. The tire with lower load reaches a higher normalized lateral force, which means a higher lateral force per load. The contact area also plays a cardinal role in the variations of tire wear, temperature, pressure and other features indirectly related to the tire. Next to increasing lateral forces is the contact area also important for tire wear, temperature etc. Misuse of tires can decrease the tire lifetime and increase locally temperatures which changes tire properties.

There are two slip definitions for tires, a longitudinal slip related to longitudinal forces, and a lateral slip or slip-angle related to lateral forces and moments. Figure 2.9 shows the combined slip of longitudinal traction and lateral force. An important feature to note is the decrease of lateral force during braking or accelerating. The shape of the circle is tire specific, for example for a wide race tire the diagram is larger

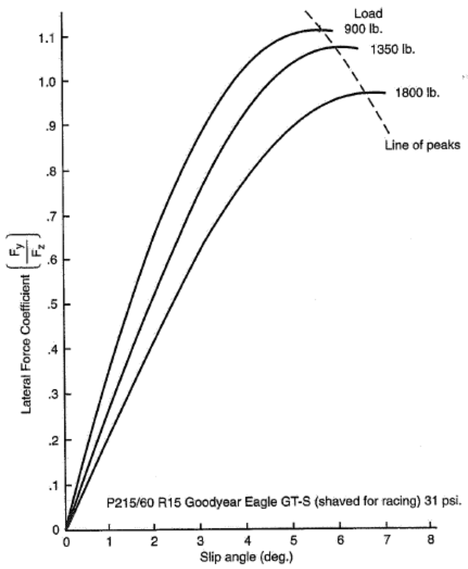


Figure 2.8: The normalised lateral force with different loads [4]. This graph shows that a better load distribution over the tires can improve the lateral force coëfficient for this race tire.

and more elliptical, so it reaches higher lateral forces. Furthermore, the camber angle can also change the friction circle. For rectangle cross-section tires the longitudinal forces will decrease faster than the lateral forces while tilting the wheel, because for the longitudinal motion it will drive on the edge, but during cornering it will deform the tire sideways and creates more contact area. Rounded tires will not see much difference in longitudinal traction, but the lateral forces will increase while tilting the wheel into the corner.

The tire characteristics responses can be simulated with tire models. There are multiple tire models available with their own method of simulating the responses and these are classified as analytical and empirical models in general. As a good example to empirical models, the Magic tire formula is a curve fitting formula and exists of 20-100 coefficients and can be extended to over 300 coefficients [3]. Where the coefficients represent the stiffnesses, compliances and many more tire properties. Therefore, this formula can fit a wide variety of tire constructions and operating conditions. These coefficients are known by the manufacturer of the tires and are obtained by tests, which cost a lot of effort to measure and the results of these tire models are not publicly available.

There are multiple studies showing that the lateral vehicle dynamics can be improved with larger camber angles. For example, the authors [1] carried out a comparison study of yaw moment on a circular track with different camber angles. In another study they compare tires under different inclination angles [11]. These studies show that a larger camber angle can increase the lateral forces, the lateral acceleration and the yaw rate. However only at the cost of increasing the vehicle roll simultaneously due to the centripetal forces depending on the height of the center of gravity.

2.4. EFFECT OF CAMBER ON VEHICLE DYNAMICS

The suspension contains two important geometric parameters: caster angle and kingpin angle which are determined by the suspension links' joints. This is presented in Figure 2.10. These parameters also determine the force arms, which are also known as the trails. These trails are important for the torques on the wheel. Different trails can change the aligning torque or the toe angle during acceleration or braking. The stiffness of the suspension links plays an important role in the influence of the applied forces. How the caster and kingpin angles change principally depends on the way of actuating the camber angle.

As shown in Figure 2.10 the scrub radius is the trail due to the kingpin angle. The forces working on this trail are due to acceleration and braking, and are effecting the toe angle. The next chapter is an overview presented with the types of actuation of the camber angle, which have different sensitivities for scrub radius change. When changing the position of the upper or lower ball joint to create camber angle there will not be so much difference in scrub radius, but when changing on the knuckle the scrub radius can change dramatically. The toe angle is accountable for the steering behavior of the vehicle. In a toe-out

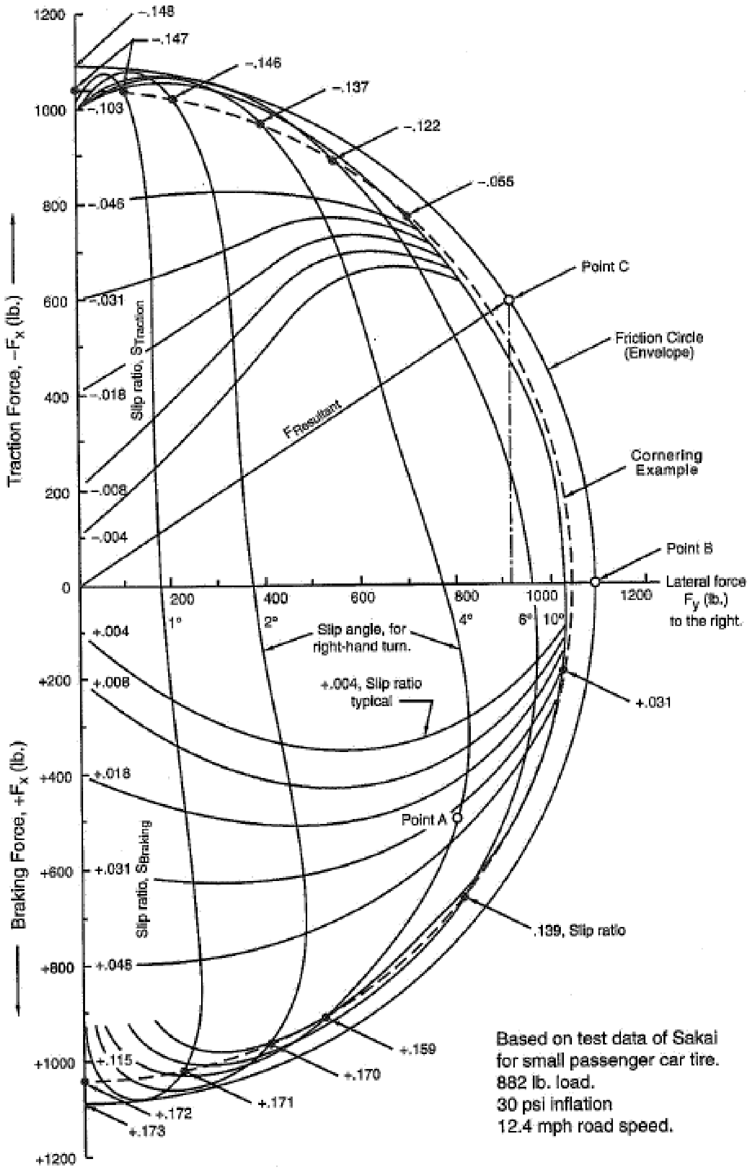


Figure 2.9: The friction circle for a small passenger car tire [4]. When accelerating or braking during cornering results in combined slip in the tires. This leads to a decrease in lateral and longitudinal forces.

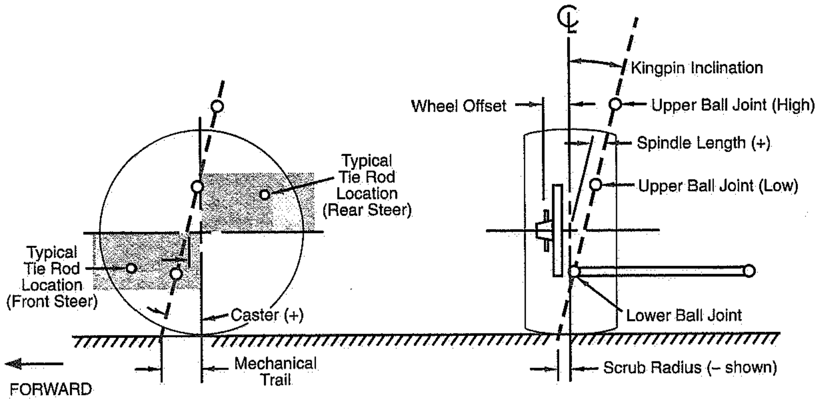


Figure 2.10: Front suspension packaging [12]. The positions of the joints on the knuckle determine the angles and offsets of the tire, but only for this position. After vertical wheel travel the joints can have different positions relative to each other dependent on the suspension design.

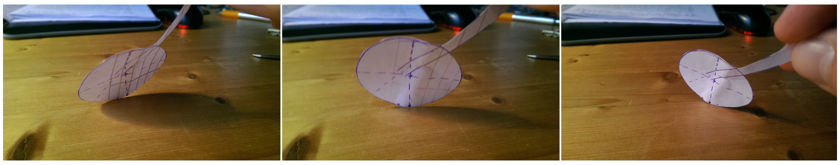


Figure 2.11: The position of the contact point is changing during the steering motion of a cambered wheel. The caster angle and mechanical trail create the axle where the tire rotates around. For example the case in this figure tilted to the left. This makes the contact point change to the front when steering to the left and change to rear when steering to the right.

configuration, the tires are both steering out of the vehicle and during cornering the outer wheel got more load, which therefore will dominate and steer the vehicle straight. While for the toe-in configuration the tires both steering into the vehicle and during cornering the vehicle steers more into the corner.

The caster angle can be kept constant, still the mechanical trail will change during steering. Existing vehicles use just a small caster angle and therefore the contact point shift is limited, but when introducing the camber angle the contact point shifts more. For a left tilted wheel steering to the left it shifts to the front, while steering to the right it shifts to the back, as illustrated in Figure 2.11. Tilting it the other way it will shift the other way for a positively caster angle. This will also have a direct influence on the aligning torque, not only the mechanical trail changes, but also the vehicle height, because when the contact point shifts to the front the vehicle lowers and if it shifts to the back it has to lift the vehicle. Lowering the vehicle will be at a lower energy state, so it prefers steering that way.

In the previous section it was shown that the force distribution through the tire was better for a tilted wheel and the tire print changed to a more symmetric shape. This symmetric shape is more efficient for the distortions and therefore it can reach higher lateral forces. This is highly dependent on the tire as discussed in the previous section. The shape, stiffness, damping, load and temperature are just examples of variables that influences the produced tire forces. Therefore, it is important to know what kind of tires are available and how these tires can be loaded, which is shown in several publications [11], [4].

The tire distortion pattern changes when tilting the tire, which leads to a change in location of the resultant force. Where the resultant lateral force in Figure 2.5 is at distance t (pneumatic trail) from the center line, it shifts closer to the center line when it is combined with a camber angle. A smaller arm to the point of rotation is realized, which leads to a smaller aligning torque. The aligning torque is created by the lateral force multiplied with the trail. This trail consists of the pneumatic and mechanical trail. The mechanical trail is shown in Figure 2.10, which is the distance from the tire center to the point where the kingpin axis under a caster angle intersects with the plane of the tire print.

The camber angle changes the contact area, which results in a more effective use for lateral force. Introducing camber to the wheel produces lateral forces without any steering input, therefore a smaller steering angle is needed to induce the remaining lateral force for driving the corner. This means the driver needs to provide less steering angle for the same curve and thus it creates a higher sensitivity in the steering. On the other hand, there is less slip angle in the tires, which makes the desired yaw rate closer to the actual

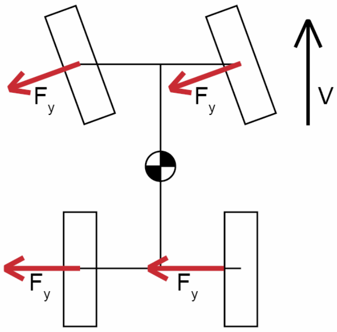


Figure 2.12: The steer angle decreases the longitudinal velocity, because the lateral forces create a longitudinal force component in opposite direction.

yaw rate, because the difference between actual and real yaw rate is the slip angle in the tire as presented in Equations 2.1 - 2.4. These equations present the yaw rate based on the bicycle model [4]:

$$\delta_{\mathrm{des}} = \frac{L}{R} \tag{2.1}$$

$$\dot{\Psi}_{\mathrm{des}} = \frac{V}{R} = \frac{V}{L} \cdot \delta_{\mathrm{des}} \tag{2.2}$$

$$\delta_{\mathrm{real}} = \frac{L}{R} - \alpha_{\mathrm{f}} + \alpha_{\mathrm{r}} \tag{2.3}$$

$$\dot{\Psi}_{\mathrm{real}} = \frac{V}{L}(\delta_{\mathrm{real}} + \alpha_{\mathrm{f}} - \alpha_{\mathrm{r}}) \tag{2.4}$$

Another effect of this decrease in steering angle is the direction of the forces. A lower slip angle and a lower steering angle results in a more aligned orientation with the vehicle. This makes the lateral forces point more sideways of the vehicle as shown in Figure 2.12, therefore there is less force in opposed longitudinal direction which would decreases the vehicle's velocity.

The roll of the vehicle is dependent on suspension characteristics and vehicle properties. During cornering the lateral forces increases due to the centripetal forces of the vehicle body. The height of the CoG with respect to the road creates the arm for this force and a roll angle is the result. When changing the camber angle by leaning the wheels into the corner the contact area of the tires shifts with respect to the vehicle's CoG and the height of the vehicle's CoG lowers as shown in Figure 2.13. This effects the equations of the roll moment. The following equation is the reference configuration in its static equilibrium, with the rotation point in the contact area of the right tire:

$$\sum M_{\mathrm{RT}} = 0 \tag{2.5}$$

$$F_{\mathrm{CoG}} \cdot h + F_{\mathrm{NL}} \cdot w - F_{\mathrm{g}} \cdot w/2 = 0 \tag{2.6}$$

This can be compared with the cambered angle configuration in its static equilibrium, with the rotation point in the contact area of the right tire:

$$\sum M_{\mathrm{RT}} = 0 \tag{2.7}$$

$$F_{\mathrm{CoG}} \cdot (h - \Delta h) + F_{\mathrm{NL}} \cdot w - F_{\mathrm{g}} \cdot (w/2 + \Delta x) = 0 \tag{2.8}$$

Δh and Δx both increases when a camber angle is introduced which means the CoG lowers in height and the contact area shifts sideways. This leads to a larger force F_{CoG} to satisfy the equilibrium. and larger lateral accelerations can be reached without increasing the roll angle. This is the case for a rigid body, but suspension is needed for performance and comfort. The suspension is a preloaded spring system and enforces roll, therefore an anti-roll bar reduces this behavior by leveling the differences in vertical wheel travel of the left and right tire. The Mercedes F400 concept vehicle [10] changes only the outer corner wheels and has

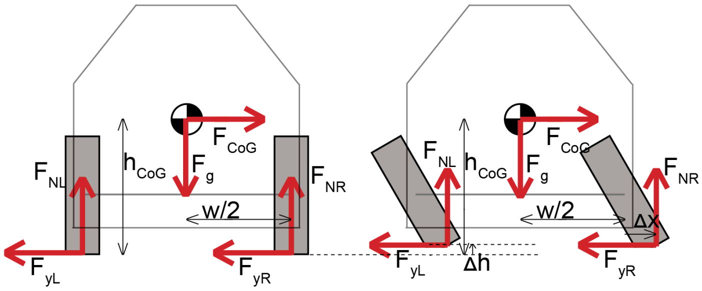


Figure 2.13: Vehicle roll imposed by the forces acting on the CoG and tires during cornering. This equilibrium of forces imposes a moment and is dependent on CoG height and lateral change of the contact point.

to counteract to the height difference with active dampers, furthermore these dampers can also adjust for the roll.

The stability can also be influenced by camber angle. By studying the steady state response [4] we find the dependency of the lateral forces and moments.

$$F = ma \quad -> \quad Y = ma_y = mV(r \cdot \dot{\beta}) \tag{2.9}$$

$$T = I\alpha \quad -> \quad N = I_z\dot{r} \tag{2.10}$$

Assuming that $\dot{r}$ and β are zero, the equations of motion become:

$$mVr = Y_\beta\beta + Y_r r + Y_\delta\delta \tag{2.11}$$

$$0 = N_\beta\beta + N_r r + N_\delta\delta \tag{2.12}$$

The forces and moments in the equation of motion can be expressed in front and rear cornering stiffness

$$Y_\beta = C_F + C_R \tag{2.13}$$

$$Y_r = \left(\frac{1}{V}\right)(aC_F - bC_R) \tag{2.14}$$

$$Y_\delta = -C_F \tag{2.15}$$

$$N_\beta = aC_F + bC_R \tag{2.16}$$

$$N_r = \left(\frac{1}{V}\right)(a^2C_F - b^2C_R) \tag{2.17}$$

$$N_\delta = -aC_F \tag{2.18}$$

Steady state turn $\dot{\Psi} = r = V/R$ can be substituted into Equations 2.11 and 2.12 and rearranged, which leads to:

$$-Y_\delta\delta = Y_\beta\beta + (VY_r - mV^2)\left(\frac{1}{R}\right) \tag{2.19}$$

$$-N_\delta\delta = N_\beta\beta + VN_r\left(\frac{1}{R}\right) \tag{2.20}$$

After solving Equation 2.20 to β and substitute it into Equation 2.19 it gives:

$$\beta = -\left(\frac{N_\delta}{N_\beta}\right)\delta - V\frac{N_r}{N_\beta}\left(\frac{1}{R}\right) \tag{2.21}$$

$$-Y_\delta\delta = \left(\frac{Y_\beta}{N_\beta}\right)N_\delta\delta - V\frac{Y_\beta}{N_\beta}N_r\left(\frac{1}{R}\right) + (VY_r - mV^2)\left(\frac{1}{R}\right) \tag{2.22}$$

And then Equation 2.22 can be rearranged to steer response.

$$\frac{1/R}{\delta} = \frac{Y_\beta N_\delta - N_\beta Y_\delta}{N_\beta(VY_r - mV^2) - VY_\beta N_r} = \frac{Y_\beta N_\delta - N_\beta Y_\delta}{V(N_\beta Y_r - N_\beta mV - Y_\beta N_r)} \tag{2.23}$$

By substituting the derivatives in Equation 2.13-2.18 the formula can be written as:

$$\frac{1/R}{\delta} = \frac{1/l}{1+KV^2} \tag{2.24}$$

with understeer gradient:

$$K = \frac{m}{l}\left(\frac{N_\beta/(-Y_\beta)}{N_\delta-(N_\beta/Y_\beta)Y_\delta}\right) \tag{2.25}$$

Finally, one can see that changing the lateral forces on front or rear tires will change the understeer gradient K of the vehicle. This is important for the characteristics regarding oversteer and understeer. For stable driving behavior vehicles need to be understeered.

When the understeer gradient is written in the form of cornering stiffness:

$$\begin{aligned} K &= \frac{m}{l}\frac{-\left(\frac{aC_F+bC_R}{C_F+C_R}\right)}{-aC_F-\left(\frac{aC_F+bC_R}{C_F+C_R}\right)(-C_F)} \\ &= \frac{m}{l}\frac{-1}{(1-a)C_F} \qquad \text{with} \quad a=b \quad \text{and} \quad C_F=C_R \end{aligned} \tag{2.26}$$

It shows that it is mainly dependent on the change in front cornering stiffness. This would mean a higher cornering stiffness in the front would decrease the K. Furthermore, this gradient is a function of the steering angle and Equation 2.24 can be written as:

$$\delta = \frac{1+KV^2}{R}L \tag{2.27}$$

which presents that the steer angle will be lower with a lower K. Thus less understeer and if it passes $K = 0$ it becomes oversteered.

2.5. MSC.ADAMS/CAR PROOF OF CONCEPT

2.5.1. MSC.ADAMS/CAR

MSC.ADAMS/Car is a multi-body software tool to simulate vehicle characteristics. In this virtual environment a vehicle can be build and modified to find the corresponding vehicle characteristics. With this vehicle model the parameters can easily be changed and tested. This makes it very useful to find sensitivities of certain parameters and see whether designs are possible to produce with desired vehicle characteristics. It is a virtual environment, therefore it can provide rapid solutions to design a mechanism and find related vehicle characteristics without building a real life vehicle and using test rigs or reserved tracks.

2.5.2. TEST SET-UP

For this test set-up the standard vehicle example is used which is supplied with the MSC.ADAMS/Car software. In Figure 2.14 the vehicle is shown with camber angle applied on the front wheels leaning into the corner. This test is focused on the changes on vehicles characteristics and therefore the camber angle is applied by varying the angle on the knuckle. This design yields to design challenges in the mudguard and damper geometries. Nevertheless, as the study's focus is on the vehicle response with cambered wheels, such geometrical challenges have been omitted for the time sake.

The tests are performed on a circular track and with increasing initial speed. A circular track with a 20m radius cornering to the left is used to see whether the lateral forces changes and what the influence on the vehicle will be. A circular track can only represent a small part of the vehicle characteristics, but is perfect to do these initial tests to give insight about camber influence. The vehicle starts at 40 km/h and accelerates to 60 km/h in the 2nd gear in 20 seconds. An acceleration is not desired for investigating the lateral forces, because this will lead to combined slip in the driven tires. The increase in speed is kept the same for all configurations and can therefore be compared with each other. It shows the differences between the configurations at what speed the vehicle loses track, because the lateral forces exceed the maximum possible.

The camber angle can be applied on four different tires in positive and negative direction. During these initial tests, the camber of left and right tires are kept in the same direction, but allowed to differ between front and rear tires. This lead to six different configurations, with positive angles leaning the wheels out of the corner and negative angles leaning the wheels into the corner.

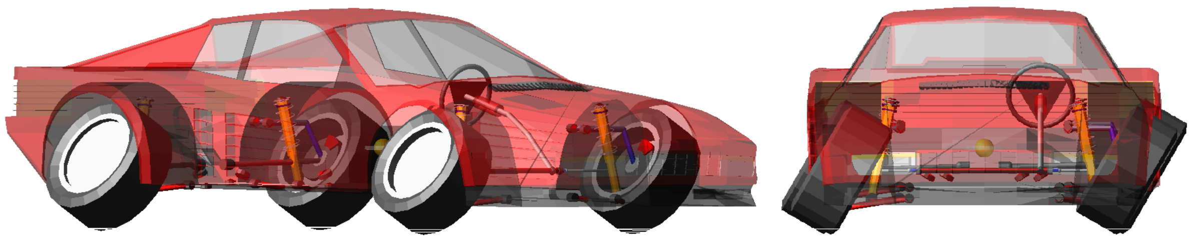


Figure 2.14: MSC.ADAMS/Car camber angle proof of concept vehicle with four wheels leaning into the left corner

2.5.3. TESTING AND RESULTS

The test was done with the following configurations to check the different characteristics.

- F 5°, R 0°
  5° positive camber angle on front wheels (−5° left, 5° right) and zero on the rear wheels.
  This resulted in losing the track with *understeer* at about 47.5 km/h.
- F 0°, R 0°
  zero camber angle on front and rear wheels.
  This resulted in losing the track with *understeer* at about 50 km/h.
- F −5°, R 0°
  5° negative camber angle on front wheels (5° left, −5° right) and zero on the rear wheels.
  This resulted in losing the track with *oversteer* at about 54 km/h.
- F −10°, R 0°
  10° negative camber angle on front wheels (10° left, −10° right) and zero on the rear wheels.
  This resulted in losing the track with *oversteer* at about 54 km/h.
- F −5°, R −5°
  5° negative camber angle on front and rear wheels (5° left, −5° right).
  This resulted in losing the track with *understeer* at about 54 km/h.
- F −10°, R −10°
  10° negative camber angle on front and rear wheels (10° left, −10° right).
  This resulted in losing the track with *understeer* at about 57 km/h.

In the list above is shown that the vehicle with 5° camber angle facing out of the corner is losing the track with the lowest velocity. Figure 2.15 shows where the lateral tire force reached its maximum and came in the slide region where it lost the track with understeer. The reference vehicle with both front and rear 0° camber angle can reach a higher force and then also slides off with understeer. When changing to negative camber leaning into the corner, the rear wheels will reach the slide region first and losing the track oversteered with a spin as shown in Figure 2.16. By setting the camber angles for front and rear on camber angles leaning into the corner the vehicle maintains its stability for a much higher speed and then slides off with understeer like the reference vehicle would do. The 10° camber angle vehicle can reach a higher lateral peak force, but the lateral force in the frictional region is lower under a larger camber angle and therefore it starts sliding further off track than the 5° type as shown after about 16 seconds in Figure 2.15.

2.5.4. DISCUSSION

In this test the vehicle accelerates on a short circle, therefore the longitudinal and lateral slip of the tires are combined. Both longitudinal and lateral forces are applied to the tires. These combined forces limits the lateral tire behavior as was shown in Figure 2.9. This is for all types equal to each other and this makes it valid to compare with each other.

These results give a good representation of change in the understeer gradient and proves that the difference in front and rear camber can definitely influence stability. The wheels where camber is applied to add lateral force. Furthermore, the front cornering stiffness has a stronger influence on the understeer gradient. When fine tuning the controllers this can be used to keep the same understeer gradient, by increasing the rear camber stronger than the front camber. This change in understeer gradient is an important characteristic

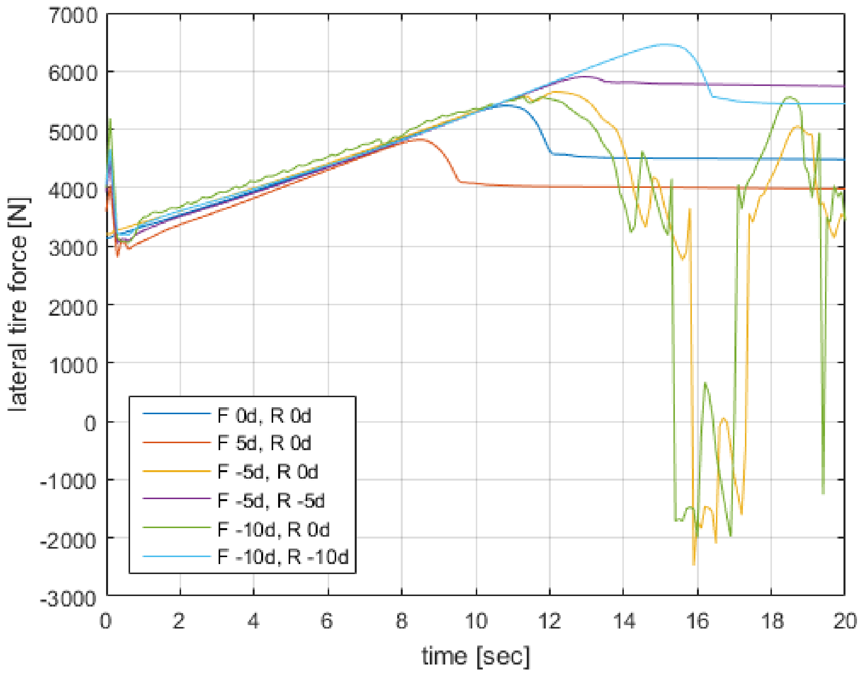


Figure 2.15: MSC.ADAMS/Car lateral tire force comparison of the front right tire for a left constant corner with an increasing velocity. The different configurations are explained in the enummeration and show that each configuration get off the track with a different speed.

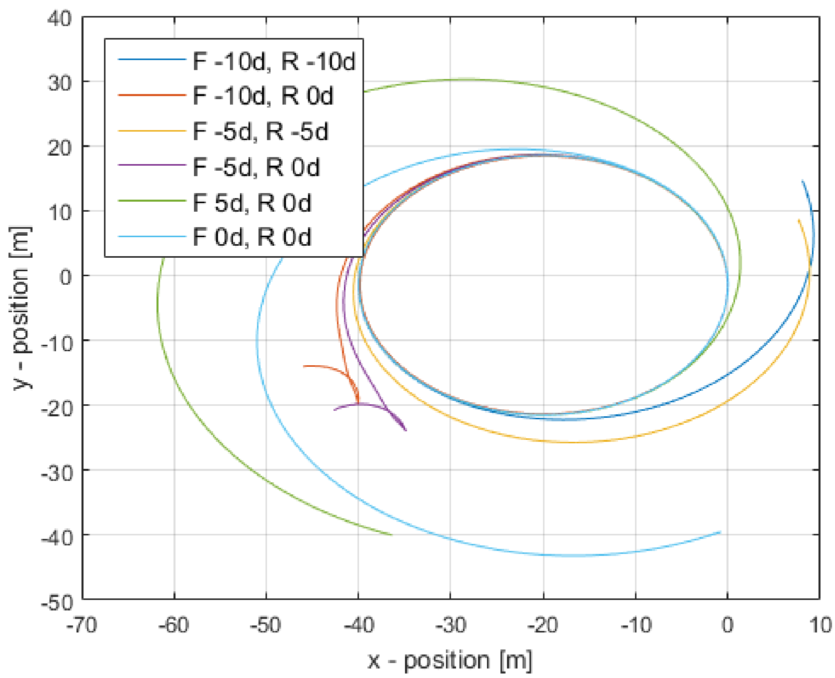


Figure 2.16: MSC.ADAMS/Car trajectory comparison for a left constant corner with an increasing velocity. The different configurations are explained in the enummeration and show that each configuration get off the track with a different speed.

which can be used in control schemes for a safe drive or change vehicle response from an oversteered to an understeered vehicle. High-end sports vehicles can even be interested to use this characteristic to trim the vehicle while driving, so for example adapting for tire wear during driving or for specific moments where for example the front of the vehicle is slightly lifted due to a curbstone and loses grip.

The interest of this thesis is in the influence of the camber on steering feel, therefore the camber angle will be actuated on all four wheels to minimize the change in understeer gradient.

2.5.5. REDESIGN TEST SET-UP

In the previous paragraph it was shown that using camber on all four wheels was the best result without changing the understeer gradient. Therefore, tests at higher speeds and a larger radius were done to see larger differences in results. The vehicle starts at 95km/h at the circle and accelerates to 96km/h in 100seconds, which is about steady state cornering.

2.5.6. TESTING AND RESULTS

For these tests three configurations are introduced:

- $0°$ of camber angle
  reference vehicle.
- $-10°$ of camber angle into the corner
  all the wheels leaning with $10°$ into the corner
- $-30°$ of camber angle into the corner
  all the wheels leaning with $30°$ into the corner.

Figures B.1 - B.10 show only the first 25 seconds of the 100sec, because there is the transient behavior shown. It is clearly shown that the overshoots and settling time decreases for an increased camber angle. The reference configuration was at its limits and could not reach a steady cornering, it did not even reach the 96 km/h and went off the road after 80 seconds.

As the previous tests showed the lateral forces increased for camber angles leaning into the corner, since the three setups could make the corner the total of the lateral forces was for the three vehicles the as same shown in Figure B.7 - B.10, therefore the lateral accelerations were equal for the three configurations. What stand out is the difference in tire load distribution, which is associated to the vehicle roll. Section 2.4 explains that the roll is associated with the centripetal force and vehicle properties. This force was equal for all configurations, because they all have the same mass and velocity in the same radius corner. The camber angles changed the height and the lateral position of the CoG above the tire contact area. This resulted in a lower roll moment and thus a lower roll for the configurations with camber angle.

The transient behavior of the three types is also different. The steer angle and yaw rate shows differences in overshoot and settling time as shown in Figures B.4 and B.5. For the $0°$ camber angle vehicle a larger steering angle was needed and this system was not damping out which leaded to getting off track. Furthermore, the steer angle decreased with camber angles leaning into the corner, as was explained by the camber thrust which already pushed the vehicle into the corner, therefore less slip angle was needed and resulted in a smaller required steering angle.

Finally, the steer torque changed due to the change in tire print. Also a lower trail resulted in a lower aligning torque for camber angles leaning into the corner.

2.6. TIRE CONCEPTS

The ideal tire would be a tire that is equal to a conventional tire when driving straight, which means a wide tire for maximum contact area and rounded edges for tilting during cornering to gain more lateral force, but still getting maximum contact area. There are multiple ideas in literature on tires for active camber systems for example:

- hybrid tires,
  these tires are a hybrid version of a car tire and a motorcycle tire as shown in Figure 2.17. In reference [11] two tires are compared with the same dimensions, but with a different carcass design. The main advantage of these tires is that the contact area is maintained when tilting the wheel as shown in Figure 2.18.

Figure 2.17: Hybrid motorcycle/car tire [11]. These tires might be a suitable option for an active camber mechanism. The smooth curve maintain the same contact area for a wide range of camber angles.

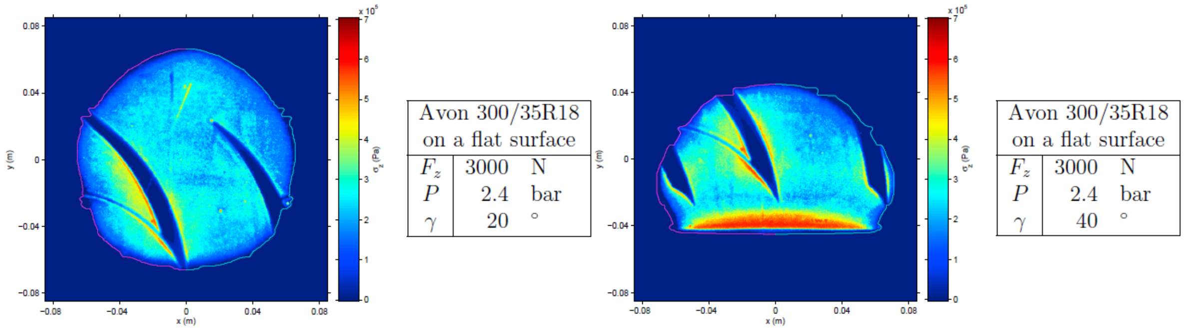


Figure 2.18: Contact patches of an Avon 300/35R18 motorcycle at 20° and 40° camber angles [11]. These figures show the change in contact area when reaching the maximum camber angle. The colors show the pressure distribution and present an accumulation near the edge of the tire for a too large angle.

- single side rounded car tires,
  Mercedes developed their own tires for their F400 concept vehicles as shown in Figure 2.19, which make use of conventional car tires with a rounded inner side. The outer corner wheels handle most of the lateral forces, therefore only these wheels are cambered and only the inner side of the wheels need a round side. Furthermore, the design of the vehicle does not change drastically, because the tires look like conventional tire from the side and the rounded part is hidden under the vehicle [10].

- conventional car tires,
  conventional tire designs aim for small camber angles to adjust for contact area and control vehicle stability [13] [2]

So there are many options for tires, but one of the most important properties is the contact area for both straight and under large camber angles. Rounded tires present a round contact area, while a conventional car tire presents a rectangular contact area as presented in Figure 2.20. This shape change does not matter, but the location and direction of the resultant force is responsible for the resulting moment, which will change the vehicle dynamics response as discussed in Section 2.4.

In addition to its shape there are multiple ways of constructing the tire. Bias ply and radial are the two most common ways of construction as shown in Figure 2.21. Both constructions have their advantages and disadvantages for using with camber angle. Since this is out of the scope of this research it is given as a recommendation for the tire design.

2.7. CONCLUSION

The camber angle can increase lateral force and this results in an influence on vehicle behavior. This increase or decrease in lateral tire forces can control for example the lateral acceleration, understeer gradient, yaw rate and roll angle. Due to these lateral forces there is already a lateral component and less steering angle is needed to make the corner. Furthermore, it will change the tire contact area which leads to a change in

Figure 2.19: Mercedes F400 concept vehicle with active camber control [10].

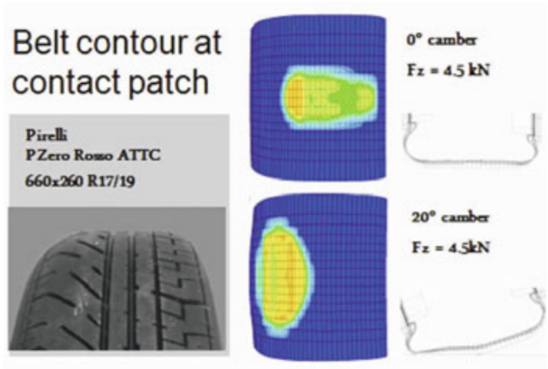


Figure 2.20: Mercedes rounded edge tire [10]. Combining the best of two tire types, a conventional tire shape for maximum traction and a rounded edge for applying camber during cornering.

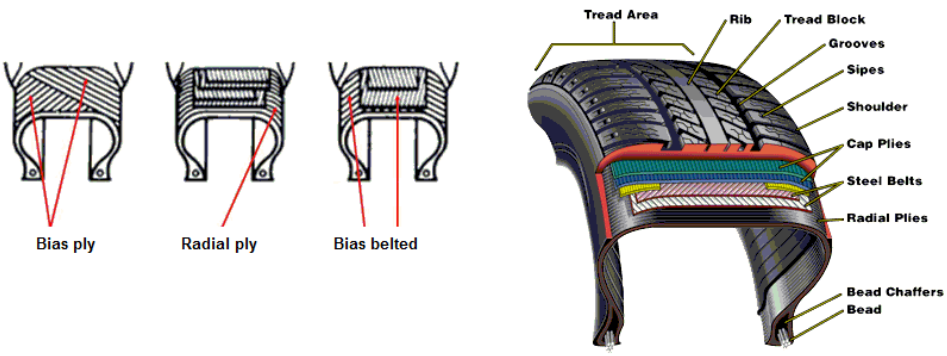


Figure 2.21: There are many tire constructions possible [14]. First, it was mainly bias ply or radial ply, with the option of belted as shown in the left three figures. Nowadays, many combinations are possible as shown in the right figure.

aligning torque.

This change in orientation requires to redesign tires. Existing tires will be misused and this leads to high temperature differences and tire wear, but also a decrease of contact area when tilting rectangular cross section tires to a too large angle. Some tire concepts were discussed to compromise this.

The tests with MSC.ADAMS/CAR showed that the sensitivity to steering can be changed by adding camber control to front or rear wheels. When applying camber on all four wheels it minimizes the changes in under- and over-steer behavior of the original vehicle. Furthermore, it shows that adding camber keeps the vehicle longer on track with camber angles leaning into the corner and losing earlier track by leaning out of the corner. This can be used in control schemes to control the understeer gradient. Oversteered vehicles can be tuned to understeered and roll-overs can be prevented by leaning the camber out of the corner at the front tires.

3

Camber mechanism

"Design is not just what it looks like and feels like, design is how it works"

- Steve Jobs

3.1. Introduction

Over the last years there are many investigations done in camber control mechanisms, which lead to numerous concept designs. Some of these concepts are prototyped and others are only a working principle. An overview is given of the concept designs found in literature to provide a good insight in possibilities and working principles. In this chapter a guideline for a feasible design is constructed for a medium to large car like the BMW 1 or 3 series. Next comes the description of the control methodology, which is essential for the position and dynamics of the camber angle.

3.2. Review camber mechanisms concepts

Researchers have proposed various mechanisms to actuate the wheel camber, which generally differ in a number of aspects including: the camber angle range, point of actuation, area of use and type of actuation. A comprehensive list of existing camber mechanisms is summarized in Table 3.1. The point of actuation is of significant influence on the design, because of space requirements, which limits the camber range. This can be divided into three positions: upper, middle and lower actuated. The upper and lower actuated mechanisms are the most straightforward designs and mostly published. In general, these two systems have contrary characteristics regarding the kinematics.

The upper actuated mechanisms turn around its lower wishbone and therefore has more displacement on the upper side of the wheel. This leads to more space requirements in the mudguard or it might collide with the spring-damper. The camber angle actuator also needs to fit in this relative small space, but does not have to be as powerful as the lower actuated version, because it has a longer moment arm. Furthermore, there is less lateral displacement with respect to the road, which leads to less friction.

The lower actuated mechanisms turn around the upper wishbone and this leads to less displacement at the upper side of the wheel, therefore the displacement with respect to the road is larger and this displacement causes friction which eventually requires a more powerful actuator. A major advantage of this design is that it can reach higher camber angles without considerable changes to the suspension and bodywork. On the other hand, the lower wishbone is the position where the largest forces are transferred to the body, which has to be handled by the actuator. Also the vertical forces are normally taken by lower wishbone, because then the spring-damper has a larger workspace. Thus the lower wishbone has a large demand of shear and normal forces. The position on the lower side is beneficial to place a larger actuator to handle this large demand, but this makes it also directly vulnerable for hitting objects on the road.

The middle actuated mechanisms apply camber by turning the wheel around its middle point. This can be established with a double actuated system, with opposing upper and lower actuators or a mechanism which makes the tire tilt at the tire center. This system has the intermediate characteristics regarding tire-road displacement and space requirements, but it takes all the disadvantages of force requirements. A double actuated system requires relatively powerful actuators as all lateral and vertical forces are taken by the actuators.

There are different design mechanisms for the wishbone to knuckle connection. Each design has its own characteristics. A wishbone can for example exist of two rods moving independent of each other, which give the wheel an extra degree of freedom and leads to a different motion than when the wishbone is of one part and it has only one joint at the knuckle. When using two independent rods with both their own actuator it introduces new opportunities, such as control of active steering and active toe angle.

The area of use is important to define the requirements of the mechanism. Sports vehicles tend to be lightweight and are usually not designed for durability or cost effectiveness. The result of this addition should increase the performance, since this is the point of focus for sports vehicles. The off-road vehicles can be considered on the other side of this picture regarding such requirements. These vehicles drive in the most unknown circumstances and should be durable and reliable when driving in remote areas. The weight is less of an issue and an easy repairable or replaceable system is more important. Urban vehicles have varying requirements and that fall in between the two ends of sports and off-road vehicles with more focus on cost. Besides that, the urban vehicles are also focused on safety and comfort.

These actuator concepts were divided into groups of five types and built in MSC.ADAMS/View [6]. This multi-body 3D software provides insight into the motion and limitations of the mechanism. That allowed us to analyze camber angles and actuator displacements which are listed in Table 3.2. Using the same parameters for each group we conducted a comparative study to present the differences in space requirements and actuator displacements.

3.3. DESIGN MODELLING

The amount of proposed active camber mechanisms makes it excessive to design another mechanism. It is more effective to design a mechanism using one of the concepts or a combination of the concepts, therefore this part is as a guide for the design of a mechanism.

3.3.1. GOAL

The design process of a mechanical design starts with the goal and refers to area of use. The mechanical design is an active camber system that can be controlled and fitted into a medium to large car with as minor as possible changes to the current configuration. This model has to made ready for a real time calculation for the vehicle model in the simulator. The requirements should be met or well considered to make full use of the functionality of the system.

3.3.2. REQUIREMENTS

Existing suspension systems have three degrees of freedom, for active suspension this will be expended to four degrees of freedom, which are the vertical wheel travel (translation in z direction), steer angle (rotation around z axis), camber angle (rotation around y axis) and wheel rotation (rotation around x axis). First, the kinematic requirements should be taken care of with the following points of consideration: mechanical forces, kinematics, tire dimensions, workspace of tire, connection to drive line, point of actuation, control and costs. Then the performance of the suspension can be tuned to the optimal desired response, with test items: wheel load variation, body isolation, handling load control, compliant wheel plane control, kinematic wheel plane control and component loading environment.

3.3.3. MODELLING

After setting up the requirements the foundation is there to start the modelling. First the basic layout with the essential parts has to get an approximate position to make it fit within the requirements. Then the modelling method is set-up and design criteria which has to be taken into account.

BASIC SUSPENSION LAYOUT

In the basic suspension layout all the essential parts, such as the wheel, knuckle, actuators and links have to get an approximate position. This will give insight if actuators or suspension links can be positioned on certain places or maybe have to be redesigned. This part is a visual reference to interpret abstract ideas for future work.

The main part where all other parts have to fit in is the workspace and this makes it a starting point. It is mostly determined on the wheel arc and the vehicle body, and therefore dependent on the vehicle. The wheel

Table 3.1: Benchmark table

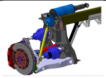
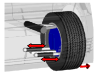
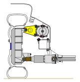
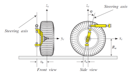
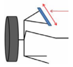
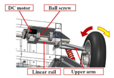
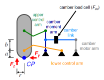
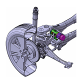
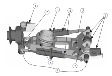
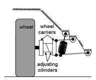
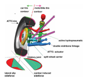
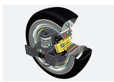
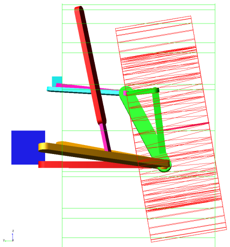
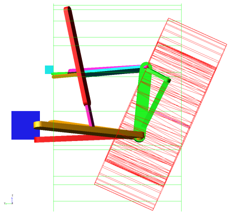
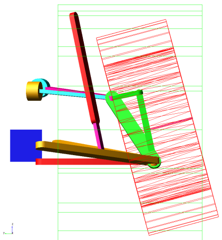
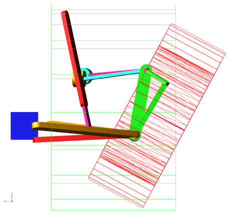
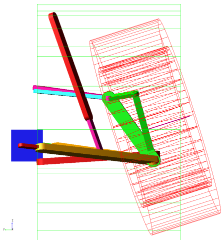
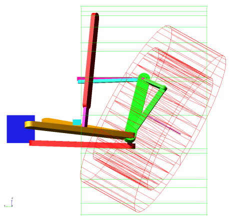
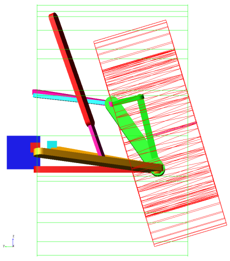
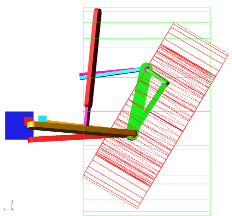
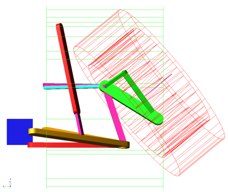
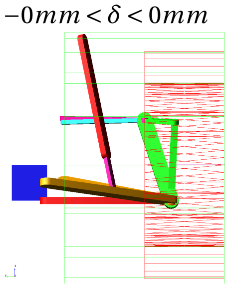
Actuation Mechanism	Figure	Max camber range	Point of actuation	Ready for active toe	Prototyped	Area of use	space requirements	additional unsprung mass weight
Ferrari [2]		$-6^\circ < \gamma < -1.5^\circ$	upper	yes	yes	sport	negligible	medium
Toyota [15]		$-5^\circ < \gamma < 5^\circ$	lower	yes	no	urban	negligible	high
Volvo [16]		$-5^\circ < \gamma < 5^\circ$	lower	yes	yes	urban	negligible	high
Caster angle [17]		$-12^\circ < \gamma < 12^\circ$	upper	no	no	road	medium	medium
UW longitudinal translation [18]		$-1^\circ < \gamma < 1^\circ$	upper	no	no	road	medium	negligible
UW lateral translation [1]		$-20^\circ < \gamma < 20^\circ$	upper	no	no	road	much	negligible
Stanford thesis [11]		$-45^\circ < \gamma < 45^\circ$	upper	no	yes	road	much	negligible
Crank bar upper wishbone [19] [20] [21] [22]		$-5.5^\circ < \gamma < 5.5^\circ$	upper	no	yes	(off-)road	negligible	negligible
Crank bar lower wishbone [13]		$-5.5^\circ < \gamma < 5.5^\circ$	lower	no	yes	urban	negligible	medium
Skew cylinders [23]		$-60^\circ < \gamma < 60^\circ$	middle	yes	yes	road	much	high
Mercedes [10]		$-30^\circ < \gamma < 0^\circ$	lower	no	yes	road/sport	much	high
Siemens [24]		$-4^\circ < \gamma < 4^\circ$	middle	yes	yes	urban	negligible	high

Table 3.2: Benchmark table from MSC.ADAMS/View

Type	Workspace / Actuator displacement without vertical wheel travel	Workspace / Actuator displacement with a minimum of -100mm - 100 mm vertical wheel travel	Camber dependency while vertical wheel travel from -100mm to 100mm	Actuation power
Upper wishbone length of rods	$-19.5° < \gamma < 30°$ $-65mm < \delta < 130mm$ 	$-10° < \gamma < 25°$ $-35mm < \delta < 100mm$ 	$< 4.5°$	low
Upper wishbone position of joint	$-20° < \gamma < 35°$ $-70mm < \delta < 150mm$ 	$-15° < \gamma < 30°$ $-55mm < \delta < 120mm$ 	$< 5°$	low
Lower wishbone length of rods	$-30° < \gamma < 40°$ $-120mm < \delta < 200mm$ 	$-18° < \gamma < 30°$ $-75mm < \delta < 150mm$ 	$< 3.5°$	high
Lower wishbone position of joint	$-27.5° < \gamma < 40°$ $-90mm < \delta < 170mm$ 	$-16.5° < \gamma < 30°$ $-60mm < \delta < 130mm$ 	$< 3.5°$	high
On the knuckle	$-40° < \gamma < 0°$ $-0mm < \delta < 0mm$ 	$-30° < \gamma < 0°$ $-0mm < \delta < 0mm$ 	$< 5°$	high

size in this workspace determines how much it can travel in vertical direction and in rotations. The actuators can be for example electric or hydraulic and both systems have their advantages and disadvantages. This is most dependent on the vehicle area of use, but in general it is desired to reduce unsprung mass for a better road handling.

One of the limitations of the tire workspace is the vertical suspension. This can be a passive or active system. When actuating the camber angle independent per tire, active suspension is desired to avoid large roll and unequal tire loading. Another wheel actuation is the rotation for the steering and camber angle. Steering can be done in the conventional way with a linear translation of the tie-rod or combined with the camber actuators. The actuators of the camber have also many options as discussed before and can be on the lower or upper wishbone or an extra knuckle is introduced. The last rotation is in the drive and brake actuation. This can be in-wheel or from the drive axle. When using an in-wheel hub motor or disk brakes the unsprung mass increases, while a drive axle comes from the sprung mass. Finally, the points at the sub frame have to be chosen. These are the points where the suspension will be connected to the vehicle. The joints of the wishbones, steer rod and drive actuation determine what the characteristics of the tire motion are. In the next paragraphs methods will be explained to choose these points.

MODELLING METHOD

When starting with modelling it is desired to simplify the model as much as possible, because there are many parameters participating in this system and the first focus is at a static, rigid, kinematic model of one wheel. Therefore, the dynamics, compliance and interaction of multiple wheels should be ignored.

The coordinate space is divided into several coordinate spaces and these are mapped into each other to let them work together. This includes a vehicle space, suspension space and tire space. By using a vehicle fixed coordinate system as a reference the influence of the vehicle orientation is removed. For the vehicle space it is important what the position to the contact point is. The tire space is a 4DOF system with vertical position, camber angle, steer angle and drive rotation. The suspension space describes the configuration and power requirements of the actuators.

3.4. CONTROL

The control system can be considered as the link between the driver and the camber mechanism. The driver is responsible for the path of the car, which means the driver should have a feeling of how the vehicle is behaving. A proper control system should assist the driver and should always behave regarding the driver's inputs. A general example is given in Figure 3.1. The driver has his own preferences and gets information from the environment, the steering wheel and the vehicle motions. Based on these inputs the driver gives an input to the steering system which will change the orientation of the wheels and this will change the vehicle motion. The camber control system needs to work in parallel with the steering system and receive information from the driver and the vehicle.

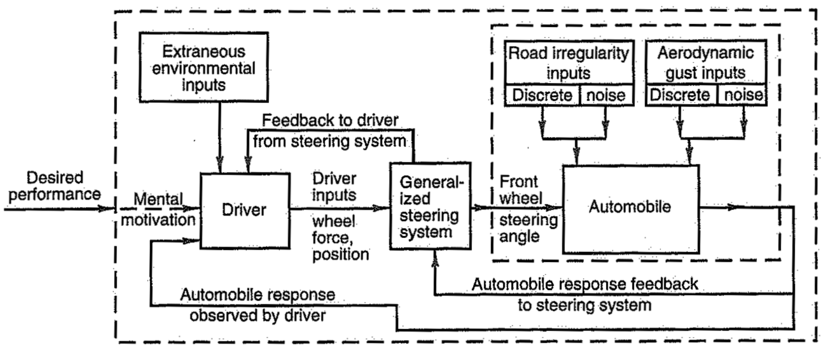


Figure 3.1: Driver-Vehicle relationship [4]. This scheme shows that the driver is part of the large loop. The vehicle with its controllers for example the active camber, which works within large loop.

3.4.1. AIM

The vehicle characteristics can be changed by actuating the camber angles of the tires as was discussed in Section 2.5. This can be done individually at each wheel, symmetrically (same per pair of wheels) or same for all four wheels. There are many options for the control and it depends on the vehicle use, just like the design of a mechanism is dependent on the type of use. For example, a sports vehicle has a different use than a city car as was discussed in Section 3.2. The part of the control which decides the camber angle can be independent on the mechanical design and the part of the desired camber angle needs to take into account the mechanism kinematics and dynamics.

The design of the controller depends on the vehicle type and type of use. Therefore, it can be tuned for:

- maximum lateral force,
  by leaning all wheels into the corner. It depends on the tire characteristics if it will be used to slightly tilt a rectangular cross section tire to maximize tire contact area or by maximize tire tilting of a rounded cross section tire to generate maximal camber thrust.

- desired corner path,
  by changing the camber angle dependent on the desired yaw rate. The aim is for an as small as possible difference in actual yaw rate and desired yaw rate.

- understeer gradient,
  by changing the camber angle on the front or rear. The understeer gradient decreases when creating more lateral force on the front wheels and it this can lead to an oversteered vehicle, while creating more lateral force to the rear wheels the understeer gradient can increase as was discussed in Section 2.5.

- preventing roll-over,
  by increasing the camber angle decreases the roll until a certain level, because the CoG lowers and the contact area of the tire shifts. The lateral force can create higher lateral forces and therefore this new equilibrium will also be overpowered. By decreasing the camber angle again or even change to positive camber angle decreases the lateral forces again, which prevent the roll-over. The roll-over originates from high lateral force and a high center of gravity. So by decreasing the lateral forces on the outer corner wheels a roll-over can be prevented. The inner corner wheels might partly compensate for the lag of lateral force of the outer corner wheels, in an attempt to keep control over the vehicle. Another method can be for example by decreasing only the front wheels' camber, which decreases the yaw rate and therefore decreases the lateral acceleration.

- emergency maneuver,
  by changing the camber angle only in a situation when it cannot avoid the hazard any more. In this case the vehicle behaves like a normal vehicle in normal use and the system intervenes only in situations where it is really needed.

A combination of controller designs would be the most appropriate way, because control for higher lateral forces increases the chance for change in understeer gradient and roll-over. In this case the system should recognize the situation, which for example can be done by the vehicle state.

3.4.2. CONTROL SYSTEMS

The track is an eight shape figure and therefore the controllers were designed to use camber throughout the track. Four different control schemes were designed, whereof two controllers have an offset and these behave like the reference configuration before they reach the offset. The controllers depend on:

- Direct lateral acceleration,
  the desired camber angle is the vehicle's lateral acceleration multiplied with a constant.

$$\gamma_{\text{active}} = a_{\text{y}} \cdot C_1 \tag{3.1}$$

- Dead zone lateral acceleration,
  this one is comparable with direct lateral acceleration controller, but with a offset so it behaves like the reference configuration before reaching the offset.

$$\gamma_{\text{active}} = \begin{cases} C_1 \cdot (|a_{\text{y}}| - C_{\text{offset}}) \cdot \text{sgn}(a_{\text{y}}) & |a_{\text{y}}| > C_{\text{offset}} \\ 0 & a_{\text{y}} \leq C_{\text{offset}} \end{cases} \tag{3.2}$$

- Hyperbolic lateral acceleration,
  this one is also comparable with direct lateral acceleration controller, but this one creates an offset and makes a smooth transition from the reference configuration behavior to the direct lateral acceleration controller behavior.

$$\gamma_{\text{active}} = a_y \cdot C_1 \cdot \left(\frac{e^{C_2 \cdot a_y} - 1}{e^{C_2 \cdot a_y} + 1} + 1 \right) \tag{3.3}$$

- Yaw error,
  the desired yaw rate will be calculated based on the speed and the steer angle, and the actual yaw rate will be subtracted to get the error. The desired camber angle is then the error multiplied with a constant.

$$\dot{\Psi}_{\text{desired}} = \frac{V}{L} \cdot \delta \tag{3.4}$$

$$\gamma_{\text{active}} = \left(\dot{\Psi}_{\text{desired}} - \dot{\Psi}_{\text{real}} \right) \cdot C_1 \tag{3.5}$$

with $C_1 = \frac{\text{max}_{\text{camberangle}}}{\text{max}_{\text{lateralacc}} - (\text{offset}_{\text{lateralacc}})}$

Newton's second law dictates that the force increases with the acceleration. Based on this linear relation three out of the four control schemes are designed to use the lateral acceleration as a control input. Section 2.4 discusses that the lateral tire forces increase due to an increase in tire slip angle, which has to be corrected by the steering angle. When increasing the camber angle the tire slip angle decreases and this should decrease the steering angle error. As an extension to this way of thinking an extra controller is designed based on the yaw error, where first the desired yaw rate is calculated using velocity and steer angle, and the difference with actual yaw rate is calculated. Thus the camber angle is based on the yaw error, which should improve the steer angle based on vehicle velocity and steering angle instead of lateral acceleration.

The camber angle is taken as the same angle for all four wheels and leans into the corner, which means a negative camber angle for the outer corner wheels and a positive camber angle for the inner corner wheels. This is to create maximum lateral force on all wheels and minimizes the change in other vehicle dynamics behavior like under- and over-steer.

3.4.3. ACTUATOR CONTROL

The actuator can be position or force controlled. The camber angle should be positioned in the desired angle and the force taken by the actuator can vary dependent on the trajectory and road surface. This makes the position control the most suitable for the designed control systems of the previous section. A feedback loop with PID controller can deliver the desired behavior as shown in Figure 3.2.

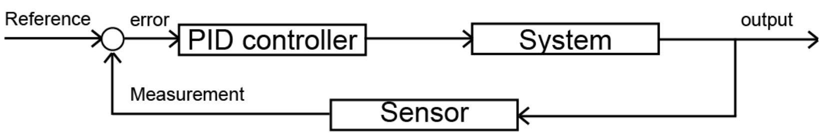


Figure 3.2: Actuator PID controller

3.5. CONCLUSION

The design and fulfillment of an active camber mechanism would be too specific for the scope of this thesis. It is not necessary for the implementation in the simulator and a specific mechanism makes it difficult for the real time implementation and robustness of the system. Therefore, an overview of the concept designs and a guide for designing a mechanism is presented. The control of the active camber is of essential use when changing from straight to corner or from one corner to the other and therefore there are four controllers proposed. Multiple controllers are essential to draw a conclusion independent on a specific controller. The controllers were designed based on lateral acceleration and yaw error, because the camber increases lateral force which is directly dependent on lateral acceleration and the yaw error controller is a continuation on the idea to decrease the steer angle correction, which was a result of decreasing the tire slip angle with the lateral acceleration controller.

4
SIMULATOR

"Imagination is more important than knowledge. Knowledge is limited. Imagination encircles the world."

- Albert Einstein

4.1. INTRODUCTION

The simulator as shown in Figure 4.2 was used in a fixed-base configuration and it is part of the Intelligent Automotive Systems (IAS) laboratories of the Delft University of Technology. The system is equipped with a steering actuator, dSPACE [25] real-time hardware and the dSPACE Automotive Simulation Models (ASM) library based on Matlab/Simulink [26]. Drivers could experience driving in a simulator with the characteristics of a real vehicle. A detailed description of the simulator parts is given in this chapter.

4.2. SIMULATOR SET-UP

The system provides a modular structure where parts of hardware or software can be replaced as needed. Furthermore, the available instrumented steering system on the simulator base allows testing with driver-in-the-loop scenarios, which was extensively utilized throughout this study.

The setup is equipped with the dSPACE Release 6.3 from November 2008, which is configured for Matlab R2008b. For a real-time interface two dSPACE boards DS1005 are used to meet the processing requirements and therefore the vehicle model is a dual core setup [27].

A decent understanding of the simulator is needed to design a proper experiment. Following sections summarize the work-flow, steer feedback model, vehicle model and visualization.

4.2.1. WORK-FLOW

The dSPACE work-flow schematic is presented in Figure 4.1. Starting with the feedback motor (Ultract motor), this motor is equipped with an encoder and torque sensor. These sensors measure the input of the driver and sending the steer angle and steer torque to the motor control, which passes it on to the dSPACE

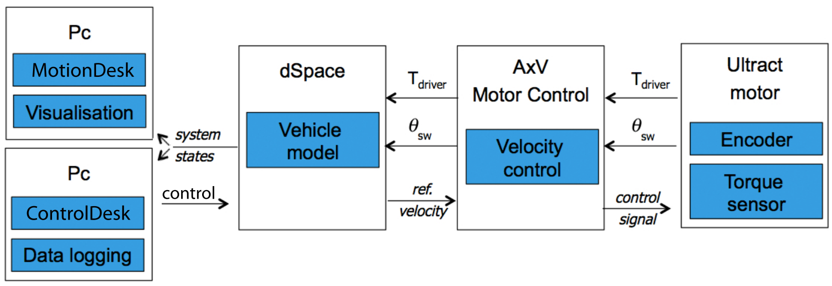


Figure 4.1: Simulator scheme, representing the connections between all the used components.

Figure 4.2: A general view of the driving simulator set-up.

hardware. These inputs run through the ASM vehicle model and calculate all vehicle parameters with a frequency of 100Hz. After the calculation it sends back control information for the motor to the motor controller, which converts this information to drive the motor. Finally, two computers are receiving the system states of the dSPACE vehicle model. One computer runs the control software (dSPACE Controldesk), which can change switches (for changing camber control systems), change variables and log all the data. Another computer is responsible of the visualization software (dSPACE Motiondesk). This is a stand alone visualization computer and the software receiving the system states, which it used to display the vehicle in the right position in the environment.

4.2.2. STEER FEEDBACK MODEL

Steering feel is an important feedback for the driver during driving, it plays a significant role for the feeling of safety and comfort. Apart from that it also contributes to the brand character, which separates the vehicle from other vehicles. Therefore steering behavior in a driving simulator is one of the most important elements to achieve a realistic driving environment. To reach realistic behavior, components such as vehicle modeling, proper visualization, motion cueing, etc. play a significant role [28]. Studies investigating the influence of motion cues on drivers have shown that having realistic steering feedback can partially compensate for the lack of motion cues [29].

The steering behavior in a simulator is determined by the used model and the hardware of the steering actuator. The steering model used in this study consists of a upper column with the steering wheel, a torque sensor and a rack-pinion unit [30].

4.2.3. VEHICLE MODEL

The ASM model is a 24 DOF multibody dynamics model [27], therefore it is divided in sub models and these models are again divided in further sub models. The top layer of this model is shown in Figure 4.3. The sensor signals of the feedback motor enter the system on the right and flows through the model, finally leaving at the left, sending signals back to the motor. The soft ECU, engine and drive-train remain unchanged, while the vehicle dynamics is extended to the active controlled camber and the environment to the desired test track.

By opening the vehicle dynamics module it shows a scheme as in Figure 4.4. In this scheme the tire forces are central in the connections and are affected by many parameters including: the tire rotation by the drive train and brake system, the tire radius by the road, the tire velocity by the vehicle body and the tire

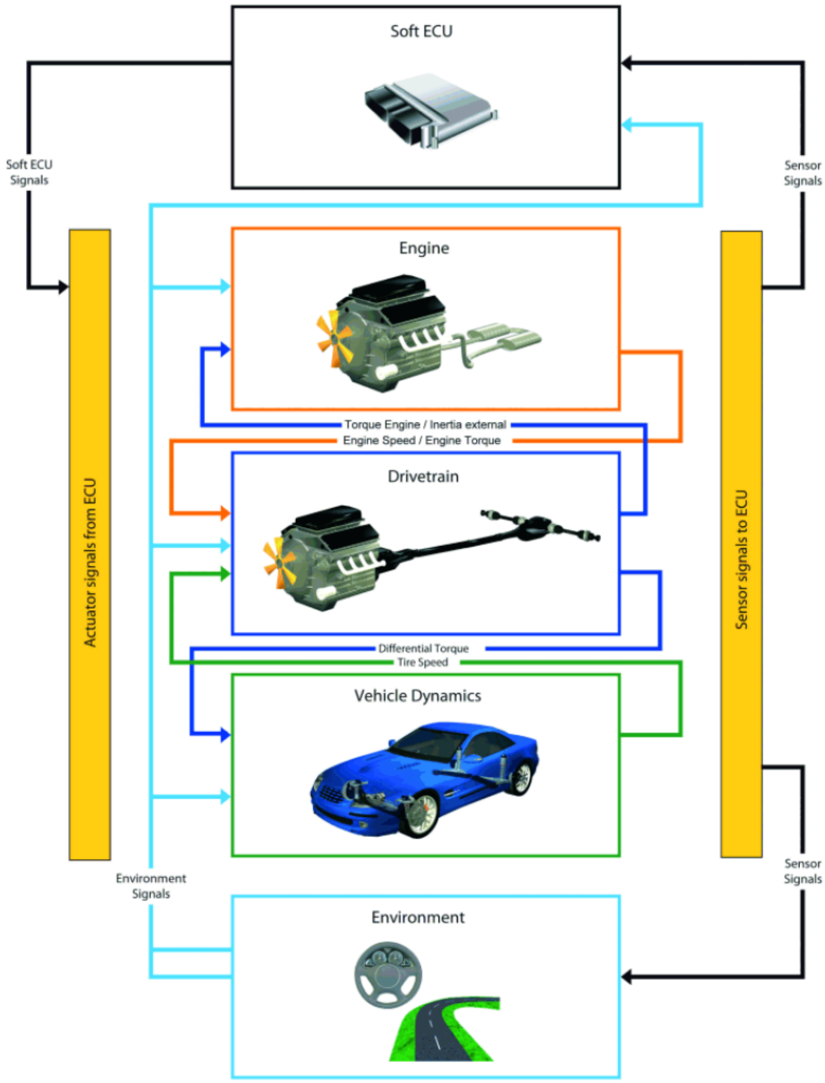


Figure 4.3: ASM vehicle model overview [25]. It shows the flow of the main parts. Each block contains the detailed equations of the simulation model.

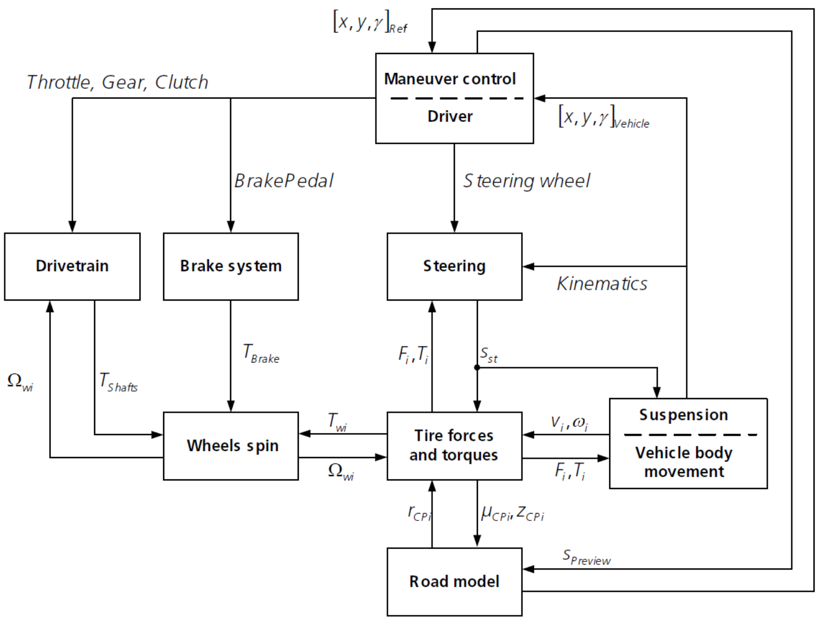


Figure 4.4: ASM vehicle dynamics model [25]. The tire forces and torques are located centrally and receive information from the powertrain, steering, suspension and road.

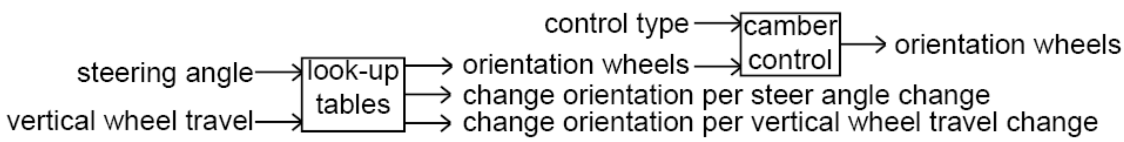


Figure 4.5: ASM suspension model including the camber controller, without change in suspension look-up tables.

orientation by the suspension. The tire model responds accordingly with a force or torque.

Suspension

The vehicle model uses the original ASM model except from the suspension model, which is modified to an active controllable camber system. Since the design of an active mechanism was out of the scope of this thesis, only the orientation of the tires was changed during control. All other vehicle variables were kept constant during the experiments. The Simulink model of the suspension is modified as shown in Figure 4.5. Where the look-up tables represent the conventional double wishbone suspension, which are depended on the steer angle input and vertical wheel travel relative to the body. The output of these look-up tables provide the new orientation of the wheels and the derivatives to steer angle input and vertical wheel travel. After which the camber controller changes the orientation to the cambered orientation. There is no direct need to also change the derivatives, because the change of orientation per steer angle and vertical wheel travel have different rotation axis. The camber angle creates a geometrical conversion factor, which is of limited addition to the current output. The change in orientation per camber angle would be a good addition, but requires more information about the suspension and the look-up tables would be extended to camber variable as shown in Figure 4.6. An analytical model would require too much calculation power, therefore these look-up tables are used and these make it possible to obtain the real-time vehicle response. The controllers were discussed in Section 3.4 and the implementation of these controllers is shown in Appendix A. The control equations are represented in the Simulink blocks. For the camber angle coefficient is a maximum camber angle of 30° used and a maximum lateral acceleration of $8m/s^2$. An offset of $2m/s^2$ is chosen, so small steering corrections do not directly active the camber angle.This new orientation of the tires is sent to the tire model, which calculates the tire forces and torques.

Tires

The tire model stands central in the vehicle dynamics model. It receives information from the vehicle body, suspension, drive train, steering and road and it sends back related forces and torques. ASM provides two

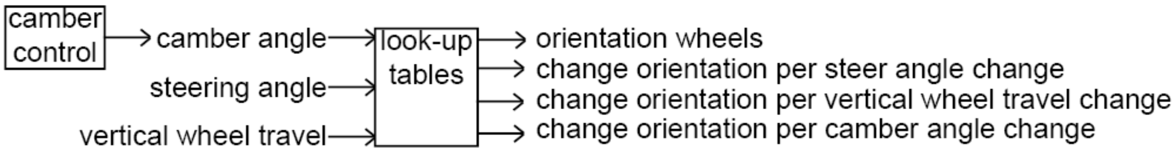


Figure 4.6: ASM suspension model including the camber controller. In this setup the look-up table should be extended to camber dependency.

types of tire models. One is the TMeasy the other is the MF, Magic Formula version MF-Tire 6.0. The magic tire formula is more accurate and using a single point model, so the tire shape does not influence the forces which is beneficial in the case of tilting car tires. The standard ASM car tire is a conventional car tire with a rectangle cross-section, but due to the limitations of the magic tire formula, this tire can be used. The magic tire formula is valid for up to 15degrees camber angles beyond which it becomes less accurate. The magic tire formula can be represented as follows [25]:

$$Y(x) = D\sin[C\tan^{-1}(BX - E(BX - \tan^{-1}(BX)))] + S_v, \tag{4.1}$$

$$X = x + S_h, \tag{4.2}$$

where B, C, D and E are factors for stiffness, shape, peak and curvature. The camber angle is in the horizontal shift factor S_h, which was already shown in Figure 2.4. This implies that a larger camber angle would increase the lateral forces, without roll-off characteristics. The magic formula could be extended to a version which is more desirable for larger camber angles, but it requires additional tire parametrization. The tire model should provide representable forces to create realistic feedback to the driver and guarantee sufficient realism of vehicle dynamics.

The ASM software provides four tire sets with different characteristics. The first tire set is used for the experiments. The track is a flat surface with friction coefficient $\mu = 1$, which means a conventional tire on dry asphalt with a very high friction coefficient. The friction coefficient of asphalt is around $\mu = 0.7$, which means the achieved tire forces will be high. This is not considered as a problem, because all configurations use the same conditions and therefore these can be compared with each other.

4.2.4. VISUALIZATION

The visualization and vehicle model are running on different computers, therefore the environment should be defined identical on both computers to get the proper feedback of the simulator. Figure 4.1 shows an unidirectional data communication from dSPACE to visualization computer and it only consists the system states. The visualization uses these system states to display the vehicle in the proper orientation in the environment, which means the environment in the vehicle model and the visualization has to be identical. To avoid a time consuming environment building process the vehicle model environment was made as a plain asphalt area, while in the visualization the road was shown with roadsides, grass and other environmental effects (e.g. houses, etc). The result of this difference in environment was that there was no difference of driving on the road or on grass and there was no collision possible with the environmental effects. The drivers will not be penalized in this manner and therefore the feel of safety is less perceptible.

It was noted that with only an asphalt road visible, the driving experience becomes inaccurate as the feel of speed disappears, therefore additional environmental effects (e.g. houses, etc) were introduced. This made it more clear when approaching a corner and with what velocity the corner is taken.

4.2.5. DISCUSSION

The present simulator was the one from Figure 4.2 and used in fixed-base configuration. The focus of this research is on steering feel, so it should not have that much difference in result compared to the moving base. The driving experience with active camber would be more realistic with moving base. Furthermore, the visualization is pretty good, but there is no difference in driving on or off road, because in the vehicle model it is just one flat area of asphalt. Still the drivers see the that they drive off road or through a house, but they do not notice any differences regarding vehicle dynamics.

The tire model is also a limitation, because it does not see any difference in tire shape. Therefore, it is possible to create large camber angles and receive values as if there are special designed tires under the vehicle.

Figure 4.7: dSPACE ASM Motiondesk image [25]. This is an impression of the road and environment. During the experiments the driver has a view from behind the steering wheel.

The camber control implementation changes only the orientation of the wheels. This change in orientation is the main contribution to the change in tire forces. Ideally the derivatives to steer input, vertical wheel travel and camber actuation would be implemented, but for these variables is more information about the suspension needed. A full suspension can be build and the look-up tables can be changed to a 3D look-up table with steer input, vertical wheel travel and camber actuation as inputs. Especially the change in camber angle would have influence in tire behavior, but this is also dependent on the way of actuation (upper wishbone actuation has less change in lateral translation in the tire/road friction). Furthermore, the actuation happens instantly, which is nearly impossible. The actuator and inertia of the wheel would be desired to be implemented to get a more realistic change in camber angle. Since this is not the main focus of the research an universal form is chosen by only change the orientation and a feed forward direct control.

5

Testing and Results

"You should never, never doubt something that nobody is sure of."

- Roald Dahl, Charlie and the Chocolate Factory

5.1. Introduction

In the previous chapters is discussed what the expected response of an additional camber angle would be and how this can be implemented in a simulator to perform tests. The influence of camber angle was first tested in the simulator before the control schemes were introduced and implemented. These first tests were done to check if the vehicle response had similar results to the MSC.ADAMS/Car tests. A different vehicle set-up is used in MSC.ADAMS/Car and dSPACE ASM which will result in different vehicle data, but it will give a sufficient insight in what happens to the vehicles responses.

The following tests were done to check how the vehicle model responses on the camber angle. In these tests the participants had to drive with a fixed camber angle on a straight line and enter a circle at a constant speed. This test was conducted to prove that camber angle was indeed increasing the lateral forces and if this was changing the steering behavior. Next to the objective data, it was already presenting results on steering feeling. The tests were then extended to a dynamic camber system and the circle was extended to a double circle with straight parts to connect them, as shown in Figure 5.1. The straight parts were introduced to have a smooth entrance into the next corner without the transient behavior of the previous corner. Different radii are used for the first and second corner. The radius of the second corner is smaller to create higher lateral forces and this is expected to show larger differences in controllers. Furthermore, the difference in corner radius avoid a monotonic steering demand and thereby the driver to lose attention.

Both objective and subjective data is recorded. The selected vehicle data can be extracted from the ASM software, which provided information on the objective differences between the controllers. After each session the participants had to fill in the questionnaire which consists of six questions before they could continue with the next session. The questionnaire consisted of questions about how they experienced the vehicle control, vehicle response, steer effort, safety, speed and fun. The objective and subjective data was then be compared with each other to check for coherence.

5.2. Experiments set-up

5.2.1. Human interaction

In a vehicle there are many impulses to the driver such as; visuals, sounds, motions, steer feel, pedals feel, which will influence the choices made by the driver. This simulator is only equipped with visual and steer feedback. From Section 4.2.2 it was already discussed that steer feedback can give sufficient driving experience.

5.2.2. Method for data evaluation

The vehicle responds regarding its properties and input of the driver. At the same time the driver experiences this behavior according to the steering wheel feedback. It is expected to find a coherence in the vehicle

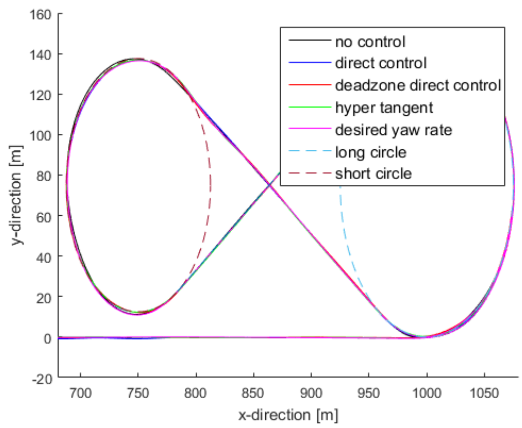


Figure 5.1: The trajectory of the experiments. A straight approach to get the vehicle up to speed and then the driver is steering it through an eight figure shape with two different radii corners.

response and the driver's experience. The questions are dependent on multiple vehicle responses and one vehicle response can be more dominant than the other one.

The analysis started by reading the overall control response, which will be mainly a result to what extend the driver has to correct the vehicle. This can be measured by the amount of steering corrections, amount of overshoot, settling time and lateral position error. A large overshoot and long settling time will point to a hard to control vehicle which requires a lot of corrections and results in a large lateral position error.

A similar case is valid for the steer response, where the time lag between steer input and vehicle response will be the most straightforward measurement, because this will show how fast it responds to a steer input. Furthermore, the steer corrections, amount of overshoot and settling time show to what extend the driver is receiving the vehicle response. The steering effort can be directly linked to the torque applied to the steering wheel as well as to the amount of work.

The camber angle influences the vehicles velocity and therefore the speed can be used as a measure, but the differences will be small. The lateral position error shows the driver it drives too fast to make the corner. The vehicle roll implies the lateral acceleration and this is function of the speed. These are also measures for the feel of safety. The feel of safety can be indicated by the overall control, steer response and experience of speed, because a hard to control vehicle will be most likely experience a too high speed and will feel unsafe, while an easy controllable vehicle can be easier to drive at high speed and therefore feel safe.

Finally, the drive fun is a personal preference and this is not directly coupled to a vehicle measure. The driver can prefer comfort and a tranquil drive, while the other can prefer high accelerations and maximum cornering. However, sports vehicles with fast steer response and good to control is in general more fun to drive especially for this group young test drivers. This is mostly coupled to steer response and overall control. A summary of the coupling is presented in Table 5.1.

Table 5.1: Coupling objective and subjective data

Subjective	Objective		
Overall control	std(Steer angle)	Settling time	Lateral position error
Steer response	Time lag	Overshoot	Settling time
Steer effort	Steer torque	std(Steer angle)	
Speed	Vehicle speed	Lateral position error	Vehicle roll
Safety	Overall control	Steer response	Vehicle speed and vehicle roll
Drive fun	Steer response	Overall control	

5.2.3. PARTICIPANTS

The participants should not differ too much in age and experience to avoid the creation of too many dependencies. Specifically with a relative small group (11 people) it is important to keep the target group in

Table 5.2: Initial experience of the participants

	Participants										
	1	2	3	4	5	6	7	8	9	10	11
game experience *	2	2	2	3	4	4	3	4	1	2	1
drive experience (x1000 km)	50	30	100	20	1	1	70	10	3	5	30
age (years)	24	24	27	26	19	20	23	25	23	25	27
license possession (years)	6	6	9	7	2	2	5	7	3	7	9
gender	M	M	M	M	M	M	M	M	F	M	M

*with game experience: 1 = never, 2 = once a year, 3 = once a month, 4 = once a week and 5 = each day.

a limited range. Therefore a specific target group is chosen. Participants should be between 18 and 28 years old (less than 10 years possession of drivers license) and have a driving experience of less than 100,000 km.

For the proof of concept three drivers participated and for the active controlled camber experiments eleven drivers participated. In Table 5.2 an overview of the initial experiences of the eleven drivers is shown.

5.2.4. LIMITATIONS

Next to limitations of the dSPACE ASM software there are limitations in the experiment setup and the introduction of the human factor. Although the number of variables is brought back to a minimum the experiment still has many variables. In order to get as close as possible repeatable tests the only property which will change between experiments is the camber angle and all other vehicle data is unchanged.

The human factor makes repeatability a significant issue. For example, the interaction of the human with the simulator using a steering wheel and visualization, which can include time lag and screen hitches due to computer processing. Some participants can adapt easier and are more familiar with the simulators. A well-known ability of humans is the learning curve; therefore, the participants start with a training session.

5.3. PROOF OF CONCEPT: CONSTANT CAMBER ANGLE

In the proof of concept participants drive a circular track with a constant camber angle. These tests were conducted to see whether the lateral forces changes and if the varying forces influence the steer torque in the vehicle model.

This proof of concept will give insight in the simulator if this is suitable for testing with active controlled camber systems and if the forces and responses are comparable with previous tests of MSC.ADAMS. Furthermore, the experience of the participants give insight in the use of the simulator and can be used for the next set of experiments.

5.3.1. PROCEDURE

For this experiment an environment is built with a circular track and a straight part. In the straight part the vehicle can get up to speed, so there is no longitudinal acceleration during cornering and the tires are only exposed to lateral slip. After the straight part the participants enter the circle with a radius of 75 m and with a speed of 95 km/h.

The first part of the experiment is about entering the curve: what is the steering feel to get the vehicle in the right curve of the track? When entering the curve, the vehicle data will show oscillations before it reaches its steady state cornering. How do the participants experience this transient response? While the second part is driving on the curve: what is the steering feel to keep the vehicle in the desired corner line and does the vehicle reach the steady state after the transient response?

The camber angles are applied on all the wheels. In this case the curve is going to the left and all the wheels are leaning into the corner (like a motorcycle would lean into the corner). This means negative camber angle on the outer corner wheels and positive camber angle on the inner corner wheels, because this would increase the lateral force on all four wheels as was concluded in Section 2.5. For this proof of concept the angles $0°$, $-10°$ and $-30°$ are used to indicate this. In order to get repeatable results all features were kept the same except the camber angles.

The experiment starts with a training session. The participants have the opportunity to drive the track four times to get familiar with the simulator before starting with the experiments. The experiment consist of three sessions, each with a specific camber angle ($0°$, $-10°$ and $-30°$). In each session the participants

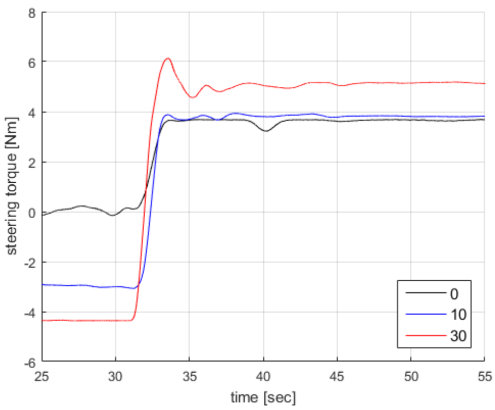


Figure 5.2: Steering wheel torque of one of the participants

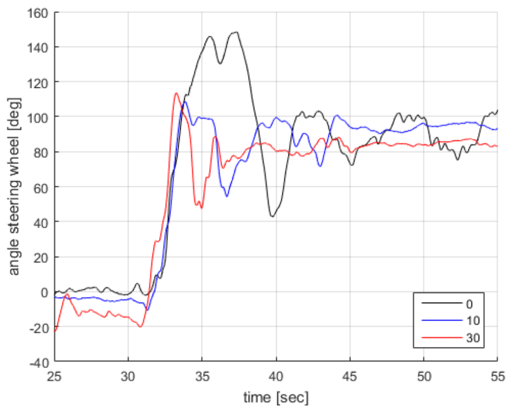


Figure 5.3: Steering wheel angle of one of the participants

have two runs. The first run is to get familiar to the new vehicle characteristics and in the second run they can adopt to these characteristics and understand them. After each session they received the questionnaire on a smart-phone, which included questions regarding steering control, response, effort, vehicle speed, safety and drive fun.

5.3.2. RESULTS

VEHICLE DATA

A selection of the vehicle data is presented in Figures 5.2-5.11. This is the data of one participant while changing from the straight part to the corner part where he/she reaches the steady state corner. A combination of the data of all the participants is shown in the boxplots Figures 5.18- 5.23. For the boxplots the dataset starts at the point where the straight part intersects with the circle at 700 m in x-direction and ends after nine seconds. This part is shown in the graphs including the overshoots up to the steady state cornering. therefore the boxplots present large deviations, which represents the overshoots. When taking an overall view on the graphs and boxplots the reference orientation with $0°$ camber shows in general large overshoots and a large transient response, while the $-10°$ and $-30°$ camber angles show smaller overshoots, a shorter transient response and less lateral error with respect to the road.

The first part of the track is the straight part and due to the camber angle lateral forces were induced. These lateral forces had to be compensated by the driver with a steer input. In the first five seconds of Figure 5.2 and 5.3 the straight part is shown and presents the steering angle and torque compensation. This is also shown in the normalized lateral tire forces in Figures 5.4 and 5.5 where the front right tire shows a negative lateral force and a positive slip angle. Thus the right front wheel compensate for the lateral force, the left wheel is around zero with negative slip angle, which is a result of the steering system leading to a toe out configuration while steering. When steering into the corner the camber thrust pulls the vehicle into the corner. The camber thrust is not sufficient to make the corner and the driver has to steer into the corner to create additional lateral forces with the slip angle. There is less slip angle needed and therefore is the steer angle is smaller than the reference configuration. A higher steering torque is shown due to the higher lateral forces.

The position of the vehicle in Figure 5.6 gives the trajectory of the vehicle. All participants were unable to keep the vehicle on track with $0°$ camber angle. A better representation is given with the absolute lateral position error in Figure 5.7, where the reference is up to six meters off the reference line. The reason for this problem can be found by the tire forces. In Figure 5.5 the normalized lateral tire forces are shown for the front right tire (front outer corner wheel). The reference configuration has large slip angles when it passes its maximum lateral force and therefore it is not able to make the corner, while the $-30°$ camber angle reaches only near the end of the linear region and can still increase the force if required.

The increased lateral force due to the camber angle results in higher lateral accelerations and yaw rate as shown in Figure 5.8 and 5.9. Due to the centripetal force this leads to a larger roll of the vehicle which is shown in Figure 5.10. The roll imposes more vertical force on the outer wheels. This new load distribution is not efficient for the lateral forces as was discussed in Section 2.3, but due to the camber angle it can still reach higher lateral forces. The front right tire in the reference configuration reaches its maximum lateral

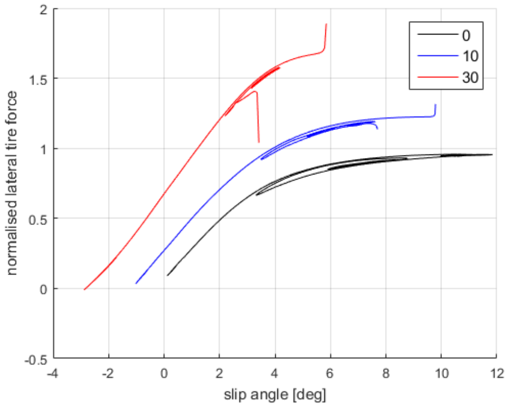


Figure 5.4: normalized lateral tire forces of the front left tire of one of the participants

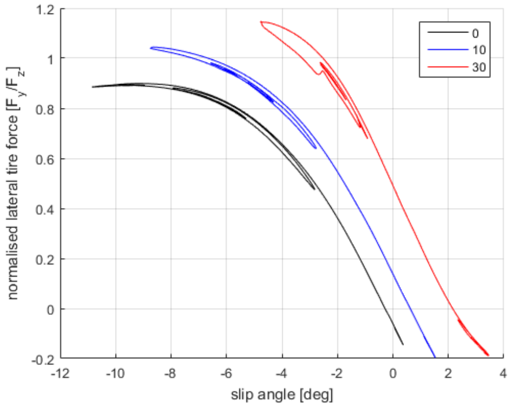


Figure 5.5: normalized lateral tire forces of the front right tire of one of the participants

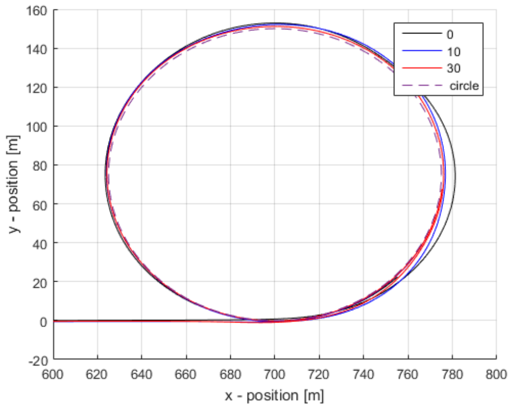


Figure 5.6: Trajectory of one of the participants

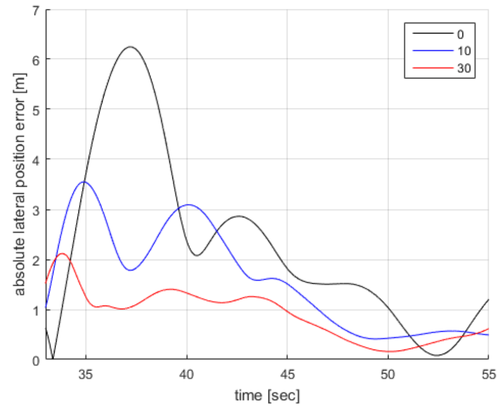


Figure 5.7: Absolute lateral position error of one of the participants

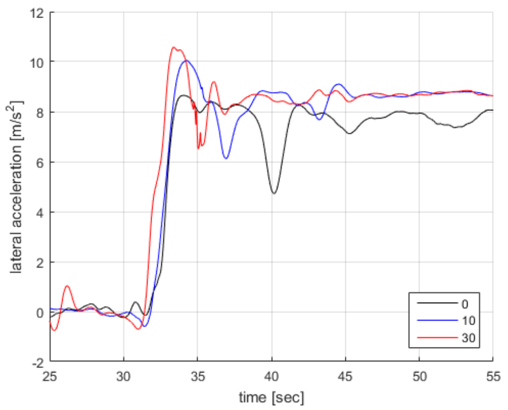


Figure 5.8: Lateral acceleration of one of the participants

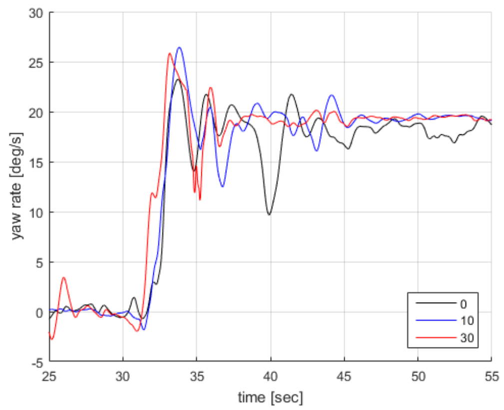


Figure 5.9: Yaw rate of one of the participants

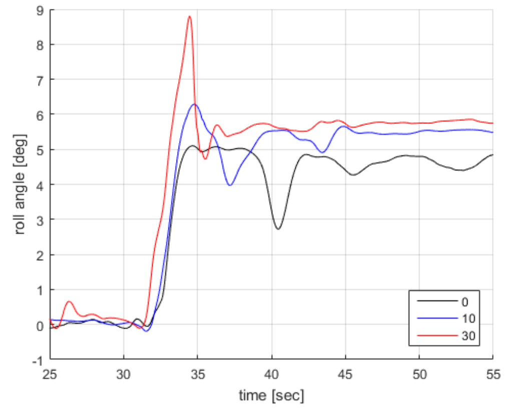


Figure 5.10: Roll angle of one of the participants

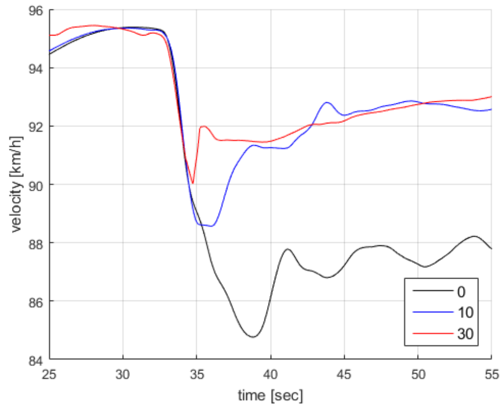


Figure 5.11: Velocity of one of the participants

force and this leads to understeer. An increase in steer angle is the result of the driver, which results in a change of orientation of the lateral forces. This is not only decreasing the lateral force component, but this is also decreasing the velocity with the longitudinal force component as shown in Figure 5.11. Due to the lower velocity the lateral acceleration decreases and the tire force falls under the maximum force, where after the lateral position error decreases. For the $-30°$ case the tires do not reach their maximum lateral force, thus the roll is dependent on the lateral acceleration. In the case of a large overshoot in the lateral acceleration this can even lead to a roll-over.

The settling time of the lateral accelerations and steering wheels angles is shown in Figures 5.12 - 5.17. These figures are a selection just after the overshoots of Figures 5.3 and 5.8. The Reference configuration has difficulties with damping out the lateral accelerations in the steady state corner at this speed and the lateral acceleration is lower than the cambered configurations. For the $-10°$ camber angle it rapidly finds the steady state, but still it irregularly bounces off, while for the $-30°$ camber angle it finds its steady state and the driver is able to hold it accurate. In the steering wheel angle similar characteristics are shown. The average steering angle of the reference configuration is misleading, because the driver has trouble to drive the desired track and the overshoots show an unstable behavior and it is not reaching a steady state cornering. The $-10°$ camber does not yet match the settling time requirements of 2%, but it shows a stable behavior. The $-30°$ camber reaches its steady state and the drivers is able to drive a steady state corner.

The boxplots in Figures 5.18- 5.23 present the average values of the participants. These figures give a better representation of the achieved values and not only of one participant. The steering wheel angle decreases and show less deviation as shown in Figure 5.18, because the camber imposes lateral forces which help the vehicle make the corner. A lower deviation of the steering wheel angle indicates more vehicle control, because less steering corrections are needed to keep the vehicle on the desired track. Less steering corrections and a lower steering angle result in less force components in longitudinal direction as was discussed in Section

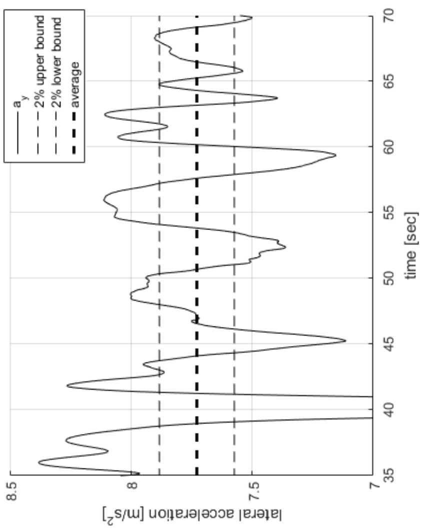


Figure 5.12: lateral acceleration with reference configuration showing the settling time with its 2% bounds around the average

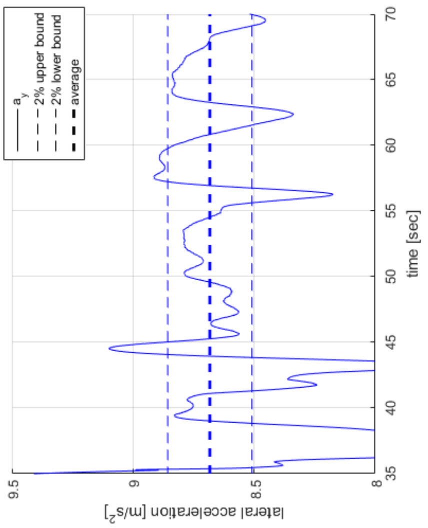


Figure 5.13: lateral acceleration with −10° camber angle showing the settling time with its 2% bounds around the average

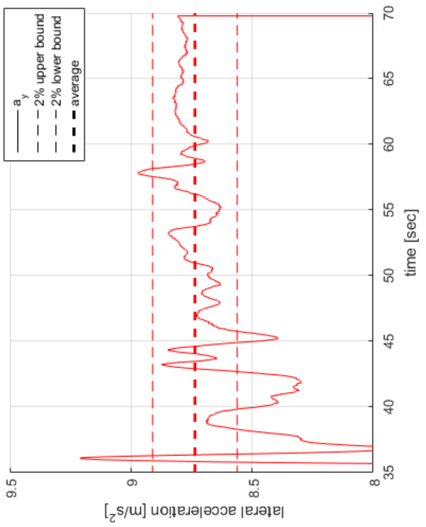


Figure 5.14: lateral acceleration with −30° camber angle showing the settling time with its 2% bounds around the average

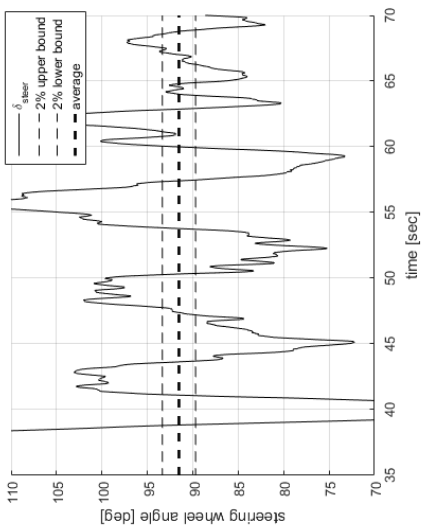


Figure 5.15: steering angle with reference configuration showing the settling time with its 2% bounds around the average

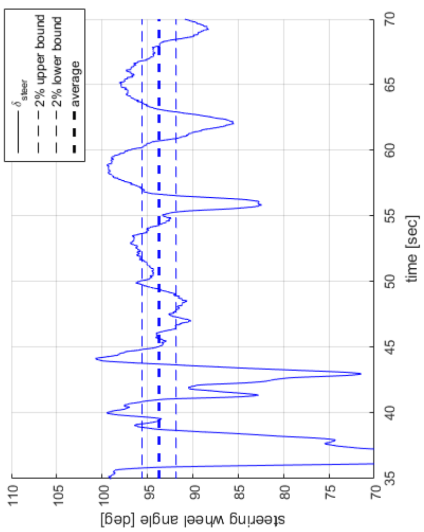


Figure 5.16: steering angle with −10° camber angle showing the settling time with its 2% bounds around the average

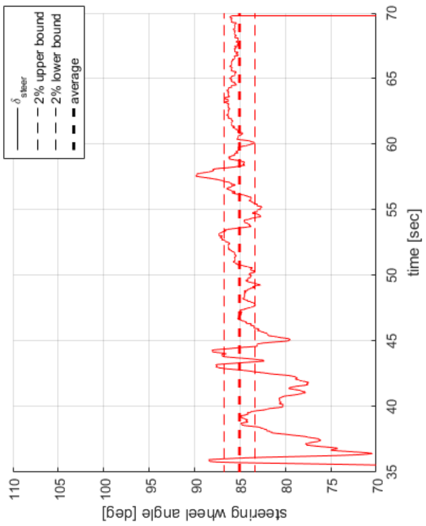


Figure 5.17: steering angle with −30° camber angle showing the settling time with its 2% bounds around the average

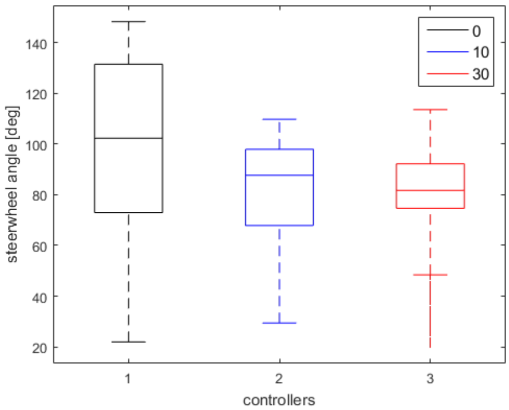


Figure 5.18: Steering wheel angle of all participants combined.

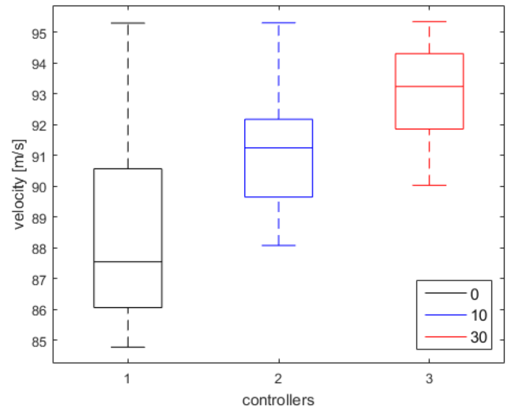


Figure 5.19: Velocity of all participants combined.

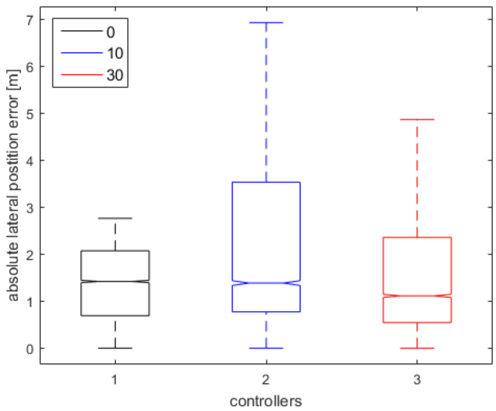


Figure 5.20: absolute lateral position error of all participants combined.

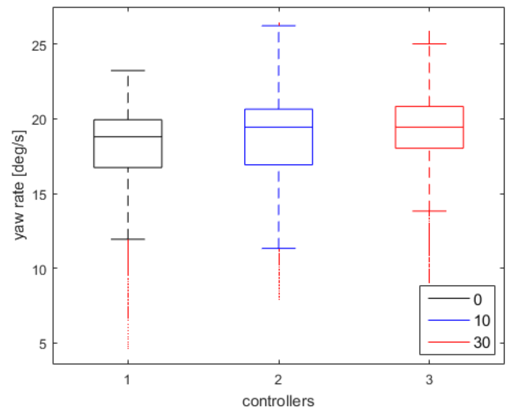


Figure 5.21: Yaw rate of all participants combined.

2.4, therefore the velocity remains more constant as shown in Figure 5.19. The absolute lateral position error in Figure 5.20 shows a misleading result, because the cambered angles have a more steady corner and the reference configuration without additional camber is more waving around. Finally, the median shows a decrease in lateral position error. Furthermore, a higher yaw rate can be established as shown in Figure 5.21, because the speed is higher. Due to the higher velocity and yaw rate, the lateral accelerations show increased values for the cambered systems as shown in Figure 5.22. At last, increasing roll angles are shown for the cambered vehicles in Figure 5.23, which is a result of higher lateral accelerations.

The data can be validated with the MSC.ADAMS/Car results. A different vehicle set-up and environment set-up is used, but the circle track radius and camber angles are the same. Table 5.3 summarizes the values of ASM and MSC.ADAMS/Car. The average value of the three participants is taken in the range of 45-55 seconds, because in this time domain they drive the steady state corner like the MSC.ADAMS/Car vehicle is driving. Two completely different vehicles driving in the same corner with the same speed should have the same lateral acceleration and yaw rate, but the forces will be different dependent on the mass. The roll and steer angle are dependent on the vehicle properties like mass, CoG height, suspension mechanism stiffness and steering ratio of the rack. The roll response is contradicting, which is a result of the method of implementation. In MSC.ADAMS the vehicle height and lateral position of the tire print with respect to the vehicle body is changing when changing the camber angle, but in ASM this is not directly connected and therefore only the angles of the wheels are changed. In ASM this leads to an increased roll in the case with a larger lateral acceleration. The steer torque is also contradicting, but this is a result of the used tire formula and root down to a numerical problem using large camber angles.

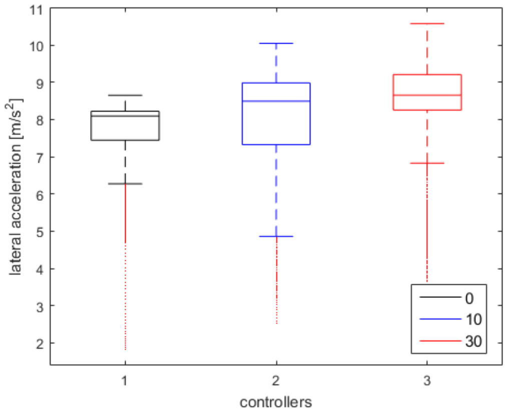


Figure 5.22: Lateral acceleration of all participants combined.

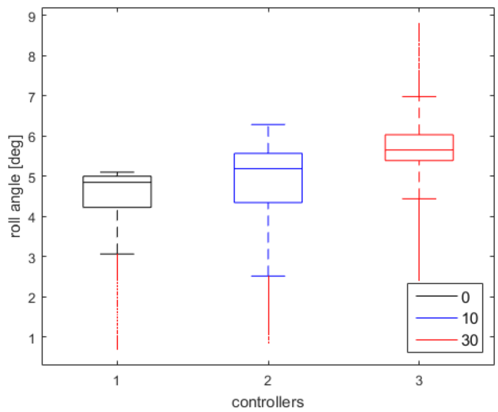


Figure 5.23: Roll angle of all participants combined.

Table 5.3: A comparison of the ASM and MSC.ADAMS/Car data.

	ASM			MSC.ADAMS/Car		
	0 degree	10 degree	30 degree	0 degree	10 degree	30 degree
lateral acceleration [m/s^2]	7.6	8.6	8.6	9.4	9.4	9.4
yaw rate [deg/s]	18.0	18.6	19.1	20.3	20.3	20.3
steer torque [Nm]	3.6	3.8	5.1	4.8	2.9	-4.5
roll angle [deg]	4.6	5.4	5.8	3.6	3.0	1.6
steer angle [deg]	96.9	92.5	82.8	97.5	80.0	67.0
velocity [m/s]	86.1	92.4	92.4	95.5	95.5	95.5

HUMAN DATA

The results of the questionnaire of the participants is divided in entering the curve and driving on the curve. Figure 5.24 shows the results for entering the curve with the three different camber angles and Figure 5.25 shows the results for driving on the curve. The values are the mean values of the results for the three participants and the bars indicates the upper and lower given values.

The marks given by the participants have in general overlap with each other, because in many cases the drivers choose the same mark for the different camber angles. The average values indicate an increase in control, response, safety and driving fun for driving with camber angles. As the number of participants were kept limited for repeatability, a statistical significance study was omitted and the given results of the marks per person are shown in Appendix D.

The responses for entering the curve and driving on the curve are very similar to each other, which is due to the number of questions. Eleven questions regarding vehicle characteristics was too much to get a reliable answer. Furthermore, the difference between entering the circle and driving on the circle is also close to each other, which makes it difficult to give a clear opinion of the experience.

There is only one outlier in the feel of safety for driving on the curve with $-30°$ camber angle, because in the second run a roll-over occurred with one of the participants, which dropped the grade to the lowest for safety. This roll-over was due to the excessive steering motion by the participant, because he/she first steered out of the corner and then abruptly tried to steer in. Due to the high lateral forces this leaded to a roll-over. This answer can be treated as an outlier and this makes the $-30°$ camber angle scores higher than the reference configuration. This experimental result was specifically considered for the design of control schemes with the aim on safety.

COMPARING THE OBJECTIVE AND SUBJECTIVE DATA

As presented in the vehicle data, a larger camber angle resulted mostly in a smaller steer overshoot, a shorter settling time and a smaller lateral position error. By using the evaluation table (Table 5.1) the vehicle data matches with the results of the overall control for driving on the curve, because Figure 5.15 - 5.17 shows a decrease in steering wheel angle corrections with a shorter settling time and this results in a smaller lateral

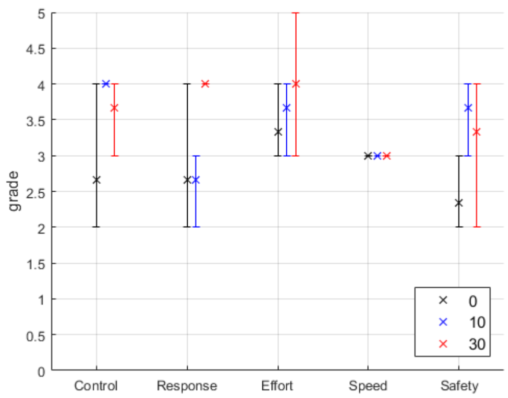


Figure 5.24: Questionnaire results for entering the curve the bars show the upper and lower bounds of the grades.

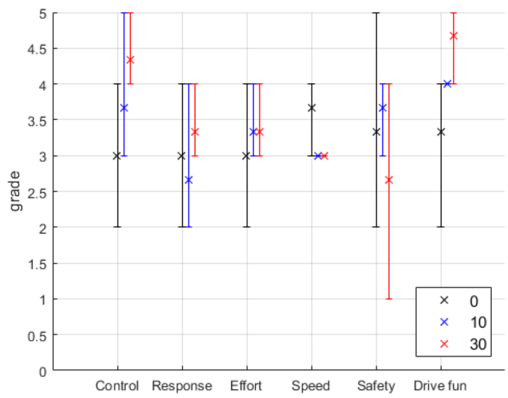


Figure 5.25: Questionnaire results for driving on the curve the bars show the upper and lower bounds of the grades.

position error for increasing camber angles. When taking a closer look at the subjective data by entering the curve the overall control and feel of safety drops at $-30°$ camber angle, which is a result of the steer response. The $-30°$ camber angle pulls the vehicle into the corner and less steer angle is needed, this was not expected by the drivers, even not in the second run with the same camber angle. At the same time a large roll developed due to the high lateral force. This steer response and roll angle resulted in less control and reduced feel of safety.

The response time of the different vehicle configurations does not change, but the camber angle pulls the vehicle into the corner, therefore it might feel as faster response. Taking the amount of steer corrections into account it suggests that the response time improves for a higher camber angle.

The steer effort is clearly increasing for the straight part, because in the straight part the drivers have to counter steer and after getting pulled into the corner the driver has to apply more force. In addition to the amount of applied torque the changes in steer angle are also maximal, including overshoots. This explains the higher effort for entering the circle than driving on the circle. Furthermore, larger camber angles result in a higher effort, as the torque of these configurations is higher. While driving on the circle, the amount of steer corrections is not that large anymore and then there is less difference in steer effort. The combination of steer corrections and steer torque might have a stronger coherence in steer effort then when using these separately of each other.

The experience of speed is surprisingly constant. When driving on the circle, the drivers seem to have more trouble to follow the track and they experienced a slightly higher speed for the reference configuration. This should be in line with the feel of safety, which is combination of the control, response and speed. It is important to note that for entering the curve the feel of safety increases, while for driving on the circle the marks had much more deviations.

Discussion

The participants were all young drivers with an age of 24-25 years old and might saw the simulator as a racing game. They all commented that they only had visual and steer feedback and therefore it is hard to say how much control they had when driving on the limit of the tires and when they start sliding off the track.

The participants had to answer eleven questions, which are a lot of variables to remember during the driving session. The results of entering the circle and driving on the circle are therefore close to each other. Still if the drivers would have experienced the two parts as completely different it would have been interesting to distinguish them. In the next experiments it can be reduced to one overall grade. The controllers only apply a camber angle when needed, thus this will not change the driving conditions radical like in the constant camber case, at most a bit nervous driving behavior. For the point to enter the circle the controllers would have more influence and this is the focus point where the questionnaire are about.

5.3.3. CONCLUSION

The goal of the proof of concept was to find out how the constant camber angle was influencing the vehicle model and how the participants would respond on this. The results of the vehicle data showed an increased lateral force by an increased camber angle, furthermore a decrease of steering angle overshoot and settling time were achieved. Therefore, the vehicle was able to drive the curve, while the vehicle with zero camber was sliding off the track. The imposed camber angle created a lateral force on the straight part for which the drivers had to correct for, which means they had to apply a counter torque for driving straight. When entering the corner this lateral force was pulling the vehicle into the corner, but this lateral force is not sufficient to drive the corner, thus an addition lateral force has to be induced from the steering angle. This resulted in a higher and opposing steer torque than the straight part.

From the questionnaire is shown that the overall control and steer response is higher valued with a camber angle than without. Better control and steer response resulted in a better feel of safety. These grades matches the vehicle data, which show improved vehicle characteristics. This makes proof of concept successful and increases the expectations of the active camber system, because the combination of driving like a conventional vehicle on the straight part and using camber control in the corner combines the advantages of both characteristics.

The constant camber angle design is only suitable for one side corner and is worthless on the straight part. This was good for the proof of concept to show that the camber angle is influencing steer feel and vehicle behavior, therefore can this be used for the design improvements of the control schemes.

5.4. REVISION SIMULATOR

In the last part we saw an increased aligning torque for cornering with constant camber, this was an unexpected result of steering torque. It was a result of the used tire formula. In Section 2.4 is explained how the aligning torque is build up, this is different than the results presented. There are two available tire formulas, both approximate the trail to create the aligning moment of the lateral force which is in both cases dependent on the tire slip angle.

- MF: magic tire formula Figure 5.26,
  this formula is a contact point method to calculate the tire forces and torques.
  In this model the camber angle is valid up to 15degrees, which leads to problems with forces and torques. The pneumatic trail is not proper changing with camber angle and due to the higher lateral force a higher torque is generated.

- easyTF: easy tire formula Figure 5.27,
  The camber angle validity is dependent on the used tire parameters, because it is dependent on contact area and therefore on tire shape. The contact area makes the easy tire formula sensitive to the orientation of the wheel and the tire forces decrease when applying a large camber angle, because the tire loses contact area and this does not compensate the increased lateral force by camber angle. A rounded tire model should then be used to find the correct tire forces.As discussed in the tire part Section 2.3 the tire has some special requirements for this application. Rounded tires would be desired to maintain the contact area. The decrease in torque using camber angle can be a result of two properties. One can be a better representation of the trails and therefore the torque decreases. Another can be the lateral force decreases and thus the aligning torque decreases.

In Figure 5.28-5.31 the comparison of the tire models is shown. In these graphs are four setups compared, "passive" means it is using the reference configuration and "active" means it is using the direct lateral acceleration controller. The passive systems are close to each other in terms of vehicle dynamics, but in the steer torque there is about 0.2Nm difference. The active systems show about 1Nm difference in steer torque during cornering and the vehicle dynamics have large differences in lateral acceleration and yaw rate. The contact area decreases during cornering due to the tire shape and the camber angle, which is decreasing the lateral forces.

5.4.1. CONCLUSION

There are two options for the tire model, both are an approximation of the expected tire forces and torques. By using the point contact model of the magic tire formula the limitations of the tire model can be used to imitate a rounded cross section tire that maintains its contact area under large camber angles, which results in the right characteristics regarding vehicle dynamics. The vehicle behavior response would be a better

In the case of **combined slip**, the input slip quantities are modified and the effect of the longitudinal force is considered.

$$M_z = M'_z + M_{zr} + s \cdot F_x$$

$$M'_z = -t \cdot F'_y$$

$$t = t(\alpha_{t,eq}) = D_t \cos[C_t \arctan\{B_t \alpha_{t,eq} - E_t(B_t \alpha_{t,eq} - \arctan(B_t \alpha_{t,eq}))\}]\cos(\alpha)$$

$$F'_y = F_y - S_{Vy\kappa}$$

$$M_{zr} = M_{zr}(\alpha_{r,eq}) = D_r \cos[\arctan(B_r \alpha_{r,eq})]\cos(\alpha)$$

$$s = R_0 \cdot \{s_{sz1} + s_{sz2}(F_y / F'_{z0}) + (s_{sz3} + s_{sz4} df_z)\gamma^*\} \cdot \lambda_s$$

$$\alpha_{t,eq} = \arctan\sqrt{\tan^2 \alpha_t + \left(\frac{K_{x\kappa}}{K_{y\alpha}}\right)^2 \kappa^2} \cdot \mathrm{sgn}(\alpha_t)$$

$$\alpha_{r,eq} = \arctan\sqrt{\tan^2 \alpha_r + \left(\frac{K_{x\kappa}}{K_{y\alpha}}\right)^2 \kappa^2} \cdot \mathrm{sgn}(\alpha_r)$$

Figure 5.26: Magic tire formula, aligning torque calculations[25]

s_y^E The lateral slip at beginning of slipping

$$\frac{n}{L} = \begin{cases} (n/L)_0 \cdot \left(1 - |s_y| / s_y^0\right) & , |s_y| \le s_y^0 \\ -(n/L)_0 \cdot \dfrac{|s_y| - s_y^0}{s_y^0} \cdot \left(\dfrac{s_y^E - |s_y|}{s_y^E - s_y^0}\right)^2 & , s_y^0 \le |s_y| \le s_y^E \\ 0 & , |s_y| > s_y^E \end{cases}$$

where L is approximated as follows:

$$L = \sqrt{8\, r_{Tire0}\, \Delta z}$$

The self-aligning torque is calculated from the dynamic lateral force and the pneumatic trail as a torque arm:

$$M_z = -n\, F_y$$

Figure 5.27: Easy Tire formula, aligning torque calculations[25]

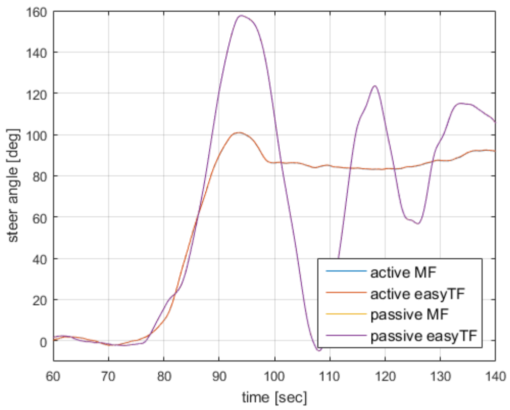


Figure 5.28: Steering angle input for the offline simulations comparing the MF and the easyTF. The steering angle inputs were recorded during the experiments driving with and without acitive camber control using the MF.

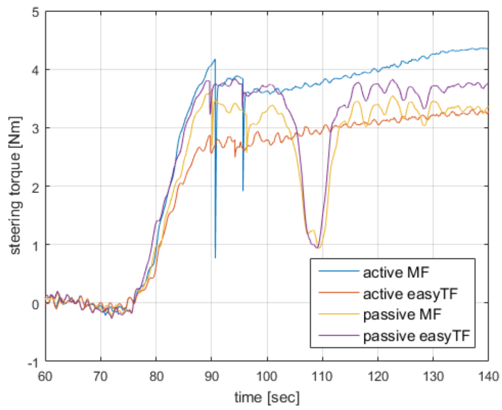


Figure 5.29: The steer torque output of the offline simulations comparing the MF and the easyTF.

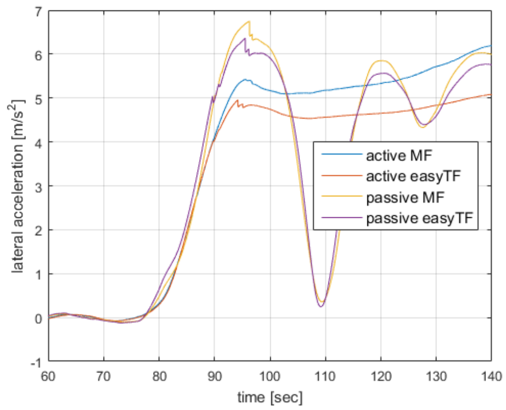


Figure 5.30: The lateral acceleration output of the offline simulations comparing the MF and the easyTF.

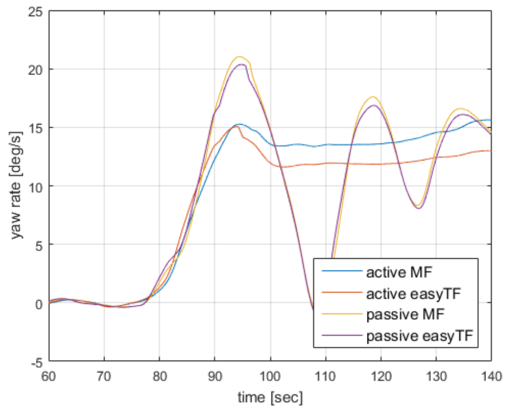


Figure 5.31: The yaw rate output of the offline simulations comparing the MF and the easyTF.

measure for the driving experience than specifically focus on the steering torque. The vehicle response is sensible in the steering wheel and therefore the experience of steering feel for changing the camber angles can be found. This increase in torque is noticeable, but the change of steer angle might have a higher factor, which can be seen in the questionnaire response where the results of steer effort are close to each other.

Further research is needed to check if this increase in steer torque give the drivers a suitable feedback and therefore have a lower overshoot and shorter settling time or that it is indeed the camber angle that is here the changing factor. By comparing the MSC.ADAMS/Car results from Section 2.5 with the experimental results it shows a decrease in overshoot and settling time, which suggests the camber angle is the changing factor.

As a final conclusion the experiments proceeds with the Magic Tire Formula, which is a better approximation of the tire forces and this would result in the driving experience of driving with cambered wheels. Therefore, the steer torque is taken as a less important factor in the research to steering feel.

5.5. Experiments: active camber control

In the previous section it was proven that the constant camber angle had an influence on vehicle behavior and steering feel. A constant camber with four wheels leaning to the left is obviously not beneficial to drive with, because it only increases lateral forces for a corner to the left and the driver has to apply a counter torque on the straight part. For a dynamic driving behavior a dynamic camber system is desired, therefore five controllers are introduced as presented in Section 3.4. These controllers make it possible to change camber accordingly to the lateral acceleration or desired yaw rate. The track is extended to an eight-shape figure track, which consists of two circles with different radii and are connected with straight parts as shown in Figure 5.1. This gives the opportunity to corner in both directions and enter the corner multiple times per session.

5.5.1. Procedure

The learning curve of the participants is a disturbing part of the simulator experiments; therefore, each participant has one training drive on a lower speed of 50km/h. This is to get familiar with the track and the simulator's vehicle model, which will flatten out the learning curve. A lower speed is used to avoid any frustration in the training session, considering the fact that the reference vehicle is driving around its limits during the tests. After this training session the speed increases to 80km/h and the test starts with the five different controllers in a semi random order. By testing in semi random order the learning curve improvements level out, thus each controller has about the same number of tests on each position in the queue. In each session the participants have to drive the eight-shape figure track for three times. In the first eight-shape they can get familiar with the controller and in the next two rounds they can focus on the controller experience. After each session the participants had to fill in the questionnaire which consists of six questions before they could continue with the next session. The questionnaire is now focused on the overall experience of the session and includes questions about how they experienced the vehicle control, vehicle response, steer effort, safety, speed and fun.

The participants were asked to drive focused and were surrounded by curtains to not be distracted during the experiments.

5.5.2. Results

In general, there is a large distinction between the controllers and the reference configuration, but not between the control systems. Most participants quickly adapt to the different controllers and therefore the objective data was close to each other.

The results are analyzed by first interpreting the tire behavior, because this is the main influence of the vehicle behavior. Then the vehicle dynamics graphs are discussed, including the yaw rate, lateral acceleration, steer wheel angle and lateral road position error. First the graphs of one test driver are discussed and then the overall picture of all participants combined is discussed. The data collection is concluded in Table 5.4 displaying the important variables. Furthermore, the questionnaire results are discussed and finally the coherence between objective and subjective data are presented.

Vehicle data

The tire forces are the source of the vehicle dynamics and these are dependent on many factors, like road surface and vehicle properties. When entering the corner, the driver gives a steer input which will change the

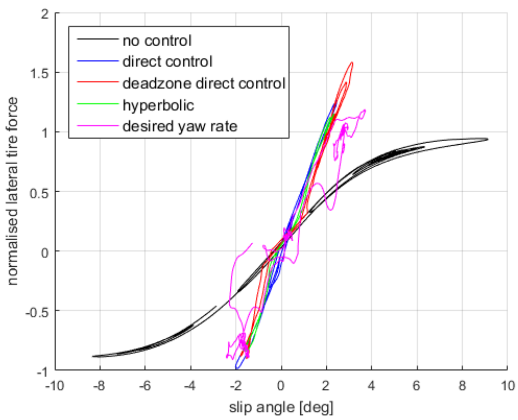


Figure 5.32: Tire forces (Front Left wheel) of the full trajectory.

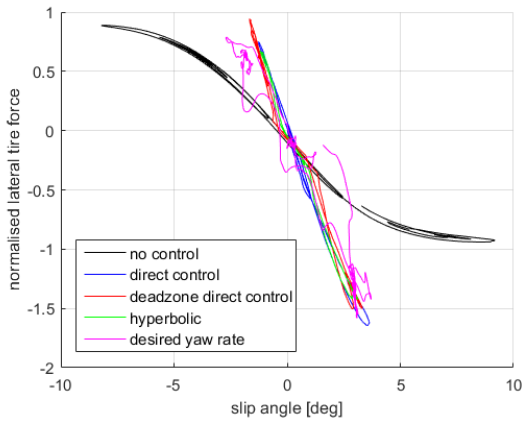


Figure 5.33: Tire forces (Front Right wheel) of the full trajectory.

orientation of the tires. This orientation difference with respect to the moving direction results in a tire slip angle and causes a lateral tire force. The tire force with a camber angle would be expected to make a shift as what happened in Figure 2.4 and as was shown in Figure 5.4 and 5.5. Due to the controllers there is no camber angle present at zero slip, because the camber angle control is dependent on lateral acceleration or steer input. In both cases there should be slip involved before a camber angle is induced and therefore there is zero camber thrust at zero slip. At maximum slip the lateral acceleration is maximum, which leads to a maximum camber angle. The progression of the tire force is dependent on the controller as will be discussed later on.

Figures 5.32 and 5.33 shows that the normalized lateral tire forces ($\frac{F_y}{F_z}$) are still in their linear region for the controlled camber systems while the reference situation reaches its maximum lateral force. A special case is the desired yaw rate. This control scheme is changing the camber angle based on the yaw rate error. Every correction of the driver is reflected to the camber angle; therefore, the line is making curly shapes. Furthermore, the graph is not symmetric in several respects. The positive and negative force are not the opposite of each other and when comparing the left and right tire graphs with each other there are also some differences. This can be explained by the different corner radii and the imposed roll changing the vertical loads. By looking at the four tires at the same time it gives a better presentation as shown in Figures C.1-C.4. A positive lateral force is for the corner to the left, while a negative force is for the corner to the right. The centripetal force imposes a roll and creates a difference in normal force for the left and right wheels. This result in more vertical force on the outer wheels, which leads to more lateral force on the outer wheels as shown in Figures C.5-C.8. Sequential the lateral forces decrease again for the inner corner wheels, due to the decrease of vertical force. By comparing the normalized forces in the graph for left and right, it shows a higher normalized force for the inner corner wheels. This is due to the force dependent friction coefficient of the tire, which is more efficient with lower forces as discussed in Section 2.3.

Finally, the longitudinal force-slip is checked for combined slip characteristics. The normalized longitudinal force is shown in Figures C.9-C.12 for only the first long radius corner. There is almost no slip on the front wheels, because it is a rear wheel driven car and this negative slip is a result of rolling resistance. On the rear wheels is slip, because it is rear wheel driven. The power-train involves a differential which equally divides the torque on the left and right wheel. The outer corner wheel has much more normal force and therefore can handle a higher force, while the inner wheel has less vertical force which leads to a much higher slip angle. These are forces in the order of 10% of the lateral force and therefore have limited effect on the maximal lateral force. The reference configuration shows higher longitudinal force and slip angle, and therefore is effecting the lateral force more.

The vehicle behavior is a response of the tire behavior, for example the yaw rate in Figures 5.34 and 5.35 and lateral acceleration in Figure 5.36 and 5.37 are responses of the tire behavior for only one driver. This driver was the closed to the average driving characteristics and he/she gives a good impression of the trend of all drivers. The reference configuration stands out in the graphs, as what was shown before in the tire graphs. The driver steers the vehicle into the corner, but the tires reaches their maximal lateral forces

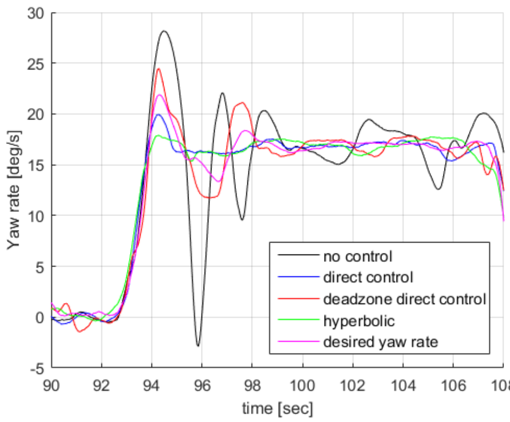


Figure 5.34: Yaw rate long corner one participant

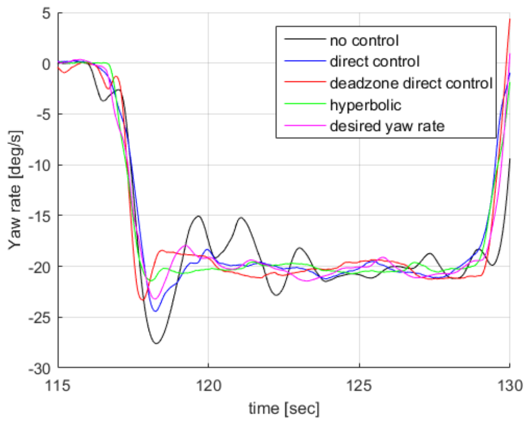


Figure 5.35: Yaw rate short corner one participant

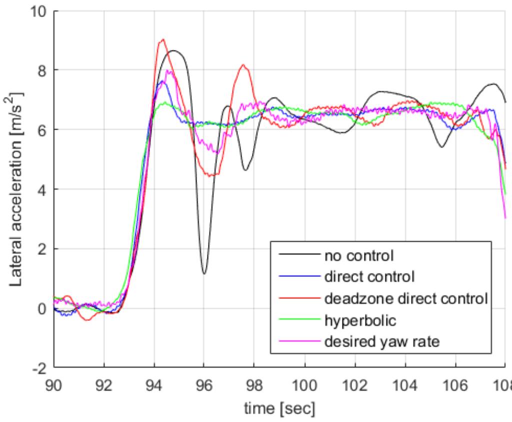


Figure 5.36: Lateral acceleration long corner of one of the participants

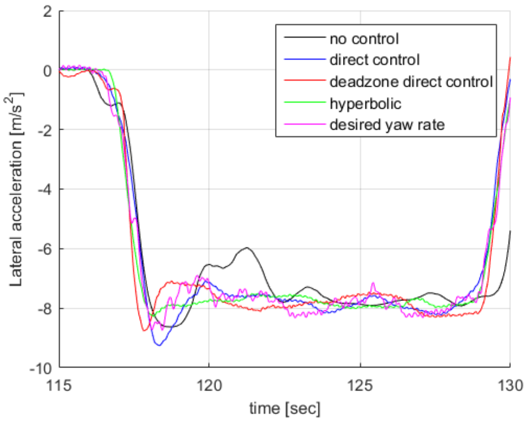


Figure 5.37: Lateral acceleration short corner of one of the participants

and the vehicle cannot follow the track and as a result the driver keeps increasing the steer angle. This is similar to the results as what was shown in the proof of concept. Also in this case the speed decreases due to force direction of the large steer angle. The lateral acceleration decreases and the tires can deliver the required force to steer the vehicle back to the track. This transition is smooth, but it happens fast. Before the driver notices it the yaw rate becomes too large and he/she has to correct for this, but this is a sensitive correction and again the vehicle loses the track. The graphs in Figures 5.34-5.39 show the large overshoot, which leads to the lateral position error which is shown in Figures 5.40 and 5.41.

This behavior of the reference configuration was what was already shown in the proof of concept. The controlled camber angles are also shown in these graphs and a clean approach of the corner is shown with around 0° steer angle. When entering the corner, the vehicle shows smaller overshoots and shorter settling times. The steering angle for the controlled camber angles shows a smoother line without too many abrupt steering motions, which means a stable and predictable behavior of the vehicle response. For the shorter radius corner even more steady steering behavior is shown, where a constant steer angle was found after a small overshoot. The speed of the controlled vehicle does not change that much compared to the reference configuration, a constant speed and a linear lateral force characteristic makes it more predictable to find the constant corner steer angle. The learning curve of the driver still plays part in this result and the differences in the control systems are visible. Since they were randomized the results differ from driver to driver, so the box plots show the results of the participants combined and this filters the learning curve.

In the boxplots three parts of the track are chosen: first the long radius corner, then the straight part in between the two corners and at last the short radius corner. These parts of the track are highlighted in Figure 5.42. For example Figure 5.43 show the boxplots of the lateral position error. The middle line

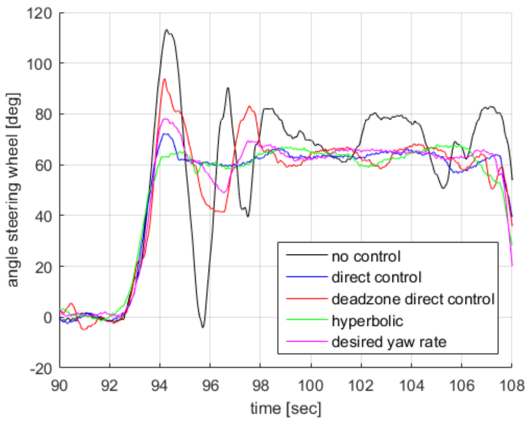


Figure 5.38: Steering wheel angle long corner of one of the participants

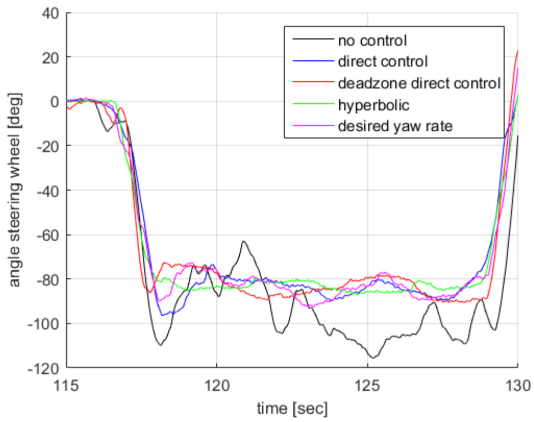


Figure 5.39: Steering wheel angle short corner of one of the participants

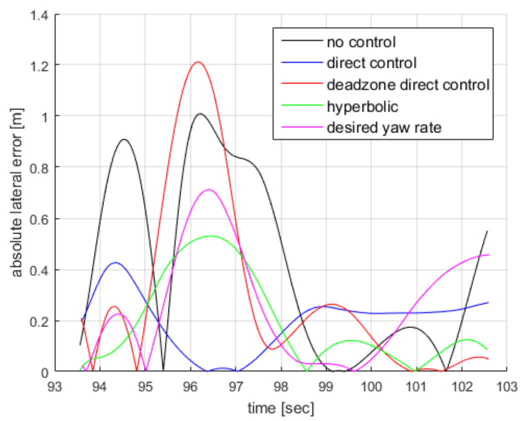


Figure 5.40: Absolute lateral position error of the long corner of one of the participants

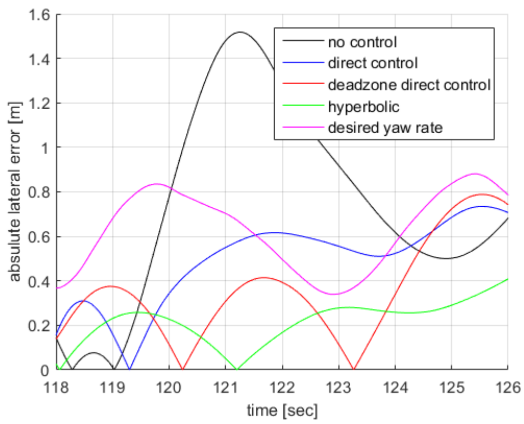


Figure 5.41: Absolute lateral position error of the short corner of one of the participants

represents the median, the box is 75% of the data and the bar shows 98% of the data. The red dots are outliers and can be a result of a numerical error or just a short peak, due to high number of data points the red dots can easily be over one hundred points and therefore it looks like a line.

The median of the lateral position error in Figure 5.43 shows an error of about 0.35 m for the reference configuration in the long corner and the box itself is more than two times larger than the boxes of the controlled systems. For the short corner the box is smaller, but the median shows an even higher error. The boxplot of the steering angle confirms the observation of the larger deviation for the reference configuration as shown in Figure 5.44, because this large deviation in steering angle causes the large lateral position error. All the controlled systems have a small deviation with the median on about the same steering angle. The lower side of the box of the reference configuration starts at the median of controlled systems, therefore large steering angles were applied in an attempt to keep the vehicle on the desired line. Furthermore, in the Appendix C more box plots are shown regarding lateral acceleration, roll angle, longitudinal velocity and camber angle. The boxplots show a clear distinction in deviations between the reference configuration and the control systems, but the control systems are all very close to each other. Therefore, the average values of the short corner are presented in Table 5.4. From this table the most distinctive values are the lateral position error, steer torque, steer angle and roll angle.

In paper [2] the investigators try to change the understeer gradient with active camber control. By applying active camber to the rear wheels they aim for more understeer. As what was shown in Section 2.5 where the stability of the vehicle can be changed by applying front or rear camber angle. In this investigation the stability was meant to be unchanged, which is investigated in the next paragraph. The results of the

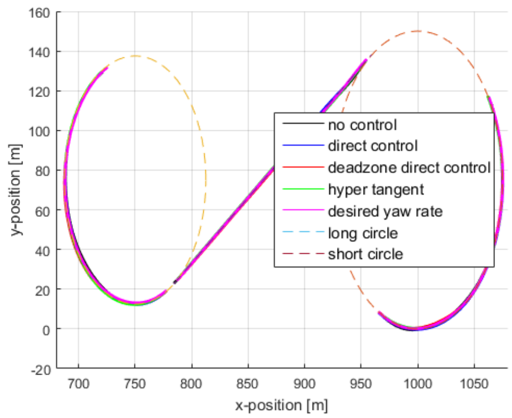


Figure 5.42: Measured parts of the track for box plots

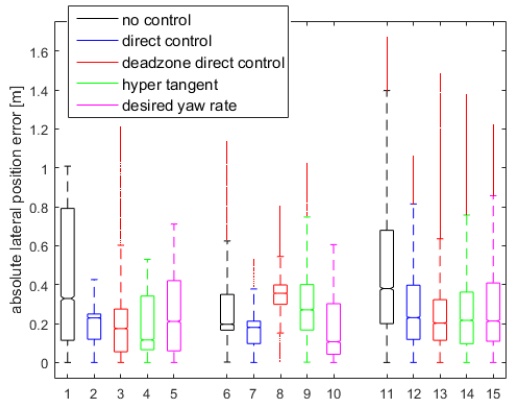


Figure 5.43: Absolute lateral error for long corner/straight part/short corner for all participants combined.

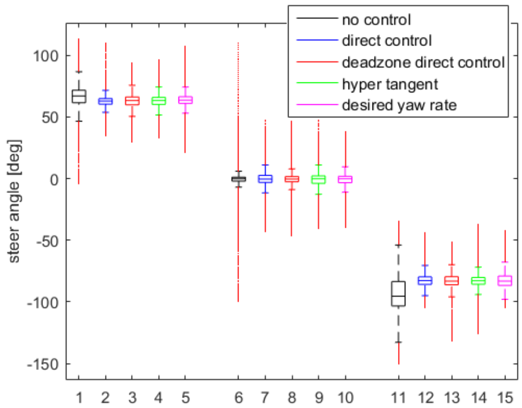


Figure 5.44: Angle steering wheel for long corner/straight part/short corner for all participants combined.

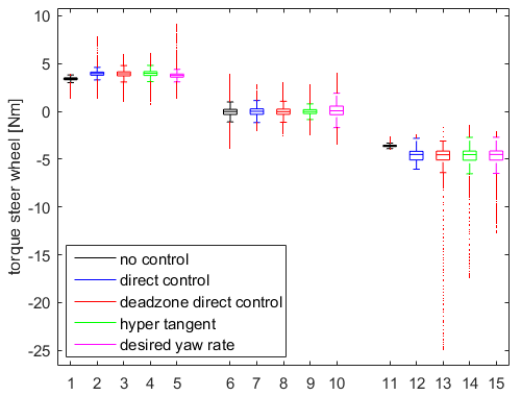


Figure 5.45: Torque on steering wheel for long corner/straight part/short corner for all participants combined.

Table 5.4: Average data of the vehicle in the short corner

	no control	direct control	deadzone direct control	hyperbolic	desired yaw rate
a_y	$7.35 m/s^2$	$+5.21\%$	$+5.66\%$	$+5.41\%$	$+4.93\%$
δ_{steer}	$93.87°$	-12.19%	-11.77%	-11.84%	-12.09%
$\dot{\Psi}$	$19.95 rad/s$	$+0.85\%$	$+1.25\%$	$+1.03\%$	$+0.50\%$
$Roll$	$4.23°$	$+16.16\%$	$+16.83\%$	$+16.35\%$	$+15.27\%$
V_x	$76.99 km/h$	$+3.03\%$	$+3.01\%$	$+3.04\%$	$+2.90\%$
$error$	$0.48 m$	-41.61%	-45.38%	-41.34%	-37.60%
T_{steer}	$3.59 Nm$	$+28.10\%$	$+30.80\%$	$+29.52\%$	$+29.91\%$

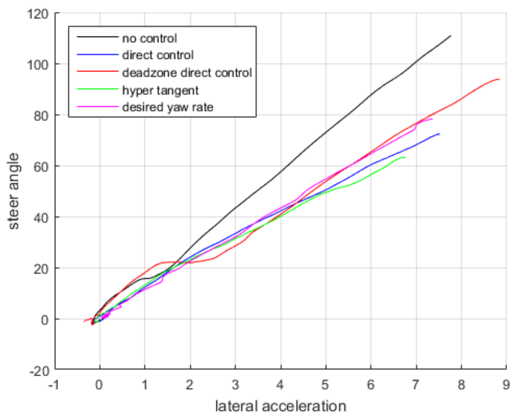


Figure 5.46: Steer angle dependency with latteral acceleration, showing a lower understeer gradient for the controlled systems. The differences in controllers are shown in the behavior of the lines. No control and direct control show straight lines, while the others change dependent on the control type.

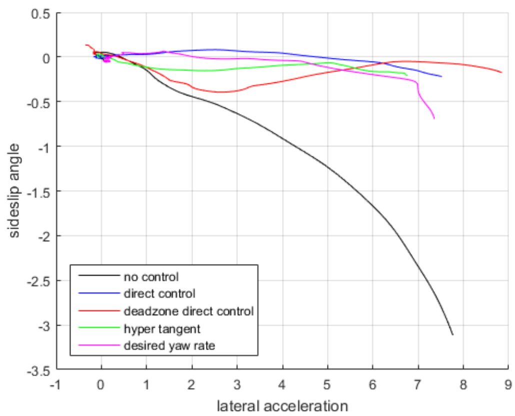


Figure 5.47: Vehicle side slip angle dependency with latteral acceleration, showing less vehicle side slip for the controlled systems.

paper and the experimental test results differ largely, because in this paper the steer angles are smaller and the used camber angles are up to 4°.

The understeer gradient of the experimental tests and the paper results are shown in Figures 5.46 and 5.48. The dSPACE ASM model is decreasing the understeer gradient when leaning with the four wheels into the corner. Even with the discussion in Section 2.5 to use camber on all four wheels it does prevent changing the stability. It shifted to a less understeered situation. In Figures 5.46 and 5.48 are the results shown of the person of the other graphs. When looking at the graphs of different people it varies more than expected. The reference configuration and direct control are in almost every case just straight lines with a different coefficient. In general, do the deadzone direct control and hyperbolic start with the same coefficient as the reference configuration before joining the direct control and desired yaw rate. The deadzone direct control starts with the reference and when the camber starts working the steer angle drops most of the times, because the camber angle coefficient is stronger and suddenly helps cornering and drivers mostly overreact to this. The hyperbolic makes in general a smooth transition from the reference to the direct control, because the camber angle becomes smoothly stronger with more lateral acceleration. The desired yaw rate is dependent on the steering behavior of the driver and this leads to curly figures in this graph with steer angle and lateral acceleration. Another measure is the vehicle sideslip with the lateral acceleration as shown in Figure 5.47. This shows that the reference configuration suffers from the lateral acceleration in an early stage, while it almost remains constant for the controlled systems. The understeer gradient of the paper is shown in Figure 5.48. It creates more understeer with a camber angle only on the rear wheels leaning into the corner. The differences are much smaller than were achieved with the ASM model, because they use camber angles up to 4°. The main advantage of this system is that they can use the regular tires and optimize the contact area.

The differences between the controllers and their dependency on lateral acceleration and yaw error are interesting measures for judging the controllers. The direct control and deadzone direct control are clearly linear dependent on lateral acceleration and the hyperbolic makes a smooth transition between the reference configuration and the direct control as shown in Figure 5.49. The desired yaw rate is dependent on how the driver makes the corner, which can differ from a linear line to a curly line. Therefore, a graph dependent on the yaw error is presented in Figure 5.50. The direct control and hyperbolic have the least yaw error and are pretty steady lines, while the deadzone direct control goes everywhere. Comparing the results with the paper it looks like they also use a hyperbolic camber controller, because smooth lines are shown in Figure 5.51 comparable with the hyperbolic controller implemented in ASM.

The desired yaw rate and yaw error are good measures to check to what extend the vehicle can follow the steer input. Equations 2.2 and 2.4 show that the difference is in the slip angle, thus there is always an error during constant cornering. The desired yaw rate is almost equal to the steering angle when assuming a constant speed and constant corner radius as shown in Figure 5.52 and 5.53. The yaw error shows the

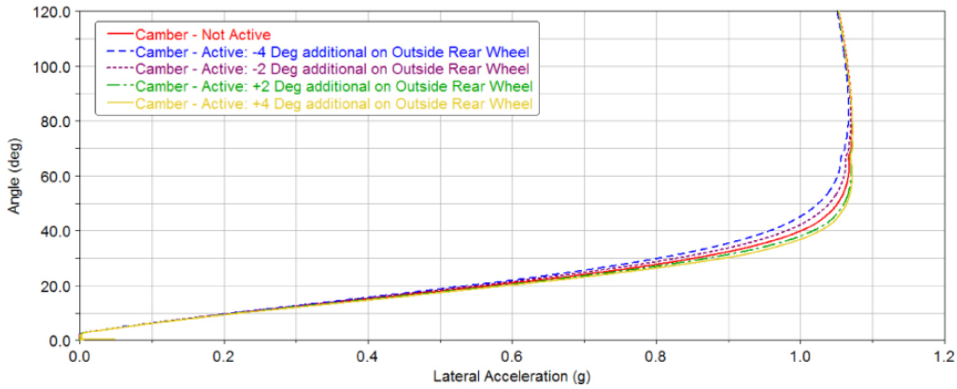


Figure 5.48: Ferrari steer dependency on lateral acceleration [2]. Camber angles upto 4° make minor changes.

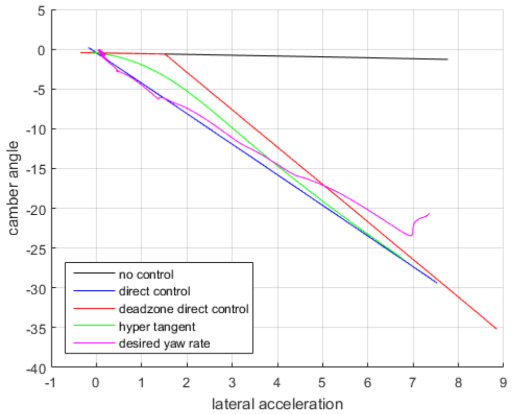


Figure 5.49: Camber angle dependency with lateral acceleration showing the characteristics of the camber angle actuation regarding the controller.

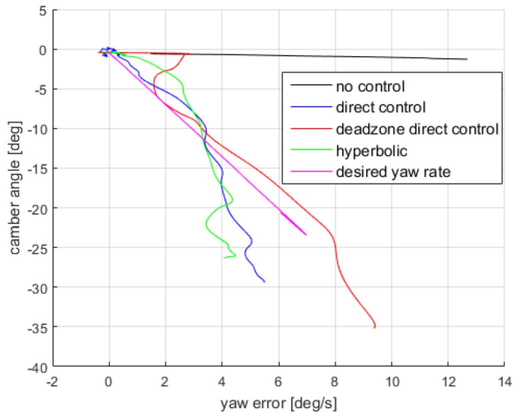


Figure 5.50: Camber angle dependency with the yaw error showing a straight line for the desired yaw rate and curly shapes for the other controllers.

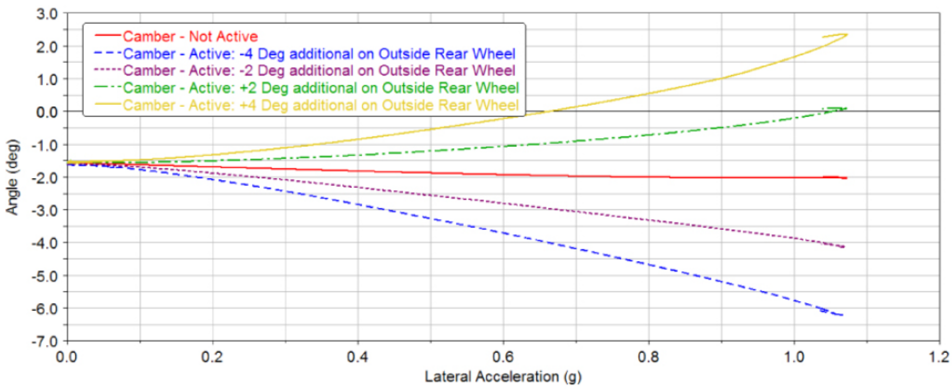


Figure 5.51: Ferrari camber dependency on lateral acceleration [2]

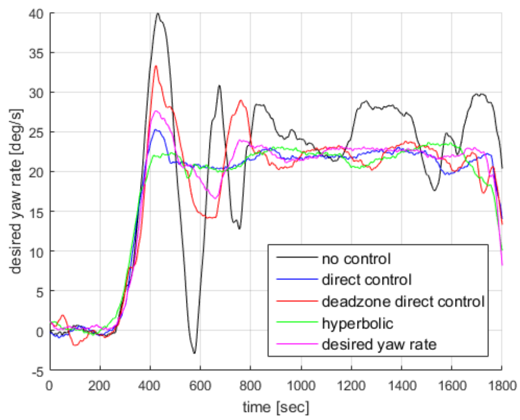


Figure 5.52: Desired yaw rate of the long corner of one of the participants.

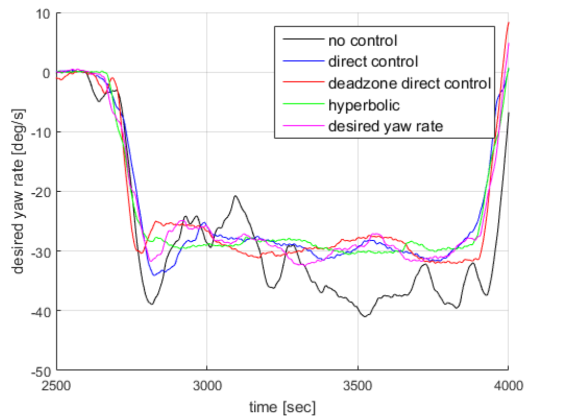


Figure 5.53: Desired yaw rate of the short corner of one of the participants.

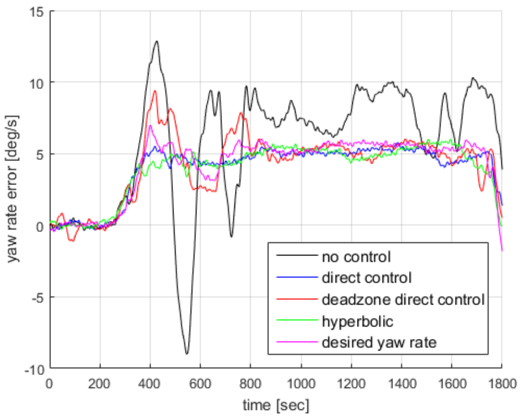


Figure 5.54: Yaw error of the long corner of one of the participants.

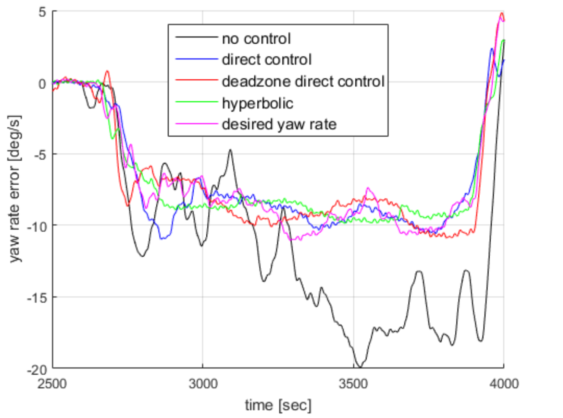


Figure 5.55: Yaw error of the short corner of one of the participants.

difference with the real yaw rate. Especially for the reference configuration very large errors are shown as shown in Figure 5.54 and 5.55.

Next to check for understeer gradient are these graphs important measures to check how the camber controllers behave on the different aspects and not only the variable where they were designed for. As an example, the desired yaw rate makes curly shapes in lateral acceleration and the deadzone makes an interesting swing in the yaw error.

QUESTIONNAIRE DATA

After each driving session the participants had to fill in the questionnaire shown in Figure D.22. These questions cover the whole session. Containing the straight part run up and three times the long corner, straight parts and short corner. The results are shown in Figure 5.56.

To show the statistical significance of the results a Wilcoxon ranksum test is performed. The results of the reference configuration versus the controllers are presented in Figure 5.57. There is no statistically significant difference between the controllers and therefore only the statistically significance is shown between the reference configuration and the controllers.

The overall control, steer response and safety showing statistically significant differences between the reference configuration and the controllers. The drivers had less control without the active camber system but the control systems were difficult to distinguish individually. The drivers felt differences, but it was hard to grade them. So the overall control and steer responses were mostly graded between 3 and 4 on a scale from 1 to 5, which means sufficient or good behavior, while the reference configuration was mostly graded between 1 and 2. For the steer effort we can see a slightly higher effort for the reference configuration. Also

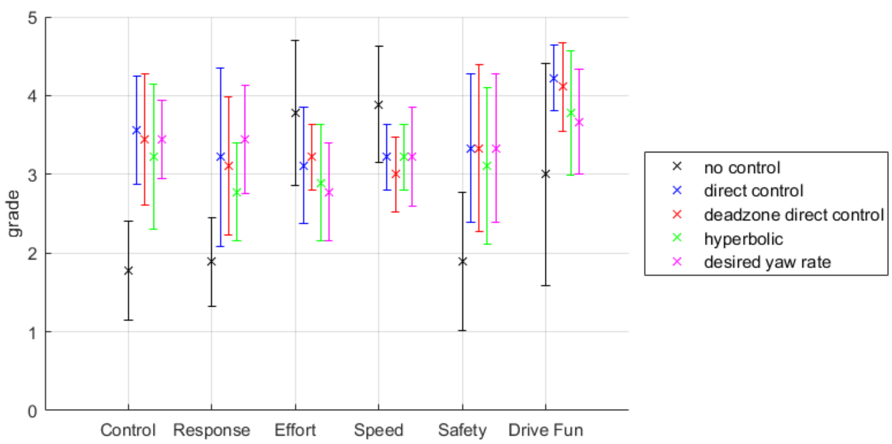


Figure 5.56: Results questionnaire

the speed is experienced higher in the reference configuration. Which leads to a lower score for the safety. The drive fun shows a large deviation in the reference configuration, but are mostly high graded for the controlled ones. By comparing the questionnaire data with the vehicle data give more insight in the answers of the questionnaire. More detailed results of the questions are presented in Appendix D.

Comparing Vehicle data and Questionnaire data

In the previous paragraph the results of the questionnaire data were presented. An overall view indicates that the controllers are all close to each other and the reference configuration mostly stands out. To find why the reference configuration stands out, the questionnaire results are compared with the vehicle data. By using Table 5.1 the data can be evaluated.

The table links the overall control with the standard deviation of the steering wheel angle, settling time and lateral position error. Looking back to the graphs of steering angle in Figures 5.38 and 5.39 the reference configuration shows a lot of overshoot and a massive settling time. This corresponds to a difficult to control vehicle, which came out of the questionnaire. The lateral error position confirms this with a larger error for the reference configuration. Also in the yaw rate and lateral acceleration this behavior of overshoots and settling times comes back. On the other hand, the controlled vehicles show smaller overshoots and a shorter settling time and this corresponds with higher grades in overall control. The drivers found the steady state cornering faster and this resulted in a better control. A similar behavior is happening with the response time, although the response time is about the same for all systems. The reference configuration lacks lateral force which makes it feel like it responds later. The desired yaw rate controller response is the quickest, because it is directly dependent on the steering angle. This is an instantly controllable input, while the lateral acceleration first has to respond on the tire forces. Due to this lack of lateral force in the reference configuration the amplitude of the steer angle becomes larger, which leads to high overshoots and a large settling times. Therefore, the grades of the steer response are comparable with the overall control.

The steer effort gives interesting results, because the question clearly asks about the force they have to apply on the steering wheel, which is lower in for the reference configuration as the steer torque shows in Figure 5.45, but still it is graded as heavier. After asking the drivers they interpreted the work they have to provide to keep the vehicle on the track as effort and they had to provide more work to keep it on the track. In other words, the standard deviation of the steering angle was more important than the higher torque.

The experience of speed and safety are very similar compared to overall control. If the controller is difficult and the lateral position error is large the speed is most likely too high for that corner with this configuration and it does not feel safe to drive that track. This finally results in the fun to drive. If it is too difficult there is no fun, but if it is not fully natural as what happens with desired yaw rate the fun also slightly drops.

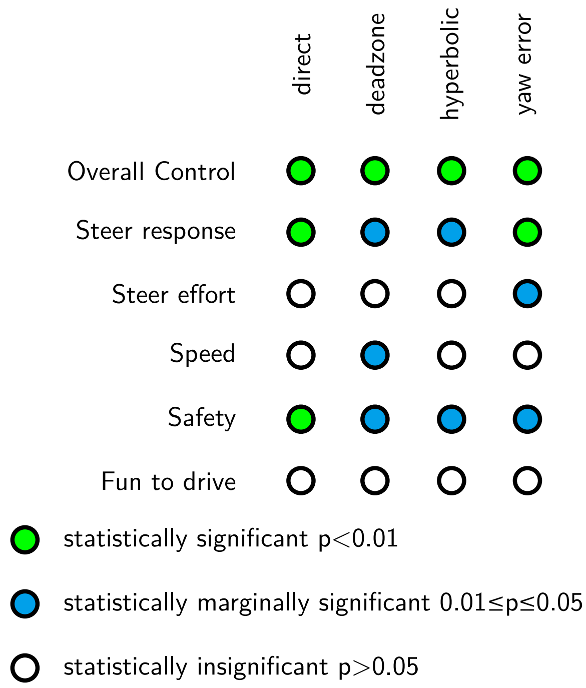


Figure 5.57: Subjective evaluation of experimental results

5.6. DISCUSSION

The **update of the circle track to the infinite loop** had some major advantages. Where in the circle track the negotiation of the corner could be done once a session, it can be done for unlimited number of times in the infinite loop. The straight parts in between the corners ensured that the transient response of the previous corner is omitted, which guaranteed a constant vehicle behavior before taking the corner and the different vehicle configurations can be compared with each other. The track consists of two different corners radii, which had two important reasons. One is the differences in vehicle dynamics results, which could provide information regarding the controller with different lateral accelerations. The second reason is the learning process and alertness of the participants. If the radius was always the same the participants could get used to the specific curve and even get disinterested. In this case with different radii in opposing directions the participants were kept alert and interested.

The **control of the active camber system** is important to be understood and accepted by the driver. A predictable behavior is desired to get the expected response after a certain steering wheel angle input. Four controllers were designed and compared with the reference configuration without camber control. One controller is completely dependent on the lateral acceleration (direct lateral acceleration controller), which amplifies every steering correction and leads to a nervous driving behavior on the straight part. There are two controllers which make a transition between the reference configuration and this first controller. One with a smooth transition (hyperbolic controller) and one with an offset followed by a stronger gradient (dead zone lateral acceleration controller). These two controllers behave like the reference configuration among small lateral accelerations, but the transition to the additional camber thrust makes it less predictable what the vehicle response would be. The hyperbolic controller can be tuned in various ways and finally a smooth balance in transition speed was selected, while the dead zone lateral acceleration consists of a sharp transition and a stronger gradient. The differences between these two controllers were too small to notice, for relative unexperienced drivers. The fourth controller imposes camber angle according to the yaw error. All controllers decrease the yaw error, but this one is specifically focused on the yaw error. This controller was evaluated as insecure. It felt unpredictable and loses its aligning torque when driving close to a straight line, because in the beginning of a steering input a peak is created in the yaw error and this directly activates the camber angle. Finally, the reference configuration was mostly judged as a worse designed controller and difficult to control. After the tests, when the drivers were told that it was the reference configuration their response was mostly surprised, because at that moment they realize that a vehicle driving on its tire limits is hard to control. The active camber controllers were impressively increasing driving performance compared to the reference configuration.

The **change in velocity** is important for the ratio of the steering angle and yaw rate, because the velocity defines the gain between those two variables. A constant velocity results in a more predictable behavior, because then the steering angle and the yaw rate are equivalent to each other. The reference configuration has much more change in velocity and this results in more corrections of the steering angle to perform the same yaw rate.

The tire force graphs show the expected **tire characteristics** as discussed in Section 2.3. For the reference configuration the maximum tire forces were reached for both corner directions, while the controlled camber systems reached till the end of the linear region. An advantages of this linear region is that it contributes to the predictability of the vehicle response, because a linear behavior is easier to predict for the driver. The desired yaw rate is different than the other camber control systems and show curly shapes in the tire forces, which would indicate an unpredictable response. This is the case as discussed before and results in an inconsistent lack of aligning torque and a feel of getting pulled into the corner. Although the tire graphs were for almost every participant equal in shape the vehicle dynamics responses were widely spread. The reason for this result is the tire formula, which responds with a certain lateral force for a slip angle and therefore the shape of the graph would not change. Only the position on the line in the graph changes, because the steering wheel input has major influences on the results of the vehicle response.

The graph of **camber angle dependency** with respect to the lateral acceleration show the behavior of the controllers. The linear response presumes a predictable driving behavior for the driver, because it is more instinctive to expect a linear response than a curve. An ever changing line like the desired yaw rate controller or a swing in the line as shown in the dead zone controller introduce a change in the lateral tire slip, which has to be corrected in the steering angle to maintain a constant yaw rate.

An investigation in **understeer gradient** was done in Section 2.5 where was concluded that camber on all four wheels would be sufficient to maintain the same understeer gradient, but the decrease of the slip angle due to camber thrust was not taken into account. This changes the vehicle side slip and leads to less understeer. Thus it changes the understeer gradient, but since there is less tire slip it does not influence the stability and increases the safety of the vehicle. For maintaining a specific understeer gradient, the ratio between front and rear camber angle should be adjusted. This can be by decreasing the camber on the front wheels and increasing it on the rear wheels.

For this investigation the **target group** were all young drivers from the university community. Only some of them had own a car and therefore the driving experience was mostly limited. Others had some gaming experience which might also have some influence on the steering wheel input behavior in the simulator. The comments of the game feel experience from the proof of concept were used to instruct the participants that it is to simulate a realistic situation and that they should drive it as if they were driving in their own car. The steering feel was mostly noticed as realistic, what enhanced the realistic feel of the simulator.

The **subjective results** of the participants were sorted on driving experience, age and gaming experience and even with a weighted sorting of these three categories, but none of these sorting's did show a clear trend in the subjective data. The active camber control assisted more with keeping the vehicle on track for the participants with less driven kilometers than for the more experienced drivers, but the differences were very close to each other. The inexperienced drivers had more problems with filling in the questionnaire and to distinguish the different controllers. This resulted in often the same grades for the inexperienced drivers, while the vehicle data showed some clear differences in vehicle response between the reference and controlled configurations. Beginner drivers are most likely more focused on driving the vehicle with all the usable feedback and therefore easily create large overshoots in the steering wheel angle. This leads to large deviations in lateral acceleration, yaw rate and lateral position error. The more experienced drivers had less difference in vehicle data and easily found the steady state corner. Some drivers were able to produce about the same graphs for with and without camber control. They could easier adapt to the changing steering characteristics. By applying the required amount of steering angle at the right moment, without overcorrecting resulted in almost equal yaw rate and lateral acceleration for all systems. These drivers remembered after the first round with the new controller which steering angle is required for the corner. Most of the experienced drivers were therefore able to distinguish the different controllers, but some of them could perfectly drive the track and had no idea what changed. Finally, this resulted in statistically significant data for the overall control, steer response and safety between the reference configuration and the controllers, while between the controllers there was almost no difference.

The simulator is a **fixed base simulator**, which limits the driver in the experience of motion cues. The lateral acceleration and vehicle roll had to be conducted from the visual effects. This might affect the result on the experience of safety, because some drivers did not even observe the roll of the vehicle. If these accelerations and vehicles roll angles would occur on a road situation the driver response might be completely different, because this can be frightening for the driver and he/she might press the brake or change the steering wheel angle. The simulator is a software simulation of reality, where the participants had the most comments about. Nevertheless, although the simulator lacked certain details in visualization and motion cues, the steering feedback was sufficient to keep the drivers focused and provide reliable answers.

The **experiments had to be performed at the tire limits** to distinguish the feel of control, steer response and safety feel, because otherwise more motion cues are needed to experience the full driving behavior of active camber systems. In a fixed based simulator, driving on the infinite loop track with 50 km/h instead of 80 km/h only the steering angle and a slight change in feel of control is noticed, because the driver does not feel the motions of the vehicle. Mercedes [10] claims a more joyful drive and more control with their active camber suspension concept vehicle, which has most likely to do with the smoother and playful motions of the vehicle.

A limitation of this research investigation is the **tire model**. The lateral force shifts due to the camber angle, which is valid up to $15°$ camber. The magic tire formula is using a point model, therefore the orientation of the tire has no influence on the contact area. Especially with higher slip angles the forces become unreliable, however the slip angle decreases due to slip and the lateral force reaches until the end of the linear region. The lateral tire forces increase to a too high level, even if rounded cross-section tire were considered, but the impression of increased lateral forces due to camber angle is correct. The **steering wheel torque** is also too high as was discussed in Section 5.4. This can lead to less overshoot in the steer corrections, because more resistance might prevent to steer too much. Less resistance in the steering wheel easily passes the desired angle and therefore creates large overshoots and a larger settling time. The results finally showed an increase in control and less steer effort for the cambered control systems with a higher steering torque, which points out that the higher torque is not influencing the perception of the driver and the amount of steering wheel angle correction plays a higher role in the steer effort. Furthermore, MSC.ADAMS/Car experiments of Section 2.5 showed a decrease in overshoot and settling time for the cambered vehicles, thus this would suggest that this is not a result of the steering torque, but a result of the additional camber thrust.

The **tire shape** and stiffness are of important matter to find the differences in vehicle response. Due to the tire model it is as if the reference configuration uses normal rectangular cross section tires and the cambered tires are special tires which maintain the contact area and increase lateral forces.

6

CONCLUSIONS

"We determine who we are by what we do"

- Starmall cinemas, *Hitman:Agent 47*

The study was set out to explore the influence of active camber control on steering feel, as an extension to the existing investigations to active camber control mechanisms, where the focus is mainly on the mechanism kinematics and vehicle dynamics. In this study the driver was taken into the loop with the focus how he/she experienced driving with active camber control.

With the evaluation of the active camber mechanisms varying design concepts were shown. These can be used for the design of a suspension system dependent on the type of use. The mechanisms showed different characteristics for example in motion and force findings, which leads to design choices in body work and actuator requirements. A trade-off is inevitable when considering an active camber mechanism.

This active camber mechanism need a control system for defining the camber angle per situation. The driver should always keep control over the vehicle, therefore the design of a control system is essential for the interaction between driver and vehicle. For these initial experiments with a simplified eight shape figure track a predictable and convenient controller is preferred. The lateral acceleration is a convenient control input, because this is a function of the required lateral forces needed to keep the vehicle on the track. Three controllers based on lateral acceleration and one controller based on desired yaw rate were designed for the experiments.

These controllers were implemented in the dSPACE ASM driving simulator to conduct the experiments. The camber control system for changing the camber angles were implemented after the look-up tables of the conventional suspension system to ensure the real-time processing of the vehicle dynamics.

After the implementation in the driving simulator the tests with the participants could be performed. The participants had to drive an eight shape figure track consisting of two different corner radii in contrary directions, therefore the drivers did not get used to a certain steering wheel angle and get disinterested. The vehicle data was recorded and a questionnaire was enquired after every session. These results were reviewed by beginning with the tire responses, which shows increased lateral forces with applied camber angles as was expected from the literature study. These increased lateral forces leaded to a decrease in overshoots of the steering wheel angle, which showed a decrease in overshoots in the yaw rate and lateral accelerations. This indicates an improvement in vehicle control, together with the responses of the drivers who gave statistically significant higher grades for the controlled systems. A similar result is shown for the steer response, because the reference vehicle without control passed the maximum lateral forces and this made it close to impossible to keep it on the track, which felt as if it was responding very slow. The improvements in vehicle control and steer response leaded to an improvement in vehicle safety, because the drivers had a better feeling of the vehicle response and less difficulties to keep the vehicle on track with the controlled camber systems. These improved vehicle dynamics leads to a decrease in tire slip angle, which have influence on the understeer gradient. By applying camber on all four wheel decreases the tire slip equally, but less tire slip leads to a lower understeer gradient. Next to the understeer gradient does the increase op the lateral tire forces also make the vehicle sensible for roll over. During the experiments the speed was just beneath the point of creating lateral accelerations to start a roll over.

As a final conclusion, it can be said that the improvements of the active camber system are in both vehicle dynamics as human interaction. The vehicle dynamics showed less overshoots and a shorter settling time and indicates better handling characteristics. This is confirmed with the questionnaire with the participants, who showed statistically significant data between the reference configuration without control and the controlled camber systems. Between the controlled systems was no statistically significant data shown, which indicates that the active system improves vehicle handling independent of the control scheme.

7

Recommendations

"One of the greatest discoveries a man makes, one of his great surprises, is to find he can do what he was afraid he couldn't do"

- Henry Ford

A simulator is the ideal setting for tests that require repeatability, which is crucial for comparative studies. However, a considerable amount of limitations is the used for the models inside. For example:

- The tire model,
  the tire model inside ASM is valid only up to 15 degrees. To get a more realistic feel the tire model can be extended to a more suitable one with more parameters regarding tire shape and camber dependency. This will also lead to a more realistic steering torque feedback.
- A real time active suspension model,
  by extending the suspension model with an extra DoF and a controller for the actuator of the camber angle.
- The motion cueing,
  if the roll and lateral acceleration were to be considered, further analysis on the motion cues and visual effects would have been required. These can change the experience of the driving behavior to a more realistic feel.
- The control schemes,
  the control schemes were focused on only camber controls for drive-ability and steering feel to drive the simple eight shape figure track. However, there are numerous other applications, such as emergency or evasive maneuvering, which can be improved by more situation dependent control strategies. The mechanisms are individual systems per wheel, which also gives the opportunity to change the camber angle per wheel and provide more precisely control compared to the vehicles using systems such as ESP. This can improve the driving performance (for racing), prevent roll over, maintain stability, etc.
- Simulation tracks,
  multiple tracks with for example double lane change give a better representation of the vehicle characteristics with different controllers.

A

Vehicle models

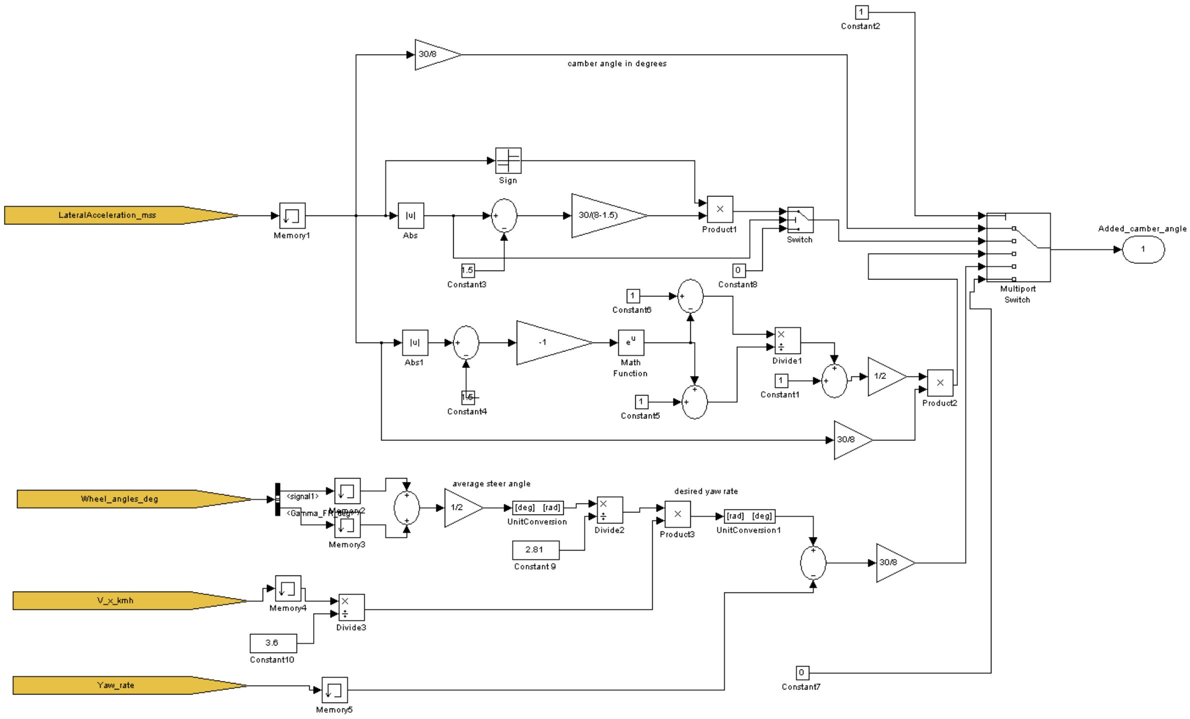


Figure A.1: Camber control systems

B

MSC.ADAMS/Car Simulation Results

B.1. Data collection of the passive camber MSC.ADAMS/Car simulation results

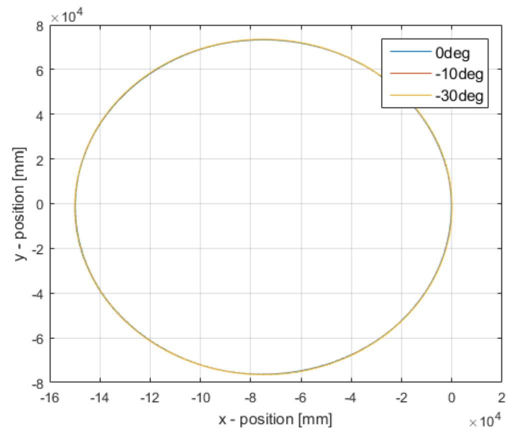


Figure B.1: Trajectory

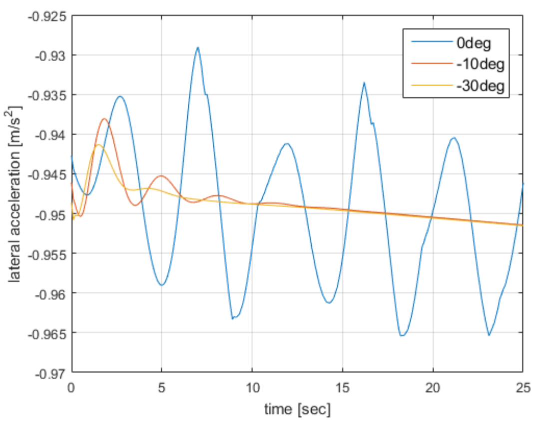


Figure B.2: Lateral acceleration

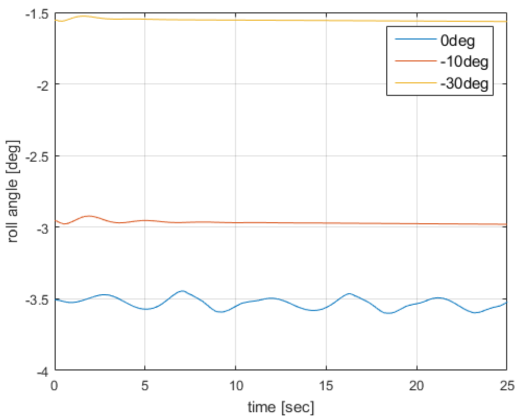


Figure B.3: Roll angle

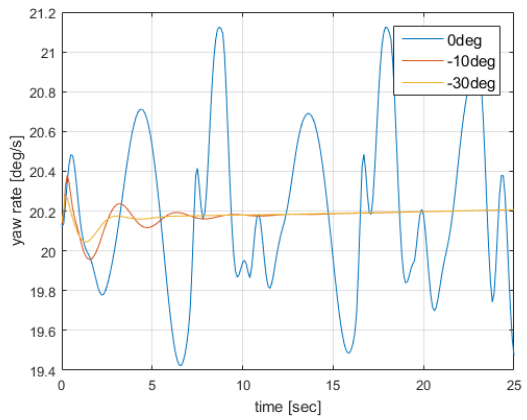


Figure B.4: Yaw rate

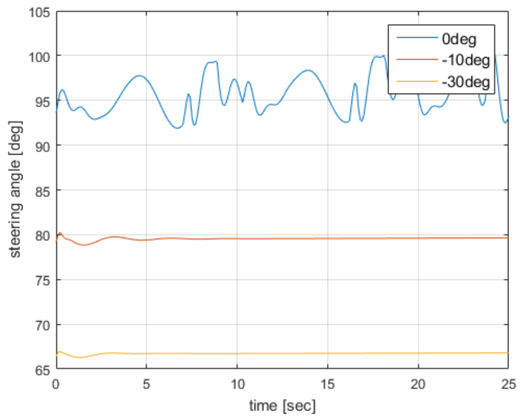


Figure B.5: Steer angle

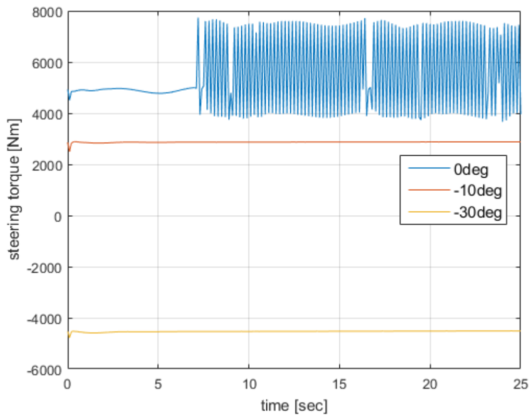


Figure B.6: Steer torque

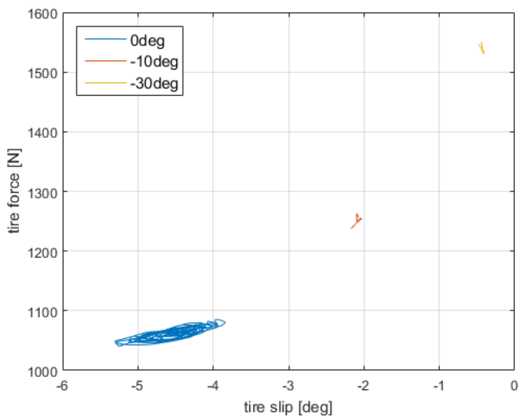


Figure B.7: Tire force vs slip FL

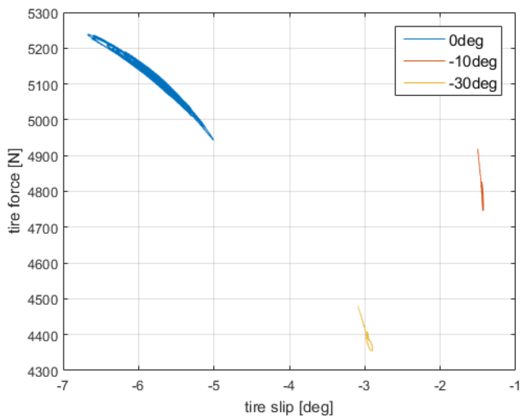


Figure B.8: Lateral tire force vs slip FR

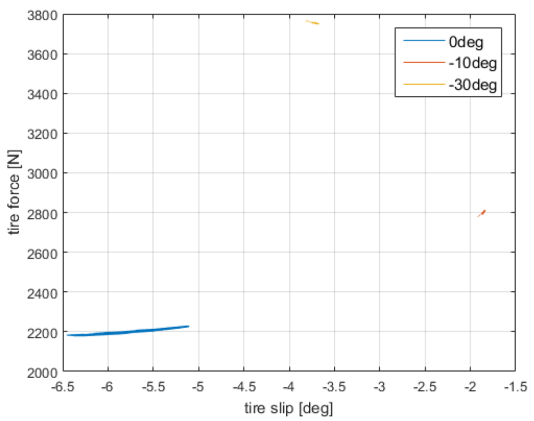


Figure B.9: Lateral tire force vs slip RL

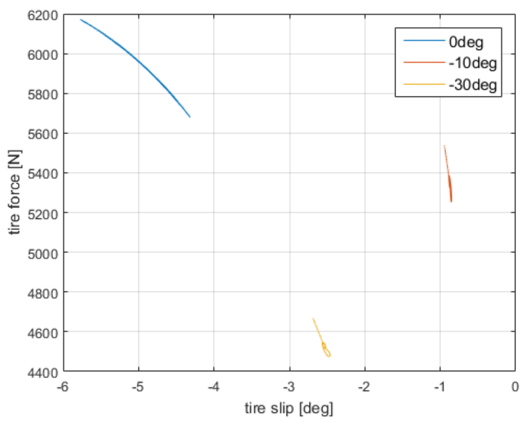


Figure B.10: Lateral tire force vs slip RR

C

DRIVING SIMULATOR RESULTS

C.1. DATA COLLECTION OF THE ACTIVE CONTROLLED CAMBER DRIVING SIMULATOR RESULTS

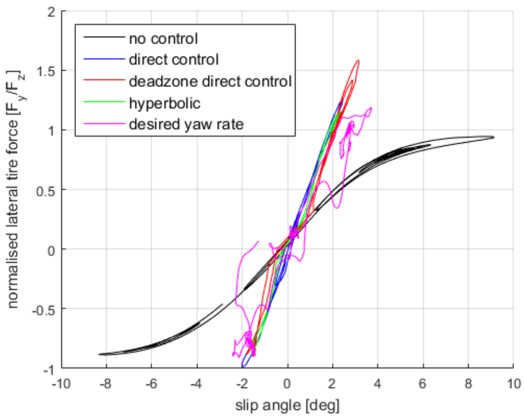


Figure C.1: Front left normalized tire force time[90-130sec]

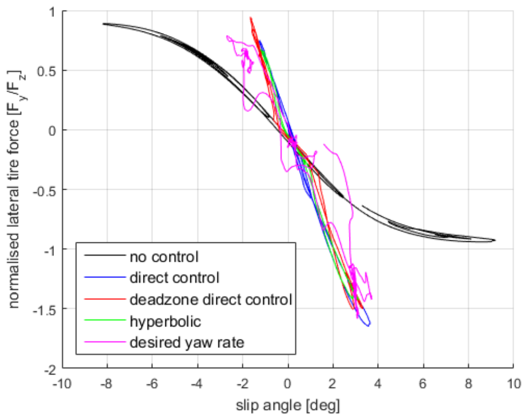


Figure C.2: Front right normalized tire force time[90-130sec]

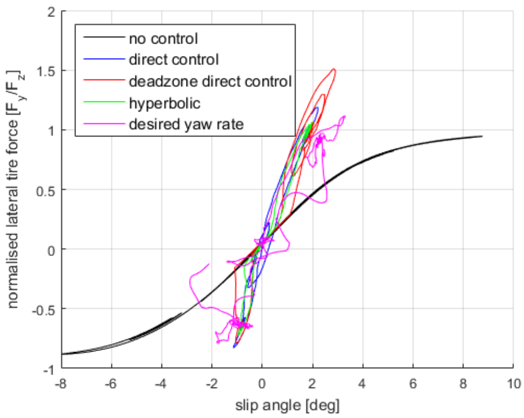


Figure C.3: Rear left normalized tire force time[90-130sec]

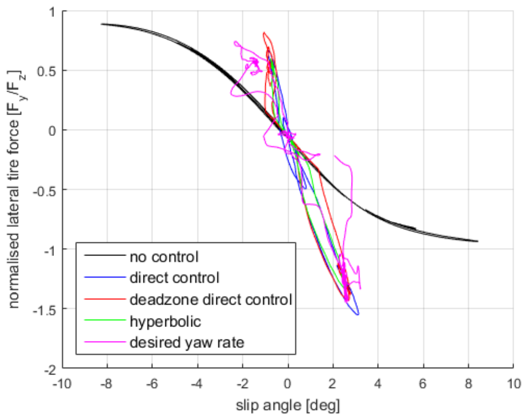


Figure C.4: Rear right normalized tire force time[90-130sec]

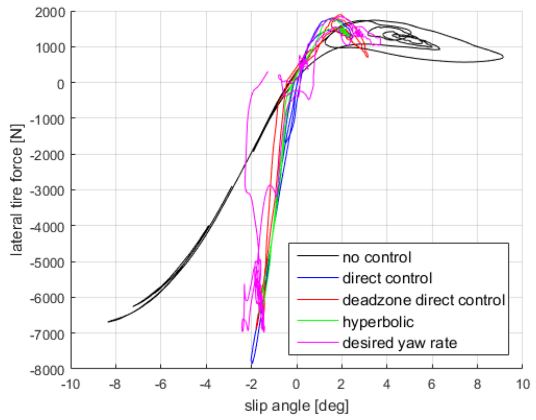


Figure C.5: Front left tire force time[90-130sec]

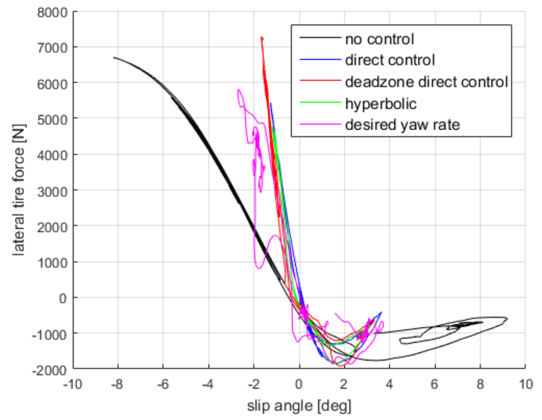


Figure C.6: Front right tire force time[90-130sec]

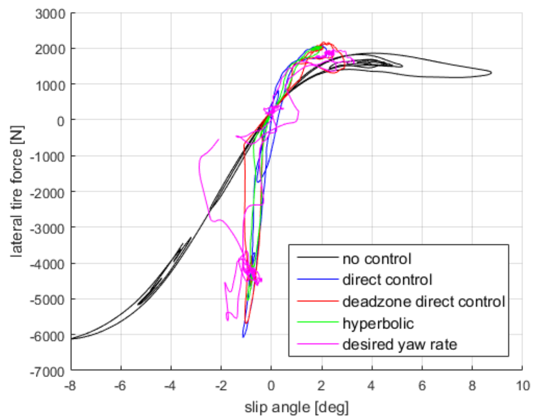


Figure C.7: Rear left tire force time[90-130sec]

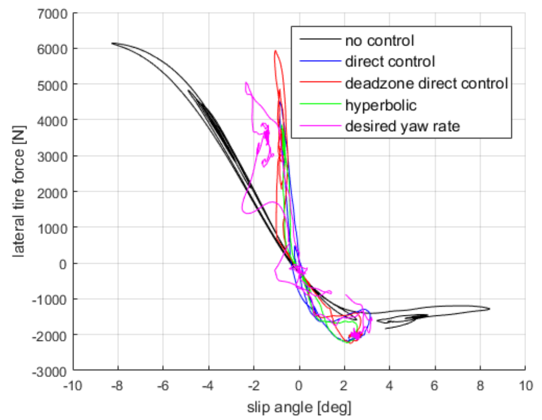


Figure C.8: Rear right tire force time[90-130sec]

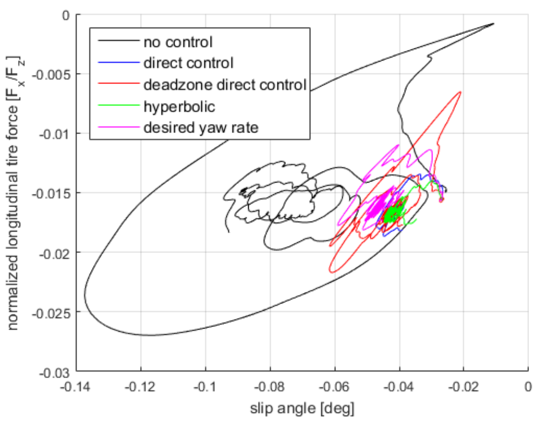


Figure C.9: Front left tire force time[90-108sec]

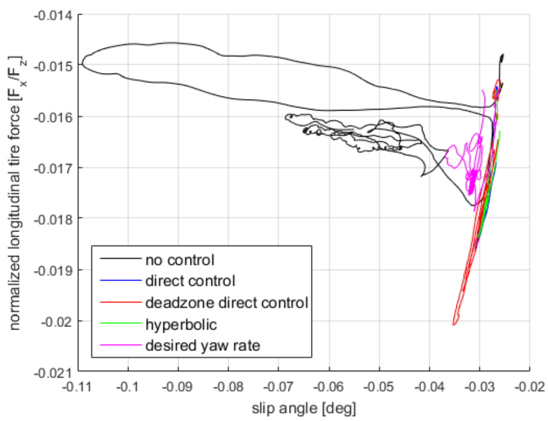


Figure C.10: Front right tire force time[90-108sec]

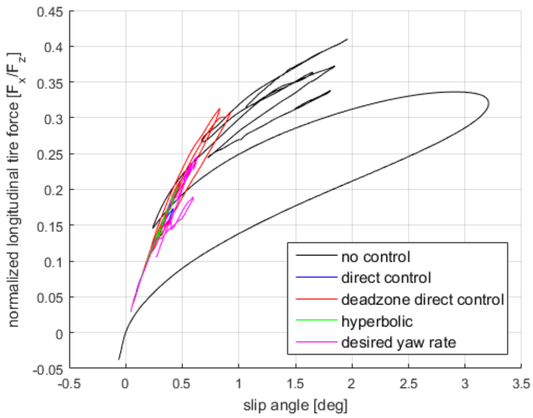


Figure C.11: Rear left tire force time[90-108sec]

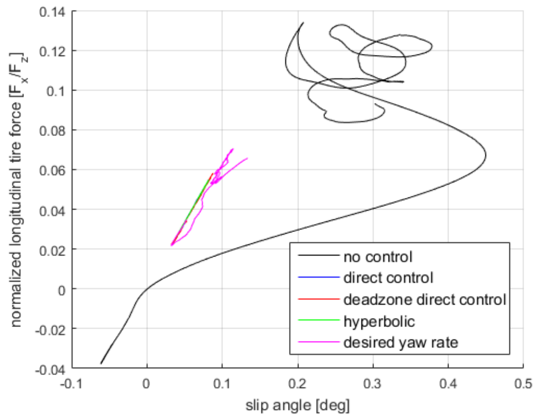


Figure C.12: Rear right tire force time[90-108sec]

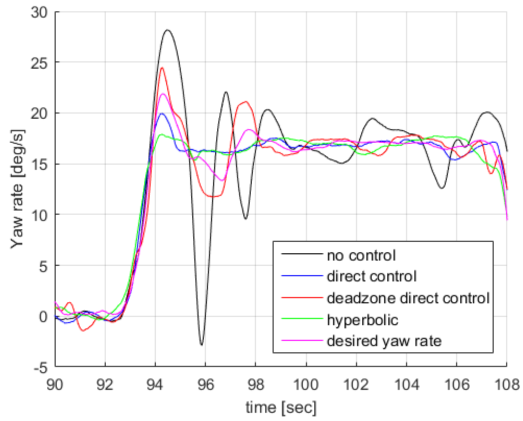


Figure C.13: Yaw rate long corner one participant

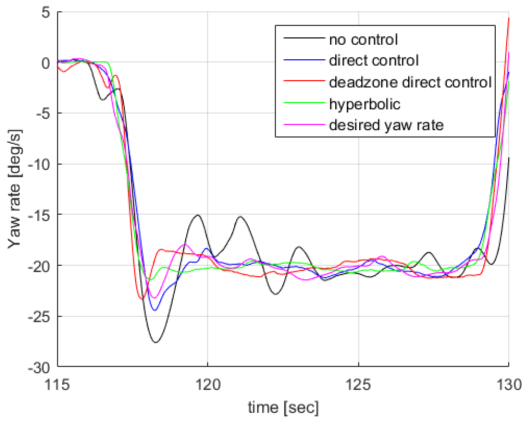


Figure C.14: Yaw rate short corner one participant

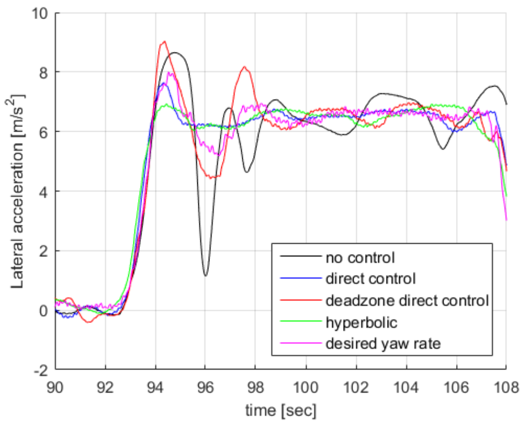


Figure C.15: Lateral acceleration long corner one participant

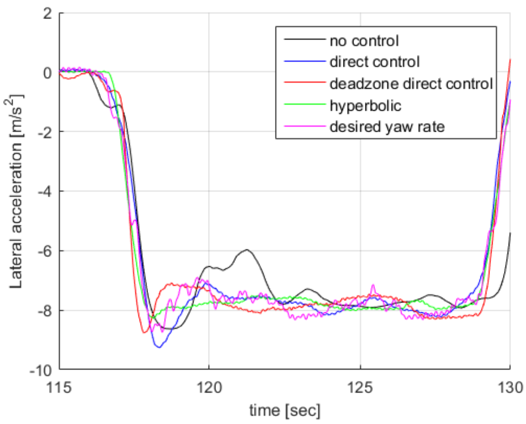


Figure C.16: Lateral acceleration short corner one participant

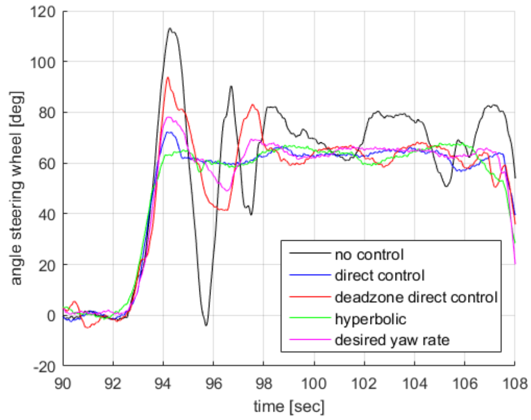


Figure C.17: Steering wheel angle long corner one participant

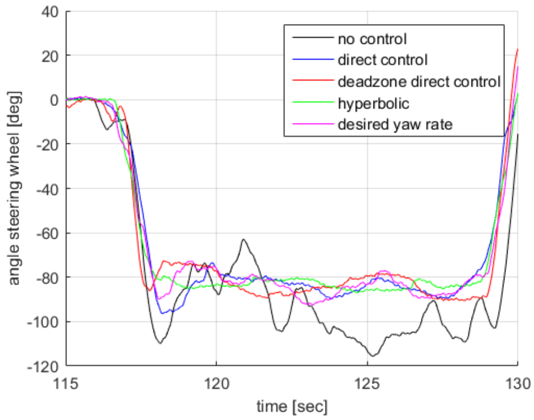


Figure C.18: Steering wheel angle short corner one participant

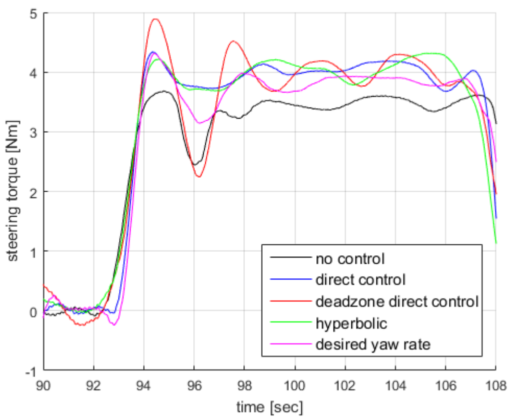


Figure C.19

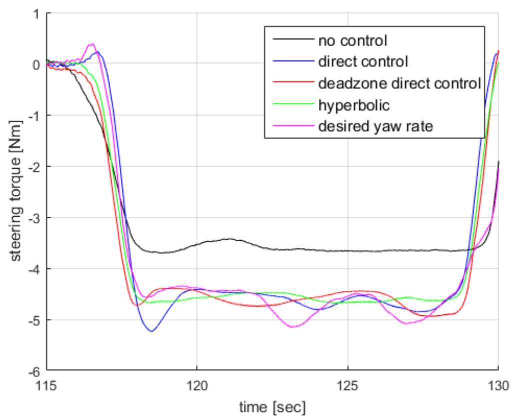


Figure C.20

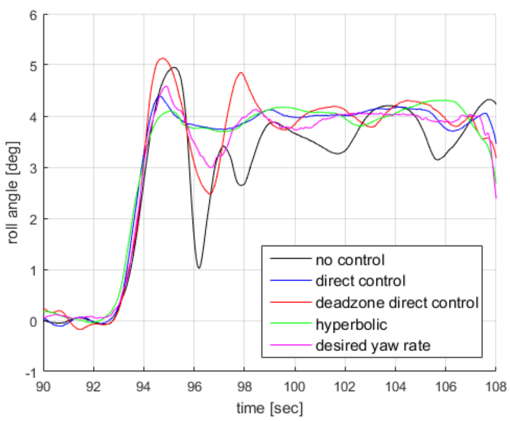
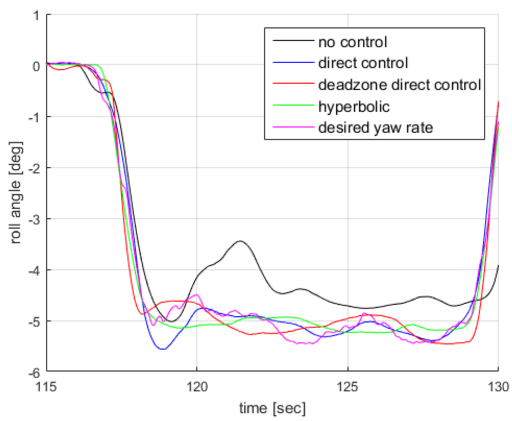


Figure C.21



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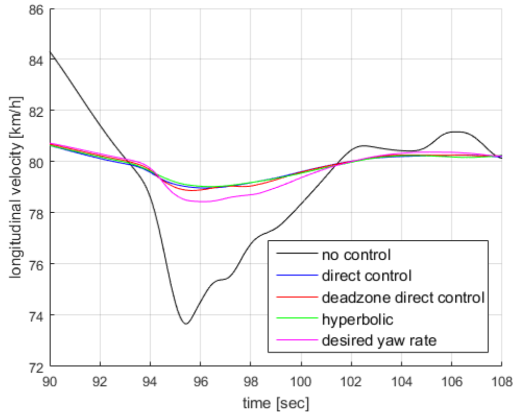


Figure C.23: Velocity of the long corner of one of the participants

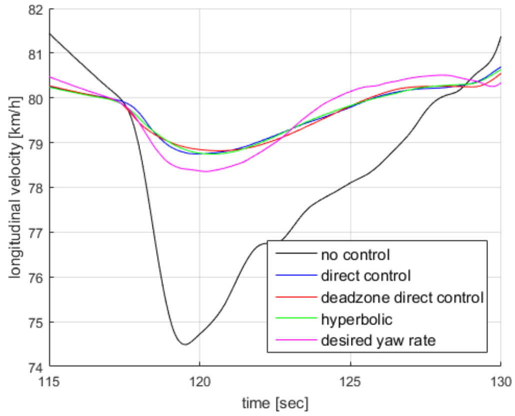


Figure C.24: Velocity of the short corner of one of the participants

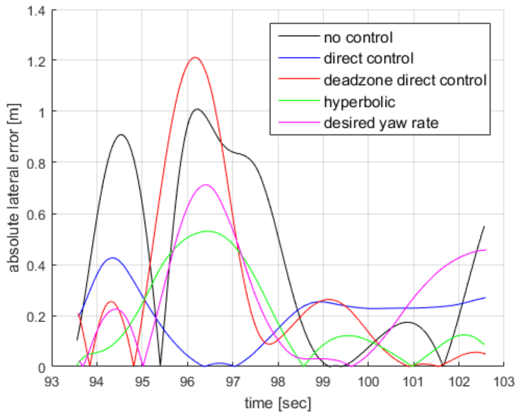


Figure C.25: Absolute lateral position error of the short corner of one of the participants

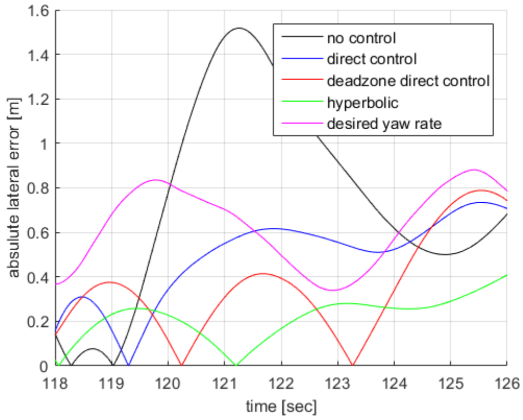


Figure C.26: Absolute lateral position error of the short corner of one of the participants

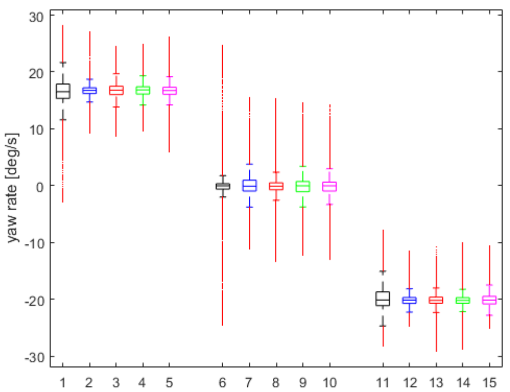


Figure C.27: Yaw rate

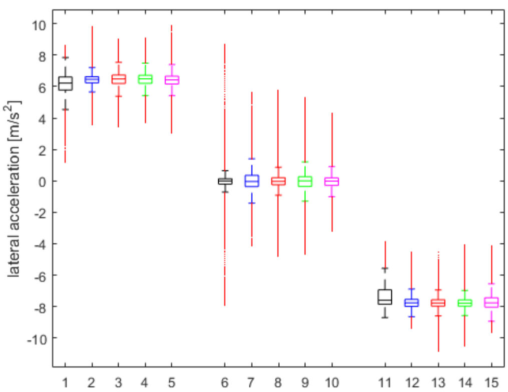
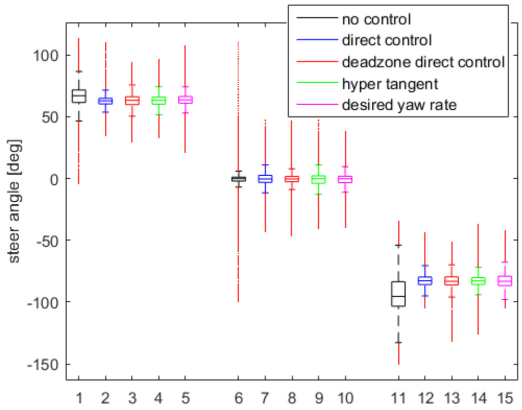


Figure C.28: Lateral accerleration



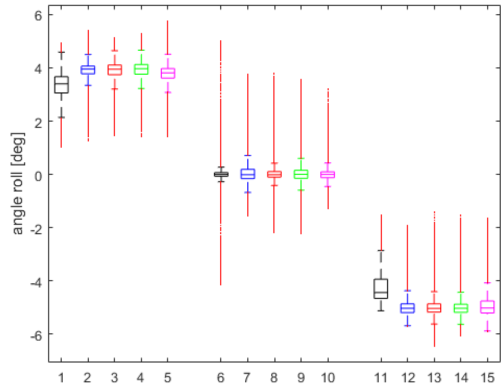
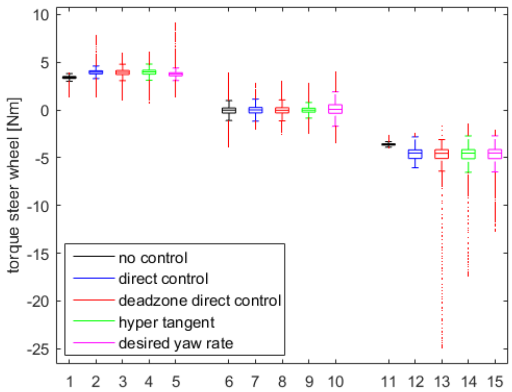


Figure C.29: Steering wheel angle for long corner/straight part/short corner

Figure C.30: Roll angle



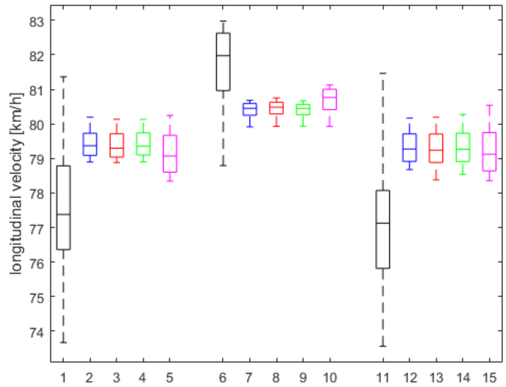
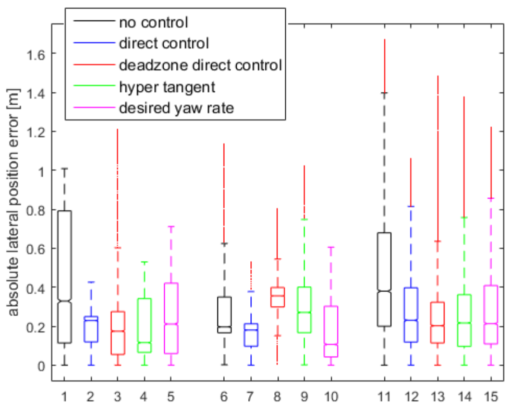


Figure C.31: Torque on steering wheel for long corner/straight part/short corner

Figure C.32: Longitudinal velocity



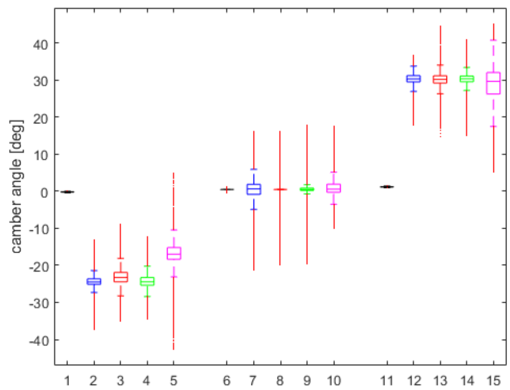


Figure C.33: Absolute lateral error for long corner/straight part/short corner

Figure C.34: Camber angle

D

RESPONSES QUESTIONNAIRE

D.1. CONSTANT CAMBER

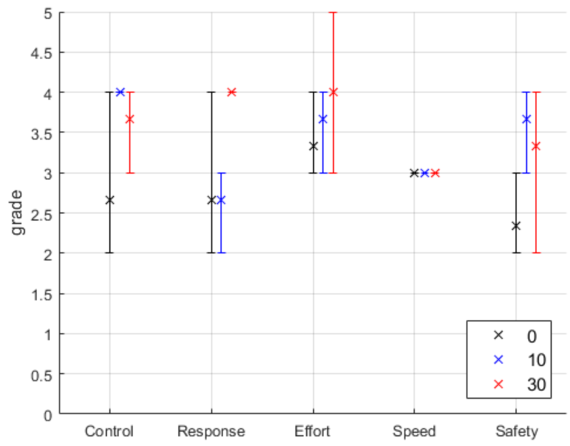


Figure D.1: Questionnaire results for entering the curve

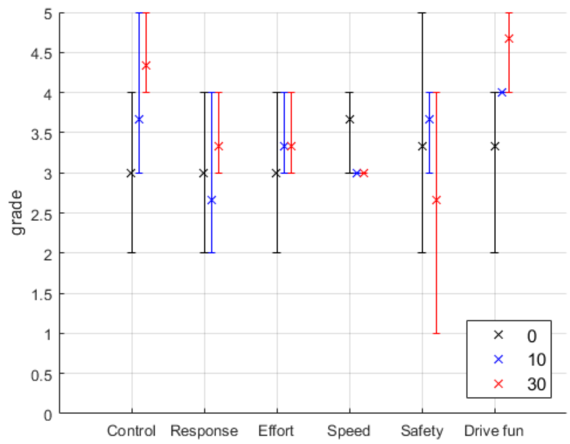


Figure D.2: Questionnaire results for driving on the curve

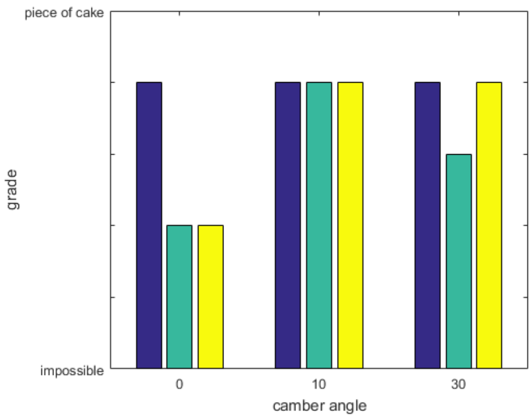


Figure D.3: Control enter the curve

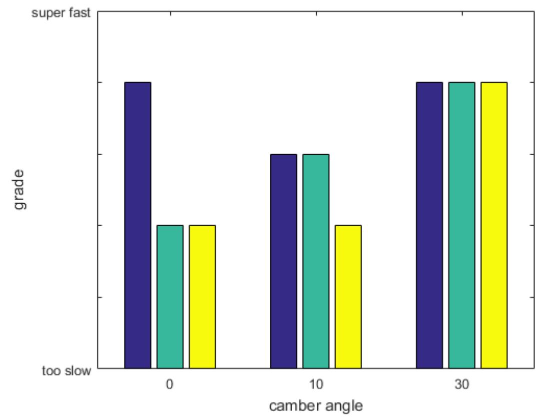


Figure D.4: Response enter the curve

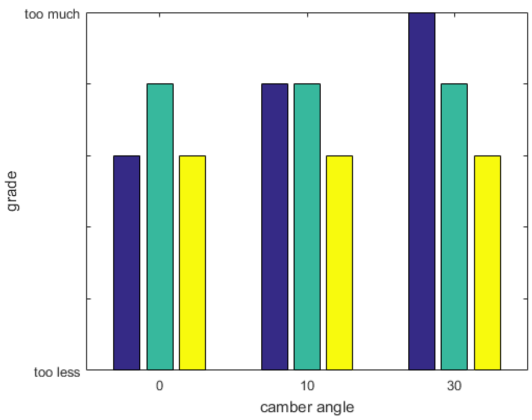


Figure D.5: Effort enter the curve

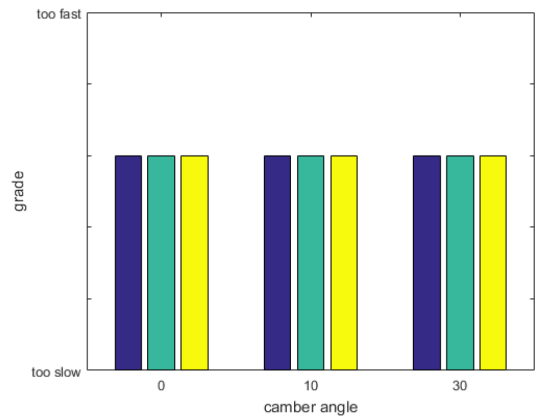


Figure D.6: Speed enter the curve

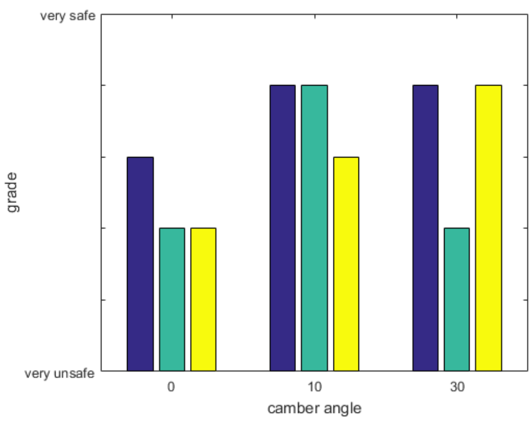


Figure D.7: Safety enter the curve

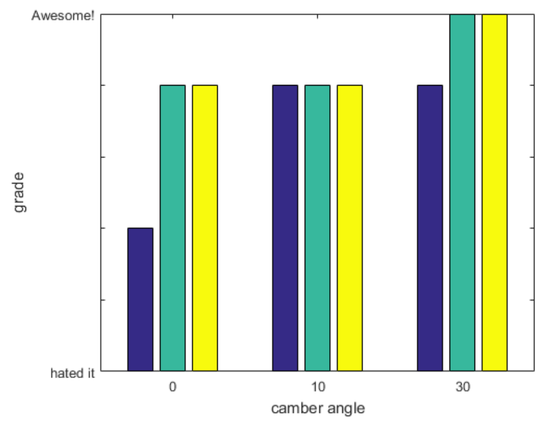


Figure D.8: Fun enter the curve

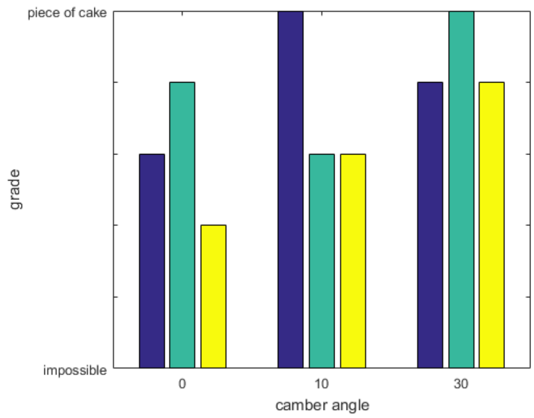


Figure D.9: Control on the curve

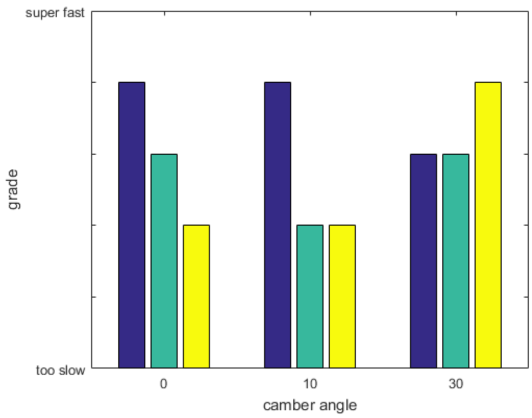


Figure D.10: Response on the curve

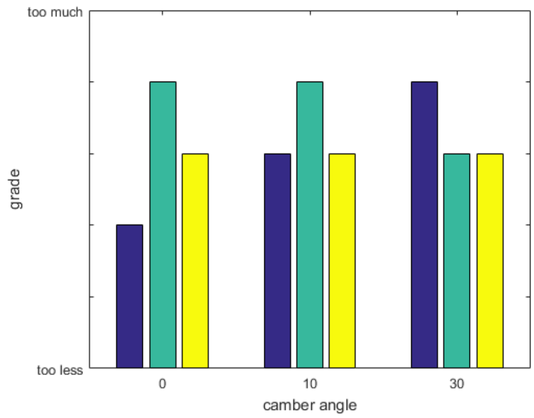


Figure D.11: Effort on the curve

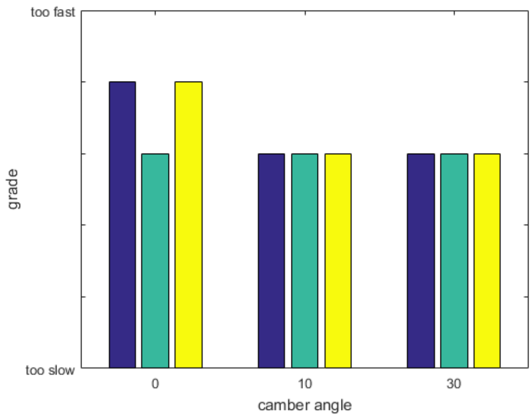


Figure D.12: Speed on the curve

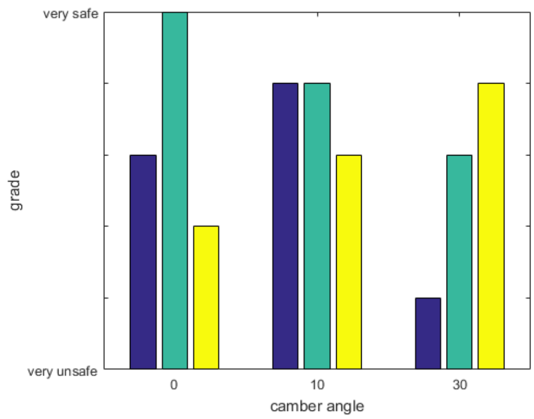


Figure D.13: Safety on the curve

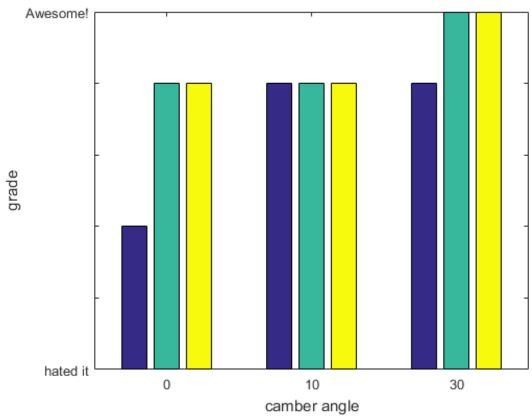


Figure D.14: Fun on the curve

D.2. ACTIVE CAMBER CONTROL

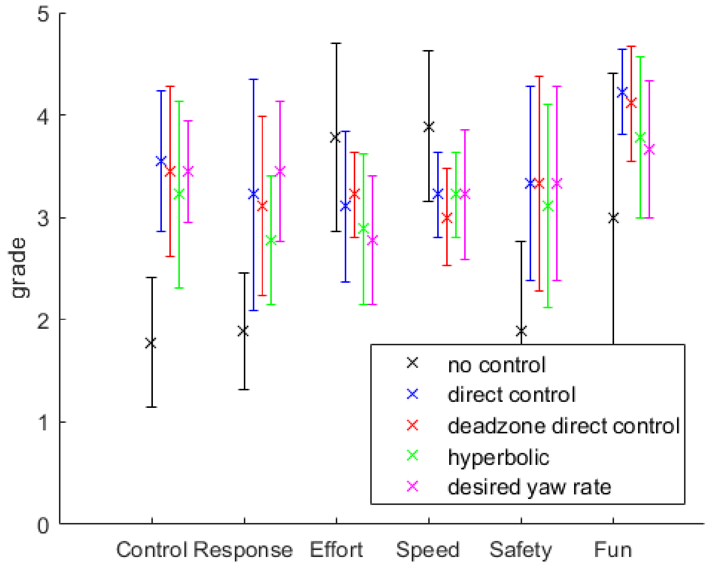


Figure D.15: Results questionaire

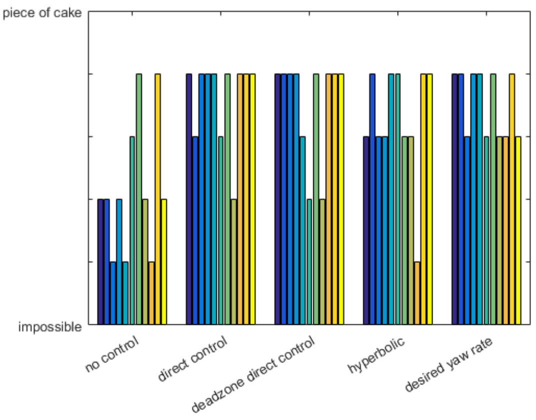


Figure D.16: Control

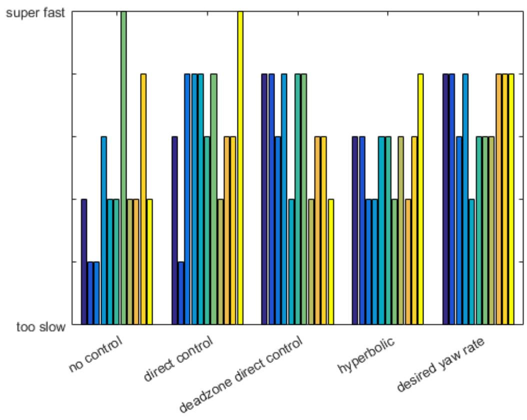


Figure D.17: Steer Response

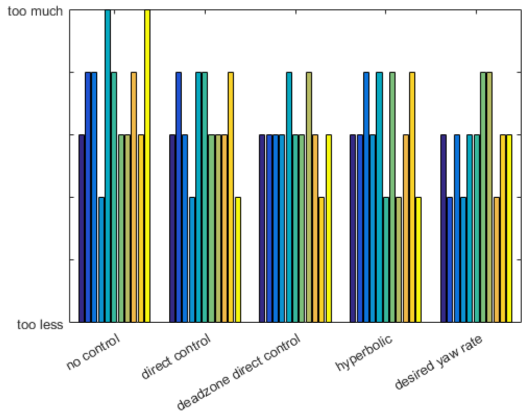


Figure D.18: Steer Effort

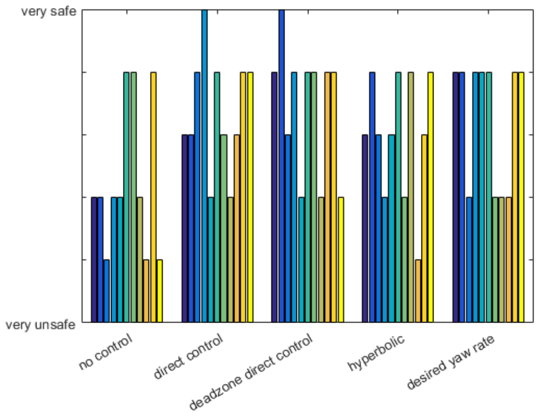


Figure D.19: Safety

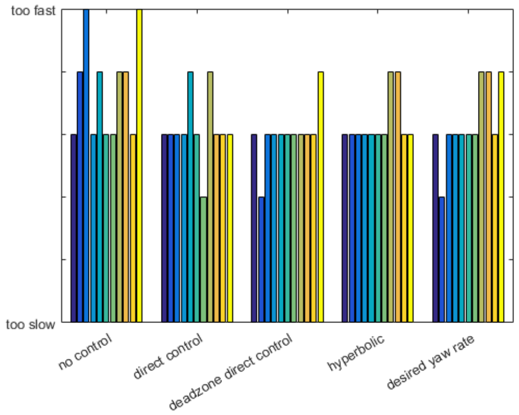


Figure D.20: Speed

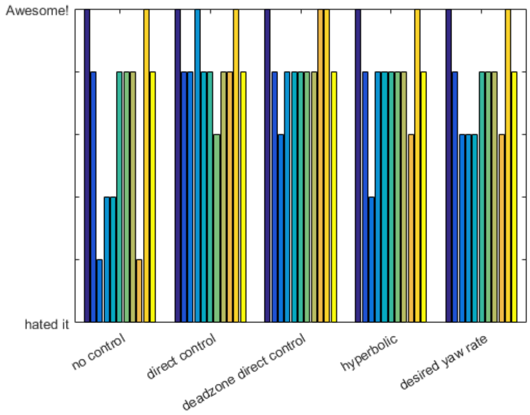


Figure D.21: Fun to drive

Overal feel of control*
How difficult was it to steer / follow the track?

	1	2	3	4	5	
impossible	○	○	○	○	○	piece of cake

Steering response*
How fast does the vehicle respond to a steer input

	1	2	3	4	5	
too slow	○	○	○	○	○	super fast

Steer effort*
How much force/effort did you had to deliver to steer the vehicle

	1	2	3	4	5	
to less	○	○	○	○	○	to much

Safety*
How safe did you feel to drive with this speed and the control you had?

	1	2	3	4	5	
very unsafe	○	○	○	○	○	very safe

Speed*
How did you experience the driving speed?

	1	2	3	4	5	
too slow	○	○	○	○	○	too fast

Drive fun*
Did you enjoy the drive?

	1	2	3	4	5	
hated it	○	○	○	○	○	Awesome!

Figure D.22: Questionnaire

Nomenclature

List of Symbols

α_f	The front tire slip angle
α_r	The rear tire slip angle
α_x	The longitudinal slip angle
α_y	The lateral slip angle
β	The vehicle slip angle
δ	The steer angle
δ_des	The desired steer angle
δ_real	The real steer angle
$\dot{\Psi}$	The yaw rate
$\dot{\Psi}_\mathrm{des}$	The desired yaw rate
$\dot{\Psi}_\mathrm{real}$	The real yaw rate
γ	The camber angle, rotation around the x-axis
C_F	The front tire corner stiffness
C_R	The rear tire corner stiffness
F_x	The tractive force
F_y	The lateral force
F_z	The vertical force
N_β	The vehicle slip moment derivative
N_δ	The vehicle steering moment derivative
N_r	The vehicle yaw moment derivative
Y_β	The vehicle slip force derivative
Y_δ	The vehicle steering force derivative
a	The length of the front wheels axis to the vehicle's CoG
b	The length of the rear wheels axis to the vehicle's CoG
F	The force
K	The vehicle's understeer gradient
k	The spring stiffness coefficient
L	The vehicle length
m	The vehicle mass

R	The corner radius
r	The yaw rate
t	The pneumatic trail
u	The displacement
V	The longitudinal velocity
x-axis	The longitudinal axis
y-axis	The lateral axis
z-axis	The vertical axis

ABBREVIATIONS

ASM	Automotive Simulation Models
BMW	Bayerische Motoren Werke
CoG	Center of Gravity
DoF	Degrees of Freedom
ECU	Electronic Control Unit
ESP	Electronic Stability Program
IAS	Intelligent Automotive Systems
LW	lower wishbone
MF	Magic tire Formula
PID	Proportional Integral Derivative controller
SAE	Society of Automotive Engineers
TMeasy	Tire Model easy to use
UW	upper wishbone
3D	Three Dimensional

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